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Benoît PAILLARD, Jacques Andre ASTOLFI, Frederic HAUVILLE - URANSE simulation of an active variable-pitch cross-flow Darrieus tidal turbine: Sinusoidal pitch function investigation - International Journal of Marine Energy - Vol. 11, p.9-26 - 2015

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## Accepted Manuscript

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PII: S2214-1669(15)00014-4  
DOI: <http://dx.doi.org/10.1016/j.ijome.2015.03.001>  
Reference: IJOME 52

To appear in:

Received Date: 9 December 2014  
Revised Date: 13 March 2015  
Accepted Date: 25 March 2015

Please cite this article as: B. Paillard, J.A. Astolfi, F. Hauville, URANSE simulation of an active variable-pitch cross-flow Darrieus tidal turbine : sinusoidal pitch function investigation, (2015), doi: <http://dx.doi.org/10.1016/j.ijome.2015.03.001>

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# URANSE simulation of an active variable-pitch cross-flow Darrieus tidal turbine : sinusoidal pitch function investigation

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## Abstract

This article describes a 2D CFD simulation implementation of a crossflow tidal turbine, the blades of which have their pitch modified during revolution. Unsteady flow around the turbine is computed with an URANSE method, using the solver ANSYS-CFX. Spatial and temporal discretizations have been studied. The pitch motion of the blades is obtained through mesh deformation, and the main rotation is implemented through sliding boundaries, with general grid interface model. The turbulence model used is  $k\omega$  SST. Langtry Menter transition model was tried but showed high discrepancies with experimental results. Five experimental cases were used to assess the accuracy of the simulation. It provided accurate computed forces for a wide range of tip speed ratios, and proved to be suitable for exploratory simulations. Harmonic pitch control was thus implemented for a tip speed ratio of 5, close to an operational value for a crossflow turbine. First, second and third harmonics pitch function were tested. It was shown that an improvement of more than 50% could be achieved with the second harmonics, with a large reduction in thrust. The flow inside the turbine and close to the blade was examined so that the case of performance improvement due to pitch control could be clearly understood. It was observed that turbine efficiency improvement requires a very slight recirculation and an angle of attack decrease on the upstream part of the turbine, and an angle of attack increase on the downstream part. The flow deceleration through the turbine was found to be a primary factor in pitch function as well. Moreover the hydrodynamic torque and thus the energy required to control the pitch were found to be insignificant.

*Keywords:* Darrieus, Variable pitch, Dynamic Stall, URANSE

## 1. Introduction

Tidal turbines are a power source that shows many significant advantages over other solutions [1]. No land is occupied unlike a dam, there is a steady predictable power output unlike with wind turbines, and low waste or side effects are generated unlike fossil fuel or nuclear power plants. These devices can consist of a classic horizontal axis systems, or cross-flow turbines which have many advantages in water : rectangular surface area helps increasing power production in shallow locations, fixed pitch devices can operate in any flow direction [2] and centrifugal loads, which evolve with  $\omega^2$ , are less severe than in air [3]. Variable pitch cross-flow turbines enable a Darrieus system to improve its performance and decrease radial forces, which do not generate a torque and are responsible for fatigue and system failure [4]. They have been studied at IRENAV since 2007 as a part of the SHIVA project, which aims at the implementation of an experimental variable pitch crossflow tidal turbine [5, 6, 7]. This device has cantilevered blades and its arms are not submersed, thereby cancelling parasitic arm drag.

The optimization of the pitch variation is of prime concern in order to take full advantage of the added mechanical complexity required to obtain these kinematics. However the complexity of the flows associated with these devices requires to choose and check the simulation tools with great care. A stream tube model coupled with the ONERA-EDLIN dynamic stall model was developed [8]. However it does not give precise information on the pattern of the flow associated with each pitch function used. Furthermore these empirical dynamic stall models are limited to the applications they were calibrated for. Finally, the stream tube model is relevant for design and overall performance assessment, but the forces and local phenomena predicted

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$c, l, r, N$	turbine chord, span, radius and number of blades	m
$C_n, C_t$	normal and tangential force coefficients, $C_{n/t} = \frac{F_{n/t}}{\frac{1}{2}\rho c l v_r^2}$ the tangential force creates torque and is the useful part of the hydrodynamic force the normal force creates no torque	/
$\eta$	power coefficient, $\eta = \frac{\text{power}}{\frac{1}{2}\rho 2r l v_\infty^3}$	/
$C_p$	pressure coefficient, $C_p = \frac{p - p_\infty}{\frac{1}{2}\rho v_r^2}$	/
$N$	turbine number of blades	/
$Re$	chord Reynolds number, $Re = \frac{r\omega c}{\nu}$	/
$S_{blade}$	blade surface, $S = cl$	m <sup>2</sup>
$S_{turbine}$	turbine surface, $S = l2r$	m <sup>2</sup>
$v_r$	relative flow velocity $v_r = v_\infty \sqrt{\cos(\theta)^2 + [\sin(\theta) + \lambda]^2}$	m/s
$v_\infty$	upstream flow velocity	m/s
$\alpha$	incidence	rad
$\lambda$	tip speed ratio $\lambda = \frac{r\omega}{v_\infty}$	/
$\omega$	rotational velocity	rad/s
$\sigma$	system solidity, $\sigma = \frac{Nc}{2r}$	/
$\theta$	main angular position	rad

Table 1: Nomenclature

20 remain questionable [9]. Being able to accurately compute the unsteady forces is mandatory in order to  
21 optimize the pitch function, which is why a RANSE simulation was developed.

22 The purpose of the present work is to implement and validate a 2D RANSE simulation of a lifting foil  
23 undergoing a Darrieus kinematic, without pitch variation. The quantities of specific interest are the forces  
24 and torques applied to the blades, and the flow around the whole system. It is the preparatory work to  
25 the development of the variable pitch cross-flow turbine simulation. First the computation methods used  
26 are introduced in comparison with the previous existing references. Then the simulation parameters and  
27 discretizations are studied and chosen, and will be used as a basis for future works. The results from the  
28 numerical simulation are then verified against experimental results from bibliographic references. Finally  
29 the flow field and pressure coefficient curves are discussed, in order to give a better insight to the reader on  
30 the flow physics.

31 The purpose of the present work is to present the implementation of a 2D unsteady RANSE simulation  
32 of a lifting surface undergoing two simultaneous motions : a Darrieus kinematic, which is a rotation around  
33 a point away from the foil ; and a dynamic pitch control, defined as a rotation around the quarter chord,  
34 adapting the angle of attack of the lifting surface to the local flow state. This second kinematics is aimed  
35 at performance improvement and radial forces reduction. The quantities of specific interest are the forces  
36 and torques applied to the blades, and the flow around the whole system.

## 37 2. Material and methods

### 38 2.1. Bibliographic background

39 Cross-flow turbines feature complex fluid phenomena, and have first been studied using momentum based  
40 models for overall performance prediction. These models cannot predict local flow accurately, and can fail  
41 in unsteady forces prediction. Unsteady CFD appeared as a good solution to overcome these limitations.  
42 First computations of this kind were implemented for cross-flow turbines in the late 1990's. Various recent  
43 projects are being carried out in this area. Allet and Paraschivoiu [10] simulated a 2D one blade turbine  
44 with NACA 0015 section. The governing equations were solved by the streamline upwind Petrov-Galerkin  
45 finite element method. Turbulence effects were introduced in the solver by the algebraic Cebeci-Smith model  
46 (CSM) and the non-equilibrium Johnson-King model (JKM). The forces oscillations were in agreement with  
47 the experimental results, even though the extremum values and some vorticity shedding could not be re-  
48 produced. Ferreira [11] compared PIV measurements with several types of CFD models : LES, DES and  
49 RANSE, combined with several turbulence models. The best model was found to be the DES model. The  
50 spatial grid did not enable the LES model to give correct results in the boundary vicinity. Consul et al.  
51 [12] assessed the influence of solidity on performance. Turbulent models Spallart-Allmaras and  $k - \omega$  SST  
52 were compared, and the latter was found to be more accurate. Qualitative agreement with experimental



53 results were obtained. Klaptocz et al. [13] used Spallart-Allmaras model to simulate a turbine fitted in-  
 54 side a diffuser. The turbine was modelled using a rotating ring. Comparison with experiment is poor for  
 55 local forces, but global efficiency curves are obtained accurately. An innovative hybrid RANSE/momentum  
 56 model was studied by Antheaume et al. [14]. The global flow through the turbine is simulated with a RANSE  
 57 model, and the local influence of the blades on this flow is computed with a stream tube model. It finds its  
 58 application mostly on farm modelization.

59  
 60 Variable pitch crossflow turbines can be divided into two categories. Active pitch controls are devices for  
 61 which the pitch angle function is constant during operation, but can be changed through a controller when  
 62 operating conditions change. Passive pitch controls are devices for which azimuthal position and rotational  
 63 velocity have an impact on the pitch function, often implemented through mass-spring arrangements [15].  
 64 The former will be studied in this paper. Several experimental projects have been carried out to evaluate  
 65 the use of variable pitch on crossflow axis turbines. Miao et al. [16] compared starting torques for 70°  
 66 and 10° maximum pitch amplitude. With 70°, a higher starting torque was reached, but a higher efficiency  
 67 in operation was anticipated numerically for 10°. Hwang et al. [17] studied a cycloidal turbine similar to  
 68 Pinson turbine, for which a cam and an eccentric were used, yielding a quasi-sinusoidal pitch function. A  
 69 power coefficient of 0.25 was achieved. Grylls et al. [18], Nattuvetty et al. [19] and Erickson et al. [20] also  
 70 tested numerically and experimentally a device similar to Pinson turbine including a pitch offset, and could  
 71 reach a  $\eta$  of 0.45. The company McDonnell tested a variable pitch wind turbine [21], for which the pitch law  
 72 was optimized numerically beforehand, and implemented by manufacturing the corresponding cam track.  
 73 They obtained a maximum  $\eta$  of 0.39. Vandenberghe et al. [22] studied a device similar to Pinson turbine,  
 74 with a second order harmonic pitch control. Asymmetric pitch law could thus be obtained, in order to adapt  
 75 the pitch to the reduced velocity encountered downstream. They reported a gross maximum experimental  
 76  $\eta$  of 0.436, after arm drag and control loss subtraction. First order harmonic was still best for  $\lambda$  above  
 77 3. Finally the company WPI studied an individual pitch control device [23]. A performance coefficient of  
 78 0.5 was measured experimentally at NTNU. It should be noted that the holding arms of this turbine are  
 79 outside of the water. This effectively cancels arm drag, which can be as high as .25 points of  $C_p$ . The same  
 80 configuration will be used SHIVA experimental platform.

81  
 82 The RANSE solver CFX was chosen because of the previous works on unsteady forces simulation on  
 83 NACA foils at IRENAV [24]. The various simulations that had been implemented were considered a valid  
 84 and strong basis for such a demanding flow simulation. The complexity and accuracy of  $k - \omega$  SST model  
 85 were seen as a valid step between the faster but less physically accurate DMST [8], and future work with  
 86 more complex fluid models such as DES or LES.

## 87 2.2. Experimental reference

88 Two sources were considered for validation [25, 26]. Both experimental devices were straight blades,  
 89 fixed pitch Darrieus turbines, with their blades connected at the quarter chord. Various solidities and  $\lambda$ s  
 90 were considered, enabling a thorough validation process. The data are gathered in table 2. Cases 1, 2 and  
 91 3 were carried out in a towing tank, hence the very low flow velocity.

92 These results were chosen because local blade loads were measured, which gives a more accurate validation  
 93 than averaged values such as the coefficient of performance. To the best knowledge of the author, there exist  
 94 no more recent publications, or pitch-controlled cases of local blade loads measurement on crossflow axis  
 95 turbines. Such an experimental device is under construction at IRENAV [7]. Even though both experimental  
 96 campaigns were carried out around  $Re = 4.10^4$  which is relatively low, they are relevant since they cover a  
 97 wide range of tip speed ratios including an operational one. No detailed experimental data could be found for  
 98 higher Reynolds numbers, at which full scale turbines will operate. No uncertainty quantification on forces  
 99 measurement were provided in these articles. Blade deformation, calibration and alignment uncertainty,  
 100 probes accuracy were not mentioned. Furthermore, the tangential force is an order of magnitude lower than  
 101 the normal force, which makes its measurement accuracy even more questionable. However these results  
 102 are the most detailed available at the time this article was written. Two separate campaigns were considered  
 103 in order to reduce as much as possible such uncertainties.

104 The measured quantities in this study are the normal and tangential force coefficients defined by equations  
 105  $C_n = F_n / \frac{1}{2} \rho c l V_\infty^2$  and  $C_t = F_t / \frac{1}{2} \rho c l V_\infty^2$ . The tangential force generates torque, whereas the normal force  
 106 does not. Hence they are relevant in the study of cross-flow turbines since they are respectively the non-  
 107 productive and the useful forces for torque production. The coordinate system used is defined in figure 1.  
 108 The turbine rotates in anti-clockwise direction. Angular datum is the position of the blade when it leaves  
 109 downstream part and enters upstream.

	Case 1	Case 2	Case 3	Case 4	Case 5
fluid	water			air	
upstream fluid velocity	0.183 m/s	0.091 m/s	0.061 m/s	3.2 m/s	6.4 m/s
blades number	2			2	1
rotor diameter	1.22 m			0.61 m	
span	1.1 m			0.61 m	
chord	0.0914 m			0.061 m	
solidity	0.15			0.2	0.1
blade section	NACA0012			NACA0018	
rotational speed	43 rpm			300 rpm	
$\lambda$	2.5	5	7.5	3	1.5
chord Reynolds number	4.10 <sup>4</sup>			3, 80.10 <sup>4</sup>	
$\eta$	0.077	0.362	-0.022	-0.129	-0.018
source	[25]			[26]	

Table 2: Experimental cases used in this article

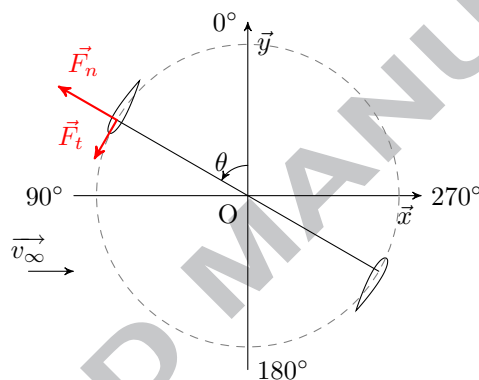


Figure 1: Coordinate system used for the present study

### 110 2.3. Numerical procedure

#### 111 2.3.1. Simulation configuration

112 The commercial RANSE solver CFX is used for this simulation. A ring rotating at constant speed is  
 113 enclosed inside a steady domain, as shown in figure 2. The flow across the non-conformal boundaries is  
 114 obtained with a general grid interface or GGI [27]. A 2D approach is relevant for both experimental cases,  
 115 since the aspect ratios are 12 for cases 1, 2 and 3 ; and 10 for cases 4 and 5, and the blades were positioned as  
 116 close as possible from the wall of the experimental infrastructure used, thereby cancelling finite wing effect  
 117 ; and the arms loss were cancelled by keeping the holding arm outside of the water in [25], and reduced  
 118 using a single streamlined arm together with strings in [26]. The low  $\lambda$  used in the latter contribute to  
 119 further arm loss reduction. Progressive start of the rotating domain was tested, but no difference in forces  
 120 and torque with step start could be observed after one revolution. The domain is 10 turbine diameters  
 121 in the transverse direction, 3 diameters upstream and 10 diameters downstream. The inlet condition is a  
 122 fluid velocity on axis x. The tip speed ratio considered defines the norm of this velocity vector, since the  
 123 rotational velocity remains constant. The inlet turbulence rate is fixed at 5%. Outlet condition is defined  
 124 with a relative pressure equal to 0 Pa. For the streamwise boundaries a symmetry condition is used. Finally  
 125 the blades are defined by a solid no slip boundary condition. The spatial resolution scheme used is a second  
 126 order Euler method, and the temporal scheme is a second order backward Euler method. The implicit form  
 127 of the solver does not impose any numerical limitation on the Courant number value. However the physical  
 128 problems associated with Darrieus kinematics requires a small enough time step in order to accurately  
 129 resolve transient flow details. This is studied in part 2.3.3. Convergence is measured through residual RMS,  
 130 for which criterion is  $10^{-4}$ .

nr. of elements	$V/V_\infty$	rel. difference
200 000	0,536	14,50%
300 000	0,471	0,64%
400 000	0,468	-

Table 3: Relative difference in flow deceleration for several spatial discretization, case 2,  $\lambda = 5$

### 2.3.2. Spatial discretization

A rotating ring inside a steady domain was used. The meshing is structured in the outside domain and in the ring, and unstructured, tet-dominant in the center part. An O-grid technique was used around the foils, and the ring width was set to 3 chords so that the mesh can deform smoothly during pitch variation, and the high gradient around the blades are kept away from the sliding interfaces. Mesh spacing continuity across the interfaces ensured an accurate simulation across them. For the boundary layer discretization, conclusions on mesh study from Ducoin [24] were used. 40 cells were used for boundary layer thickness, and 300 cells in the chordwise direction. First cell size was adapted to the experimental cases in order to account for diameter and flow differences. The dimensionless wall distance  $y^+$  was always kept below 1 in order to compute viscous layers accurately. A convergence study on spatial discretization was carried out. Three different meshes were built, including respectively 200 000, 300 000 and 400 000 elements. Flow at turbine center divided by inlet flow is the quantity used as the convergence criteria, in order to accurately measure the upstream-downstream interaction. Mesh independence is obtained for 300 000 elements. Table 3 shows the results for case 2. The final mesh consists of 50 000 cells in the center, 100 000 in the outer part, 150 000 in the ring.

The blades pitch angles are changed over time. Previous studies implemented that feature through rotation of circular regions enclosing each blades [17]. Another GGI method was then used. In the present study this is obtained through mesh deformation in the ring containing them. The solver deforms the mesh at every time step. The mesh is treated as a deforming structure, with rigidity increasing with smaller cell size, so that the small cells in the boundary layer remain undeformed. An Arbitrary Lagrangian-Eulerian (ALE) formulation is used to solve the motion of the mesh. The Lagrangian deformation enable easy and precise boundary and interface conditions, but may create severe mesh distortion. On the other hand Eulerian deformation creates no distortion, but solid boundary and interfaces are difficult to apply, since boundary and mesh nodes do not necessarily coincide. With the ALE approach both technique are used in a single domain, so that boundaries can be precisely tracked, and cell quality can be optimized away from them. The motion of each node is taken into account by modifying the conservation equations. The control volume changes over time, and the velocity of its boundaries is used in the ALE formulation. An illustration of the meshing strategy and deformation can be found in figure 2.

### 2.3.3. Temporal discretization

The time step is controlled through the angular step since rotational velocity is constant. The following azimuthal steps were tested :  $0.5^\circ/\text{step}$ ,  $1^\circ/\text{step}$  and  $2^\circ/\text{step}$ , which gave CFL-RMS values of 3, 5 and 10 respectively. The forces coefficients do not change below an angular step of  $1^\circ$ . This value will be used in the following study. The results for case 4 after 9 revolutions can be found figure 3.

### 2.3.4. Wake survey

An accurate description of the impact of upstream part on downstream part requires a fully developed turbine wake. For 3 different experimental cases the evolution of tangential force was inspected for convergence as the wake develops. The results are shown in table 4. The wake became fully developed for all cases at the ninth revolution. This value will be used in further study. The number of revolutions required increases with  $\lambda$ . Higher  $\lambda$  leads to faster rotation, hence less time per revolution for the wake to convect.

### 2.3.5. Turbulence modelization

The low chord Reynolds number and the unsteadiness of the flow raise questions about the choice of turbulence and transition models. A Large Eddy Simulation (LES) or Detached Eddy Simulation (DES) model has shown better results for Darrieus turbine simulation at the cost of longer computation time [9]. However this work aims at the optimization of a pitch angle function, which will require many cases to be considered. DES is at least one order of magnitude more computer intensive than RANSE models [27], which makes the cpu time prohibitive for our goal. Furthermore this study was carried out in 2D, whereas DES and LES were developed to simulate the 3D nature of turbulence. Even though promising results were

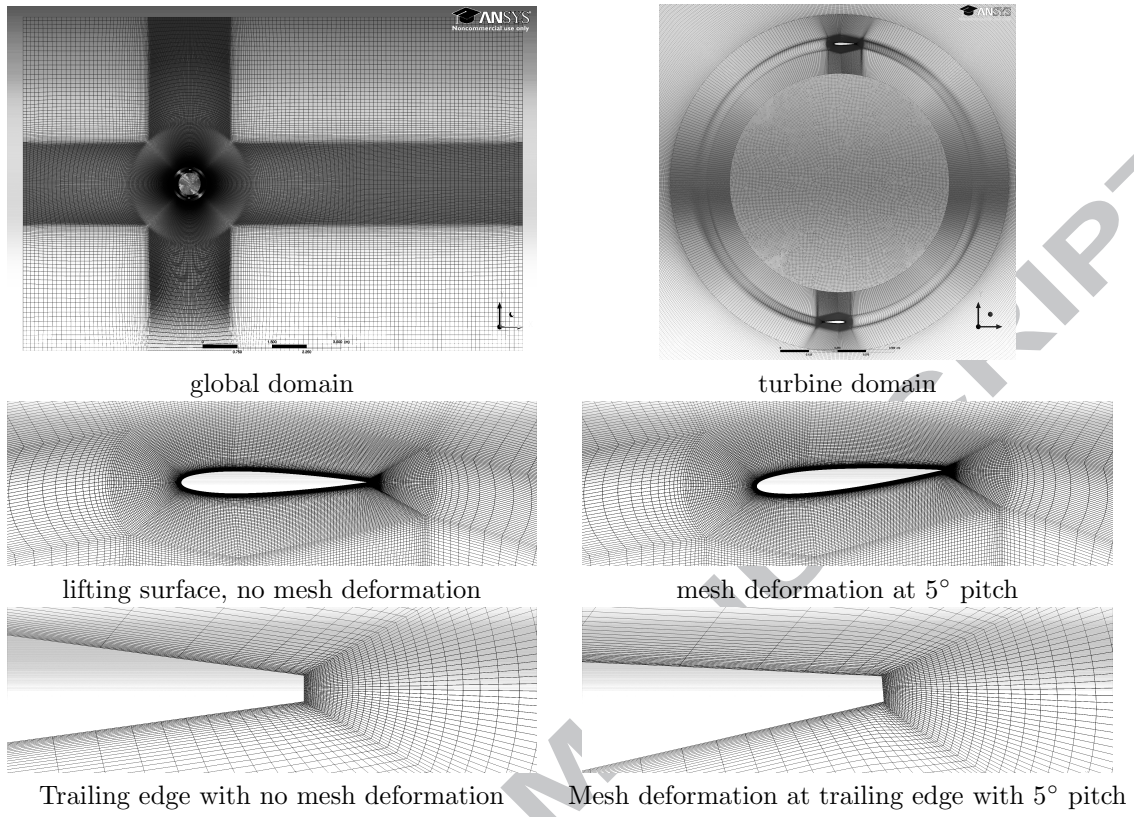
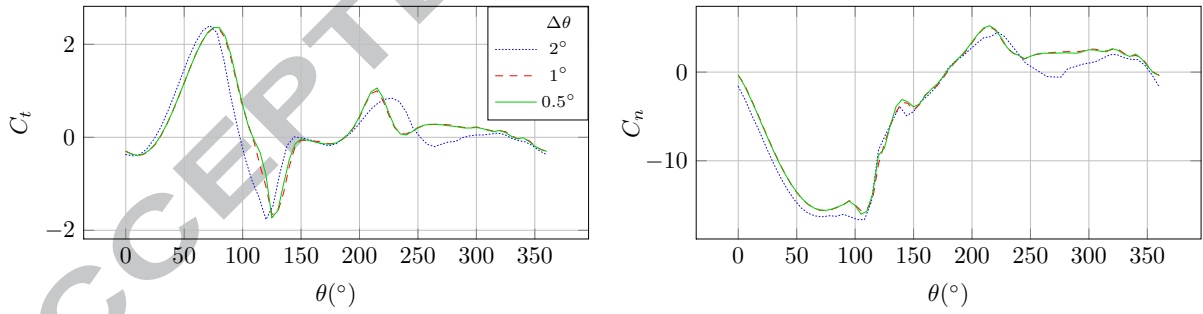


Figure 2: General meshing configuration

Figure 3:  $C_t$  and  $C_n$  for case 4 ( $\lambda = 3$ ) as a function of angular position for three angular steps ( $\Delta\theta = r\Delta t$ )

revolutions	case 1	case 2	case 3
5	1,34%	6,40%	8,03%
6	0,56%	4,11%	5,03%
7		2,50%	3,14%
8		1,33%	1,83%
9		0,54%	0,82%

Table 4: Relative tangential coefficient variation with wake development for cases 1, 2 and 3

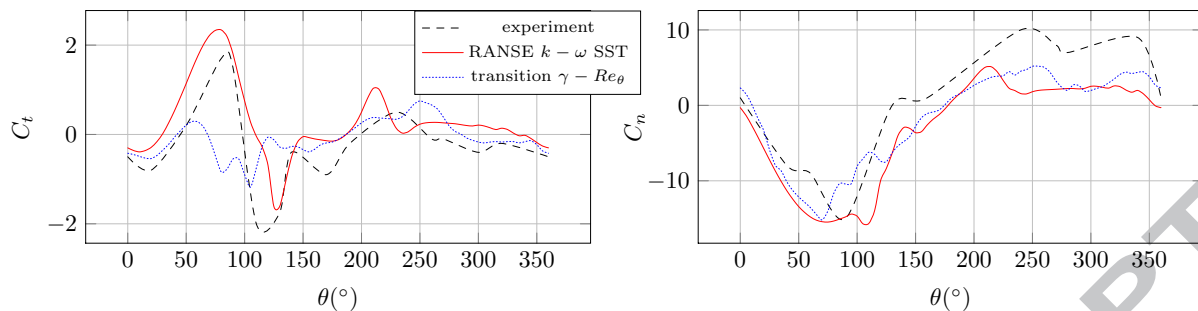


Figure 4:  $C_t$  and  $C_n$  for case 4 ( $\lambda = 3$ ), with/without transition model

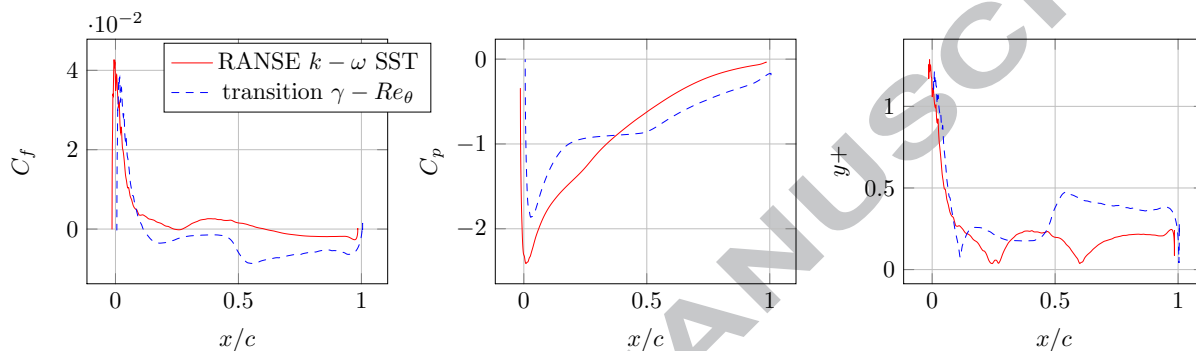


Figure 5: Pressure, friction coefficients and  $y^+$  parameter for case 4 ( $\lambda = 3$ ) at  $\theta = 50^\circ$

178 obtained using LES or DES in 2D when compared to RANSE and 3D LES or DES [9], [28], it was decided  
 179 not to use it for the present study. The turbulence model used in this study is  $k - \omega$  SST, which combines  
 180  $k - \omega$  model close to boundaries and  $k - \epsilon$  in further domain, with blending functions between them. It is  
 181 a two equation model, which showed good results on solid boundary flows involved in cross-flow turbines  
 182 simulation [29].

183 Transition from laminar to turbulent boundary layer [30] can have a very strong influence on flows  
 184 around lifting surfaces. It can improve performance by delaying separation [31], and it can degrade it when  
 185 a separation bubble occurs [32].  $\gamma - Re_\theta$  transition model was tested [30, 33]. It simulates transitional  
 186 behavior by comparing a local Reynolds number to a reference value  $Re_\theta$  correlated experimentally, and by  
 187 using an intermittency number  $\gamma$ . Fully turbulent  $k - \omega$  SST and turbulent with transition  $\gamma - Re_\theta$  model  
 188 were compared in the present study. Force results from simulation of case 4 are given in figure 4. Unlike  
 189 numerical results, experimental normal force coefficient between  $0^\circ$  and  $90^\circ$  does not decrease steadily, as  
 190 should be anticipated from a constant incidence increase. This is a typical result during laminar separation  
 191 bubble formation [34]. However the transition model could not reproduce it, and an early stall is simulated  
 192 on the upstream part, clearly visible on  $C_t$  curve.

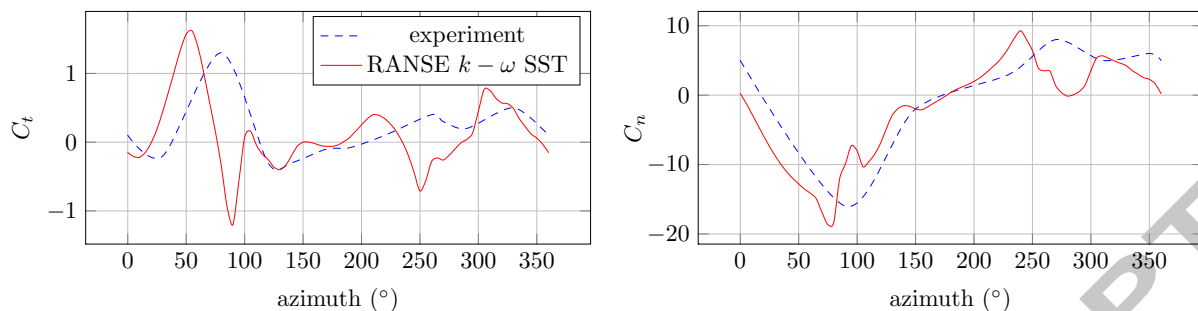
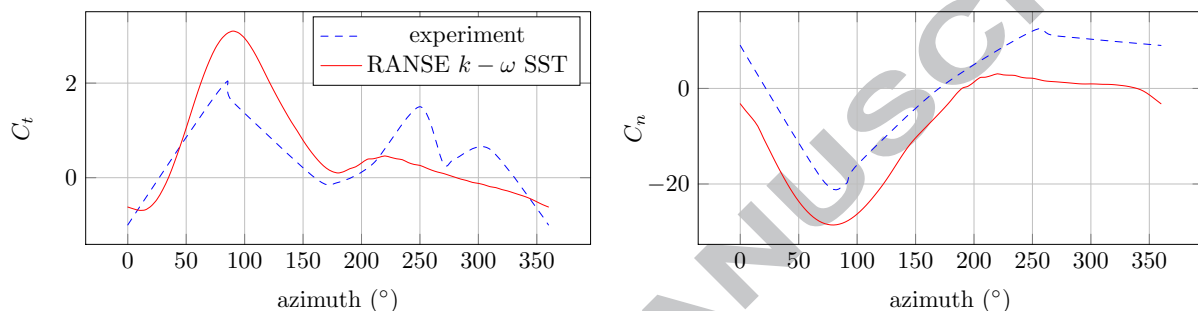
193 Friction and pressure coefficients  $C_f$  and  $C_p$ , and  $y^+$  are shown in figure 5 for  $\theta = 50^\circ$ . Transition  
 194 model reaches a negative  $C_f$  value at  $0.1c$ , and  $C_p$  curve has a plateau, two signs of stall onset.  $y^+$  curve  
 195 rise steeply at  $0.5c$  which shows the transition onset. This leads to the conclusion that laminar stall is  
 196 predicted by the transition model, unlike what the experimental results show. The inlet turbulence rate  
 197 has a strong influence on the transition model behaviour and has not been studied here. Stall consequences  
 198 are over predicted with the transition model, which leads to the choice of the turbulent  $k - \omega$  SST model  
 199 in the following study. The transitional flow remains an important phenomenon for this range of Reynolds  
 200 number. Numerical and experimental testing is currently being carried out at IRENAV on that topic [35].  
 201 The first results show an important influence of the laminar separation bubble on transition and stall, which  
 202 might explain the early stall computed by the model (figure 4).

### 203 3. Results and discussion

#### 204 3.1. Validation

205 Comparison with experiments on force coefficient is studied in this section. Figure 6 shows the results  
 206 for case 1, which is close to where most crossflow axis turbines usually operate. Agreement is average. An



Figure 6: Validation on  $C_t$  and  $C_n$  for case 1 ( $\lambda = 2.5$ ,  $\sigma = 0.15$ )Figure 7: Validation on  $C_t$  and  $C_n$  for case 2 ( $\lambda = 5$ ,  $\sigma = 0.15$ )

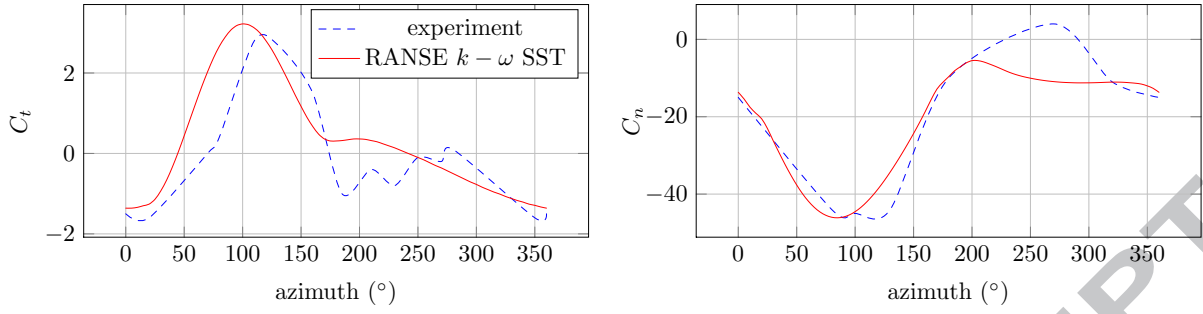
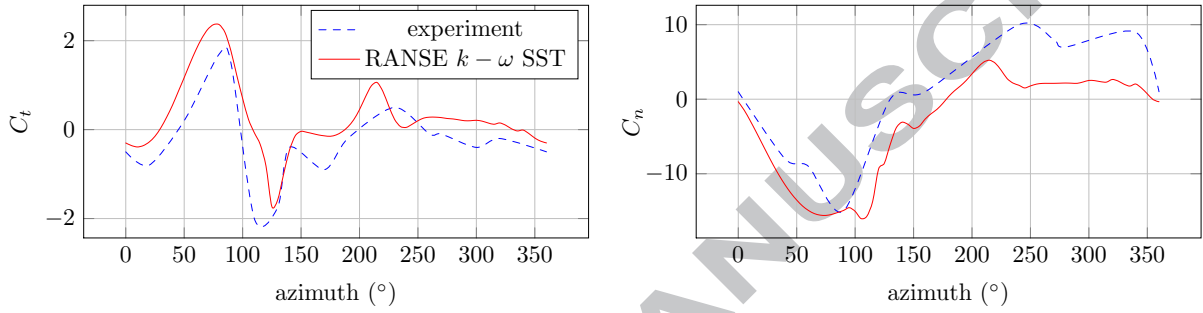
207 angular offset is observed which could be explained by a slight blade misalignment. Stall is predicted early  
 208 and stronger. A large  $C_t$  drop at  $\theta = 90^\circ$  is predicted, a sign of a deep stall, not observed experimentally.  
 209 Numerical and experimental turbines thus operate in complete different conditions at that azimuth. This is  
 210 the only case where such a difference occurs. It can be noted that with  $\lambda$  and solidities of 3 and .2 respectively  
 211 for case 4, experimental stall is observed (see figure 9). Reducing solidity creates less flow blockage and thus  
 212 less flow deceleration ; and lower  $\lambda$  gives a higher incidence. These characteristics increase stall inception,  
 213 and one should then expect case 1 to exhibit stall as well. A blade roughness difference between the  
 214 experimental sources can be incriminated. The dynamic stall modelization and transitional behaviour are  
 215 also questionable. Further numerical testing are currently being carried out at IRENAV.

216 Case 2 is shown in figure 7. Computed upstream  $C_t$  is 50% higher than measurements. In the downstream  
 217 region, numerical  $C_t$  remains very low compare to the experiment.  $C_n$  results show an offset as well. The  
 218 discrepancies can be due to a slight misalignment of the blade pitch in the experiments as suggested by similar  
 219 results obtained by [9, 36, 37] where Darrieus turbine for different conditions of pitch and pitching axis were  
 220 studied. It was observed that a small pitch misalignment can create such a deviation. Indeed when the foil is  
 221 pitched nose-out the incidence decreases upstream and increases downstream, which changes the tangential  
 222 force and normal forces as described above. Numerically, an inaccurate estimation of upstream/downstream  
 223 interaction can be incriminated. Spatial discretisation study showed that further wake refinement did not  
 224 increase accuracy. That phenomenon is currently under investigation at IRENAV.

225 Case 3 is shown in figure 8. In that case the agreement is fairly good, particularly in the upstream  
 226 part of the turbine. The upstream  $C_t$  peak is well reproduced. However discrepancies are observed in the  
 227 downstream part. Experimental oscillations originating probably from stall vortices are observed in the  
 228 experiments were not simulated. In the same manner as case 2, it can be argued that the influence of the  
 229 upstream part on the downstream part is not accurately reproduced.

230 Case 4 is shown in figure 9. The agreement is fairly good. The evolution is well reproduced showing the  
 231 ability of the model to account for the unsteady stalled behaviour of a Darrieus turbine at low  $\lambda$ .

232 Case 5 is shown in figure 10. This case involves large incidence variations and deep stall. The agreement  
 233 is very good.  $C_t$  oscillation frequency is reproduced accurately, and the amplitudes differs of about 10 to  
 234 20%. The simulated  $C_n$  oscillates more than the experimental results, but the mean values are close. These  
 235 oscillations are due to stall vortices convected along the chord. For both coefficients, the simulation is more  
 236 accurate in the upstream part where angle of attack is large, showing the ability of this model to simulate  
 237 deep stall. Again discrepancies are observed rather in the downstream part of the turbine.

Figure 8: Validation on  $C_t$  and  $C_n$  for case 3 ( $\lambda = 7.5$ ,  $\sigma = 0.15$ )Figure 9: Validation on  $C_t$  and  $C_n$  for case 4 ( $\lambda = 3$ ,  $\sigma = 0.2$ )

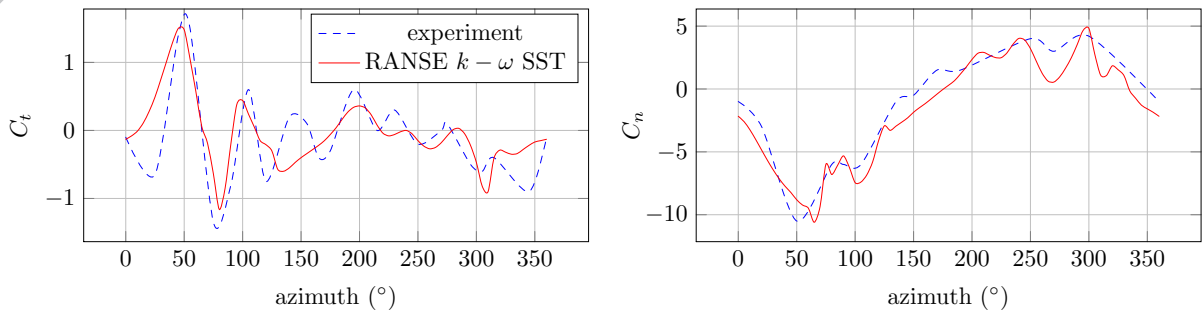
### 238 3.2. Variable pitch

239 In order to optimize the device, a variable cyclic pitch is added in the computation. The main goal  
 240 of pitch control are to increase torque and decrease non-productive  $C_n$  ; to smooth the forces during the  
 241 turbine rotation ; and to control cavitation inception.

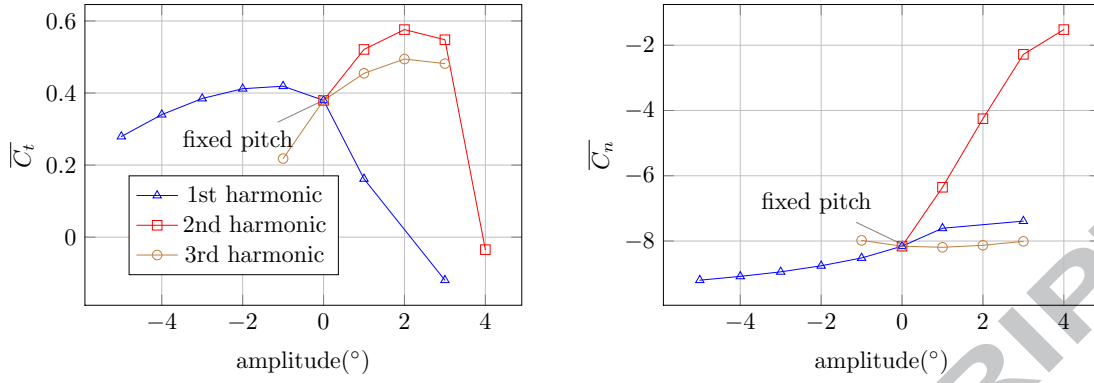
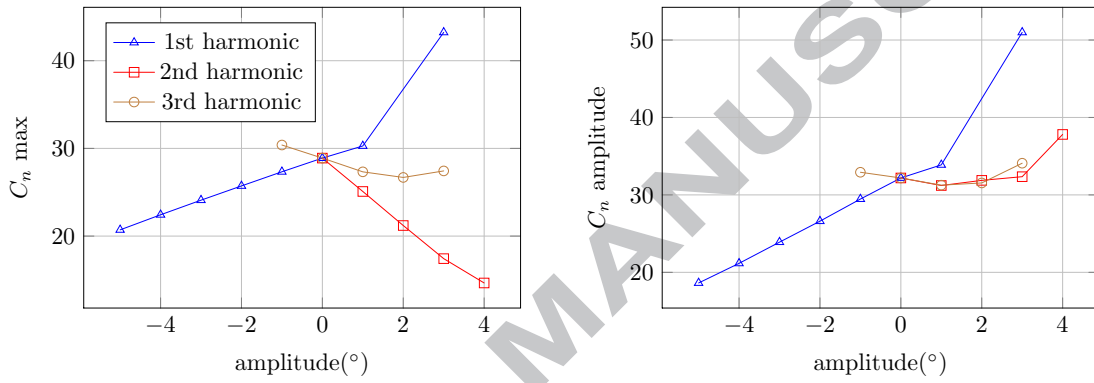
242 Sinusoidal pitching was implemented in the present simulation for case 2,  $\lambda = 5$ . This case was chosen  
 243 since it has the highest  $\eta : .362$  (see figure 3). This value is high compared with existing crossflow turbines,  
 244 for which TSR usually range from 3 to 4 for low solidity turbines [38]. The aim of the present paper is not  
 245 to provide solution to the pitch law optimization problem, but to validate a model and a workflow. This is  
 246 why it was not mandatory to choose a nominal  $\lambda$  value.

247 However, unlike all crossflow wind turbines and most tidal turbines, the targeted configuration of this  
 248 project is a tidal turbine where holding arms are outside of the water, and the blades are cantilevered.  
 249 The parasitic arm and junction drag is thus almost cancelled, which enables a turbine  $\lambda$  increase. Even  
 250 without this feature, recent researches on strut arrangements showed that inclined struts can be beneficial  
 251 to efficiency [39]. In addition to that, it is believed that multi-objective optimization of the pitch law would  
 252 result in a nominal  $\lambda$  increase, since main shaft torque, thus generator or gearbox cost would decreased.  
 253 This would require  $\lambda$  to be a parameter of the optimization, which is outside the scope of this paper.

254 Three frequencies were tested, namely the first, second and third harmonics, corresponding respectively to

Figure 10: Validation on  $C_t$  and  $C_n$  for case 5 ( $\lambda = 1.5$ ,  $\sigma = 0.1$ )



Figure 11: Influence of pitch function amplitude on  $\bar{C}_t$  and  $\bar{C}_n$ Figure 12: Influence of pitch function amplitude on maximum and amplitude value of  $C_n$ 

255 the turbine frequency, twice the turbine frequency and three times the turbine frequency. Various amplitudes  
 256 from  $-5^\circ$  to  $4^\circ$  were tested for each frequency.

### 257 3.2.1. Performance study

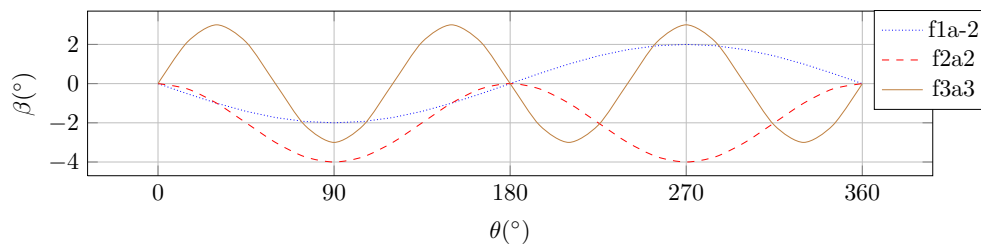
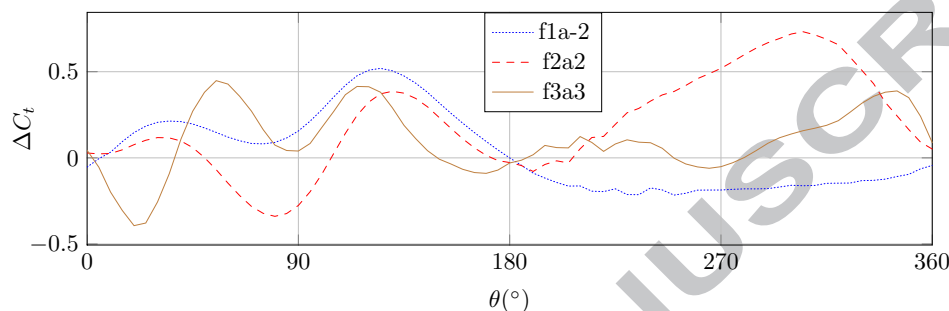
258 Mean forces coefficients as a function of pitch amplitude and frequency are displayed on figure 11. Fixed  
 259 pitch is displayed as a zero amplitude function. As observed, the second harmonic function shows the best  
 260 results. The best  $\eta$  is obtained for an amplitude of  $2^\circ$  with an increase of 52% in tangential force, in addition  
 261 to a decrease of 49% in normal force. An amplitude of  $3^\circ$  gives almost the same performance, with a further  
 262 50% decrease in normal force.

263 Maximum and amplitude values of normal force coefficient are shown in figure 12. A large reduction  
 264 can be obtained on the maximum value with the second order functions, which will decrease the ultimate  
 265 load. However the amplitude load remains almost constant below an amplitude of  $3^\circ$ . The  $C_n$  reduction in  
 266 upstream part results in an increase downstream. This pitch control strategy will not decrease vibratory or  
 267 fatigue constraints.

### 268 3.2.2. Pitch angle influence on unsteady $C_t$

269 The best pitch functions for each harmonic are closely studied now. f1a-2 stands for the first harmonic  
 270 function with an amplitude of  $-2^\circ$ ,  $\beta_{f1a-2} = -2 \sin(\omega t)$ ; f2a2 stands for the second harmonic function with  
 271 an amplitude of  $+2^\circ$ ,  $\beta_{f2a2} = 2(\cos(2\omega t) - 1)$  (notice the -1, required to reach  $0^\circ$  at upstream-downstream  
 272 transitions. It yields a minimum incidence of  $-4^\circ$ , not  $2^\circ$ ); f3a3 stands for the third harmonic function with  
 273 an amplitude of  $+3^\circ$ ,  $\beta_{f3a3} = 3 \sin(3\omega t)$ . These functions are shown figure 13, where  $\theta = \omega t$ . Positive pitch  
 274 angles stand for an inward rotation of the leading edge.

275 Absolute difference between  $C_t$  with fixed and variable pitch is shown in figure 14. Both f1a-2 and f2a2  
 276 exhibit similar behavior on the upstream part.  $C_t$  is increased at the beginning of the upstream section, and  
 277 after  $40^\circ$  the difference decreases, and becomes negative for f2a2. At  $85^\circ$  it starts increasing again, and both  
 278 functions give the most increase for upstream part at  $135^\circ$ . For f3a3 the steep rise in pitch at the start and  
 279 end of the upstream part decreases the tangential load. For the downstream part, performance is improved

Figure 13: Sinusoidal pitch functions  $\beta(\theta)$ Figure 14: Difference in  $C_t$  for variable pitch compared with fixed pitch

for all pitch functions when pitch angle is decreased, thus increasing the angle of attack. This figure leads to several conclusions. First increasing incidence on the upstream part is disadvantageous for this regime, and a decrease of  $-2^\circ$  in pitch looks optimal. Second, decreasing pitch below  $-2^\circ$  is not beneficial as shown for f2a2 for which minimum pitch is  $-4^\circ$ . Third an incidence increase on downstream part is beneficial, which is due to flow deceleration. The strong coupling between local incidence, local flow velocity and pitch angle function requires further investigation on flow deceleration.

### 3.2.3. Flow velocity reduction

A tidal turbine converts fluid kinetic energy into solid rotational kinetic energy. The amount of reduction in flow velocity occurring when the fluid travels through the turbine is crucial. A small reduction results in a low transfer between fluid and solid. A large reduction results in much energy being withdrawn from the fluid, but not necessarily transferred to the solid, and can create undesirable high disturbance in the flow. According to the momentum theory, the energy extraction is maximum when the flow velocity downstream of the turbine is one third of the upstream flow velocity [40].

The overall reduction can be illustrated by the axial velocity reduction along the axial centerline, shown figure 15. All pitch functions reduce the angle of attack at upstream part, thus reducing flow disturbance and inducing a faster upstream flow velocity compared with fixed pitch. On downstream part f2a2 is the only function for which flow is decelerated below fixed pitch values. Indeed,  $\beta_{f2a2} < 0$  for all  $\theta$  unlike the others laws, resulting in f2a2 being the only function increasing angle of attack everywhere downstream. The other two functions show faster flow. f3a3 is the function creating fastest flow on the central axis of the turbine. A slight velocity increase is observed downstream. This can be explained by the fast rotation of the blade around its quarter chord, creating a propelling effect on the flow. Another conclusion arises from the comparison between f1a-2 and f2a2. On upstream part these functions create very similar deceleration, despite f2a2 reaching a pitch of  $-4^\circ$  and f1a-2 reaching  $-2^\circ$ . The least flow disturbance by f2a2 should result in a faster flow, however the high incidence at downstream part also disturbs the upstream part. This shows how crucial the correlation between upstream and downstream functions is, and why the flow velocity variation needs to be precisely assessed.

### 3.2.4. Pressure field

The pressure fields compared between fixed and variable pitch around one blade are shown in figure 16. For case 2, figure 16a, the pressure on the upstream part remains quite low and constant at the suction side between  $\theta = 40^\circ$  and  $120^\circ$ . The low pressure area is much smaller on the downstream part however. This is due to the velocity reduction induced by the upstream part, which reduces the angle of attack.

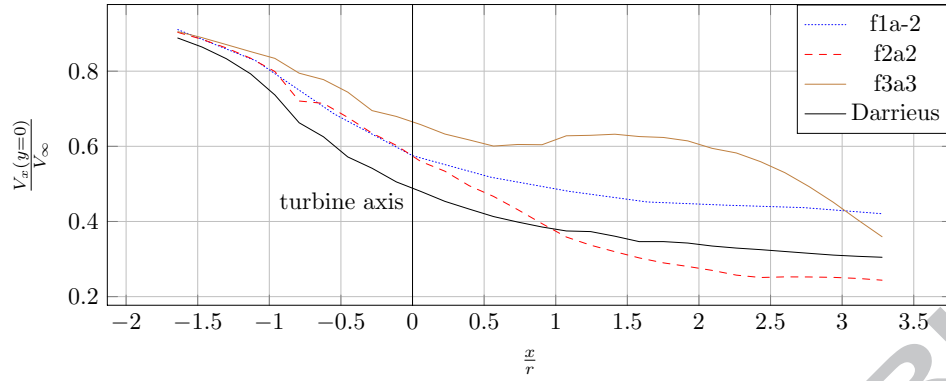


Figure 15: Flow deceleration through turbine center

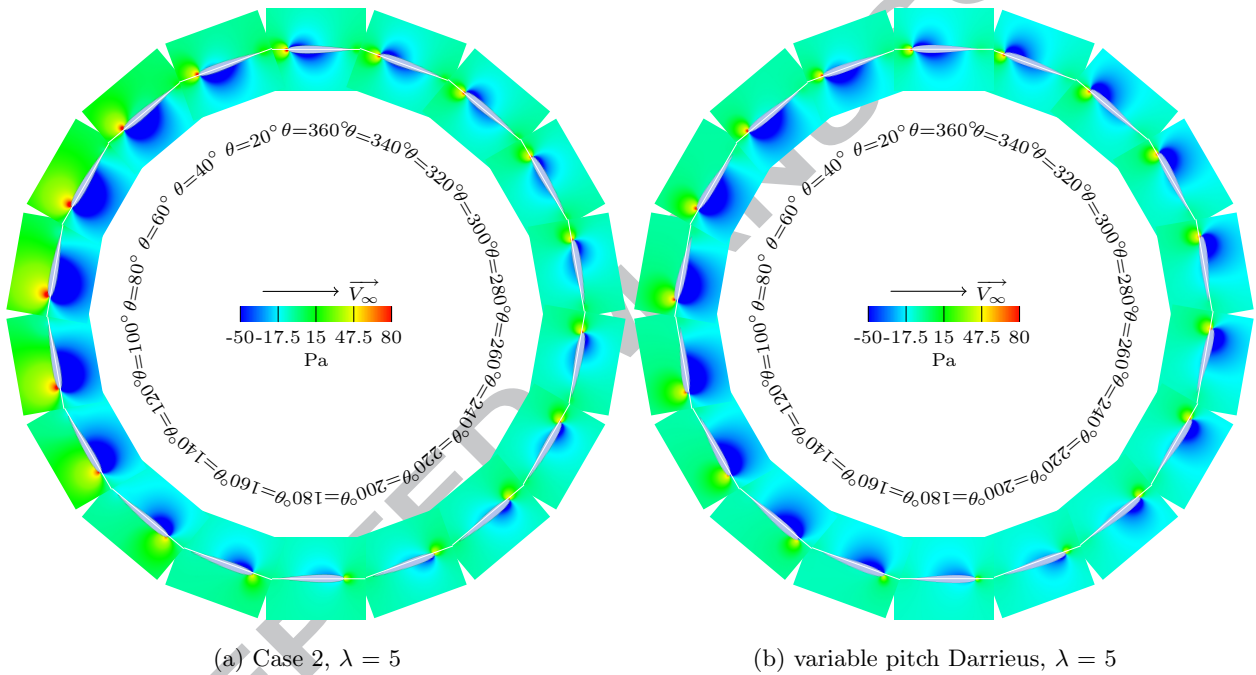


Figure 16: Pressure field

311 The tangential and normal force coefficient (figure 7) are thus much smaller for the downstream part. For  
 312 variable pitch, figure 16b, the law considered is the optimal for this study, f2a2. The low pressure at the  
 313 suction side is weaker at the upstream part, and stronger at the downstream part. This results in a more  
 314 levelled and better energy conversion. The upstream part extracts less power, which means power can be  
 315 extracted more efficiently from the downstream part.

### 316 3.2.5. Axial velocity field

317 The axial velocity fields compared between fixed and variable pitch are shown in figure 17. The axial  
 318 rather than the transverse component or the magnitude was chosen, since it is the energy source from which  
 319 the turbine can produce power. Fixed pitch is shown in figure 17a, and variable pitch f2a2 is shown in figure  
 320 17b. The first difference observed is the wake at the transition between upstream and downstream parts,  
 321 top and bottom parts of the illustrations. With fixed pitch the blades create a wake where flow velocity is  
 322 increased, which translates into energy loss. A different pitch angle could result in better efficiency. On the  
 323 other hand with variable pitch f2a2, this wake is reduced greatly. This pitch law enables the blades to go  
 324 through zero angle of attack at a better azimuthal position during the transition where pressure and suction  
 325 side are reversed. The flow velocity reduction due to energy conversion is clearly visible for all angles. It is  
 326 thus obvious that during the downstream pass, the blade moves in a much slower fluid, and hence has much  
 327 lower energy available for conversion, which results in lower tangential force coefficient and torque.

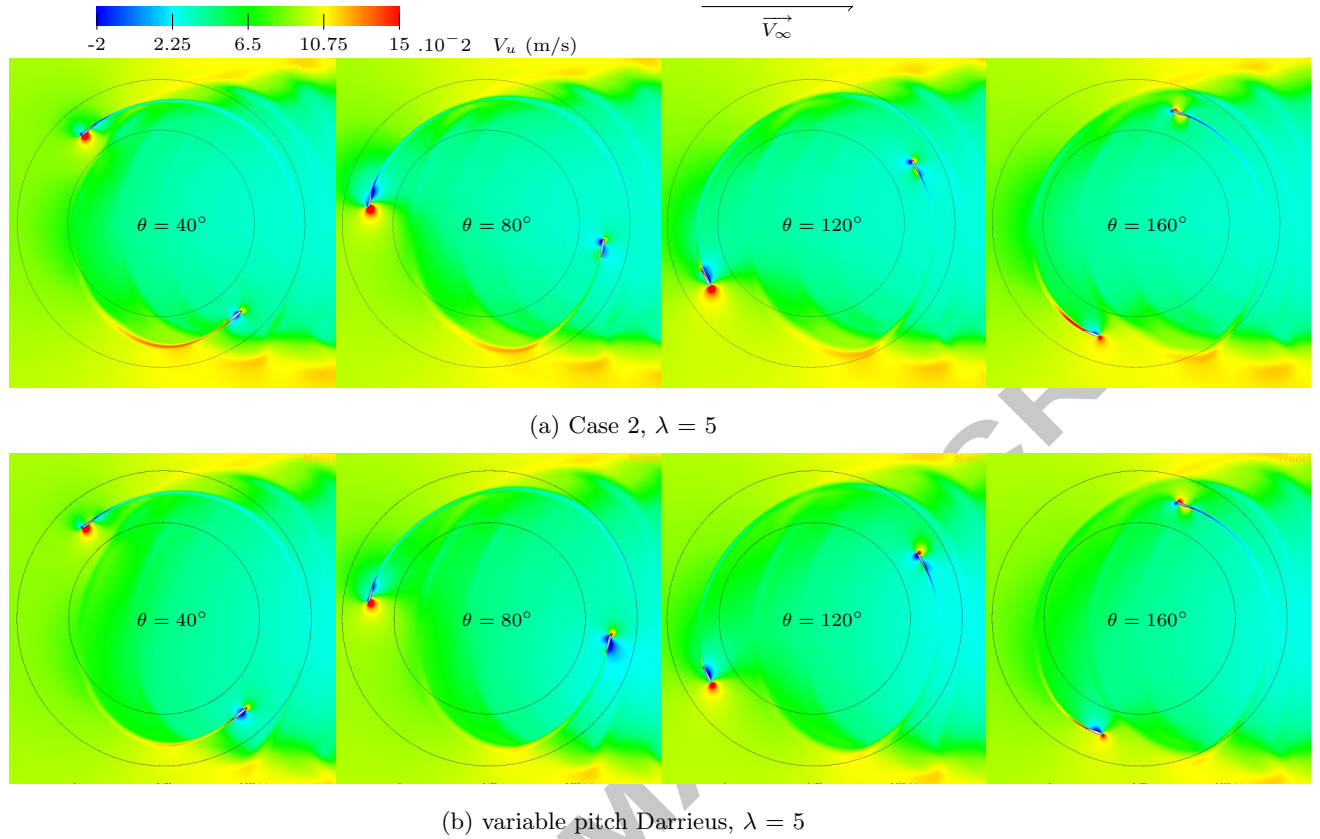


Figure 17: Axial velocity field

### 3.2.6. Chordwise pressure coefficient distribution

Pressure coefficient  $C_p$  along blade boundary is shown figure 18. The x coordinate is the adimensional chord position, and the curves are given for several azimuthal positions. The pitch functions are drawn along the  $\theta$  axis at the bottom right of the chart. A circle on each curve indicates the outer part of the turbine. The fixed pitch case and the three pitch functions introduced earlier are shown. The fixed pitch turbine creates a lower minimum  $C_p$  than variable pitch, hence higher cavitation sensitivity. The higher incidence of f3a3 at the beginning and end of the upstream section translate into two slightly smaller peaks. For the fixed pitch, f1a-2 and f3a3, plateaux can be noticed. They are the sign of stall inception through vortex formation. These plateaux remains quite thin, which means complete stall is avoided. However these  $C_p$  singularities create disturbance in the flow which should be avoided. f2a2 shows no sign of stall, and its lower  $C_p$  peak would result in lower cavitation sensitivity. On the downstream part however, the higher angle of attack associated with f2a2 results in a higher  $C_p$  peak, with a maximum value equal to the maximum upstream value. Again, this means that a more levelled energy extraction is obtained with this law. f2a2 creates a much greater suction at the downstream part. It is the main reason for its higher performance. It can finally be noticed that the curves shown between  $220^\circ$  and  $340^\circ$  are very different from a pitching blade, with a  $C_p$  lower on the intrados than on the extrados on the rear part of the blade. This illustrates the flow curvature effect and is consistent with previous studies [41, 42].

### 3.2.7. Energy consumption of pitch control

The torque required to set the blades in motion results in additional energy transfer and needs to be assessed. The comparison between the energy extracted by the device and this pitch moment energy is carried out by reducing the power needed to drive the pitch to a value dimensionally equivalent to the performance coefficient  $\eta$ , defined as  $\eta_\beta$ . It is negative when power is needed to set the blades in motion, hence reducing the global performance. Hydrodynamic moment on blades  $M_{\text{hydro}}$  is given by URANSE computation.  $\eta_\beta$  is then computed by the following equation :  $\eta_\beta = \frac{M_{\text{hydro}} \dot{\beta}}{\frac{1}{2} \rho 2 r l v_\infty^3}$

Two approaches can be used to assess this influence. If a mechanical device, or a hydraulic device is used to obtain pitch variation, then when  $\eta_\beta$  is positive, additional power can be fed back to the primary power extraction. This is not true when servomotors are used, and in this case only negative values of

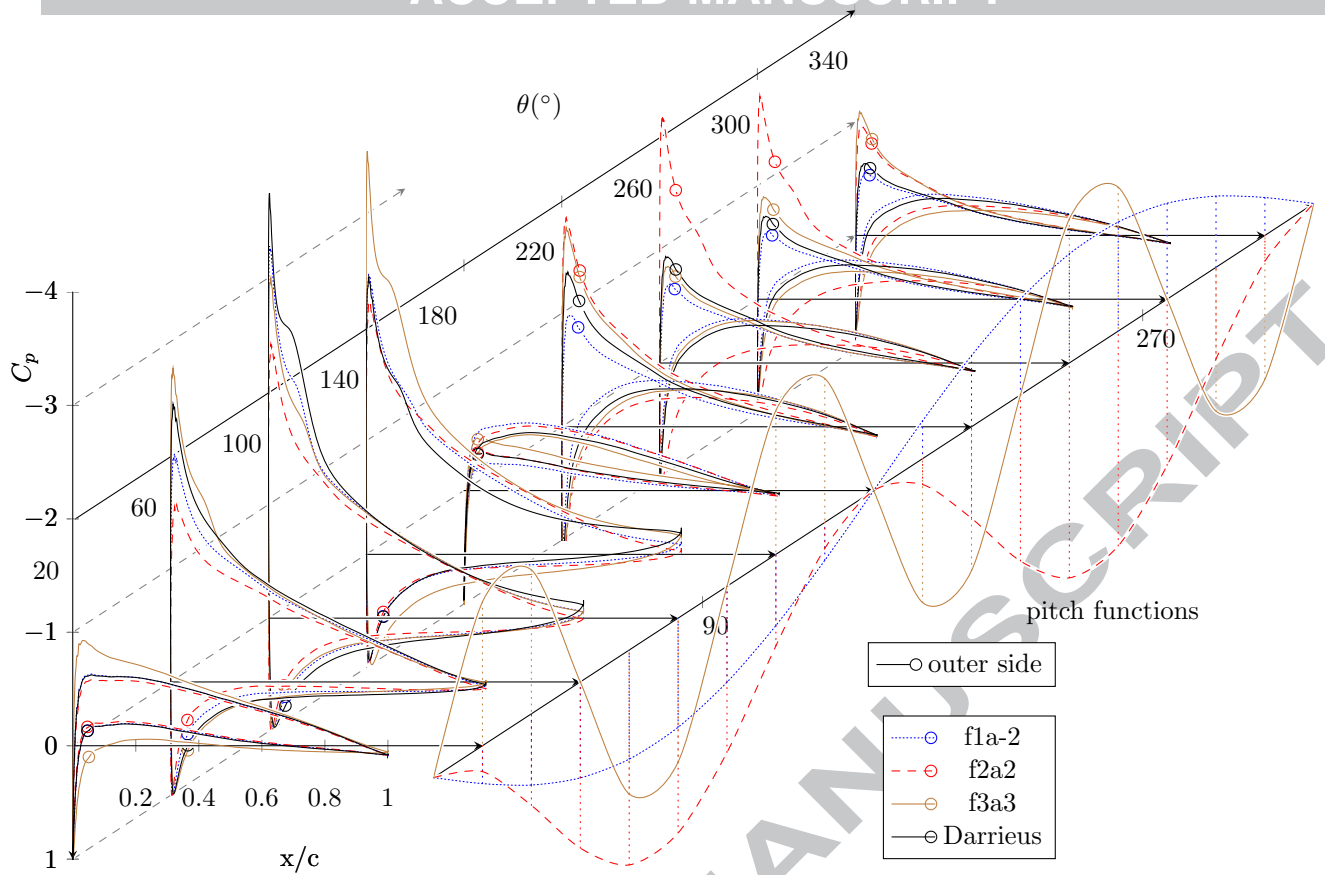


Figure 18: Pressure coefficient for f1a-2, f2a2, f3a3

loi	$\eta$ (%)	$\eta_\beta$	$\eta_\beta$ with negative values
Darrieus	28,45	-	-
f2a1	39,03	0,20	-0,29
f2a2	43,16	0,49	-0,47
f2a3	41,06	0,71	-0,66
f2a4	-2,60	-1,14	-2,64
f3a1	34,07	0,05	-0,73
f3a2	37,03	0,04	-1,52
f3a3	36,09	-0,08	-2,38

Table 5: Global and moment performance coefficients

$\eta_\beta$  need to be considered. Results for f2 and f3 can be found in table 5. For the approach consisting of averaging all the values, the moment performance coefficient is mainly positive. f2a2 would benefit from an additional 0.49% in  $\eta$ . Only f3a3 would get a negative value of only 0.08% which is very low. When using only negative values, f3a3 would have its performance decreased by 2.38%, f2a2 by 0.47%. The higher moment influence on f3 functions is explained by a faster pitch velocity. The third harmonics are thus less desirable for further studies. The hydrodynamic moment associated with lift generation is considered here at quarter chord. For energy consumption reduction this location may be optimized, which is outside the scope of this work. It was however important to demonstrate that, even with the quarter chord as pitch center, the energy consumption remains very low compared with the gain associated with active variable pitch.

#### 4. Conclusion

The simulation of a cross-flow turbine has been implemented in a URANSE solver. The validation of the model is based on the comparison with experimental data obtained from the literature. The validation against experimental data was performed on the tangential and normal force comparison for five configurations corresponding to various tip speed ratios and solidities. The computation is based on the fully



370 turbulent  $k\omega$  SST model. However because of moderate chord length Reynolds number in the experiments,  
 371 a transition model was also tested. The computation domain was built with a rotating ring containing  
 372 blades, enclosed inside a steady domain with sliding interfaces at the boundaries. The boundary layer could  
 373 thus be meshed with high quality, independently of the turbine rotation. Thorough determination of simu-  
 374 lation parameters and discretizations has been carried out and shows that a  $1^\circ$  discretization of the turbine  
 375 rotation is necessary, that  $y^+$  has to be lower than unity to correctly compute the boundary layer close to  
 376 the foil surface, and that 300 cells are required in the chordwise direction along the blades. It is observed  
 377 that the agreement is rather good for all cases for the fully turbulent model. Even though the experimental  
 378 forces results led the author to believe transition might have occurred, it could not be obtained with the  
 379  $\gamma - Re_\theta$  transition model. The complexity associated with the upstream-downstream interaction is found to  
 380 be the main source of discrepancies. The close analysis of wall pressure coefficient and flow field at various  
 381 stages of the turbine rotation highlights boundary layer events such as separation, stall, vortex shedding  
 382 and flow reattachment. This allows the authors to believe that the present simulation is accurate enough  
 383 to be used for the simulation of variable pitch cross-flow turbines. With such devices flow events can be  
 384 controlled and power production can be optimized with the use of sinusoidal pitch functions, by varying the  
 385 frequency and amplitude. Variable pitch was implemented for a tip speed ratio of 5, aiming at performance  
 386 improvement primarily. Sinusoidal functions of different orders were tested. The second harmonic functions  
 387 resulted in a performance increase of 52%. For this regime optimal pitch variation seems to require a very  
 388 slight recirculation and an incidence decrease on upstream section, and an angle of attack increase on down-  
 389 stream section. The flow deceleration through the turbine was found to be a primary factor in pitch function  
 390 performance evaluation. Finally the power required to set blades into motion around their quarter chord  
 391 was compared with the power extracted by the turbine. The ratio was found to be lower than 3% for third  
 392 harmonics, and lower than 0.5% for second order harmonics. The performance gain associated with variable  
 393 sinusoidal pitch control is thus relevant for further study and optimization. The future steps consists in the  
 394 assessment of composed sinusoidal pitch functions ; the addition of a pitch offset ; the analysis of the pitch  
 395 center location ; the addition of  $\lambda$  as an optimization parameter ; the evaluation in inclusion of arm and  
 396 junction drag in the optimization ; and finally the computation and optimization of arbitrary pitch functions.  
 397

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RANS modelization of a Darrieus crossflow turbine where blades' pitch angle changes over time to improve performance > Complete validation on a fixed pitch system, with models comparison and temporal/spatial discretization study > Influence of periodical pitch variation on performance including power required for pitch variation > Local pressure coefficient description

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