Numerical and experimental studies of a small vertical-axis wind turbine with variable-pitch straight blades

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Abstract
This paper presents a numerical simulation and experiment on the effect of the variable pitch angle on the performance of a small vertical-axis wind turbine (VAWT) with straight blades. The power coefficient of the VAWT was measured in an open-circuit wind tunnel. By conducting two-dimensional unsteady computational fluid dynamics simulations using the RNG k-ε, Realizable k-ε, and SST k-ω models, the power and torque of the VAWT and the flow around the straight blades were also analyzed. The numerical simulation of the power performance results were validated using wind tunnel experimental data. The results of both the numerical simulations and experiments showed that a VAWT with variable-pitch blades has better performance than a VAWT with fixed-pitch blades. The numerical simulation of the performance using the RNG k-ε turbulence model had good qualitative agreement with the experimental results. The numerical simulation was able to capture the flow separation on a blade, and it was shown that a variable-pitch blade can suppress the flow separation on its blades at a tip speed ratio lower than that of fixed-pitch blades.

Key words: Vertical axis wind turbine, Variable pitch, Numerical simulation, Wind tunnel test, Power coefficient, Turbulence model, Dynamic stall

1. Introduction

Wind energy is one of the most promising renewable energy resources. Wind power generation does not cause the emission of carbon dioxide and pollution, and thus the capacity of wind power generation has increased rapidly in recent years. Global warming and climate change will have a decisive impact on the mid-term and long-term prospects of wind power, which has increased awareness of potential contributions of wind energy, including economic, social, and ecological sustainability, to the total energy supply (World Wind Energy Association, 2011).

There are two types of wind turbines: horizontal axis wind turbines (HAWTs), such as a propeller-type wind turbine, and vertical axis wind turbines (VAWTs), such as the Darrieus wind turbine. The power coefficient of HAWTs is generally larger than that of VAWTs, since the angle of attack of a VAWT blade changes significantly in one revolution. A majority of the large wind turbines that are used for power generation are HAWTs. However, VAWTs have many advantages, such as being omni-directional without needing a yaw control system, having better aesthetics for their integration into buildings, having more efficiency in turbulent environments, and having lower sound emissions (Mertens, 2006; Mewburn-Crook, 1990); therefore, VAWTs are expected to be used in urban areas.

Numerical and experimental studies of VAWTs have been carried out by several researchers, including Castelli et al. (2011), Takao et al. (2008), Howell et al. (2010), Sato et al. (2011) and Chong et al. (2013). Due to the improving performance of VAWTs with straight fixed-pitch blades, Tanaka et al. (2011) have studied the effects of blade profiles (NACA0018, NACA4418 with its camber facing outward, and NACA4418 with its camber facing inward) and setting angle on the performance of a small straight-bladed VAWT. They indicated that the starting performance of the VAWT...
with the NACA0018 can be further improved by setting the wings at an outward angle of 5 degrees. Yamada et al. (2011) have also studied the effects of the camber and thickness of a blade (NACA0020, 3520, 6518, 6520, 6525, 6530, and 8520) on the performance of a small straight-bladed VAWT. Their report showed the mean and temporal torque variation at any azimuth angle of one and two blades.

As part of a numerical simulation, Chen and Kuo (2013) have studied the effects of pitch angle and blade camber on the unsteady flow characteristics and the performance of a small-size Darrieus wind turbine with the NACA0012, 2412, and 4412 blade profiles. Their results indicated that the self-starting ability and the moment coefficients of a VAWT that has blades with large cambers (NACA4412) are better than those with the other blades (NACA0012 and 2412). Chen and Zhou (2009) used the SST $k$-$\omega$ model to investigate the aerodynamic performance of a VAWT via a two-dimensional numerical simulation. The results showed the optimum pitch angle for the power coefficient of the VAWT. Aresti et al. (2013) conducted two- and three-dimensional numerical simulations of the performance and flow through a small scale H-type Darrieus wind turbine by using the RNG $k$-$\varepsilon$, standard $k$-$\varepsilon$, and standard $k$-$\omega$ turbulence models. The self-starting capabilities of the VAWT were found to increase with the increasing mount angle of attack of the blades. Almohammadi et al. (2013) investigated the mesh independence of the predicted power coefficient of a VAWT with straight blades by employing a two-dimensional numerical simulation. Roh and Kang (2013) investigated the effects of a blade profile, the Reynolds number, and the solidity on the performance of a straight-bladed VAWT by using the numerical procedure of the multiple stream tube method. McNaughton et al. (2014) presented a two-dimensional numerical investigation of a VAWT, which had a high solidity of 1.1, and investigated the effects of the original and a modified version of the SST models for low Reynolds numbers on the flow structure, dynamic stall, and blade-vortex interaction.

Due to the phenomenon called a dynamic stall, which is a major component of the unsteady aerodynamics of a Darrieus wind turbine with a low tip speed ratio (TSR) (Nobile et al., 2011; Parashivouli, 2002), an aerodynamic analysis of a VAWT is complex. The dynamic stall of a blade can produce a lift force and pitching moment values that exceed static values. The dynamic stall process depends on both the amplitude and the history of angles of attack of the airfoil (Spera, 2009). Due to the formation of vortices, which reduce the static pressure on the suction side of the rotor blade, the presence of a dynamic stall at low TSRs can have a positive impact on power generation; however, the formation of vortices can have negative consequences such as vibration, noise, and a reduction in the fatigue life of the components of the VAWT (Fujisawa and Shibuya, 2001) during operation. A VAWT with straight variable-pitch blades can change the amplitude and the rate of increase of the angle of attack in one revolution, compared to a VAWT with straight fixed-pitch blades. Simão Ferreira et al. (2007, 2009, and 2010) investigated the dynamic stall of a two-dimensional single-bladed VAWT via PIV and evaluated the differences between the commonly used turbulence models. Zhang et al. (2013) predicted the aerodynamic performance and the flow field of a straight-bladed VAWT. The power coefficient of the two-dimensional computational fluid dynamics (CFD) agreed well with the experimental data, and it was demonstrated that the RNG $k$-$\varepsilon$ turbulent model is a useful resource to predict the tendency of aerodynamic forces; however, the model requires a high estimate value of the turbulence viscosity coefficient.

The development of VAWTs with variable-pitch straight blades has been carried out to obtain a high starting torque and high efficiency during operating speeds. A few of these VAWTs include the ASI/PINSON wind turbine (Noll and Zvara, 1981), a Giromill wind turbine (Anderson, 1981), a straight-bladed cycloturbine (Nattuvetty and Gunkel, 1982), a VAWT with a self-acting variable pitch system (Pawsey, 2002), and the Orthoptere wind turbine (Shimizu et al., 1997). Kiwata et al. (2010) and Yamada et al. (2012) developed a VAWT with variable-pitch straight blades that have a slight camber using a four-bar linkage without actuators. It was found that the power coefficient of this VAWT is better than that with fixed-pitch blades. Furthermore, this VAWT can direct itself towards the wind, which enables it to utilize the rotational speed control in the turbine via its tail vanes.

The present paper describes the results of our numerical and experimental studies on the performance of a VAWT with variable-pitch straight blades, which are symmetrical airfoils (NACA0018). A two-dimensional numerical simulation of the flow around a VAWT was performed using the commercial CFD software ANSYS FLUENT 13.0. The aim of this paper is to clarify the reason, in terms of aerodynamics, for improving the performance of a straight-bladed VAWT that will be installed with a variable-pitch mechanism. The effects of the variable-pitch angle, TSR, and the turbulent models, i.e., the RNG $k$-$\varepsilon$ (Yakhot and Orszag, 1986), Realizable $k$-$\varepsilon$ (Shih et al., 1995), and SST $k$-$\omega$ (Menter, 1994) turbulence models, on the performance of the VAWT were investigated. In addition, a
numerical simulation of the unsteady flow around the blades was conducted for the VAWTs with fixed- and variable-pitch straight blades.

2. Methodology

2.1 Model geometry

A VAWT with variable-pitch straight blades utilizing a linkage mechanism was designed and built as shown in Fig.1 (Kiwata et al., 2010). The main geometrical features of this turbine are shown in Table 1. The wind turbine had a diameter of \( D (= 2R) = 800 \) mm and a height of \( h = 800 \) mm. The section of the VAWT that was analyzed was the symmetrical airfoil of its NACA 0018 blade, which had a maximum lift coefficient \( C_{L_{\text{max}}} \) at an angle of attack of \( \alpha \approx 15^\circ \) and a maximum lift-to-drag ratio \( C_{L/C_D} \) at an angle of attack of \( \alpha = 5.5^\circ \) (Sheldahl and Klimas, 1981). The rotor of the VAWT was composed of three blades with a chord length of \( c = 200 \) mm and a main shaft with a diameter of 0.06 m. The solidity was \( \sigma (= n_c/\pi D) = 0.239 \), which is defined as the ratio of the total blade area \( (n_c) \) to the circumference area \( (\pi D) \). Figure 2 shows a top view of the VAWT and its variable-pitch mechanism, which consists of a four-bar linkage with an adjustable eccentric link \( l_e \). This turbine had an eccentric rotational point \( O_e \) that was different from the main rotational point \( O \). Point \( P_1 \) is near the leading edge of the blade, and point \( P_2 \) is near the trailing edge of the blade. The angle between the main-link \( (l_m) \) and the \( x \)-axis is the azimuthal angle \( (\theta) \). This mechanism is able to vary the pitch angle \( \alpha_p \) according to the rotation of the main link without utilizing the actuators. The geometry of the pitch angle \( (\alpha_p) \) is the sum of the blade offset angle \( (\alpha_c) \) and the blade pitch angle amplitude \( (\alpha_w) \). The optimum blade offset pitch angle was chosen from the experimental results to be \( \alpha_c = 11.9^\circ \) (Kiwata et al., 2010). The blade pitch angle amplitude was changed from \( \alpha_w = \pm 0^\circ \) to \( \pm 15.0^\circ \) by increasing the length of the eccentric link from \( l_e = 0 \) m to 22 mm.

2.2 Experimental apparatus

Figure 3 shows a schematic diagram of the experimental apparatus. The experiments were carried out in a circuit wind tunnel with an open test-section that had a cross-sectional area of 1250 mm \( \times \) 1250 mm. The open test-section has the capability to allow the conditions inside the test section to be largely unaffected by larger blockage percentage models because of the ability to leak flow and expand the flow around objects within the test-section (Ross and Altman, 2011). The uniform flow in the analyzed area was measured by an ultrasonic anemometer (Kaijo Sonic, DA-650-3TH (TR-90AH)). The turbulence and non-uniform level variation in the exit of nozzle at a wind speed \( V_\infty = 8 \) m/s was less than 0.5% and \( \pm 1.0\% \), respectively. A geared motor (Mitsubishi Electric, GM-S) and a frequency inverter with aerodynamic breaking resistor (Hitachi Industrial Equipment Systems, SJ200) were used to drive the wind turbine. The torque \( T \) and rotation speed \( N \) of the wind turbine were measured in each case in order to calculate the power.
coefficient \( C_p = T \omega / 0.5 \rho D h V_\infty^3 \); \( \rho \), air density; \( \omega \), turbine angular velocity) by using a torque transducer (TEAC TQ-AR5N with a rate capacity of 5 N·m) and a digital tachometer (ONO SOKKI HT-5500) under a constant wind velocity of \( V_\infty = 8 \) m/s.

2.3 Numerical Simulation

2.3.1 Computational domain and meshing

The two-dimensional computational domain, boundary conditions, and the mesh structure are shown in Figs. 4 and 5. The computational domain had a radius of 24\( R \), and the inlet and outlet boundary conditions were placed upwind and downwind of the rotor, respectively, as shown in Fig. 4. The inlet of the computational domain corresponded to the uniform flow condition at \( V_\infty = 8 \) m/s with a turbulence intensity of 1.0\% and turbulence viscosity ratio of 5.0\%. The pressure outlet condition for \( \Delta p = 0 \) was specified at the downstream boundaries of the computational domain. The computational domain consisted of three mesh zones, i.e., one fixed sub-domain outside the rotor, one dynamic sub-domain around the rotor with a radius of 3\( R \), and three dynamic sub-domains around the blades with a diameter of 3\( c \) and an equal spacing of 120\(^\circ\), as shown in Fig. 5. The mesh interfaces consist of one mesh interface around the rotor and three around blades 1, 2, and 3. A dynamic mesh technique was used for this simulation. The dynamic meshes around the rotor and blades rotate according to the TSR, while the mesh on the exterior of the rotor remains stationary. The rotation of the four dynamic sub-domains can be controlled independently by a user-defined function (UDF), which was programmed in C++ code. The surfaces of the blades and main shaft were set as a wall, i.e., no-slip boundary conditions. The fixed sub-domain was meshed using quadrilateral grids, and the dynamic sub-domain around the rotor was meshed using triangular grids. The three dynamic sub-domains around the blades were meshed using a combination of triangular grids and quadrilateral grids, especially near the surfaces of the blades. In order to construct a high density mesh on the surfaces of the blades, the layers of the cells were generated by using an inflation tool that had a growth rate of 1.2. The distance of the first layer was 0.01mm, and \( y^+ \leq 1 \) was obtained. The fixed sub-domain on the exterior side of the rotor had 116,808 elements, the sub-domain around the rotor had 21,680 elements, and the three dynamic sub-domains around the blades had 66,389 elements. The total number of elements was about 3.37\( \times \)10\(^5\).

2.3.2 Turbulence model and solver setting

A two-dimensional numerical simulation of the VAWT was performed using FLUENT 13.0 software with the flow conditions corresponding to the experiment. The governing equations were the continuity equation and the unsteady Reynolds-averaged Navier-Stokes (URANS) equation, in which the Reynolds stresses were solved by using the three different turbulence models (the RNG k-\( \varepsilon \), Realizable k-\( \varepsilon \), and SST k-\( \omega \) models). The computational domain was discretized using the finite volume method. The convection terms of the governing equation were discretized using the second-order upwind discretization method. The implicit algorithm of the PISO method was applied for the pressure-velocity coupling. Table 2 shows a summary of the parameters employed in the six different cases where different time steps and TSRs were defined. The TSR (\( \lambda = R \omega / V_\infty \)) was varied from \( \lambda = 0.5 \) to 2.0. For all of the transient simulations, the total time was defined to allow enough time for the formation of flow to develop around the rotor. Consequently, the fluctuating torque coefficient became periodically stable when the turbine minimum operated more than three revolutions.

Table 1 Main geometrical features of the tested wind turbine

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine diameter, ( D )</td>
<td>800 mm</td>
</tr>
<tr>
<td>Blade span length, ( h )</td>
<td>800 mm</td>
</tr>
<tr>
<td>Blade chord length, ( c )</td>
<td>200 mm</td>
</tr>
<tr>
<td>Blade airfoil</td>
<td>NACA0018</td>
</tr>
<tr>
<td>Number of blades, ( n )</td>
<td>3</td>
</tr>
<tr>
<td>Aspect ratio, ( AR = h/c )</td>
<td>4</td>
</tr>
<tr>
<td>Turbine solidity, ( = nc/\pi D )</td>
<td>0.239</td>
</tr>
<tr>
<td>Main link length, ( l_m )</td>
<td>373 mm</td>
</tr>
<tr>
<td>Second link length, ( l_s )</td>
<td>365 mm</td>
</tr>
<tr>
<td>Blade link length, ( l_c )</td>
<td>85 mm</td>
</tr>
<tr>
<td>Eccentric link length, ( l_e )</td>
<td>0 mm, 15 mm, 22 mm</td>
</tr>
</tbody>
</table>

Fig. 3 Schematic diagram of the experimental apparatus
2.4 Numerical conditions
Three different rotor blade configurations were evaluated. Case 1 had a fixed-pitch blade with a pitch angle of $\alpha_w = 0^\circ$. Case 2 had a blade pitch angle of $\alpha_w = \pm10.2^\circ$; this blade pitch angle changed from $\alpha_p = 1.8^\circ$ to $\alpha_p = 22.1^\circ$ in one revolution. Case 3 had a variable-pitch blade with an angle of $\alpha_w = \pm15.0^\circ$; this blade pitch angle changed from $\alpha_p = -3.1^\circ$ to $\alpha_p = 26.9^\circ$ in one revolution. The blade offset pitch angle was chosen to be $\alpha_c = 11.9^\circ$ from the experimental data. The variation of pitch angle, $\alpha_p$, will be shown in next section.

3. Geometrical characteristics of variable-pitch blades

3.1 Variations of the angle of attack and pitch angle of a blade
Figure 6 shows the local velocity vectors and the transient aerodynamics forces exerting on the rotating blade. The local relative velocity vector of blade $W$ is dependent on both the local wind velocity vector $V_i (|V_i| = V_i)$ and the tangential velocity vector $V_t (|V_t| = R\omega)$, as shown in the following equations.

$$W = V_i + V_t$$  \hspace{1cm} (1)

The length of the local velocity vector (relative inflow velocity) $|W(\theta)|$ at an arbitrary azimuth angle $\theta$ is defined as

$$|W(\theta)| = |V_\infty|^2 \left( a^2 + \lambda^2 + 2a \cos(\theta - 90) \right)^{0.5}$$  \hspace{1cm} (2)

where $a (= V_i/V_\infty)$ is the ratio of the induced velocity to the free-stream velocity, and $\lambda (= R\omega/V_\infty)$ is the TSR. The variation of the relative velocity of the blade with the azimuth angle $(\theta)$ is shown in Fig. 7 with the assumption that $a = 1$. The relative velocity has a maximum value at $\theta = 90^\circ$ and a minimum value at $\theta = 270^\circ$.

![Fig. 4 Computational domain and boundary conditions](image1.png)

![Fig. 5 Mesh structure in the computational domain](image2.png)

<table>
<thead>
<tr>
<th>Tip speed ratio, $\lambda$</th>
<th>Turbine rotational speed, $N$ (min$^{-1}$)</th>
<th>Number of time steps for one cycle</th>
<th>Time step size (s)</th>
<th>Time for one cycle (s)</th>
<th>Time steps for total unsteady calculation (Minimum)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>95.5</td>
<td>720</td>
<td>$8.73 \times 10^{-4}$</td>
<td>0.628</td>
<td>2160</td>
</tr>
<tr>
<td>1.0</td>
<td>191.0</td>
<td></td>
<td>$4.36 \times 10^{-4}$</td>
<td>0.314</td>
<td></td>
</tr>
<tr>
<td>1.2</td>
<td>229.2</td>
<td></td>
<td>$3.67 \times 10^{-4}$</td>
<td>0.262</td>
<td></td>
</tr>
<tr>
<td>1.5</td>
<td>286.5</td>
<td></td>
<td>$2.91 \times 10^{-4}$</td>
<td>0.209</td>
<td></td>
</tr>
<tr>
<td>1.7</td>
<td>324.7</td>
<td></td>
<td>$2.57 \times 10^{-4}$</td>
<td>0.185</td>
<td></td>
</tr>
<tr>
<td>2.0</td>
<td>382.0</td>
<td></td>
<td>$2.18 \times 10^{-4}$</td>
<td>0.157</td>
<td></td>
</tr>
</tbody>
</table>
The torque coefficient of each blade was calculated by the moment of the blade, which is the difference between the tangential component of lift and the drag of the blade against the center point of the rotor. The tangential component of lift and drag are perpendicular to the radius of the rotor. The expression of the torque coefficient is given by

\[ C_t = C_L \sin (\alpha + \alpha_p) - C_D \cos (\alpha + \alpha_p) \]  

(4)

The blade pitch angle \( \alpha_p \) depends on the blade offset pitch angle \( \alpha_c \) and the blade pitch angle amplitude \( \alpha_w \). The geometrical angle of attack of the blade \( \alpha \) is defined as a function of the TSR \( \lambda \) and the azimuthal angle \( \theta \), as given by

\[ \alpha = \tan^{-1}[\cos \theta \cdot (\sin \theta + \lambda \alpha) \pm \alpha_p] \]  

(5)

In the upstream area (90° ≤ \( \theta \) ≤ 270°), the value of \( a \) is greater than in the downstream area (−90° ≤ \( \theta \) ≤ 90°).

Figure 8 shows the variations in the blade pitch angle \( \alpha_p \) and the geometrical angle of attack \( \alpha \) in a uniform flow at a constant TSR of \( \lambda = 1.5 \), on the assumption that \( a = 1 \). The blue and green dashed lines show the variations in the blade pitch angle amplitudes \( \alpha_w = \pm 10.2^\circ \) and \( \pm 15.0^\circ \) with the blade offset pitch angle of \( \alpha_c = 11.9^\circ \), respectively. The red solid line shows the variations in the geometrical angle of attack \( \alpha \) of \( \alpha_w = 0^\circ \). The amplitude of the geometrical angle of attack of the fixed-pitch blade is larger than that of the variable-pitch blade. If the variable-pitch mechanism is installed in a straight-bladed VAWT, the amplitude of the geometrical angle of attack of the variable-pitch blade will become small. The variable-pitch mechanism has potential to reduce the peak angles of attack of the blades, alleviating the time it spends in stalls. Figure 9 shows the geometrical angle of attack of three different TSRs for \( \alpha_w = \pm 10.2^\circ \). The amplitude of the geometrical angle of attack of the variable-pitch blade decreases as the TSR increases.

[Fig. 6 Schematic diagram of the angles and vectors of the velocity, drag, and lift coefficient of a rotating blade]

[Fig. 7 Variation in the relative inflow velocity for \( a = 1 \) and \( V_\infty = 8 \text{m/s} \)]

[Fig. 8 Variation in the blade pitch angle and geometrical angle of attack of a blade at \( \lambda = 1.5 \) for \( \alpha_c = 11.9^\circ \) and \( a = 1 \)]

[Fig. 9 Variation in the tip speed ratio and geometrical angle of attack of a blade for \( \alpha_w = \pm 10.2^\circ \) and \( a = 1 \)]
4. Results and Discussion

4.1 Evaluation of the turbulence model in numerical simulations

Three types of turbulence models, i.e., the RNG $k$-$\varepsilon$, Realizable $k$-$\varepsilon$ and SST $k$-$\omega$, were examined for the two-dimensional numerical simulation of flow and performance of a VAWT with variable-pitch blades. Figure 10 shows the prediction of the torque coefficient $C_{\text{trk}}(\theta)$ for one of the variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ at $\lambda = 1.5$ for one revolution. One of the advantages of using a numerical simulation is the ability to predict the generation of torque on one blade rotor despite the simultaneous operation of multiple rotors. The torque coefficient of one blade predicted by using the SST $k$-$\omega$ model is higher at any azimuthal angle than those of the RNG and Realizable $k$-$\varepsilon$ models. However, the torque coefficient curves pattern are similar for the three turbulence models. The torque coefficient of one blade predicted by using the RNG $k$-$\varepsilon$ model has maximum torque at $\theta = 200^\circ$ for the variable-pitch blade with $\alpha_w = \pm 10.2^\circ$ at $\lambda = 1.5$. A positive torque was generated except when $90^\circ < \theta < 135^\circ$ and $270^\circ < \theta < 360^\circ$. The torque has a high positive value in the upper range of $135^\circ < \theta < 270^\circ$, which is referred to as the high-torque zone (HTZ). The range and peak value of the positive torque coefficient in the HTZ depends on the relative velocity and amplitude of the angle of attack of the blade. Negative and low torques were generated in the downstream areas and when $90^\circ < \theta < 135^\circ$ even though the largest relative velocity of the rotating blade occurred at an azimuth angle of $\theta = 90^\circ$ due to both of low angle of attack and domination of drag. Therefore, to increase the performance of a straight-bladed VAWT, the angle of attack of the blades is one of the most important factors. The angle of attack can be controlled by varying the blade pitch angle $\alpha_w$ with the azimuthal angle $\theta$.

Figure 11 shows the power coefficients $C_p$ of the VAWT with variable-pitch blades with $\alpha_w = \pm 10.2^\circ$. The power coefficients of the SST $k$-$\omega$ and the Realizable $k$-$\varepsilon$ models were predicted to be higher than those of the RNG $k$-$\varepsilon$ model. The power coefficient of the RNG $k$-$\varepsilon$ model is comparatively close to the experimental value. Qin et al. (2011) reported that the predicted torque of the VAWT with fixed-pitch straight blades obtained by using the Realizable $k$-$\varepsilon$ model was much higher than that when using the RNG $k$-$\varepsilon$ model. The Realizable $k$-$\varepsilon$ model has a limitation to produce non-physical turbulent viscosities when the computational domain contains both rotating and stationary fluid zones (ANSYS Theory Guide, 2012). However, they found that the RNG model predicted the flow fields to be relatively more compact than the standard $k$-$\varepsilon$ model, but with strong vortices downstream of the trailing edge of the blade. Howell et al. (2010) also showed that the RNG $k$-$\varepsilon$ model is able to predict flow fields that include large flow separations more accurately than the standard $k$-$\varepsilon$ model. The aerodynamic performance prediction of a straight-bladed VAWT was investigated by Zhang et al. (2013). It was demonstrated that the RNG $k$-$\varepsilon$ model is able to predict the tendency of aerodynamic forces, but with a high estimate value of the turbulence viscosity coefficient. Almohammadi et al. (2013) showed that the predicted maximum power coefficient obtained using the SST $k$-$\varepsilon$ model was higher than that obtained using the RNG $k$-$\varepsilon$ model. Thus, although turbulence models have an influence on the resultant flow field and performance of the VAWT, the effect of blade pitch angle amplitude on the performance of a VAWT with variable-pitch blades was investigated in the present study using the RNG $k$-$\varepsilon$ model. Furthermore, a VAWT with limited-span blades was also investigated by Howell et al. (2010), and their results showed that the presence of vortices along the tips of the blades in their 3D simulation was responsible for producing an efficiency that was significantly different compared to the 2D simulation. The difference in the power coefficient between the 2D numerical simulation and the experiment in the present study is considered to be an effect of the blade tips, i.e., the small aspect ratio of the blade $[AR = h/c = 4]$ and the additional loss from the support arms.

4.2 Power and torque coefficients

4.2.1 Power coefficients of the VAWTs with variable- and fixed-pitch blades

The power coefficients $C_p$ of the VAWTs with variable- and fixed-pitch blades for the numerical simulation and experiment are shown in Figs. 12 and 13, respectively. These figures present the effects of the blade pitch angle amplitude $\alpha_w$ on the power coefficient of the wind turbine. The effects of the blade pitch angle amplitude qualitatively agree with the two-dimensional numerical simulation using the RNG $k$-$\varepsilon$ turbulence model and the experiment. The power coefficient of the VAWT with variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ is higher than that with fixed-pitch blades. The peak power coefficients of the variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ and the fixed-pitch blades occur at a higher
4.2.2 Effect of variable pitch angle on torque coefficients

The torque of a straight-bladed VAWT is generated by the moment of the rotating blades, which results from the tangential component of lift and drag forces on the airfoil. The generation of fluid forces on the blades depends on the relative inflow velocity and the angle of attack, i.e., the azimuthal angle position. The variation of the torque coefficients $C_{TBK}$ for one of the blades at $\lambda = 1.5$ with an azimuthal angle $\theta$ is shown in Fig. 14. The maximum torque for the variable-pitch blade with $\alpha_w = \pm 10.2^\circ$ was generated at $\lambda = 1.5$. The generated torque on a blade of a straight-bladed VAWT varies for one cycle. The variation of the torque coefficient is classified into two zones for one cycle: an upstream zone at $90 \leq \theta \leq 270^\circ$ and a downstream zone at $270^\circ \leq \theta \leq 360^\circ$ and $0^\circ \leq \theta \leq 90^\circ$. For the fixed-pitch blades ($\alpha_w = \pm 0^\circ$), a negative torque was generated at azimuthal angles of $0^\circ \leq \theta \leq 105^\circ$ and $240^\circ \leq \theta \leq 360^\circ$. The maximum positive torque value occurred at $\theta \approx 190^\circ$. On the other hand, for the variable-pitch blades ($\alpha_w = \pm 10.2^\circ$ and $\pm 15.0^\circ$), the azimuthal angle of the maximum torque increased slightly at $\theta \approx 200^\circ$ as compared to that of the fixed
blades. The difference in the azimuthal angle of the maximum torque between the fixed- and variable-pitch blades is related to the azimuth angle at the optimum angle of attack, which approaches an angle of attack of the maximum lift coefficient of about $\alpha = 15^\circ$ for the NACA0018 airfoil ($Re = 1.5 \times 10^5$). The maximum and minimum torques of the variable-pitch blades with $\alpha_w = \pm 15.0^\circ$ are smaller than those with $\alpha_w = \pm 10.2^\circ$. However, there is only a slight difference in the torque coefficients between $\alpha_w = \pm 10.2^\circ$ and $\pm 15^\circ$ at $55^\circ < \theta < 140^\circ$. Negative torque occurred for $90^\circ \leq \theta \leq 135^\circ$ for all cases. However, for the fixed-pitch blade, the negative torque decreased, as seen in Fig.14.

Figure 15 shows the velocity contours around the rotor at $\lambda = 1.5$. For the fixed-pitch blade with $\alpha_w = 0^\circ$, the low velocity area (blue color) was more widely distributed than that of the variable-pitch blades. Figures 16 shows the pressure coefficient contours around the rotor at $\lambda = 1.5$, which is defined $(p - p_\infty)/(0.5p_\infty V_a^2)$. The difference in the pressure coefficient between the suction and pressure sides on the fixed-pitch blade at $\theta = 100^\circ$ was bigger than that for the variable-pitch blade. Consequently, the negative torque of the fixed-pitch blade for $100^\circ < \theta < 140^\circ$ decreases more than that of the variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ and $\pm 15^\circ$, as shown in Fig. 14.

4.2.3 Effect of TSR on torque coefficients

The power and torque coefficients of the VAWT with variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ are shown in Fig. 17. The total power and torque coefficients are divided into two components in the upstream and downstream areas. The power coefficient in the upstream area (curve $C_{\mu u}$) increases as the TSR increase. However, the total power coefficient (curve $C_p$) increase until a TSR of $\lambda = 1.5$, then decreases as the power coefficient in the downstream area decreases (curve $C_{\mu d}$). The torque coefficient for $\lambda \leq 1.5$ is positive in both the upstream and downstream areas (curves $C_{\alpha u}$ and $C_{\alpha d}$). However, for $\lambda > 1.5$, the torque coefficient in the downstream area becomes negative, and the torque coefficient decreases with an increase in the TSR. The total torque coefficient (curve $C_t$) increases until a TSR of $\lambda = 1.0$, then decreases due to the decrement of the torque coefficient in both the upstream and downstream areas.

Figure 18 shows the effects of the TSR on the torque coefficient $C_{TBK}(\theta)$ for one cycle of the variable-pitch blades with $\alpha_w = \pm 10.2^\circ$. The torque coefficient fluctuations of one blade at four different TSRs are also presented. For a TSR of $\lambda = 0.5$, the maximum torque coefficient was generated at $\theta \approx 180^\circ$, and the torque coefficient is nearly positive at all azimuth angles (except $90^\circ < \theta < 125^\circ$). The maximum value and peak angle of the torque coefficient varies with the TSR (shown as the inside of circle $b$ in Fig.18). For a low TSR of $\lambda = 0.5$, the maximum value of the torque coefficient was higher than that for a high TSR and shifted to a large azimuth angle as the TSR increased. The azimuth angle for the maximum torque coefficient increases as the TSR increases. The shift of the azimuthal angle for the maximum torque coefficient is related to the angle of attack of the rotating blade, i.e., the azimuthal angle where the angle of attack becomes the maximum lift coefficient ($\alpha \approx 15^\circ$) or the minimum angle of attack (see Figs. 8 and 9).

Figures 19, 20, and 21 show the velocity vectors, the velocity contours, and pressure coefficient contours around variable-pitch blades for TSRs of $\lambda = 0.5$ and 2.0. Velocity vectors near blades at $\theta = 80^\circ$, 200°, and 320° for low and high tip speed ratio as shown in Fig. 19. The velocity vectors near the blade at $\theta = 200^\circ$ at low TSR ($\lambda = 0.5$) are larger than at high TSR ($\lambda = 2.0$) as seen in Fig.19 (a-2) and (b-2). In contrast, the velocity vectors near blade at $\theta = 80^\circ$ for a low TSR ($\lambda = 0.5$) are smaller than at high TSR ($\lambda = 2.0$) as seen in Fig. 19 (a-1) and (b-1). The separated flow and high velocity from the leading edge of blade caused low pressure coefficient, as shown in Figs. 20(a) and 21(a). Consequently, the maximum torque coefficient is generated at $\theta = 200^\circ$, as shown in Fig.18 (circle b). For the blade at $\theta = 320^\circ$, the flows from suction and pressure side of the blade are observed, because the rotating velocity of the blade is smaller than the velocity of the uniform flow (see Fig. 20(a)). The pressure decreases at the suction side of the blade, and a positive torque is generated near $\theta = 320^\circ$ for $\lambda = 0.5$ as shown in Fig.18 (circle c). Although the wake of the blades is generated in the upstream area for $\lambda = 0.5$, this wake has a smaller influence on the blades in the downstream area compared to that with $\lambda = 2.0$. For a high TSR at $\lambda = 2.0$, the blade in the upstream area creates a wider wake region (blue areas in Fig. 20(b)) in the downstream area compared to that with $\lambda = 0.5$. The wake from the blades has influence pressure coefficient on pressure and suction side. Therefore, the torque at $\lambda = 2.0$ is not generated in the downstream area (see Fig. 18). It was confirmed by Nobile et al. (2011) that the wakes from the blades in the upstream area develop in the downstream area and a mast interacts with the downstream blades.

A negative torque coefficient was generated at an azimuth angle of $90^\circ \leq \theta \leq 135^\circ$ for all TSRs. It can be seen in Fig. 21(b) that the difference in the pressure coefficient around a blade at $\theta = 130^\circ$ between $\lambda = 0.5$ and 2.0 becomes large. The lift force is generated at a low TSR of $\lambda = 0.5$ and it is reduced to a tangential force. In contrast, for a high...
TSR of $\lambda = 2.0$, there is a slight difference between the pressure coefficients on the suction and pressure sides of the blade due to the low angle of attack, and a negative torque coefficient was consequently generated.

![Graph showing torque coefficients $C_{TBK}(\theta)$ for one of the blades at $\lambda = 1.5$ (RNG $k-\varepsilon$ model)](image)

Fig. 14 Effects of blade pitch angle amplitude on torque coefficients $C_{TBK}(\theta)$ for one of the blades at $\lambda = 1.5$ (RNG $k-\varepsilon$ model)

![Velocity contours around the rotor at $\lambda = 1.5$ (RNG $k-\varepsilon$ model)](image)

Fig. 15 Velocity contours around the rotor at $\lambda = 1.5$ (RNG $k-\varepsilon$ model)

![Pressure coefficient contours around the rotor at $\lambda = 1.5$ (RNG $k-\varepsilon$ model)](image)

Fig. 16 Pressure coefficient contours around the rotor at $\lambda = 1.5$ (RNG $k-\varepsilon$ model)
Fig. 17 Total power coefficient, and the power coefficients in the upstream and downstream areas, $\alpha_w = \pm 10.2^\circ$ (RNG $k$-$\varepsilon$ model)

Fig. 18 Effects of the TSR on the torque coefficients $C_{TBR}(\theta)$ for one of the variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ (RNG $k$-$\varepsilon$ model)

Fig. 19 Velocity vectors around the VAWT with $\alpha_w = \pm 10.2^\circ$ at low and high tip speed ratios (RNG $k$-$\varepsilon$ model)

(a-1) $\theta = 80^\circ$, $200^\circ$, and $320^\circ$  
(a-2) $\theta = 200^\circ$  
(a) $\lambda = 0.5$

(b-1) $\theta = 80^\circ$, $200^\circ$, and $320^\circ$  
(b-2) $\theta = 200^\circ$  
(b) $\lambda = 2.0$

Fig. 20 Velocity contours around the VAWT with $\alpha_w = \pm 10.2^\circ$ (RNG $k$-$\varepsilon$ model; $\theta = 80^\circ$, $200^\circ$, and $320^\circ$)

(a) $\lambda = 0.5$  
(b) $\lambda = 2.0$

Fig. 21 Pressure coefficient contours around the VAWT with $\alpha_w = \pm 10.2^\circ$ (RNG $k$-$\varepsilon$ model)

(a-1) $\lambda = 0.5$  
(a-2) $\lambda = 2.0$  
(b-1) $\lambda = 0.5$  
(b-2) $\lambda = 2.0$  
(a) $\theta = 80^\circ$, $200^\circ$, and $320^\circ$  
(b) $\theta = 10^\circ$, $130^\circ$, and $250^\circ$
4.3 Flow around the rotor blade

Figures 22 and 23 show the vorticity contours around the VAWT with fixed- and variable-pitch blades with $\alpha_w = \pm 10.2^\circ$ at an operating TSR of $\lambda = 1.0$, 1.5, and 2.0 as obtained by using the RNG $k-\varepsilon$ model. For the low tip speed ratio of $\lambda = 1.0$, Fig. 22(a) shows the presence of vorticity on the fixed blade. As shown in Fig. 23(a), the variable pitch blade of VAWT can suppress the formation of vortices around blades as compared with the fixed- pitch one. The difference between the presence and absence of vortices at the same TSR for variable- and fixed-pitch blades is due to the amplitude and the rate of increase of the angle of attack. As shown in Figs. 22(b)(c) and 23(b)(c), the flow separation on the blade weakens progressively with increasing the tip speed ratio. The presence of a vortex on the suction side of a blade creates a low pressure, and consequently, the lift force increases (Larsen et al., 2007). There are four stages of a dynamic stall: initiation of a leading-edge separation, vortex build-up at the leading edge, detachment of the leading-edge vortex (LEV) and build-up of the trailing-edge vortex (TEV), and detachment of the TEV and breakdown of the LEV (Soerensen et al., 1999). The feature of the dynamic stall that distinguishes it from a static stall is the shedding of significant concentrated vorticity from the leading-edge region. This vortex disturbance subsequently sweeps over the airfoil surface causing a change in pressure and results in a significant increase of the airfoil lift and large nose-down pitching that exceeds static values (Paraschivoiu, 2002). Fujisawa and Shibuya (2001) showed via PIV that two pairs of counter-rotating vortices develop in the wake of a NACA0018 blade in a straight-bladed VAWT. Thus, a numerical simulation can predict the presence of vorticity behavior on the fixed blade at the low tip speed ratio. It is obvious that a VAWT with variable-pitch blades can significantly suppressed the leading and trailing-edge separation on its blades at a tip speed ratio lower than that of fixed-pitch blades.

Fig. 22 Vorticity contours around the VAWT with fixed-pitch blades ($\alpha_w = \pm 0^\circ$; RNG $k-\varepsilon$ model)

Fig. 23 Vorticity contours around the VAWT with variable-pitch blades ($\alpha_w = \pm 10.2^\circ$; RNG $k-\varepsilon$ model)
5. Conclusions

The effects of the variable pitch angle, TSR, and turbulence model on the performance of a VAWT and the unsteady flow around the blades were investigated using a two-dimensional numerical simulation. The numerical simulation of the power performance results were validated using wind tunnel experimental data. The following conclusions were drawn:

(1) The prediction of performance by numerical simulation using the RNG $k$-$\varepsilon$ turbulence model qualitatively agreed with the experiment. It was found that a VAWT with variable-pitch blades has better performance than a VAWT with fixed-pitch blades.

(2) The performance of a VAWT is influenced by the amplitude and the rate of increase of the angle of attack. Reducing the angle of attack improves the torque coefficient, especially in the upwind area and some areas downstream, where a positive torque is generated on the VAWT with variable-pitch blades.

(3) For a low TSR of $\lambda < 1.5$ in a VAWT with variable-pitch blades, positive power is generated in both the upstream and downstream areas. For a high TSR of $\lambda > 1.5$, the downstream power coefficient becomes negative, and the power coefficient decreases as the TSR increases.

(4) The RNG $k$-$\varepsilon$ turbulence model can capture the presence of a vortex on the blade rotor at low TSRs. A VAWT with variable-pitch blades can significantly suppress the leading and trailing-edge separation as compared to a VAWT with fixed-pitch blades.

Nomenclature

- $\alpha$: Ratio induced velocity to free-stream velocity
- $AR$: Aspect ratio of blade ($= h/c$)
- $C_{Li}$: Tangential lift coefficient
- $C_{Di}$: Tangential drag coefficient
- $C_p$: Turbine power coefficient ($= T\omega/\rho R \omega h V^3$)
- $C_{pu}$: Turbine power coefficient in the upstream area ($= \frac{\pi}{2} \int_0^{\pi/2} C_{TBRK} d\theta$)
- $C_{pd}$: Turbine power coefficient in the downstream area ($= \frac{\pi}{2} \int_{\pi/2}^{\pi} C_{TBRK} d\theta$)
- $C_{T}$: Turbine torque coefficient ($= T/l \rho h V^2$)
- $C_{ta}$: Turbine torque coefficient in the upstream area ($= \frac{\pi}{2} \int_0^{\pi/2} C_{TBRK} d\theta$)
- $C_{td}$: Turbine torque coefficient in the downstream area ($= \frac{\pi}{2} \int_{\pi/2}^{\pi} C_{TBRK} d\theta$)
- $C_{TBRK}$: Turbine torque coefficient of one blade
- $c$: Blade chord length
- $D$: Turbine diameter
- $h$: Blade span length
- $l_c$: Blade link length
- $l_e$: Eccentric link length
- $l_m$: Main link length
- $l_s$: Second link length
- $N$: Turbine rotational speed
- $n$: Number of blades
- $p$: Static pressure at a point
- $p_\infty$: Static pressure at free stream
- $R$: Turbine radius
- $RNG$: Renormalization group
- $SST$: Shear stress transport
- $T$: Turbine torque
- $T_c$: Turbine mechanical loss torque
- $V_\infty$: Wind speed
- $V_i$: Induced Velocity
\[ V_t \]: Tangential velocity (\([V_t] = R\omega\))

\[ W \]: Local relative velocity vector (\([W = V_t + V]\))

\[ \alpha \]: Geometrical angle of attack

\[ \alpha_0 \]: Blade offset pitch angle

\[ \alpha_p \]: Blade pitch angle

\[ \alpha_a \]: Blade pitch angle amplitude

\[ \theta \]: Azimuth angle (angle between the main-link and x-axis)

\[ \lambda \]: Tip speed ratio (\([= R\omega / V_{\infty}\])

\[ \rho \]: Air density

\[ \rho_{\infty} \]: Air density of free stream

\[ \sigma \]: Turbine solidity (\([= nc/2\pi R]\))

\[ \omega \]: Turbine angular velocity (\([= 2\pi N/60]\))

References


