

Development of a Desmodromic Variable Pitch System for Hydrokinetic Turbines

Pier Francesco MELANI^{1a}, Francesco BALDUZZI^{1b}, Giovanni FERRARA^{1c},
Alessandro BIANCHINI^{1d*}

¹⁾ Department of Industrial Engineering, Università degli Studi di Firenze
Via di Santa Marta 3, 50139, Firenze, Italy
Phone +39 055 275 8773
Fax + 39 055 275 8755

Abstract

The use of vertical-axis turbines is raising interest in the field of hydropower production from rivers or water channels, where suitable mass flows are available, without the need of high water jumps or large construction sites. Like their wind-based counterparts, these machines have many advantages in terms of installation and maintenance, but suffer from an inherently complex rotor hydrodynamics, characterized by a continuous variation of the blade angle of attack beyond the static stall limit and the corresponding onset of dynamic instabilities. In virtue of their typically lower rotational speed nonetheless, hydrokinetic machines can benefit from the possibility of active blade pitch control, which is barely feasible in case of wind turbines due to their high characteristic operation frequencies. Moving from this background, a novel control strategy, based on a double-cam desmodromic pitching system, has been developed in the present study by means of a combination of medium- and high-fidelity numerical approaches. In order to tailor the corresponding pitching law, here implemented in a parametric form, to this class of machines, a dedicated automatic optimization framework has been developed in the ANSYS® WORKBENCH® 20.2 suite. The latter combines a Multi-Objective Genetic Algorithm (MOGA) optimizer with an advanced two-dimensional Actuator Line Model (ALM) model of the turbine, developed by the authors. This optimization approach has then been applied to a test rotor with the objective of maximising the turbine power coefficient while containing the rotor stream- and cross-wise force oscillations. The resulting optimal configurations have been eventually verified via high-fidelity, fully-resolved CFD simulations. Results show that variable pitch can represent a promising solution for increasing the performance of vertical-axis hydrokinetic turbines, leading to a +35% efficiency enhancement as well as to a reduction of the machine unsteady loads.

Keywords: Actuator Line Model, Hydrokinetic Vertical Axis Turbine, Active Pitch Control, Computational Fluid Dynamics

1. Introduction

The increasing investment costs and the lowering availability of favourable sites for traditional hydropower generation [1], which is based on the exploitation of the potential energy accumulated by the water previously collected inside a storage reservoir, has shifted lately the interest of the industry towards non-conventional installations, such as small natural streams, canals or artificial water networks [2]. For this application, the use of hydrokinetic turbines (HKTs), which, analogously to their more widespread wind counterparts (WTs), extract power from the stream kinetic energy content, is becoming increasingly relevant [3].

a) *pierfrancesco.melani@unifi.it*

b) *francesco.balduzzi@unifi.it*

c) *giovanni.ferrara@unifi.it*

d) *alessandro.bianchini@unifi.it* (* = corresponding author)

The characteristics of water flows in fact make the use of the technology developed for the wind energy sector even more convenient than in its original field of application, overcoming many of its shortcomings. First, water has a density almost three orders of magnitude higher than air. Correspondingly, water flows have a higher energy density, which is made available at a much lower inflow speed [1]. For instance, at given rotor swept area, a water velocity of 0.85 m/s is sufficient to provide the same input power of a standard wind velocity of 8 m/s [4]. From a design point of view, this leads to more compact machines, from 3 to more than 10 times smaller, rotating at a lower rotational speed. In spite of their smaller dimensions, these rotors tend also to be more efficient than the WT ones, since they can easily operate at a blade Reynolds number 2-3 times higher and, in virtue of the higher Tip-Speed Ratio (TSR) achievable, tend to have a lower solidity [5], although sufficiently long chords are needed to ensure structurally robust blades. The second main advantage of this energy source is that, especially in the case of artificial water canals, the inflow velocity is more or less constant in magnitude and direction during operation and has a very low turbulence content [6]. As a consequence, the design of these machines can be optimized in a small range around a single operating condition, rather than over the whole expected inflow velocity spectrum as in wind applications, and no additional control devices (e.g., yawing system) are required. Moreover, self-starting issues are overcome. The benefits described so far come nonetheless at the price of, aside the classic issues given by water such as corrosion and cavitation, higher hydrodynamic loads and more demanding structural design.

As in wind energy, one of the HKT architectures with the highest potential is the vertical-axis one (HVAT), whose main advantage is represented by the possibility to keep the generator unit outside the water, notably facilitating both installation and maintenance [6]. This concept suffers nonetheless from a lower efficiency compared to its horizontal-axis counterpart [7], because of the complex rotor aerodynamics involved. Their operation is in fact intrinsically unsteady, with large oscillations of the angle of attack experienced by the blades during one rotor revolution, often exceeding the static stall limit. If filling this efficiency gap is still not possible for wind applications, in hydrokinetic ones the yet smaller gap can be further reduced – or even closed - with auxiliary *pitching systems*, i.e. devices able to vary the blade angle of attack over one rotor revolution according to a predefined control strategy. In virtue of the low rotational speed characterizing HKTs, the responsiveness and acceleration required by these systems are in fact in the range of those effectively achievable at an industrial level. Furthermore, because of the constant inflow speed of the water stream, such devices would need to be optimized only for one operating condition and wind direction, notably easing the design process. The adoption of this solution is not only justified from an aerodynamic point of view, but also from a structural one. It can be in fact successfully employed to moderate the large oscillation of the forces acting on the rotor shaft over one rotor revolution that characterize vertical-axis machines (*shaking*) [6,8], which can be critical for the design of HVATs due to the magnitude of hydrodynamic loads.

1.1 Pitching systems

In general, pitching systems can be classified in *passive* and *active* ones. The first ones rely on the self-regulation of the blade pitch based on the balance between local hydrodynamic, inertial and elastic forces. These solutions have a great potential, but their design is extremely challenging [9]. The second ones instead make use of an external control action to regulate the blade position. The most advanced version of this technology, which makes use of actuators such as servo-motors, is usually not employed on vertical-axis wind turbines (VAWTs), especially small-scale ones, due to the elevated cost and complexity [10]. A promising solution for hydrokinetic applications could instead be represented by pitching systems based on *eccentric/cams*, which stand out in terms of robustness and cost-effectiveness. Their main drawback is that the variety of achievable pitching laws is limited and constrained by the system configuration. The latter must be therefore selected *a priori* and becomes an integral part of the machine design process. As mentioned in the previous section,

this is not a problem for hydrokinetic turbines, since, in contrast with wind ones, their performance needs to be optimized only for the design condition.

As evident in Table 1, most of the studies on this topic focused on the development of systems capable of achieving a *sinusoidal* pitching law of fixed amplitude. A representative example is the *Linkage* one [11], where the blade pitch is regulated by a crank-rod mechanism connected to the rotor shaft (see Figure 1a). The success of this strategy derives from its simplicity and effectiveness in improving the machine efficiency and reducing rotor load oscillations, especially for Tip-Speed Ratios (TSRs) lower than the optimum one, where dynamic stall control is a relevant issue. For $TSR > TSR_{opt}$ on the other hand, this solution tends to lower the performance with respect to the fixed-pitch case. In the *downwind* ($180^\circ \leq \theta < 360^\circ$) section of the rotor in fact, where the blade angle of attack is already well below the static stall limit due to the *upwind* ($0^\circ \leq \theta < 180^\circ$) blockage effect, the pitching action results in a further load reduction [12]. A second issue of the sinusoidal law, as pointed out by Staelens et al. [13], is the inability to achieve the optimal blade loading in the *upwind* sector during the *pitch-up* (increasing AoA) and *pitch-down* (decreasing AoA) phases, i.e. before and after reaching the static stall point. This limits the power enhancement potentially obtainable. In this context, no additional benefit can be observed when using higher-order sinusoidal functions.

In order to overcome the two issues described so far, more complex pitching approaches have been proposed over the years. Chen et al. [12] introduced a *truncated sinusoidal* function, with a flat characteristic in the *downwind* sector, not to interfere with the blade loading in that part of the revolution. In their case, the result is an efficiency increase with respect to the fixed-pitch turbine of 63%, 20% more than the pure sinusoidal case. Staelens et al. [13] on the other hand attempted to address the second issue by adopting a strategy based on *stall control*, i.e. pitching the blade only when the angle of attack exceeds a predefined target value, usually close to the static stall limit. In both cases, due to the complexity of the corresponding pitch law, the eccentric crank-rod mechanism needs to be replaced with a cam [14], as in Figure 1b, which can be profiled to achieve the desired motion. Nonetheless, as a single-cam can assume only convex shapes, the pitching action is limited to only half of the rotor revolution, since during normal HVAT operation the angle of attack changes sign from the *upwind* to the *downwind* sector. Thus, the pitching action in the *downwind* region can be either negative or neutral from a blade loading point of view.

Table 1 – Literature review of the current pitching solutions for small VAWT/HVATs.

Pitch law	Source	Pitch system	$\Delta C_{P,opt}$	Benefits	Issues
Fixed	[14–17]	n/d	+ 2-13%	- simple - cost-effective performance increase	- limited improvement in start-up - limited load oscillations reduction
Sine	[9,11–14,16,18–21]	eccentric, single-cam	+ 17-53%	- improved start-up - reduced load oscillations - performance increase for $TSR < TSR_{opt}$ - easy to regulate with speed	- performance degradation for $TSR > TSR_{opt}$ - low power augmentation potential
Sine (high-order)	[20,22]	n/d	+ 34-52%		
Sine (truncated)	[12]	n/d	+ 63%	- sine law benefits - performance increase for $TSR > TSR_{opt}$	n/d
Stall control	[13]	n/d	n/d	higher performance than sine	discontinuous

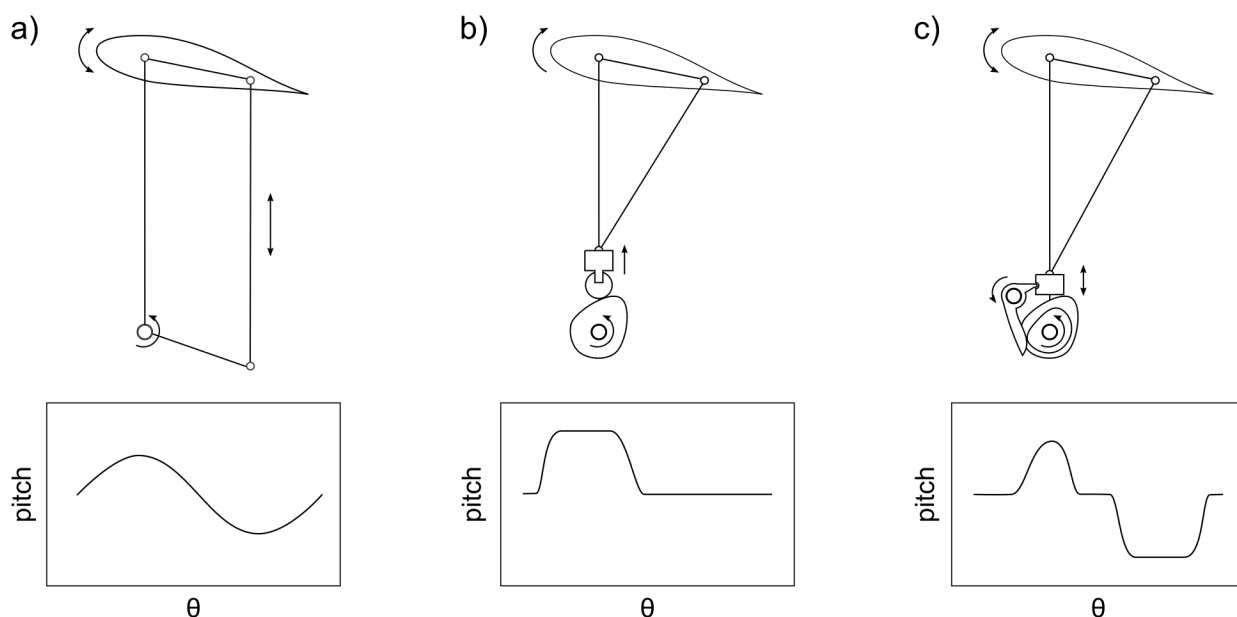


Figure 1 – Pitching systems and the corresponding pitching laws: a) linkage b) single-cam c) desmodromic double-cam.

The maximum flexibility can be achieved with a *desmodromic double-cam* system, originally developed for automotive applications [23]. To the authors' knowledge, this control strategy has never been applied so far to vertical-axis turbines. As schematically represented in Figure 1c, the use of a second profiled cam and a set of leverages allows to pitch the blade with different laws in the clockwise and anti-clockwise directions, creating the opportunity of a positive pitching action to the *downwind* sector and thus unlocking the maximum potential of this kind of control strategy.

1.2 Outline of the study

Starting from this background, in this study the possibility of improving the performance of a vertical-axis hydrokinetic turbine by means of an active blade pitching control system is investigated. More specifically, an innovative family of motion laws are developed by means of numerical techniques in a way that they can be obtained in real turbines by means of a mechanical *desmodromic double-cam* system; the use of this system could indeed allow one achieving the largest variety of motion laws without the cinematic limitations typical of other active systems found in literature. In order to fully exploit the potential of this technology for the present application, a novel parametric pitching law, tailored to the specific characteristics of HVATs, has been introduced, together with a dedicated framework for the automatic optimization of its tuning parameters. The latter, implemented in the ANSYS® WORKBENCH® 20.2 suite, combines a Multi-Objective Genetic Algorithm (MOGA) optimizer with an advanced two-dimensional hydrodynamic model of the turbine, developed within the same simulation environment and based on a formulation of the Actuator Line Method (ALM) specifically tailored to vertical-axis machines in a recent work by the authors [24]. This model is able to solve the flow field across the rotor at a reduced computational cost in comparison to blade-resolved Computational Fluid Dynamics (CFD), while preserving the benefits of this latter in terms of flow field description.

The proposed optimization approach has then been applied to a selected test rotor, for which high-fidelity, blade-resolved CFD benchmark data were available, to find the value of the pitching law parameters maximising turbine power coefficient at the optimum operation point. Given the elevated shaft load oscillations (*shaking*) characterizing HVATs, the minimization of the rotor stream- and cross-wise force oscillations has been also taken into consideration as a lower priorities objective. The *variable-pitch*, i.e. exploiting the desmodromic system, solution has been also compared with the *fixed-pitch* one, i.e. with a constant pre-pitch applied to the rotor blades.

Due to the approximation introduced by the adoption of a lower order method such as the ALM, the optimal configurations resulting from the *variable-pitch* optimization have been eventually verified via a high-fidelity, blade-resolved CFD simulation.

2. Study case

The test case selected for the present study is a two-blade H-shaped hydrokinetic turbine (see Figure 2), developed by the authors in a previous work with an industrial partner [4].

2.1 Baseline design

A heuristic approach was followed, starting the design process with the definition of the main geometrical parameters (rotor diameter, blade chord and height), whose values are reported in Table 2, and selection of the blade hydrofoil, in this case a DU-06-W-200. The latter is mounted with outward concavity to enhance the power extraction during the *upwind* part of the rotor revolution. In this phase, the BEM code VARDAR [25] available at Università degli Studi di Firenze for the simulation of vertical-axis machines was employed. For the second part of the design process, different candidates, distinguished in terms of number and arrangement of the rotor blades, were generated and their behaviour simulated for different inflow water speeds. For the purpose, the two-dimensional blade-resolved URANS CFD approach developed by the authors over years of research on VAWTs [26,27] was employed. The reliability of the latter in simulating hydrokinetic machines, with all the known limitations of the two-dimensional formulation, has been assessed via the validation against both towing tank [28] and on field [17] experimental tests. For this reason, as later explained in Section 5, the same model served as a basis for the development of the variable-pitch one used in the current application.

Eventually, the design process converged towards a two-blade rotor configuration (see Figure 2), as the one yielding the highest *power coefficient* C_P and the lowest *rotor shaft load oscillations* in the *streamwise* (ΔC_X) and *cross-wise* (ΔC_Y) directions, according to the definitions of Section 2.3.

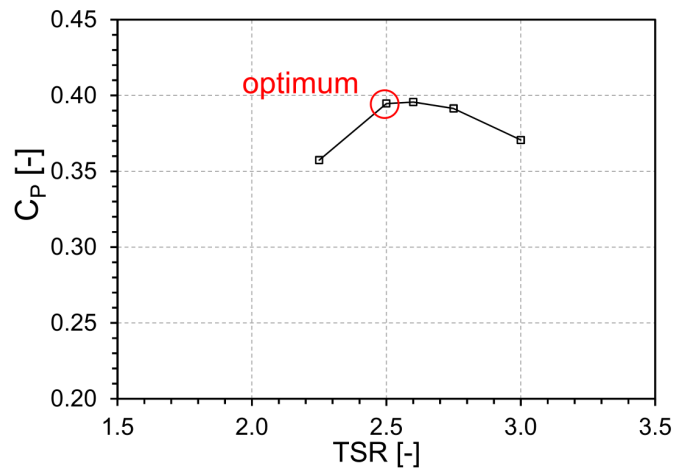


Figure 2 - Test case: schematic view in the mid plane (up) and power coefficient curve (down).

Table 2 - Details of the test rotor selected for the investigation.

Dimension	Value
Number of blades [-]	2
Radius [m]	1
Chord [m]	0.25
Airfoil	DU-06-W-200
Blade-spoke connection [x/c]	0.25

2.2 Operating conditions

In virtue of the characteristics of hydrokinetic machines, which, as discussed in Section 1, usually work at a nearly constant water flow speed, only one operating point was considered for this study, corresponding to the *optimum* (TSR=2.5), i.e. the highest efficiency condition, of the *baseline* defined in Section 2.1. The corresponding inlet conditions are reported in Table 3.

Table 3 – Test operating conditions.

Inlet quantity	Value
Tip-speed ratio [-]	2.5
Water speed [m/s]	2
Water density [Kg/m ³]	998.2
Turbulence intensity [%]	0.24

2.3 Performance indexes

For the design of the baseline and successive optimization of the desmodromic pitching system, *ad hoc* performance indexes were defined. More in detail, the rotor aerodynamic efficiency was quantified via the *power coefficient* C_P , while the rotor mechanical loading via the *shaft load oscillation coefficients* in the *streamwise* (ΔC_X) and *crosswise* (ΔC_Y) directions, respectively. The corresponding definitions are reported in Equations 1-3:

$$C_P = \frac{P}{0.5\rho AV_0^3} \quad (1)$$

$$C_X = \frac{F_X}{0.5\rho AV_0^2} \quad (2)$$

$$\Delta C_X = |C_{X,max(\theta)} - C_{X,min(\theta)}| \quad (3)$$

where ρ is the freestream flow density, A the turbine swept area, P the shaft power and V_0 the freestream wind speed. The subscripts $min(\theta)$ and $max(\theta)$ refer instead to the minimum and maximum values over one rotor revolution, respectively. It must be noted that the definition in Eqs. 2-3 is also valid for the crosswise load component C_Y .

3. Pitch law

The introduction of a desmodromic double-cam system, such as the one proposed here, could notably expand the range of achievable pitching laws, paving the way to the development of a control strategy, which better suits the characteristics of vertical-axis turbines.

In the present work, a *piece-wise* pitching law is proposed, which, as shown in Figure 3, was built according to the specific behaviour of the rotor blade in the different phases of one rotor revolution. In the *upwind* region ($0^\circ \leq \theta < 180^\circ$), where the dynamic stall phenomenon is predominant, the pitching action is employed to keep the blade AoA below a specified *target angle of attack*, here defined as the incidence maximizing torque production. It must be noted that this quantity, that is regulated by the parameter C_{up} , in unsteady conditions does not necessarily coincide with the static stall limit [19] and must be therefore considered an object of the optimization. The corresponding pitching law is obtained by subtracting the angle of attack in ideal conditions α_{id} , i.e. in absence of dynamic and blockage effects [13], from the target one:

$$\gamma_{upwind} = \begin{cases} (C_{up} - \alpha_{id}) \cdot \xi, & \alpha_{id} < C_{up} \\ 0, & \alpha_{id} \geq C_{up} \end{cases} \quad (4)$$

$$\alpha_{id}(\theta) = -\operatorname{atan}\left(\frac{\sin(\theta)}{TSR + \cos(\theta)}\right) \quad (5)$$

$$\xi = 1 - \tanh^8 \left[1.14 + 0.722 \left(\frac{\pi(\theta - \theta_i)}{\theta_f - \theta_i} - \pi \right) \right] \quad (6)$$

where $\xi(\theta)$ is a damping function that is applied at the intersection points θ_i and θ_f between the target and ideal AoA curves to avoid discontinuities. This strategy aims at combining the advantages of the *constant* and *sinusoidal* pitching strategies outlined in [13], smoothly raising the angle of attack as long as the flow is attached to the blade, for then preventing any further increase leading to performance degradation.

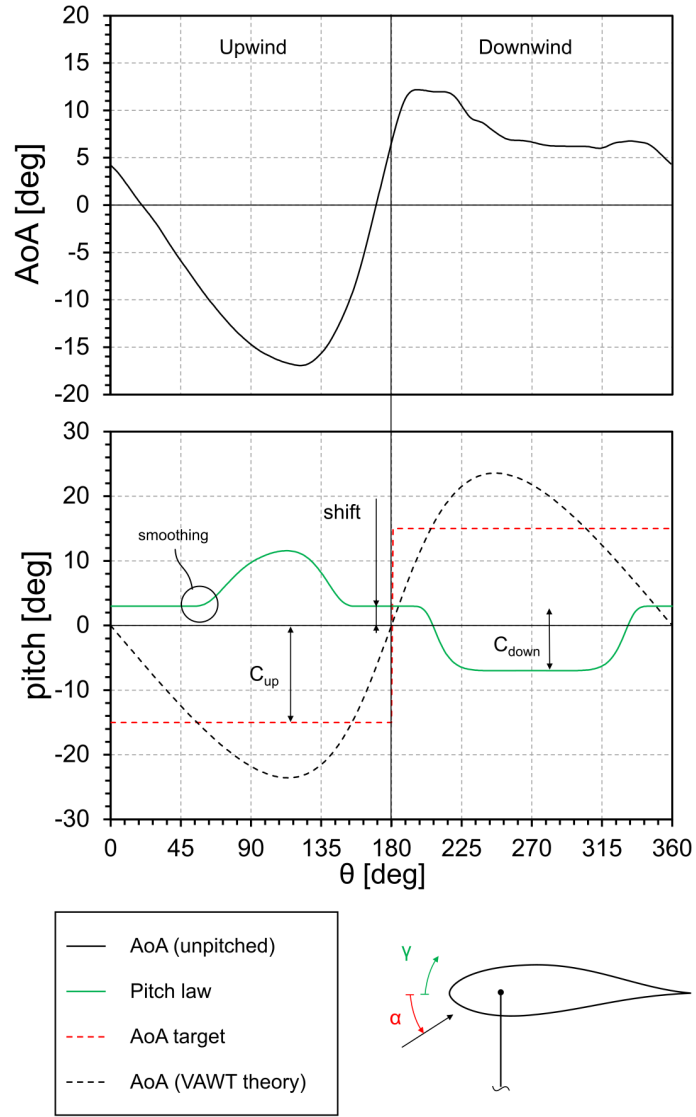


Figure 3 – Parametric pitch law.

In the *downwind* region ($180^\circ \leq \theta < 360^\circ$), on the other hand, due the presence of massive flow blockage, the un-pitched case presents a fairly “flat” AoA profile, well below the static stall limit (see Figure 3a). In this case, the pitching action is required to increase the blade loading as much as it is allowed by the local flow conditions. The corresponding law is built accordingly using a function from the *sigmoid* family:

$$\gamma_{downwind} = C_{down} \cdot [1 - \tanh^{16}(1.111 + 0.706(\theta - \pi))] \quad (7)$$

where pitching magnitude C_{down} is the second optimization parameter.

The final pitching law is eventually achieved by applying a constant offset (*shift*) to the combination of the upwind and downwind profiles, as in Eq. (8):

$$\gamma(\theta) = shift + \begin{cases} \gamma_{upwind}, & 0^\circ \leq \theta < 180^\circ \\ \gamma_{downwind}, & 180^\circ \leq \theta < 360^\circ \end{cases} \quad (8)$$

4. Optimization workflow

In order to find the optimal pitching law for the considered test case, dedicated tools were developed and combined into a workflow within the ANSYS® WORKBENCH® 20.2 framework.

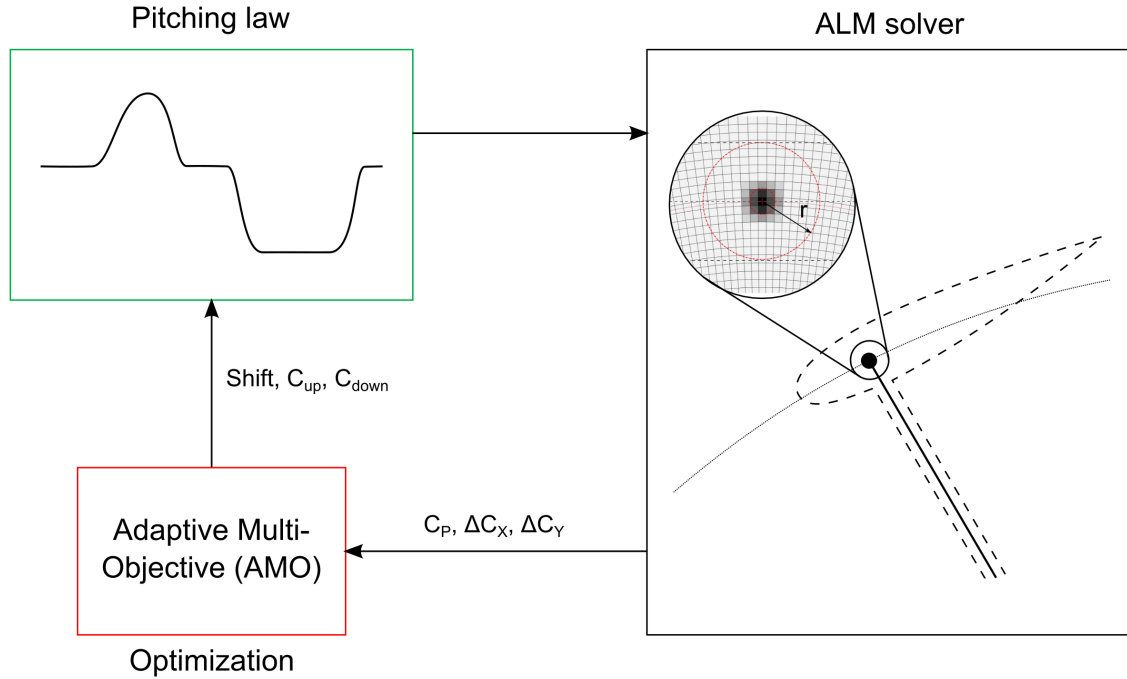


Figure 4 – Schematic representation of the adopted optimization algorithm.

As schematically shown in Figure 4, the latter consists essentially of three sub-modules, whose formulation is described in detail in the following sections:

- Pitching law*: For each set of the *shift*, C_{up} and C_{down} parameters given by the optimizer, this module generates a new azimuthal pitching law, as outlined in Section 3. The corresponding effect on turbine performance is then assessed via ALM simulation.
- ALM solver*: An Actuator Line model of the turbine, combining a lumped-parameter representation of the rotor blades with the CFD resolution of the flow field, is used to evaluate the performance of each pitching set-up in terms of *power coefficient* C_P (Eq. 1) and *rotor shaft load oscillations* in the *streamwise* (ΔC_X) and *crosswise* (ΔC_Y) directions, respectively. The definition of the last two quantities is reported in Equations 2 and 3.
- Optimizer*: The optimization module (see Section 4.2) elaborates the performance data coming from the ALM solution and adjusts the pitch law tuning parameters accordingly, until the optimal configuration maximizing the hydrodynamic efficiency and minimizing rotor shaft oscillations is achieved.

4.1 Actuator Line Method

Thanks to its accuracy and reduced computational cost, the Actuator Line Method (ALM) [29] was selected as the core tool for the proposed optimization procedure. In this approach, the rotating blade forces are estimated from the local inflow conditions, based on input tabulated polar data, and then plugged into the computational grid as equivalent volume momentum sources. A CFD-based approach is used, on the other hand, to solve the corresponding flow field. Since the actual geometry of the turbine blade is not numerically resolved, but modelled via a lumped-parameter representation, the overall grid density of the simulation, together with the corresponding computational cost, is notably reduced. The corresponding workflow can be synthetically described as follows. For each timestep: (a) the Aerodynamic Center (AC) of each blade element, i.e. the corresponding airfoil

quarter-chord point, is located on the grid based on the turbine characteristics and operating condition; (b) the local undisturbed velocity V is extracted from the resolved CFD flow field and is used to evaluate the blade angle of attack α and chord-based Reynolds number Re ; (c) the airfoil lift (C_L) and drag (C_D) coefficients are obtained via linear interpolation of the available tabulated aerodynamic data; (d) the blade element forces are inserted into the computational domain as volume momentum sources; (e) the modified unsteady Navier-Stokes set of equations is solved.

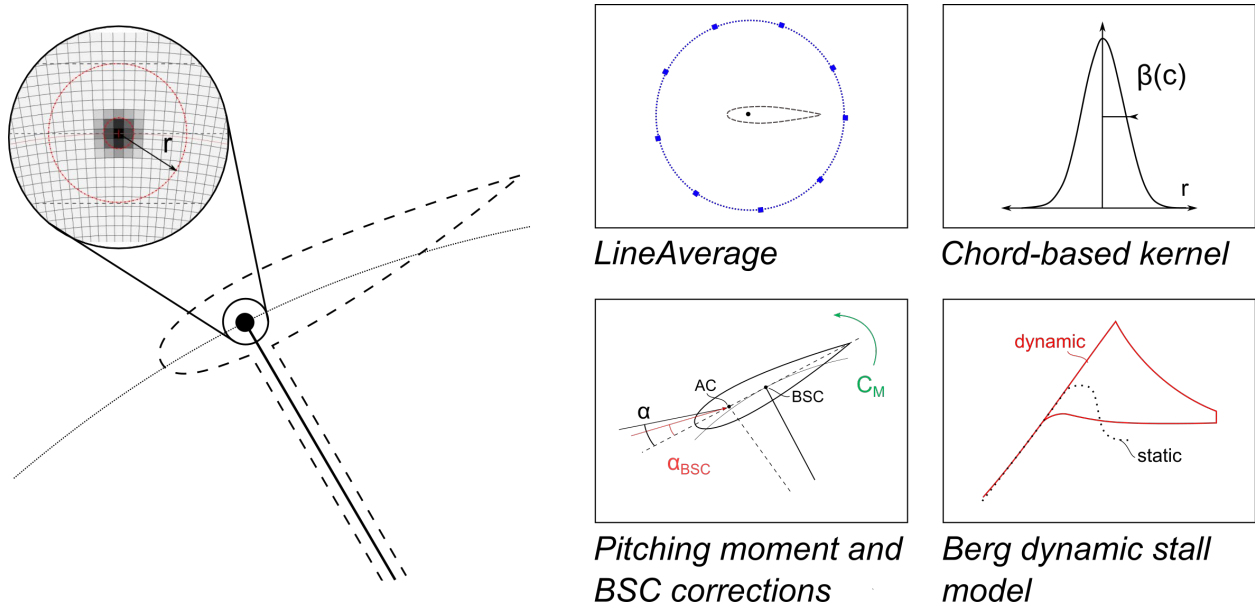


Figure 5 - ALM tool used for the optimization.

In the present work in particular, the ALM module developed by the authors in the ANSYS® FLUENT® solver, version 20.2, and specifically tailored for the simulation of vertical-axis machines [24], was adopted. A schematic overview of the proposed methodology is shown in Figure 5. The main features of the code are:

- *LineAverage method for sampling the angle of attack:* Originally proposed for horizontal-axis machines by Jost et al. [30], the *LineAverage* method (LineAve), computes the undisturbed velocity V as the integral average of the flow velocity field along a closed line around the airfoil, as shown in Figure 5. As verified by the authors in a recent work on VAWTs [31], this approach provides a reliable estimation of the blade angle of attack when applied to vertical-axis machines, under both unstable and stable operation. In the proposed tool, the latter is coupled with a low-pass filtering of the sampled data to remove the corresponding background noise [32]. This is achieved with a dedicated *Cubic Spline Smoothing (CSS)* algorithm [33], which at the beginning of each rotor revolution is applied to the AoA signal coming from the previous one, so that no *real-time* processing of the signal is required. The resulting signal is in this way smooth enough to be numerically derived over time to be given as an input to a dynamic stall model. Regarding the method set-up, it has been chosen for the present study to use a circular sampling line, centered in the aerodynamic center of the airfoil (assumed as the quarter-chord) and with a radius $r=lc$ and a resolution of $N=80$ equally spaced points along the sampling line.
- *Chord-based regularization kernel:* In order to spread the computed aerodynamic forces into the domain, the standard *isotropic two-dimensional Gaussian kernel* is used, as schematically depicted in Figure 5. The corresponding width β is related to an airfoil characteristic dimension [34,35], i.e. chord c , via an ad hoc calibration campaign performed by the authors in a previous work [36]. The resulting set-up corresponds to a $\beta/c=0.2$ and a minimum ALM grid resolution of $\beta/h=2$.

- *Pitching moment and BSC corrections:* The predicted torque is corrected to account for two phenomena specific of vertical-axis machines. On one hand, since the *airfoil pitching moment* C_M operates in the same plane of the turbine rotation, an additional contribution must be introduced into the shaft torque computation [37]. On the other hand, for turbine designs in which the Blade-Spoke Connection point (BSC), i.e. its point of connection to the corresponding support strut, does not coincide with the blade aerodynamic center (AC), the blade angle of attack and load repartition need to be modified accordingly [24,38]. It must be noted how for the selected test case, since the blade is mounted at quarter-chord (see Table 2), the latter correction is not necessary.
- *Berg dynamic stall model:* The dynamic stall model from Berg [39] is used in the code to account for the effect of unsteadiness on the blade loads, in the formulation proposed by the authors in [40]. More in detail, dynamic effects are embedded in the solution by introducing a *delay* in the measured angle of attack that is proportional to the flow reduced frequency via a coefficient γ_i . It must be noted how the latter, which is given in the model as a semi-empirical function of the airfoil thickness and the freestream Mach number, was treated in this work as a free parameter. The selected DU-06-W-200 profile in fact is way thicker than those tested in the original work of Gormont [41] for the derivation of this correlation. Lift and drag coefficients are eventually obtained as a linear interpolation between the static and dynamic ones, weighted over the constant A_M . For the present study, a dedicated calibration was performed for this parameter and the γ_i constants, as reported in Section 6.1.

For further details about the adopted ALM tool, please refer to [24].

Polar data

As for every aerodynamic code based on the blade element theory, the accuracy of the ALM methodology is largely based on the quality of the input polar data. For the small HKT considered for this optimization study, the pre-stall aerodynamic coefficients were obtained via blade resolved CFD for $Re \in [750 \times 10^3 \ 2000 \times 10^3]$. All simulations were performed with the two-dimensional RANS approach developed by the authors over the years in ANSYS® FLUENT® 20.2 solver [42]. Since at the given Reynolds numbers, the effect of laminar separation bubbles on the performance of thick airfoils as the one considered is still relevant [43], the *4-equations γ - Re_θ transition model* [44] was employed for the closure of the turbulence problem. The resulting polar data was then extended over the deep stall region, i.e. up to $AoA = \pm 180^\circ$, via the Viterna-Corrigan model [45], as recommended by Bianchini et al. [46] for VAWT applications.

It must be noted how, in order to take the *flow curvature* phenomenon into account in the simulation campaign, the tabulated data rotors were derived for a virtually cambered airfoil, obtained by proper *conformal mapping* from the geometric one according to the rotor chord-to-radius ratio c/R , blade-spoke connection point (BSC) and TSR [47]. As observed in several studies [48–50] in fact, due to the deformation of the local streamlines induced by its cycloidal motion, the turbine blade behaves aerodynamically (i.e. in terms of lift, drag and moment coefficients) like a virtually transformed equivalent airfoil, with a camber line defined by its arc of rotation and oriented outwards with respect to the turbine axis, as in Figure 6.

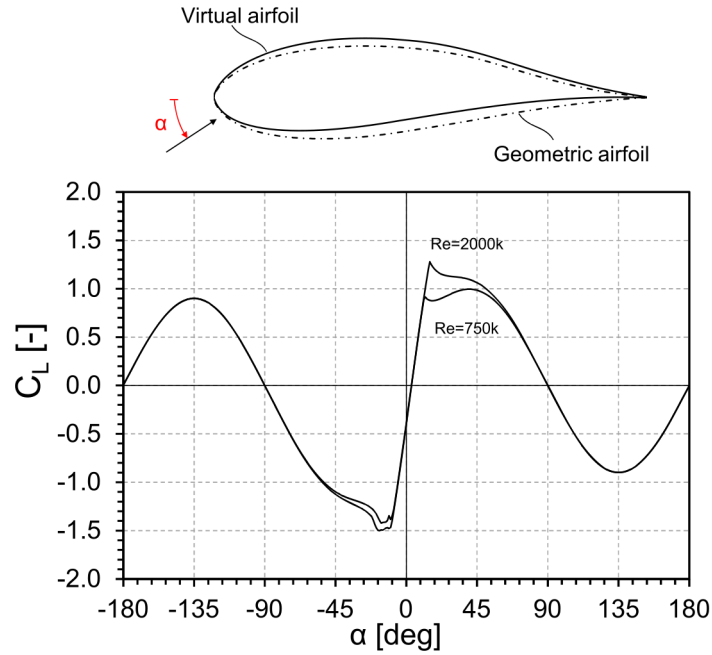


Figure 6 - Polar data of the virtual airfoil used for the ALM simulations.

Numerical set-up

All two-dimensional simulations were carried out with an unsteady Reynolds-Averaged Navier-Stokes (URANS) approach via the commercial solver ANSYS® FLUENT®, in the 20.2 release. For turbulence closure, the $k-\omega$ Shear Stress Transport (SST) model was selected: compared to the other models available, it provides better accuracy and stability in unsteady flow conditions such as the ones experienced by VAWT blades, with massively separated flows and strong pressure gradients [26]. For detailed information about the settings for adopted numerical schemes, convergence definition, maximum inner iterations and residuals, please refer to [26].

An open-field domain was chosen for the analysis. The corresponding dimensions, parameterized with respect to turbine diameter D , were adjusted to minimize the disturbance induced by the turbine on the freestream flow, justifying the application of the standard *far field* boundary conditions commonly adopted for external flows. More in detail, a uniform velocity, turbulent kinetic energy k and frequency ω were applied at the inlet based on the testing conditions of the considered test case (see Table 3). The turbulent quantities were estimated according to the test inlet turbulent intensity and length scale. At the outlet, a uniform atmospheric pressure was imposed, while for the other patches the *symmetry* condition was adopted.

The computational domain was discretized with a hybrid meshing strategy. For the toroid encompassing the turbine rotor (*ALM rotor*), where the actuator line forces are applied, a structured mesh was employed, according to the requirements of the ALM code. For the *wake and external* regions instead, an unstructured tetrahedral mesh was used. As shown in Figure 7, the element sizing corresponding to each region was evaluated based on the blade chord length, according to a dedicated sensitivity analysis performed by the authors in a previous work [24]. The resulting mesh consisted of approximately 81'000 elements.

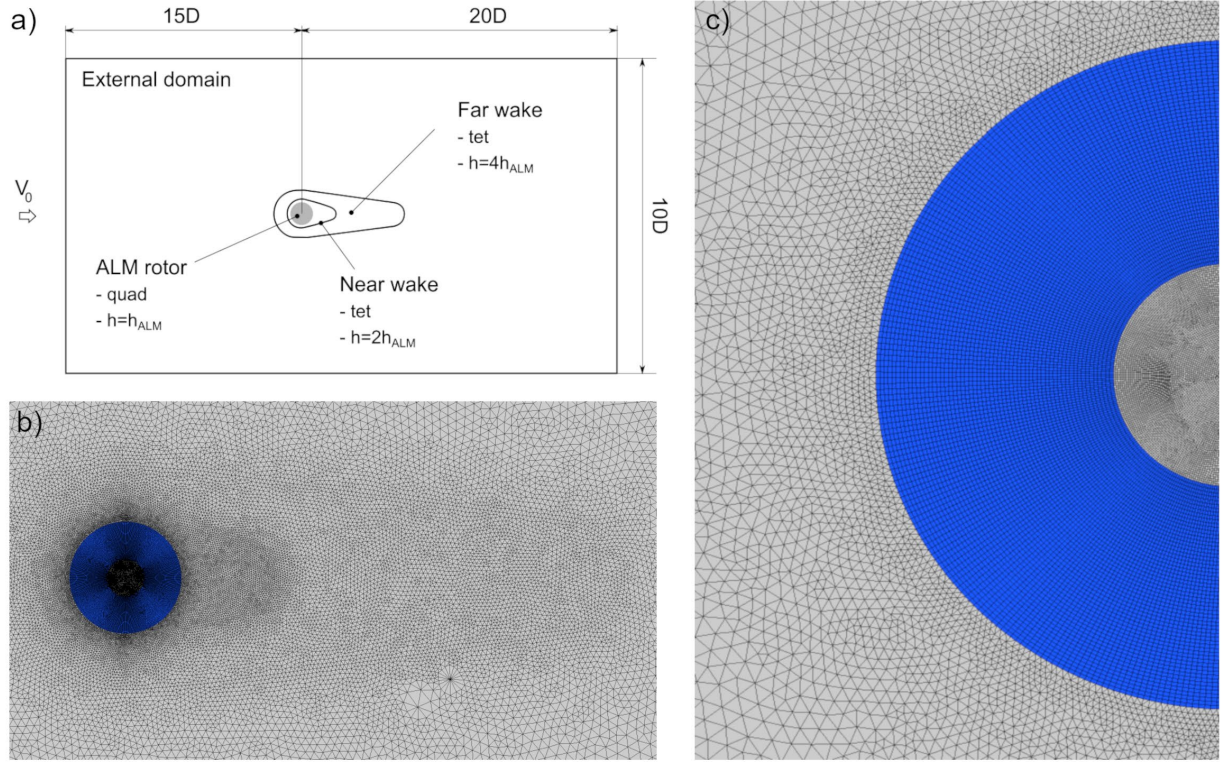


Figure 7 – Overview of the mesh used for ALM simulations: (a) computational domain (b) wake refinement region (c) ALM rotor region.

4.2 Optimization set-up

The optimization algorithm selected for the present work is the *Adaptive Multi-Objective (AMO)* approach included in the ANSYS Workbench Direct Optimization package. This routine allows to converge directly to the optimal configuration, given one or more criteria to be satisfied, via a series of sequential refinement steps. More in detail:

1. Given the initial range for each of the design parameters, a first set of configurations is generated via an *Optimal Space Filling* algorithm.
2. The response surface corresponding to the generated data is obtained using a Kriging interpolation.
3. A set of optimal configurations for the current data is extracted from the response surface via a *Multi-Objective Genetic Algorithm (MOGA)*, representing the best trade-off among the different objectives given as an input.
4. A new investigation domain is created from the results of the previous optimization, in order to locally refine the reference response surface, and the steps 1-3 are repeated until convergence.

For the current case the initial dataset was generated starting from the investigation range reported in Table 4, using 35 samples. The selection of the initial values was mainly based on the test case characteristics and the authors' experience. For every following iteration of the optimization routine, the number of available samples was restricted to 20. As anticipated in Section 4, the performance of each configuration was evaluated in terms of *power coefficient* C_P and *rotor shaft oscillations* ΔC_X and ΔC_Y . It must be noted that, as reported in Table 5, a higher priority was given to the maximization of hydrodynamic efficiency, given the framework in which this study takes place.

Table 4 - Initial investigation domain of the optimization

Design parameter	Range
C_{up} [deg]	[-24 -5]
C_{down} [deg]	[-10 10]
Shift [deg]	[-5 5]

Table 5 – Objectives of the optimization.

Objective	Range	Priority
Maximize C_p [-]	[0 1]	High
Minimize ΔC_x [-]	[0]	Medium
Minimize ΔC_y [-]	[0]	Medium

5. Pitching CFD Model

Due to the approximation introduced with the adoption of a lower order model such as the ALM, the results of the optimization were verified via an ad hoc blade-resolved U-RANS CFD model, able to reproduce the imposed blade pitching motion by means of a *sliding mesh technique*. The CFD modeling is based on a consolidated numerical approach that has been developed by the authors for VAWTs [26,27], and recently it has been soundly validated on HVATs upon comparison with towing tank [28] and on field [17] experiments (see Figure 8). For additional info on the validation, please refer to cited references. The *coupled* algorithm for pressure-velocity coupling is used, and the 2nd order upwind scheme for both RANS and turbulence equations. For turbulence closure, the *k- ω SST* model was adopted, in virtue of its suitability to VAWT simulation observed by the authors in previous works [26,51]. For detailed information about the settings for convergence definition, maximum inner iterations and residuals, please refer to [26].

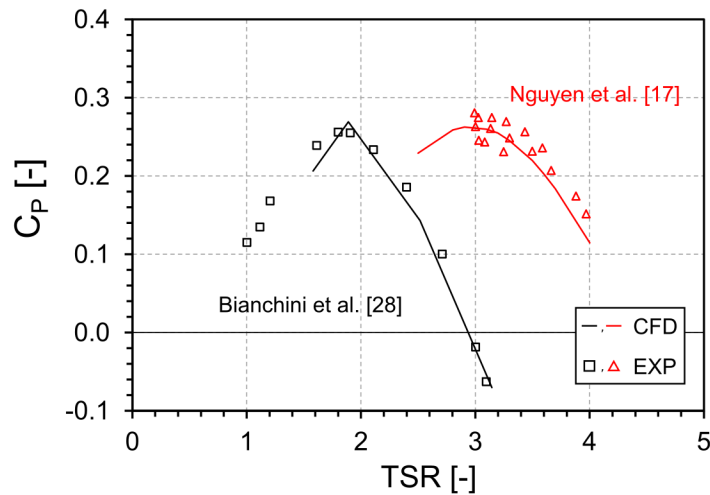


Figure 8 – Comparison between 2D blade resolved CFD simulations and experiments from [17,28].

5.1 Computational Domain

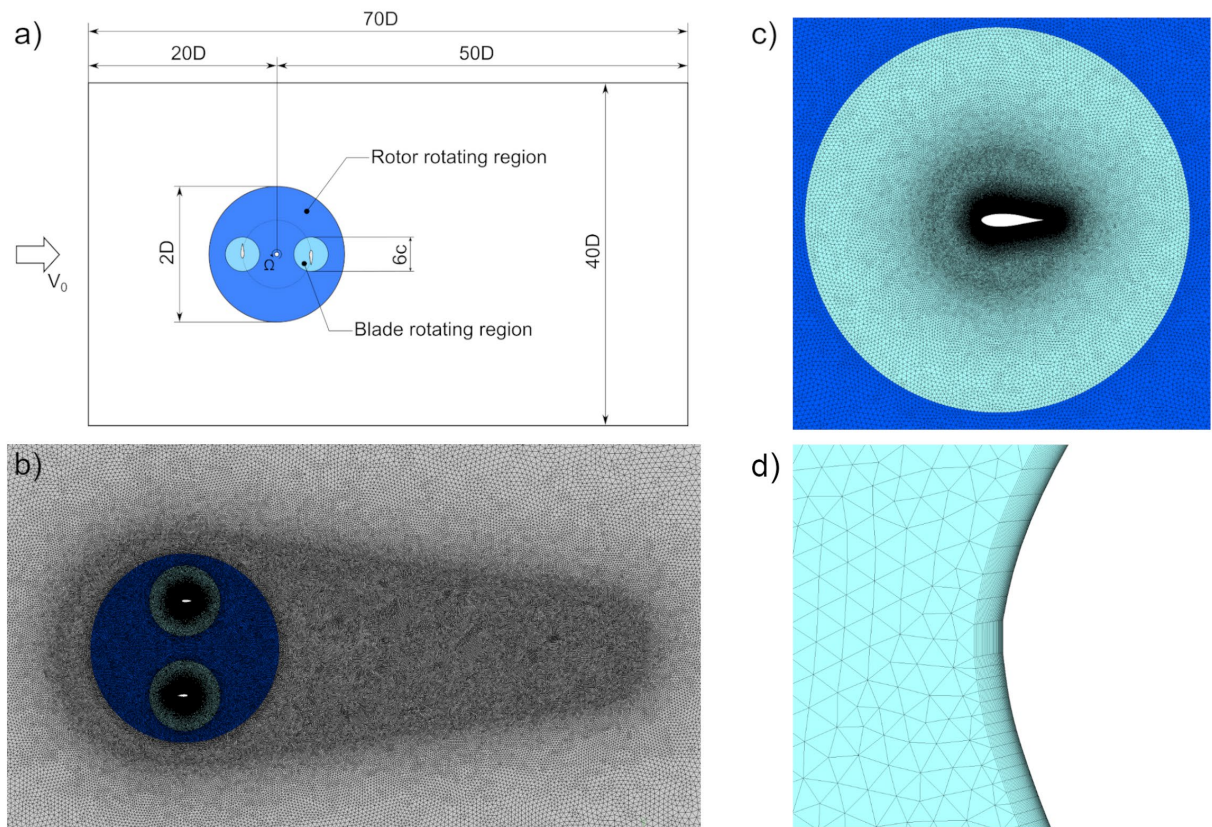
Figure 9 shows the computational domain adopted in the present study. All boundaries were distanced from the turbine axis in such a way that the standard *far-field* boundary conditions could be applied. Reference length unit for the analysis is the rotor diameter D . More in detail, a uniform flow velocity, corresponding to the freestream wind speed V_0 , was imposed at the inlet section, the ambient pressure condition at the outlet. For the lateral boundaries, a *symmetry* condition is defined. The computational domain was divided in four regions, coupled together via a *sliding mesh* technique:

407 a) the external domain b) the rotor region, rotating at a constant rotational speed Ω c) the two blade
 408 regions (see Figure 9c), whose motion is controlled by a FLUENT[®] User Defined Function (UDF),
 409 based on the pitch law defined in section 3 and taking as input parameters *shift*, C_{up} and C_{down} .

410 5.1 Mesh

411 The selected mesh refinement level and timestep were selected based on the experience of the
 412 authors on similar test cases and were verified to be compliant with the results of the sensitivity
 413 studies presented in [26] and particularly with the thresholds for the dimensionless vorticity and the
 414 Courant number proposed in [52]. An unstructured mesh, composed by triangular elements, was
 415 adopted for the whole domain, progressively refining towards the rotor region. In order to properly
 416 capture the development and mixing of the wake behind the rotor, an ad hoc drop-shaped refinement
 417 region was defined, as shown in Figure 9b. For the blade discretization on the other hand, an O-grid
 418 of quadrilateral elements was used, as visible in Figure 9d, so that the boundary layer around the
 419 airfoil could be adequately resolved. Complying with the requirements of the $k-\omega$ SST turbulence
 420 model, the size of the wall-adjacent cell was dimensioned to guarantee a dimensionless wall distance
 421 (y^+) lower than ~ 1 . An overview of the main settings is reported in Table 6.

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424 **Figure 9 – Overview of the mesh used for blade-resolved CFD simulations: (a) computational domain (b) wake refinement**
 425 **region (c) blade rotating region (d) airfoil leading edge boundary layer.**

426 **Table 6 - Spatial and temporal discretization settings of the turbine CFD model.**

Setting	Value
Grid size [# elements]	992'334
# elements (blade) [-]	720
# layers (boundary layer) [-]	40
angular timestep [deg]	0.2

6. Results

In this section, the main outcomes of the current study are presented. After a preliminary section regarding the validation of the Actuator Line model used for the rotor simulations, the results of both the *fixed-* and *variable-pitch* optimization procedures are illustrated in terms of blade loads and rotor shaft oscillations. The sensitivity of the rotor aerodynamic and mechanical performance to the pitch law tuning parameters is also discussed. Eventually, the reliability of the obtained results is verified via dedicated pitching blade resolved CFD simulations.

6.1 Validation of the ALM model

As a preliminary step for the pitch law optimization, the Actuator Line model used for the evaluation of rotor performance was subjected to a dedicated validation, so that its suitability for this kind of analysis could be assessed.

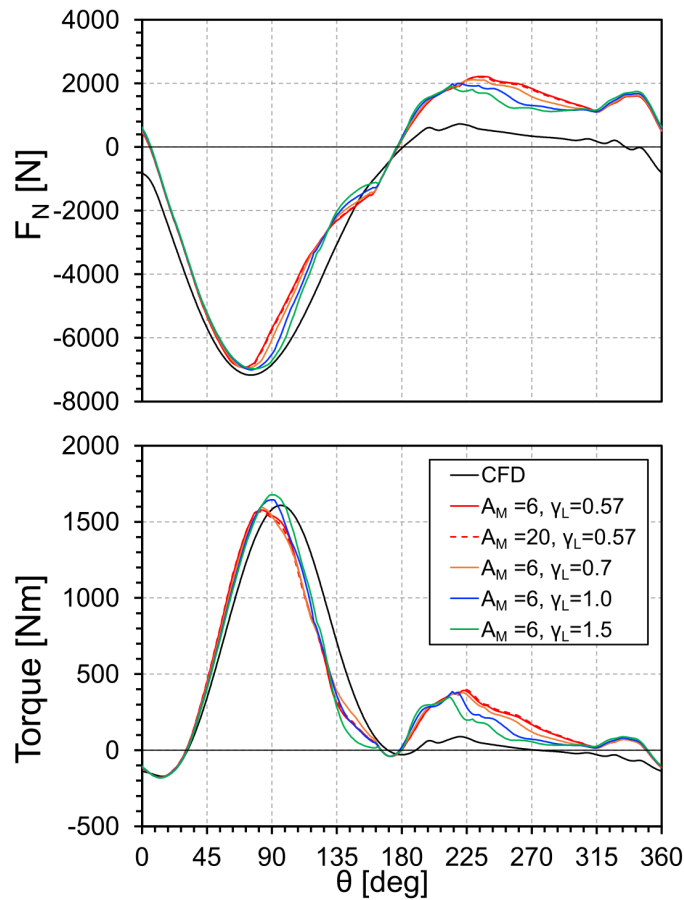


Figure 10 – Validation of the rotor ALM model on reference blade-resolved CFD data.

In this frame, the calibration of the Berg dynamic stall model, which, as anticipated in Section 4.1, requires as input two empirical coefficients (i.e. the constants A_M and γ_i) was also performed. It must be noted that for the coefficient γ_i , which quantifies the sensitivity of the blade loads to the flow unsteadiness, only the value referring to the lift (γ_L) was varied. The one related to the drag (γ_D) was instead taken as $\gamma_L/2$, as recommended by [41].

Figure 10 shows the comparison in terms of blade normal force (F_N) and torque between the ALM and the blade resolved CFD simulations from Balduzzi et al. [4], which were taken as a benchmark for the present work. In general, the agreement between the two is good, especially in the *upwind* sector during the *pitch-up* phase ($\theta < 90^\circ$). In the *pitch-down* one instead ($90^\circ < \theta < 180^\circ$), the ALM

struggles to capture the *lag* between angle of attack and blade loads normally observed on VAWT blades during unsteady operation [31]. This effect, which is notoriously hard to describe with simpler dynamic stall models such as the Berg one, leads to an underestimation of the blade loads by the numerical model. The opposite issue can be highlighted in the *downwind* region ($180^\circ < \theta < 360^\circ$), where the ALM is known to slightly underestimate the rotor blockage phenomenon, i.e. the slowing down of the flow due to energy extraction, happening in the upwind one.

Despite the problems described so far, the adopted numerical model is believed to be sufficiently accurate to be employed in the proposed optimization, also considering the comparative nature of the analysis. For the latter, the set-up with ($A_M=6$, $\gamma_L=1.5$) was employed, since it provides the best matching with the CFD benchmark data, as clearly shown in Figure 10.

6.2 Variable-pitch optimization

Both *fixed-* (FP) and *variable-pitch* (VP) optimization routines were run with the set-up described in Section 4.2, reaching convergence after 65 and 79 iterations, respectively. This corresponds approximately to 9 days of computation time on a 32-core workstation. Only difference between the two is the number of free pitch law parameters, which in the FP case was reduced from three to one, i.e. the pre-pitch *shift*. The obtained optimal configurations are numerically compared with the *baseline* one in terms of aggregated performance indexes (C_P , ΔC_X and ΔC_Y) in Table 7. This dataset is also graphically shown in Figure 11, where it is sided by the system response surfaces to the variation of the pitch law tuning parameters.

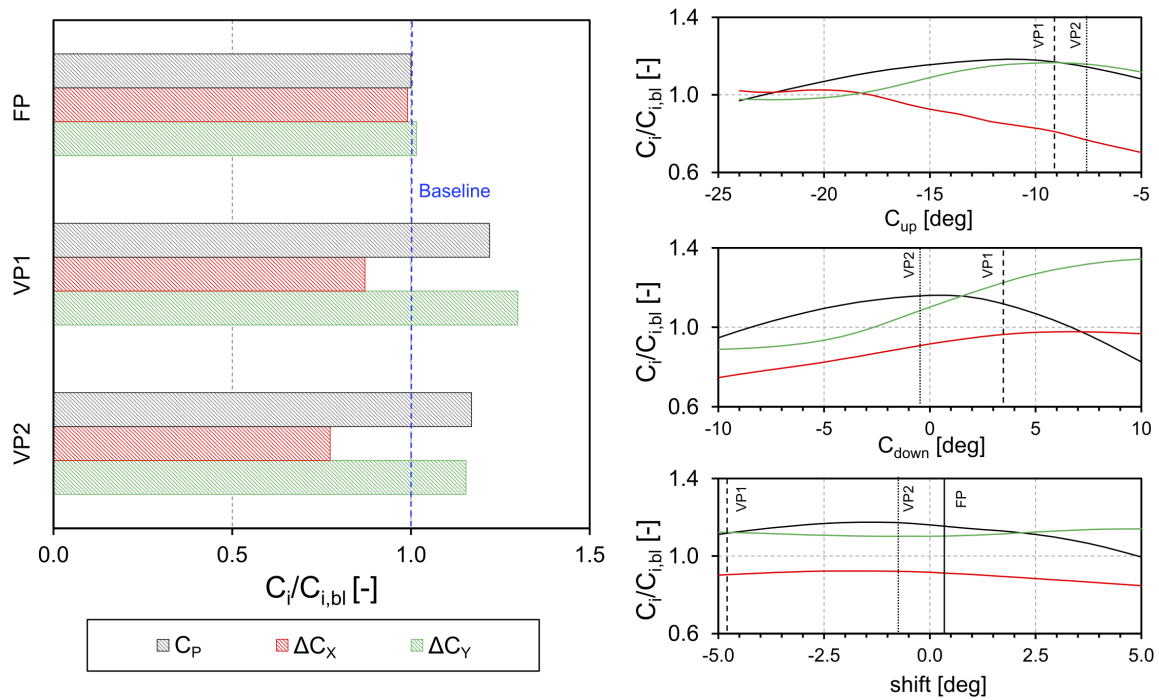


Figure 11 – ALM Optimization results.

Table 7 – ALM Optimization results.

	Baseline	FP	VP1	VP2
C_{up} [deg]	-	-	-9.0714	-7.4235
C_{down} [deg]	-	-	3.3586	-0.2337
shift [deg]	-	0.87	-4.7337	-1.400
C_P [-]	0.4562	0.4561	0.5566	0.5336
ΔC_X [-]	2.0771	2.0580	1.8103	1.6079
ΔC_Y [-]	2.0929	2.1259	2.7185	2.4147

At first glance, one can observe how the *fixed-pitch* strategy, i.e. applying a pre-set pitch to the turbine blades, leads in this case to no benefits in terms of performance. On the contrary, a -1% reduction of the power coefficient with respect to the *baseline* is obtained, while the shaft load oscillations are substantially unvaried. This can be understood upon examination of the blade loading trends over one rotor revolution reported in Figure 12. As a matter of fact, since in the *baseline* configuration *upwind* dynamic stall is already well-developed, as evident from the lift (C_L) and drag (C_D) coefficients hysteresis loops, no further increase in the blade angle of attack can be applied without limiting its maximum value. All negative values of the shift parameter lead therefore to a performance degradation. Positive values on the other hand, as the one used in this case, can improve *downwind* torque production, but at the cost of reducing the blade loading in the *upwind* sector. The final result is no neat improvement of the rotor power extraction. One should remember, however, that this outcome is strongly dependent on the two-fold scope of the optimization (i.e. maximization of power extraction while keeping loads low), while the literature [15] suggests that a non-null pitch could improve the power coefficient only.

The introduction of the double-cam *variable-pitch* system is able to remove the constraint described so far. This possibility was exploited in two different ways in the frame of the proposed optimization.

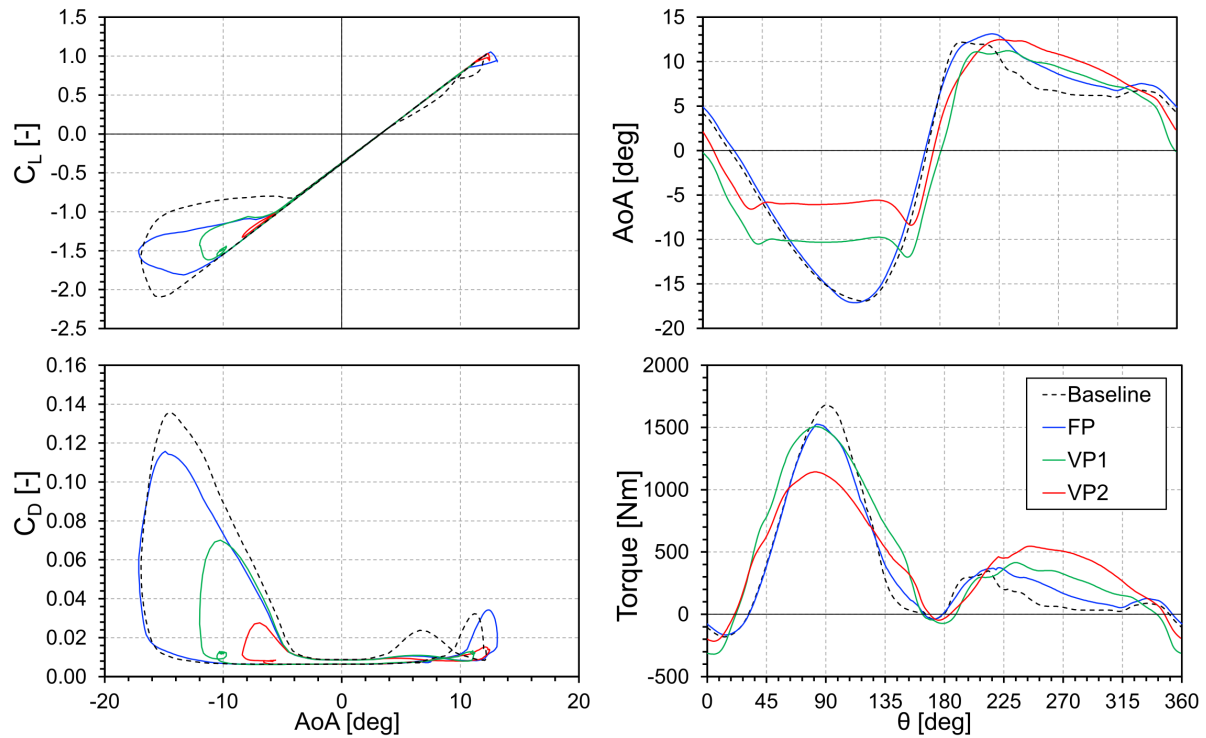


Figure 12 – Comparison in terms of blade aerodynamic performance between the baseline, the optimum of the fixed-pitch optimization (FP) and the two candidate points from the variable-pitch one (VPi).

The first (VP1) can be referred as the “*maximum power*” approach, as it privileges energy extraction over mechanical resistance. As shown in Figure 12, this strategy tends to maximize *upwind* torque production by raising the shift parameter as much as possible, up to ca. -4.7° , for then saturating the blade angle of attack to a target value of ca. -10° . Doing that, the blade load is increased with respect to *baseline* both in the *pitch-up* and *pitch-down* phases, controlling the dynamic stall phenomenon in such a way, that the typical lift overshoot is obtained without an excessive penalty coming from the drag increase. In the *downwind* sector instead, where an enhanced performance degradation is observed due to the increased rotor blockage, the pitching action is employed to both compensate the negative shift applied to the *upwind* sector and keep the angle of attack below its

target limit, resulting in a C_{down} value of ca. 3.3° . The final result is an increase in the overall C_P of ca. +22% and a reduction of the streamwise shaft load oscillation ΔC_X of ca. -13% with respect to the baseline, as reported in Figure 11. The corresponding large blade load unbalance between the *upwind* and *downwind* sectors leads nonetheless to an increment of the shaft oscillations in the crosswise direction, as clearly visible in Figure 13. ΔC_Y is therefore +29% higher than that of the *baseline*.

The latter issue was addressed by switching to the VP2 approach, or “*balanced design*”. This strategy relies completely on controlling dynamic stall in the *upwind* sector of one rotor revolution, lowering the target angle of attack – which is approximately halved with respect to the VP1 configuration - until unsteady effects are almost nullified. This can be clearly seen in the lift and drag loops reported in Figure 12, which show almost no signs of hysteresis. By notably unloading the rotor blade in the *upwind* sector, thus reducing the corresponding blockage, it becomes possible to achieve a good torque production in the *downwind* one as well. In this case, almost no pitching action is required, so that the C_{down} value is slightly lower than 0. The result is an increase in the power coefficient of ca. +17%, which, although slightly lower than the one achieved with VP1, comes with a reduction around -23% for ΔC_X and an increment of +15% for ΔC_Y , ten percent points more and fourteen less than those of the “*maximum power*” approach, respectively. The latter load balancing effect can also be observed in Figure 13, where the oscillation of the rotor shaft load coefficients is reported on the X-Y plane.

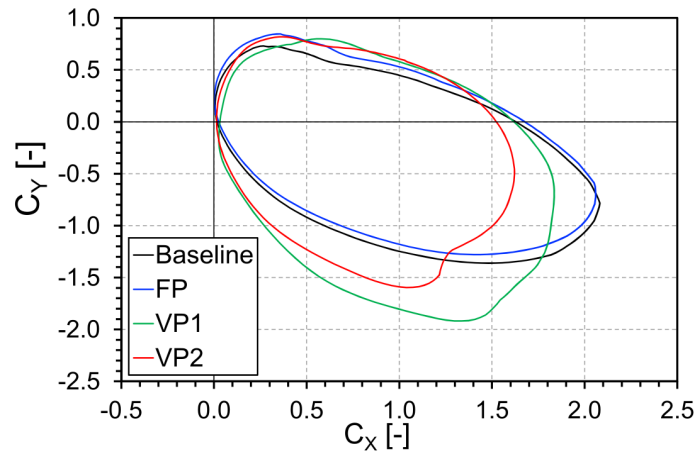


Figure 13 – Comparison in terms of rotor shaft load coefficients between the baseline, the optimum of the fixed-pitch optimization (FP) and the two candidate points from the variable-pitch one (VPi).

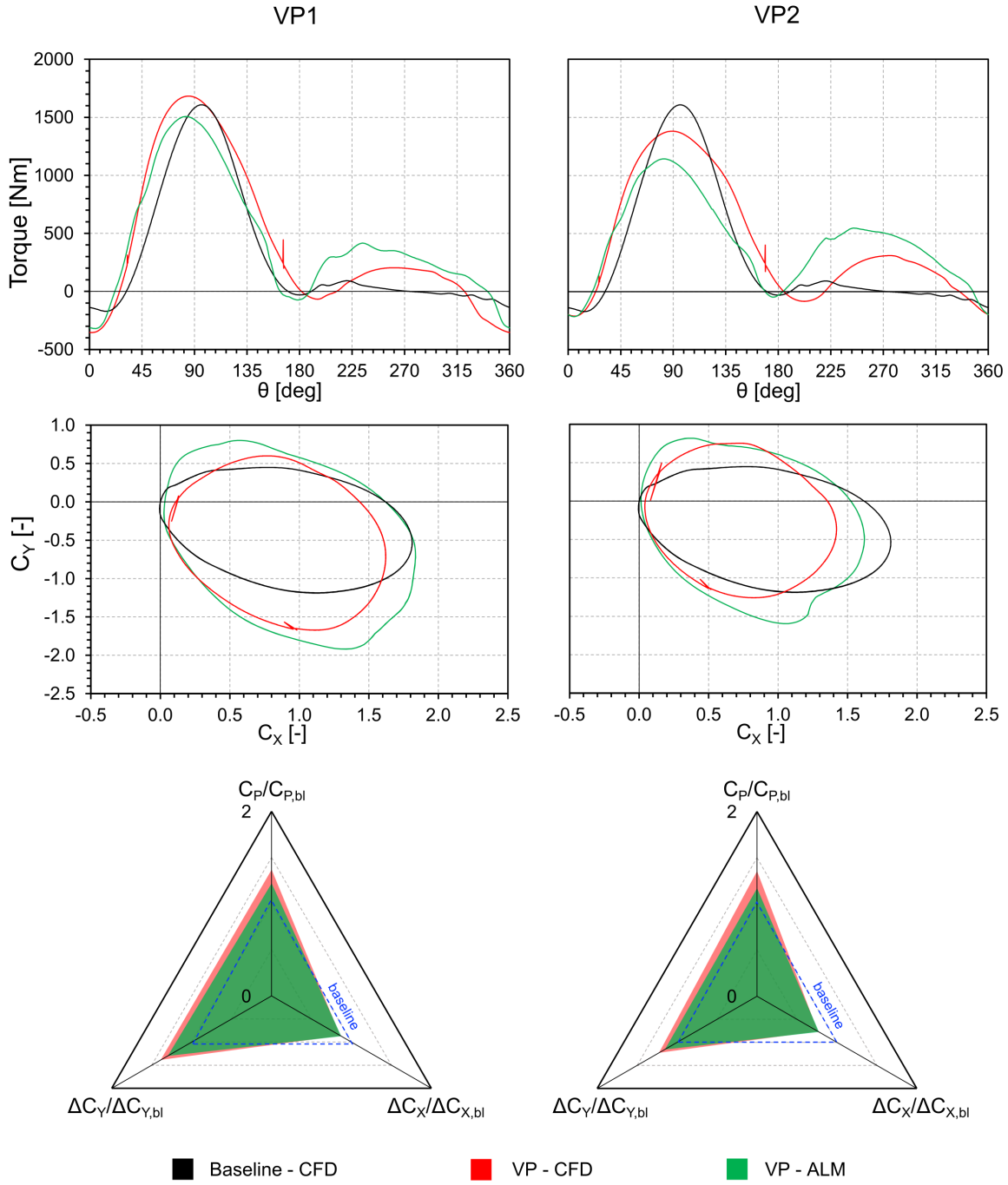
6.3 Blade-resolved CFD verification

In order to understand whether the configurations coming from the ALM-based optimization could actually lead to an improvement in the test machine performance, or were mere results of the approximations intrinsically related to the ALM method and outlined in Section 6.1, an *ad hoc* validation was performed via the pitching blade-resolved model described in Section 5.

Figure 14 reports in a synthetic way the comparison in terms of blade torque and shaft load coefficients over one rotor revolution between the *baseline* and the optimized designs VP1 and VP2, simulated with the adopted blade resolved CFD approach. The corresponding ALM data is also shown as a reference. This visual dataset is completed by the aggregated performance indexes used for the current work, i.e. C_P , ΔC_X and ΔC_Y , computed for both ALM and CFD and normalized over the correspondent baseline values. In general, the same biases coming up from the validation of the baseline model and outlined in Section 6.1 can also be observed in the active pitching case. Once again, the ALM tends to deviate from the blade-resolved data in the *upwind pitch-down* phase ($90^\circ < \theta < 180^\circ$), where performance-boosting unsteady effects are relevant also in absence of dynamic stall, and in the *downwind* region ($90^\circ < \theta < 180^\circ$), where torque extraction is overestimated. The rotor

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blockage effect is in fact usually underestimated by the ALM, especially for medium-to-high solidity rotors such as the one under consideration.



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Figure 14 – Comparison in terms of blade torque, rotor shaft oscillations and aggregated performance indexes between blade-resolved CFD and ALM for the optimum design points VP1 and VP2.

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Due to these issues, the ALM predicts a higher power extraction and shaft load oscillations than CFD, as evident from Figure 14 and the numerical data reported in Table 8. For the *baseline* configuration, the computed C_P , ΔC_X and ΔC_Y are +15%, +14% and +27% higher than the CFD ones, respectively. Considering a variable-pitch case such as the maximum power design (VP1), on the other hand, this deviation is strongly reduced, with an overestimation of the power coefficient and the crosswise oscillations of +1% and +19%, in the order given. In the authors' experience, this might be related to the specific stall control strategy adopted for the pitching system and the consequent

lowering weight of dynamic stall on the blade loads. The latter is in fact responsible for a large part of the uncertainty associated to ALM simulations.

Analyzing the same dataset in relative terms instead, the two methods seem to have the same behavior. As clearly shown in the spider diagrams in Figure 14, the percentage gain predicted by ALM and blade-resolved CFD with respect to the corresponding baselines is in fact quite similar for the different performance indexes, in particular the stream-wise oscillation coefficient ΔC_X . The other two coefficients, on the other hand, are slightly higher for CFD, which predicts for instance a +37% gain on the CP of the VP1. For the ALM the latter was +22%. Although this verification should be performed on other points of the optimization space, these results are a first hint of the reliability and robustness of the proposed ALM-based approach. It must be also noted that the latter provides a conservative result with respect to higher-order methods.

Another interesting outcome of this CFD verification step is that the efficiency gap between VP1 and VP2 is smaller than the one predicted by ALM, but the corresponding benefits in terms of shaft load oscillations reduction remain more or less unchanged. As reported in Table 8, CFD-VP2 has a C_P only -1% lower than CFD-VP1, which comes though with a -13% ΔC_X and a -13% ΔC_Y . In the frame of the present work, it represents thus the best trade-off between power and mechanical resistance.

Table 8 – Results of the blade-resolved CFD simulations.

	Baseline	VP1	VP2
C_{up} [deg]	-	-9.0714	-7.4235
C_{down} [deg]	-	3.3586	-0.2337
shift [deg]	-	-4.7337	-1.400
C_P [-]	0.3950	0.5410	0.5355
ΔC_X [-]	1.8167	1.5611	1.3797
ΔC_Y [-]	1.6386	2.2719	2.0119

7. Prospects and Discussion

Based on the presented results, the proposed pitching laws developed in view of a possible actuation using a *double-cam desmodromic* pitching system have shown the potential to be an effective strategy for the power augmentation of HVATs. The obtained efficiency increase of ca. +35% with respect to the *baseline* for both the “*maximum power*” and “*balanced design*” solutions is higher than the majority of those reported in the literature for the family of sinusoidal functions (see Table 1). Considering the maximum value of this coefficient nonetheless, which can go up to +53% for *higher-order sine* in the study of Paillard et al. [20] and +63% for the *truncated sine* according to Chen et al. [12], the *double-cam* solution still falls behind. These data can be deceiving though. In fact, the baseline design and operating conditions are not uniform among the different studies selected for the comparison. The rotor solidity in particular, as demonstrated by Sagharichi et al. [21], plays a fundamental role in defining the power increase potentially achievable with a variable-pitch system, thus introducing a certain amount scattering between the different studies. It must also be noted that the *baseline* selected for the current work already derives, as described in Section 2.1, from a dedicated optimization, which raised the maximum C_P to almost 0.4. In this case, a +35% increase corresponds to a C_P of ca. 0.54, which is in line with that computed by Paillard and ten points higher than the 0.45 obtained by Chen.

From the point of view of load reduction, on the other hand, a direct benchmarking with the existing technologies is hindered by the lack of quantitative data and standardisation in the definition of the load indexes. This study has demonstrated nevertheless the capacity of the proposed pitching

system to notably reduce the rotor shaft load oscillations in the streamwise direction, up to -23%, at the cost of a slight increase in the cross-wise one (+15%).

Despite the promising characteristics outlined so far, the development of this technology is still at an early stage. As demonstrated by Kirke [6] in fact, the real behaviour of variable-pitch HVATs is notably different from the one predicted by two-dimensional CFD, mainly due to the effects of tip vortexes and of the blades support struts, which in hydrokinetic applications are even more relevant than in wind ones. The combination of these factors usually leads to a measured performance that is lower than the one predicted by numerical simulations. In order to embed these phenomena in the design of the pitching system, a thorough validation of the numerical approach will be performed, based on the experimental data that is going to be collected once a first prototype of the baseline turbine is built. The same dataset might also serve as a support for the extension of the prediction capabilities of the ALM tool, so that the proposed optimization framework could be used in the future for the design of a real *desmodromic* pitching system.

8. Conclusion

In the present study, a novel control strategy, based on a *double-cam desmodromic* pitching system, has been proposed for hydrokinetic vertical-axis turbines. The objective of the optimization was the maximization of the turbine's power coefficient, while keeping the rotor stream- and cross-wise force oscillations low. The results of the study can be synthetized in the following outcomes:

- The optimized *fixed-pitch* configuration proved to be unsuccessful for this specific case, leading to a slight performance degradation with respect to the *baseline*, i.e. no pitch applied.
- The first design ("*maximum power*") coming from the optimization routine has maximized power production by loading the blade to its limit in the *upwind* sector, thus penalizing torque production in the *downwind* one. The result is a C_P of ca. 0.54, corresponding to a +37% increase with respect to the *baseline*, and a reduction of the streamwise shaft load oscillations ΔC_X of -16%. Cross-wise shaft load oscillations nonetheless are enhanced of approximately +29%.
- The second design generated by the optimization, or "*balanced design*", has proved to be the best trade-off between machine performance and mechanical loading. Although slightly less efficient than the "*maximum power*" one, with a C_P of 0.535, this configuration is able to further reduce streamwise load oscillations (ΔC_X -23%) and better control the increase in the cross-wise one, which amounts to +15%. This is achieved by allowing for a higher energy extraction in the *downwind* sector by limiting the *upwind* blade loading below its maximum potential.
- Dedicated pitching blade resolved CFD simulations confirmed the results of the ALM-based optimization, providing a first hint of the reliability of the ALM simulation technique for this kind of applications.

On this basis, the present study is thought to prove how the application of relatively simple pitch-variation systems could bring hydrokinetic turbines to unprecedented efficiency levels, making the technology extremely attractive for a future expansion of energy harvesting in flowing water applications.

9. Acknowledgements

The authors would like to acknowledge Dr. Mattia Pampaloni for its support to the present study.

10. Nomenclature

Acronyms

AC	Aerodynamic Center
ALM	Actuator Line Method
AMO	Adaptive Multi-Objective
AoA	Angle of Attack
BEM	Blade Element Momentum
BSC	Blade-Spoke Connection
CFD	Computational Fluid Dynamics
CSS	Cubic Spline Smoothing
FP	Fixed Pitch
VP	Variable Pitch
HKT	Hydro-Kinetic Turbine
HVAT	Hydrokinetic Vertical-Axis Turbine
MOGA	Multi-Objective Genetic Algorithm
SST	Shear Stress Transport
TSR	Tip-Speed Ratio
URANS	Unsteady Reynolds-Averaged Navier-Stokes
VAWTs	Vertical Axis Wind Turbines
WTs	Wind Turbines
UDF	User Defined Function

Greek symbols

α	Angle of Attack (symbol)	[deg]
β	Gaussian Kernel Width	[m]
γ	Blade Pitch	[deg]
ϑ	Blade Angular Position	[deg]
ρ	Freestream Flow Density	[Kg/m ³]
Ω	Rotational speed	[rad/s]

Latin symbols

c	Blade Chord	[m]
C_D	Drag Coefficient	[-]
C_L	Lift Coefficient	[-]
C_M	Moment Coefficient	[-]
C_P	Rotor Power Coefficient	[-]
C_{up}	Pitch Law Target Angle of Attack	[deg]
C_{down}	Pitch Law Downwind Gain	[deg]
D	Rotor Diameter	[m]
ΔCX	Shaft Streamwise Load Oscillation Coefficient	[-]
ΔCY	Shaft Crosswise Load Oscillation Coefficient	[-]

677	F_N	Blade Normal Force	[N]
678	h	Cell Characteristic Dimension	[m]
679	R	Rotor Radius	[m]
680	Re	Chord-based Reynolds Number	[-]
681	$shift$	Pitch Law Pre-pitch	[deg]
682	V	Undisturbed Flow Velocity Vector	[m/s]
683	V_0	Freestream Wind Speed	[m/s]

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843