CFD Investigation of Separation Control on a Vertical Axis Wind Turbine: Steady and Unsteady Suction

Abdolrahim Rezaeiha\textsuperscript{1,2}, Hamid Montazeri\textsuperscript{1}, Bert Blocken\textsuperscript{1,2}

\textsuperscript{1} Eindhoven University of Technology, Eindhoven, The Netherlands
\textsuperscript{2} KU Leuven, Leuven, Belgium

Author contact email: a.rezaeiha@tue.nl

\textbf{Abstract.} High-fidelity two-dimensional unsteady Reynolds-averaged Navier-Stokes (URANS) simulations are employed to investigate the influence of boundary layer suction through a slot located near the leading edge of a vertical axis wind turbine operating in dynamic stall. The analysis includes both steady and unsteady suction with different frequencies. The results show that: (i) when the suction slot is located within the chordwise extent of the laminar separation bubble, dynamic stall can be avoided with minimal suction amplitude; (ii) the most promising suction location is the most upstream suction location studied, at 8.5\%c where c is the blade chord length; (iii) the suction only needs to be applied during the azimuthal angles when dynamic stall occurs; (iv) the oscillation frequency of the suction velocity has insignificant influence on the obtained turbine power gain; (v) applying unsteady suction is interesting as it reduces the energy consumption of the suction system, thus the net power gain.

Keywords: Active flow control, power enhancement, smart rotor, boundary layer suction, aerodynamics

\section{1. Introduction}

Flow separation is an undesired flow phenomenon, which negatively influences the aerodynamic loads on wind turbine blades, by limiting the maximum lift, increasing the drag and resulting in fluctuating loads, and consequently abates the turbine power performance. In addition, it generates noise and results in structural vibration, where the latter could limit the turbine lifetime due to fatigue. Therefore, separation control is highly desired on wind turbine blades [1-3].

Active flow control (AFC) has emerged as a mechanism for flow separation control and has already been extensively studied for aeronautical applications [4]. However, it has not yet received sufficient attention for wind turbines [5, 6].

The current study intends to investigate the impact of slot suction, as one of the most classic AFC methods for flow separation control, on the aerodynamic performance of a vertical axis wind turbines. The suction is applied through a slot positioned near the blade leading edge, where the suction amplitude (A\textsubscript{s}) and location (X\textsubscript{s}/c) vary within 0.5\% \leq A\textsubscript{s} \leq 10\% and 8.5 \leq X\textsubscript{s}/c \leq 28.5, (c is the blade chord length).
The focus of the analysis will be on the turbine power enhancement due to the boundary layer suction. The studied wind turbine is an H-type Darrieus vertical axis wind turbine (VAWT), which suffers from a poor power performance at low tip speed ratios ($\lambda = 2.5$ and $3.0$) due to massive flow separation on blades.

This work is a continuation of our previous work already published in Ref. [7]. The earlier work focused on steady suction applied during the whole turbine revolution, where the influence of suction location, suction amplitude was investigated. In addition, the dependency of the optimal suction settings on the operational parameters, namely tip speed ratio, Reynolds number and turbulence intensity was also analyzed. In the present study, the following two scenarios are studied:

1) An investigation is performed to compare the impact of suction on the turbine power performance between two cases: (i) suction is applied during the whole turbine revolution, (ii) suction is applied only during the azimuthal angles where dynamic stall occurs.

2) The impact of unsteady suction on the turbine power performance is also investigated. For this case, the suction velocity is a sinusoidal time-varying function with different frequencies.

The analysis is based on high-fidelity unsteady Reynolds-Averaged Navier-Stokes (URANS) simulations extensively validated with experiments [7].

The outline of the paper is as follows: computational settings and parameters are presented in Sec. 2. The analysis of the impact of steady and unsteady suction are discussed in Sec. 3. Discussion and conclusions are provided in Sec. 4 and 5, respectively.

2. Computational settings and parameters

Table 1 lists the turbine geometrical and operational characteristics and the suction settings. The turbine characteristics are the same as the one employed in the experiment by Tescione et al. [8], which is also used for validation of our CFD simulations, see Ref. [9]. Note that in order to reduce the computational cost, the turbine is simplified with only one blade, no shaft and no spokes [10, 11]. The suction width, i.e. 2.5% of the chord, is based on the recommended values in the literature [12]. The suction is modeled as a uniform velocity inlet imposed along the slot width. For the case of steady suction, the velocity magnitude is constant in time while for the unsteady suction, the velocity magnitude is a time-varying sinusoidal function, see Eq. 2 in Sec. 3.

The suction slot is positioned on the blade’s inner side. Based on the knowledge gained through the prior simulations for a similar case without suction [13, 14], the suction slot location is positioned within the laminar separation bubble (LSB) on the blade’s inner side, which corresponds to the airfoil suction side during the first half revolution of the turbine. Note that considering the approach-flow turbulence level and Re, a laminar boundary layer could form along the leading edge of an airfoil. The laminar boundary layer might separate further downstream due to the adverse pressure gradient. The disturbances within the boundary layer, which are known to exist prior to the separation point, are then amplified due to the Kelvin-Helmholtz instabilities within the separated shear layer and this generally serves as the mechanism to trigger the transition to turbulence [15]. The separated shear layer might reattach due to the higher momentum in the turbulent part and this could lead to the formation of a closed LSB.
Table 1. Geometrical and operational characteristics of the turbine and the suction settings.

| Type                                      | Darrieus H-type |
|-------------------------------------------|-----------------|
| Number of blades, \( n \)                 | 1               |
| Diameter, \( d \) [m]                     | 1               |
| Height, \( h \) [m]                       | 1               |
| Swept area, \( A \) [m\(^2\)]            | 1               |
| Solidity, \( \sigma \)                    | 0.06            |
| Airfoil                                   | NACA0018        |
| Airfoil chord, \( c \) [m]                | 0.06            |
| Blade aspect ratio, \( h/c \)             | 16.67           |
| Location of blade-spoke connection         | Half-chord, \( c/2 \) |
| Rotation direction                         | Counter-clockwise|

**Operating conditions**

| Freestream velocity, \( U_\infty \) [m/s] | 9.3            |
| Turbine rotational velocity, \( \Omega \) [rad/s] | 46.5 – 55.8 |
| Tip speed ratio, \( \lambda \)              | 2.5 – 3.0      |
| Chord-based Reynolds number, \( Re_c \)     | \( 1.03 \times 10^5 – 1.21 \times 10^5 \) |
| Approach-flow total turbulence intensity, \( TI \) [%] | 5             |

**Suction characteristics**

| Start location, \( X_{s1} / c \) [%] | 7.33, 12.33, 17.33, 27.33 |
| Center location, \( X_{c} / c \) [%] | 8.50, 13.50, 18.50, 28.50 |
| End location, \( X_{s2} / c \) [%] | 9.83, 14.83, 19.83, 29.83 |
| Width, \( W / c \) [%]               | 2.5              |
| Suction velocity, \( V_s \) [m/s] (inward) | 0.0465, 0.093, 0.2325, 0.465, 0.93 |
| Amplitude, \( A_s = V_s / U_\infty \) [%] | 0.5, 1.0, 2.5, 5.0, 10.0 |
| Momentum coefficient, \( c_m = \frac{2V_s}{c} \left( \frac{V_s}{U_\infty} \right)^2 \) [%] | 0.000125, 0.005, 0.003125, 0.0125, 0.05 |

Incompressible 2D URANS simulations are performed using the commercial CFD software package ANSYS Fluent v2020R1. The computational domain size is \( 35d \times 20d \) (d: turbine diameter), where the turbine center is 10d from the domain inlet and the sides. A hybrid computational grid consisting of 366,976 quadrilateral cells with \( y^+_{\text{max}} < 1.0 \) is employed. The number of cells along the blade circumference is 1,100. This is excluding 150 cells along the suction slot. The grid is based on a grid-sensitivity analysis presented in Ref. [7]. Turbulence is modeled using the four-equation transition SST turbulence model, also known as \( \gamma \)-\( \text{Re}_\theta \) [16]. The choice of the turbulence model is based on a critical analysis of the predictions of seven commonly-used Reynolds-averaged turbulence models for flow around VAWTs against three different experimental data sets [9], also verified by comparison against the more advanced scale-resolving simulations [13, 14]. It is also validated for turbulent boundary layer along a flat plate with slot suction [7]. Second-order spatial and temporal discretization and SIMPLE pressure-velocity coupling scheme are employed. The azimuthal increment \( \delta \) is 0.1° with 20 iterations per time step. Twenty turbine revolutions are performed to reach statistical convergence. The computational settings are according to the best-practice guidelines for CFD simulations of VAWTs (e.g. [9, 17, 18]). In addition, a time-step sensitivity analysis for the VAWT with suction is also presented in Ref. [7].

Details of the solution verification analysis, the three sets of the validation studies for the VAWTs and the validation study for slot suction over a flat plate with turbulent boundary layer are presented in Refs. [7, 9], and for brevity are not shown here.

3. Results

3.1. Steady suction (during the whole turbine revolution)

Fig. 1 shows the contour maps of the turbine power gain, in terms of \( \Delta C_T \), in ‘suction amplitude – suction location’ space for steady suction being applied during the whole turbine
revolutions. The maps help to identify the optimal suction characteristics as a function of tip speed ratio. Within the studied range, the following observations are made:

- In general, the power gain due to suction remarkably increases with decreasing $\lambda$. For example, with a suction amplitude of 0.5% and $X_s/c = 8.5\%$, the turbine power coefficient ($C_P$) at $\lambda = 2.5$ increases from 0.045 to 0.156, corresponding to a power gain of 246%. At $\lambda = 3.0$, $C_P$ value increases from 0.137 to 0.251, which corresponds to a power gain of 83%. The more prominent influence of suction at lower $\lambda$ is because at lower $\lambda$ the turbine aerodynamic performance is highly dominated by flow separation, thus, its suppression will have a strong impact on the overall power performance.

- For $\lambda = 2.5$ and 3.0, the optimal suction location is at $8.5\%c$, where $c$ is the blade chord length.

- The optimal range of suction location, which can be inferred from the red regions in contour plots in Fig. 1, becomes wider as $\lambda$ increases. This is because of the increase in the chordwise extent of the laminar separation bubble (LSB) for higher $\lambda$, see [7]. Note that based on the approach-flow turbulence level and Re, a laminar boundary layer could form along the leading edge of an airfoil. The laminar boundary layer might separate further downstream due to the adverse pressure gradient. The disturbances within the boundary layer, which are known to exist prior to the separation point, are then amplified due to the Kelvin-Helmholtz instabilities within the separated shear layer and this generally serve as the mechanism to trigger the transition to turbulence [15]. The separated shear layer might reattach due to the higher momentum in the turbulent part and this could lead to formation of a closed LSB.

- The optimal suction location is independent of the suction amplitude.

Applying suction at $X_s/c = 8.5\%$ is found to be the most effective for the studied tip speed ratios. Therefore, it can be concluded that for the studied turbine at the given operating condition, i.e. $T_l = 5\%$ and $Re_c \approx 10^5$, a single leading-edge suction slot location can be selected to effectively suppress the flow separation at different tip speed ratios and to enhance the VAWT power performance. The sensitivity of the optimal suction location and amplitude to tip speed ratio can be considered insignificant. In this regard, for typical constant-speed VAWTs, whose tip speed ratio inevitably changes with varying freestream velocity, the location of the suction slot can be configured near the blade leading edge to align the LSB location in order to effectively suppress the flow separation at low tip speed ratios. Table 2 presents the optimal suction location as a function of tip speed ratio for two different suction amplitudes.
Table 2. Optimal suction location and the respective turbine power gain as a function of tip speed ratio. The optimal values at each \( \lambda \) are based on 20 URANS simulations. (TI = 5\%, Re_c \approx 10^5)

| Parameter                | \( \lambda = 2.5 \) | \( \lambda = 3.0 \) |
|--------------------------|----------------------|----------------------|
| Optimal \( X_s/c \) [%] | 8.5                  | 8.5                  |
| \( \Delta C_P \) [%] at \( \Lambda_s = 0.5\% \) | 246                  | 83                   |
| \( \Delta C_P \) [%] at \( \Lambda_s = 10.0\% \) | 259                  | 85                   |

Fig. 2 shows the turbine instantaneous moment coefficient, \( C_m \), during the turbine half-revolution for different tip speed ratios with suction off and on \((X_s/c = 8.5\%\) and \( \Lambda_s = 0.5\%\)). It can be seen that by applying the suction, at \( \lambda = 2.5 \) where the turbine experiences a deep dynamic stall, the turbine \( C_m \) is unchanged for \( \theta \) up to \( \approx 50^\circ \). For \( \theta > 50^\circ \), the turbine \( C_m \) is substantially improved due to the suction and consequently, the turbine \( C_P \) is significantly enhanced by 246.6\%. The peak in \( C_m \) is also shifted from \( \theta = 68^\circ \) to \( 91^\circ \) due to the suction. This shift is because of the delayed stall on the blade. The sudden reduction in \( C_m \) (at \( 68^\circ < \theta < 103^\circ \)) and the subsequent fluctuations (at \( 103^\circ < \theta < 145^\circ \)), which occur for the turbine with suction off due to the dynamic stall on the blade, are visibly prevented by suction. At \( \lambda = 3.0 \), where the turbine goes into a comparatively light dynamic stall, the trend of the change in \( C_m \) due to the suction is similar to \( \lambda = 2.5 \), however, the turbine power gain is comparatively less. The turbine \( C_P \) values at \( \lambda = 3.0 \) are increased by 83.2\% due to suction.

\[ V_{tan,n} = \frac{(u \cos(\theta) + v \sin(\theta))}{U_\infty} \]

Fig. 3 illustrates contour plots of the instantaneous dimensionless tangential velocity \( V_{tan,n} \), defined using Eq. 1 which is based on the fixed coordinate system denoted by \( X'' \) and \( Y'' \) in the schematic shown on top of Fig. 3, with superimposed streamlines at different azimuthal positions for \( \lambda = 2.5 \). Note that similar observations, but to a lesser magnitude, are also observed for the other tip speed ratio of 3.0, which for brevity is not shown here. A schematic of the coordinate system and the presented azimuthal positions is shown on top of the figure. The figure is presented to highlight the reverse flow regions along the blade surface with suction off and on. It can also be seen that the unsteady separation and the dynamic stall phenomenon on the blade occur for \( \theta > 50^\circ \), in line with the trend observed for \( C_m \) values in Fig. 2.
3.2. Steady suction (only during the dynamic stall)

Further investigation is performed to compare two cases: (i) the suction is applied during the whole turbine revolution, which is the case in Fig. 1-3; (ii) the suction is applied only during the azimuthal angles during which dynamic stall occurs, $50^\circ < \theta < 180^\circ$. Fig. 4 shows the instantaneous moment coefficient during the turbine half-revolution for both cases. The comparison shows a negligible difference. This reveals that applying the leading-edge suction is only beneficial for the turbine power performance when the unsteady separation and the dynamic stall is occurring. This finding is of interest because this way the energy consumption of the
suction system will be almost 36% in comparison to applying the suction during the whole turbine revolution.

![Figure 4](image_url)

Figure 4. Instantaneous moment coefficient during the turbine half-revolution for the baseline (suc. off), steady suction during the whole turbine revolution (0°-360°) and only during the dynamic stall process (50°-180°) at tip speed ratios of 2.5 and 3.0.

3.3. Unsteady suction

In order to further reduce the power consumption of the suction system, an analysis is performed to compare the results of the steady suction versus the unsteady suction for separation control on VAWTs in dynamic stall. In the latter, the suction velocity is a time-varying sinusoidal function with the same amplitude as the steady suction ($A_s = 0.5\%$), see Eq. 2. The frequency of the sinusoidal oscillation, $f$, given in Table 3, and yields the dimensionless frequencies, $f^+$, of 1, 2, 5 and 10. The $f^+$ is defined using Eq. 3. Note that the timestep for the unsteady suction is based on a sensitivity analysis using a timestep two times finer, where negligible difference between the two cases are observed.

$$V_s(t) = -|U_s A_s \sin(ft)|$$  \hspace{1cm} (2)

$$f^+ = \frac{2\pi f}{360\Omega}$$  \hspace{1cm} (3)

| $\lambda$ | 2.5 |
|----------|-----|
| $\Omega$ [rad/s] | 46.5 |
| $f^+$ | 1 | 2 | 5 | 10 |
| $f$ [Hz] | 2664 | 5328 | 13321 | 26642 |

| $\lambda$ | 3.0 |
|----------|-----|
| $\Omega$ [rad/s] | 55.8 |
| $f^+$ | 1 | 2 | 5 | 10 |
| $f$ [Hz] | 3197 | 6394 | 15985 | 31971 |

Fig. 5 shows the normalized suction velocity, $V_s^+$, defined using Eq. 4, versus dimensionless time, $t^+$ defined using Eq. 5, for different $f^+$ values of 1, 2, 5 and 10. Note that $d\theta$ and $dt$ are azimuthal increment and time step, respectively.

$$V_s^+ = \frac{|V_s|}{A_s}$$  \hspace{1cm} (4)
\[ t^* = \frac{td\theta}{2\pi dt} \]  

Figure 5. Normalized suction velocity magnitude versus dimensionless time for different dimensionless frequencies \( f^* \) values of 1, 2, 5 and 10.

Fig. 6 shows the instantaneous moment coefficient for the baseline case (suc. off), steady suction and unsteady suction with different \( f^* \) values of 1, 2, 5 and 10. The comparison reveals that the influence of the oscillation frequency of the suction velocity on the turbine power performance is insignificant. The fact that the unsteady suction reproduces the power gain of the steady suction for the turbine in dynamic stall is important because using the unsteady suction, the power consumption of the suction system can be reduced to almost 65% of the steady suction. The value is calculated based on the flow rate needed for a given suction slot for steady and unsteady suction systems.

Figure 6. Instantaneous moment coefficient during the turbine half-revolution for the baseline (suc. off), steady suction (\( f^* = 0 \)) and unsteady suction with different frequencies (\( f^* = 1, 2, 5 \) and 10).

4. Discussion
Note that the present CFD modeling of a VAWT with suction is validated using two separate sets of validations: (i) VAWT without suction; (ii) slot suction along a flat plate with turbulent boundary layer. Ideally, the validation study needs to be performed for a VAWT with suction, however, no such an experiment is found in the literature and this has been a limitation of this study.

The aerodynamic forces are calculated by integration of the pressure and viscous forces along the airfoil walls, excluding the suction slot, which is defined as a velocity inlet. Therefore, the calculated forces are not influenced by the momentum of the sucked air along the suction slot. The release of the sucked air is also not considered in the modeling.

In the present study, the simulations are performed in 2D, assuming an infinite blade span with a suction slot along the whole blade span. In practice, the blade span has a finite length and the suction slot is typically along a fraction of the span. Therefore, any possible three-dimensional effects are not taken into account. In addition, other practical aspects of a suction system, such as the release of the sucked air, the losses through the suction ducts, the pump efficiency, and a non-uniform velocity profile along the suction slot are not considered. Future studies are recommended to estimate the impact of the aforementioned points on the CFD predicted turbine power gain due to the suction in order to provide a more realistic estimation.

Including a suction system on the blades of a wind turbine might impose several practical concerns regarding the structural strength of the blade surface near the suction location, the space needed for installation of the pumping system and the connecting tubes, the location to exhaust the pumped air and the input power for the pumping system. Note that the sucked air can be exhausted via the blade/shaft tips on the top and bottom, which will negligibly influence the turbine structural and aerodynamic performance.

In order to estimate the required input power for the suction system, a potential pumping system is considered (see Table 4). Note that the pressure losses in the tubing are not considered. For the reference operating condition, see Table 1, an estimation of the new power gain due to the steady suction at \( \lambda = 2.5 \) and 3.0 is 219.6\% and 74.3\%, respectively. The net power gain due to the suction increases to 240.1\% and 81.2\% for the unsteady suction applied only during the azimuthal angles where dynamic stall occurs. Due to the low volume flow rate needed for the vacuum pump, the turbine net power gain by the suction is still highly significant and, thus, well justifies its employment.

| Parameter                                      | Value   | Unit  |
|------------------------------------------------|---------|-------|
| Width, \( W \)                               | 0.0015  | m     |
| Suction velocity, \( V_w \)                  | 0.0465  | m/s   |
| Volume flow rate per blade with unit span     | 4.185   | l/min |
| Power consumption of an off-the-shelf vacuum pump (flow rate 5 l/min, vacuum deeper than 80 kPa, voltage 12 V, e.g. Ref. [19]) | 4 – 6   | W     |

Table 5. Estimation of the net power gain due to suction.

| \( \lambda \) | \( C_P \) | \( \Delta C_P \) [%] | \( P \) [W] | Pump input power | Net power gain | Net power gain [%] |
|---------------|---------|---------------------|------------|-----------------|---------------|-------------------|
|               | suc. off| suc. on             |            | su. off         | su. on        |                   |
| 2.5           | 0.045   | 0.156               | 246.6      | 22.2            | 76.9          | 6                 | 70.9             | 219.6            |
| 3.0           | 0.137   | 0.251               | 83.2       | 67.5            | 123.7         | 6                 | 117.7            | 74.3             |

5. Conclusions

The main conclusions are as follows:
- Applying suction through a slot near the blade leading edge, along the chordwise extent of the laminar separation bubble (LSB) and upstream of its bursting location, can effectively suppress the dynamic stall on the turbine blade.
- The most promising suction location is the most upstream position studies, i.e. $X_S/c = 8.5\%$.
- Applying suction with an amplitude of $A_S = 0.5\%$, near the blade leading edge at $X_S/c = 8.5\%$, at the reference conditions, $Re_c = 10^5$ and $T_l = 5\%$, results in a power gain of $\Delta C_p = 247\%$ and $83\%$ at different tip speed ratios of $\lambda = 2.5$ and $3.0$, respectively.
- Applying the leading-edge suction is only beneficial at the azimuthal angles where the dynamic stall occurs.
- Unsteady suction can result in similar power gain for the turbine in dynamic stall while the power consumption of the suction system can be reduced to $65\%$.
- The influence of frequency of the time-varying sinusoidal function for the unsteady suction is insignificant on the turbine power gain.

Acknowledgements

The first author is currently a postdoctoral fellow of the Research Foundation – Flanders (FWO) and is grateful for its financial support (project FWO 12ZP520N). The authors acknowledge the partnership with ANSYS CFD. This work was carried out on the Dutch national e-infrastructure with the support of SURF Cooperative.

References

[1] M. Gad-el-Hak and D. M. Bushnell, "Separation control: Review," Journal of Fluids Engineering, vol. 113, no. 1, pp. 5-30, 1991.
[2] M. Raiola, S. Discetti, A. Ianiro, F. Samara, F. Avallone, and D. Ragni, "Smart rotors: Dynamic-stall load control by means of an actuated flap," AIAA Journal, pp. 1-14, 2018.
[3] D. Greenblatt and R. Lautman, "Inboard/outboard plasma actuation on a vertical-axis wind turbine," Renewable Energy, vol. 83, pp. 1147-1156, 2015.
[4] A. Shmilovich and Y. Yadlin, "Flow control techniques for transport aircraft," AIAA Journal, vol. 49, no. 3, pp. 489-502, 2011.
[5] T. K. Barlas and G. A. M. van Kuik, "Review of state of the art in smart rotor control research for wind turbines," Progress in Aerospace Sciences, vol. 46, no. 1, pp. 1-27, 2010.
[6] S. J. Johnson, J. P. Baker, C. P. van Dam, and D. Berg, "An overview of active load control techniques for wind turbines with an emphasis on microtabs," Wind Energy, vol. 13, no. 2-3, pp. 239-253, 2010.
[7] A. Rezaeiha, H. Montazeri, and B. Blocken, "Active flow control for power enhancement of vertical axis wind turbines: leading-edge slot suction," Energy, vol. 189C, p. 116131, 2019.
[8] G. Tescione, D. Ragni, C. He, C. Ferreira, and G. J. W. van Bussel, "Near wake flow analysis of a vertical axis wind turbine by stereoscopic particle image velocimetry," Renewable Energy, vol. 70, pp. 47-61, 2014.
[9] A. Rezaeiha, H. Montazeri, and B. Blocken, "On the accuracy of turbulence models for CFD simulations of vertical axis wind turbines," Energy, vol. 180, no. C, pp. 838-857, 2019.
[10] A. Rezaeiha, I. Kalkman, H. Montazeri, and B. Blocken, "Effect of the shaft on the aerodynamic performance of urban vertical axis wind turbines," Energy Conversion and Management, vol. 149, no. C, pp. 616-630, 2017.
[11] A. Rezaeiha, H. Montazeri, and B. Blocken, "Towards optimal aerodynamic design of vertical axis wind turbines: Impact of solidity and number of blades," Energy, vol. 165, no. B, pp. 1129-1148, 2018.
[12] L. Huang, P. G. Huang, R. P. LeBeau, and T. Hauser, "Numerical study of blowing and suction control mechanism on NACA0012 airfoil," *Journal of Aircraft*, vol. 41, no. 1, 2004.

[13] A. Rezaeiha, H. Montazeri, and B. Blocken, "Scale-Adaptive Simulation (SAS) of dynamic stall on a wind turbine," in *Notes on Numerical Fluid Mechanics and Multidisciplinary Design - Progress in Hybrid RANS-LES Modelling (HRLM 2018)*, vol. 143, 1 ed.: Springer International Publishing, 2020, pp. 323-333.

[14] A. Rezaeiha, H. Montazeri, and B. Blocken, "CFD analysis of dynamic stall on vertical axis wind turbines using scale-adaptive simulation (SAS): Comparison against URANS and hybrid RANS/LES," *Energy Conversion and Management*, vol. 196, no. C, pp. 1282-1298, 2019.

[15] T. Michelis, S. Yarusevych, and M. Kotsonis, "On the origin of spanwise vortex deformations in laminar separation bubbles," *Journal of Fluid Mechanics*, vol. 841, pp. 81-108, 2018.

[16] F. R. Menter, R. B. Langtry, S. R. Likki, Y. B. Suzen, P. G. Huang, and S. Völker, "A correlation-based transition model using local variables—part I: model formulation," *Journal of Turbomachinery*, vol. 128, no. 3, pp. 413-422, 2006.

[17] A. Rezaeiha, H. Montazeri, and B. Blocken, "Towards accurate CFD simulations of vertical axis wind turbines at different tip speed ratios and solidities: Guidelines for azimuthal increment, domain size and convergence," *Energy Conversion and Management*, vol. 156, no. C, pp. 301-316, 2018.

[18] A. Rezaeiha, I. Kalkman, and B. Blocken, "CFD simulation of a vertical axis wind turbine operating at a moderate tip speed ratio: guidelines for minimum domain size and azimuthal increment," *Renewable Energy*, vol. 107, pp. 373-385, 2017.

[19] "KNF micro- and mini-pumps for air and gas: https://www.knfusa.com/en/micro-air/" (accessed.