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Original Paper

Performance and Flow Condition of Cross-Flow Wind Turbine with a Symmetrical Casing Having Side Boards

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Abstract

A cross-flow wind turbine has a high torque coefficient at a low tip speed ratio. Therefore, it is a good candidate for use as a self-starting turbine. Furthermore, it has low noise and excellent stability; therefore, it has attracted attention from the viewpoint of applications as a small wind turbine for an urban district. However, its maximum power coefficient is extremely low (10 %) as compared to that of other small wind turbines. In order to improve the performance and flow condition of the cross-flow rotor, the symmetrical casing with a nozzle and a diffuser are proposed and the experimental research with the symmetrical casing is conducted. The maximum power coefficient is obtained as C_{pmax} = 0.17 in the case with the casing and C_{pmax}=0.098 in the case without the casing. In the present study, the power characteristics of the cross-flow rotor and those of the symmetrical casing with the nozzle and diffuser are investigated. Then, the performance and internal flow patterns of the cross-flow wind turbine with the symmetrical casings are clarified. After that, the effect of the side boards set on the symmetrical casing is discussed on the basis of the analysis results.

Keywords: Cross-flow wind turbine, Power coefficient, Torque coefficient, Symmetrical casing, Numerical analysis

1. Introduction

With the increasing research and developments in the field of renewable energy, the wind-generated electricity has attracted considerable attention as a form of natural energy. A cross-flow wind turbine has a high torque coefficient at a low tip speed ratio, good self-starting, low noise, and high stability characteristics. Therefore, it has attracted attention as a small wind turbine for an urban district. However, it has a serious drawback in that its maximum power coefficient is extremely low (10%) as compared to that of other small wind turbines. The internal flow condition of the cross-flow wind turbine has not been clarified in detail, although systematic researches on the changes in design parameters related to the blade number and setting angle have been conducted[1,2]. On the other hand, researches on the use of a guide vane and a ring diffuser to improve the performance of the cross-flow turbine have been conducted[3,4]. In our previous research, we clarified that the power coefficient increased with the adoption of the casings, which was limited to wind in one direction and took into account the effect of air accumulation[5]. However, the wind in urban and coastal regions comprises prevailing winds in two directions, such as land breeze and sea breeze, that frequently occur in a specific period and under specific conditions. Vertical-axis wind turbines like the cross-flow wind turbine have the advantage of high stability against a turbulent flow, which means that these turbines are not affected by the wind direction. Therefore, in order to improve the turbine performance while making use of the good self-starting characteristics of the cross-flow wind turbine, the casing suitable for these prevailing winds have been designed and manufactured in this research. And the effect of the side boards set on the casing has been investigated by the experimental and numerical analysis.

In the present paper, in order to improve the performance of the cross-flow wind turbine, the symmetrical casing with the nozzle and diffuser is proposed to adapt to the prevailing winds in two directions. Then, the performance and internal flow characteristics of the cross-flow wind turbine with the casing are clarified. Furthermore, the influence of the side boards set on the symmetrical casing on the performance of the turbine is discussed, and the optimum position of the side boards that achieves a better performance of the cross-flow wind turbine is considered.

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2. Experimental Equipment and Measurement Method

Figure 1 shows a schematic representation of the experimental equipment. In the experiment, a wind tunnel with a square discharge of 500mm×500mm was used, and the main flow velocity was set to 20m/s in order to reduce the effect of the measurement error on the performance of the turbine. The intensity of turbulence at the outlet of the wind tunnel was 1.2%. A torque meter, a rotational speed pickup sensor, and a motor were assembled vertically with the same axis at a low position on a test wind turbine. The rotational speed and tip speed ratio could be changed using the rotational speed controller. The test wind turbine was set 600mm downstream of the wind tunnel discharge in order to take the blockage effect into account. The turbulent intensity at the position of the test wind turbine was lower than 1.2%, and the velocity distortion at the same position was lower than 3.8% in 90% of the area of the wind tunnel discharge. The experimental Reynolds number based on the blade chord length with a uniform incoming flow was approximately Re=$U_\infty l/\nu$=2.27×10^4.

A rotor used in the experiment and its primary dimensions are shown in Fig.2 and Table 1. A thin circular arc blade was used. The blade number was Z=24, outer diameter was D_1=150mm, inner diameter was D_2=120mm, and the inner/outer diameter ratio was D_2/D_1=0.80. The inlet angle was β_1=40º, and the outlet angle was β_2=80º. The rotor diameter was the same with the outer diameter of the rotor (D=D_1=150mm) and the rotor height was H=400mm. Figure 3 represents the symmetrical casing. The casing were set around the cross-flow rotor and insertion holes for a pitot tube measurements were made at the upper end wall of the rotor, 5mm outside of the rotor in circumferential positions at interval of 10º. The holes were used for measuring the pressure and the velocity at the rotor inlet and outlet. The circular arc shape was applied for the nozzle shape of the symmetrical casing in the previous research[6]. On the other hand, the nozzle shape of the symmetrical casing in this paper was designed based on the theory of the cross-flow hydroturbine[7]. In this casing, the decrease ratio of the nozzle throat width from the inlet of the nozzle to the outlet of the nozzle was kept constant to make the uniform flow condition at the inlet of the rotor. Further, the experiment using the side boards, as shown in Fig.3 was conducted. The side boards were used for suppressing the influence of the external flow on the outlet flow from the rotor and for increasing the incoming flow to the rotor. The side boards had a thickness of 15mm and width of 37mm. The set position of the side boards was defined as the distance s from s=0mm, as shown in Fig.3. The side board was set at five positions in the experiment: s=0mm which was named P1, where the influence by the external flow would be the least, s=37.5mm named as P2, s=75mm named as P3, s=112.5mm named as P4 and s=150mm named as P5, which was the maximum s position. In this experiment, the casing was inclined against the main flow as Θ=15° as shown in Fig.3. The flow distributions at the rotor inlet and outlet were measured using the one-hole pitot tube having an outer diameter of 3.0mm. This tube was used as the substitution of a three-holes pitot tube by rotating it by ±30°. The measurement was conducted at 1/3H height from the upper end wall of the rotor. The circumferential position θ for the measurement was defined as 0° at the inlet of the rotor, and the clock-wise direction was a plus.

3. Numerical Analysis Conditions

A two-dimensional numerical flow analysis was conducted using a commercial code (Fluent6.3) for the purpose of investigating the internal flow pattern that was hard to measure by the experiment. A large rectangular region(20D×15D) shown in Fig.4 was used for the numerical analysis in order to suppress the inlet and the outlet boundary conditions influence the results. A uniform flow of U_∞=20m/s at the inlet region and the constant pressure with a parallel flow at the outlet region were set as boundary conditions. Multiple reference frames were used for the rotating rotor region and the stationary casing region. The numerical analysis of the region except the rotor and the rotor region were conducted using an absolute reference frame and a relative reference frame, respectively, in which the rotational speed was taken into consideration. Information was exchanged between the absolute and relative reference frames after a frame transformation. Both the unstructured triangle mesh and the structured rectangle mesh were used for the complex shape, i.e., the rotor and region around the rotor respectively. The standard wall function and k-ε turbulence model were applied.

Table.1 Primary dimensions of cross-flow rotor

D_1 [mm]	D_2 [mm]	D_2/D_1	l [mm]	Z	β_1 [deg]	β_2 [deg]	β' [deg]
150	120	0.80	17.3	24	40	80	63.5

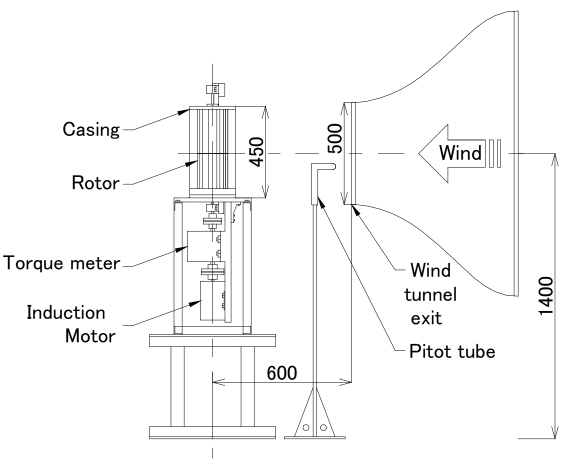
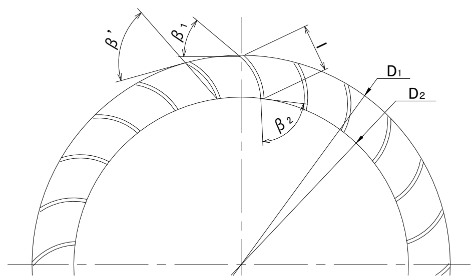


Fig.1 Experimental equipment



D_1 : Outer diameter
D_2 : Inner diameter
β_1 : Inlet angle
β_2 : Outlet angle
β' : Pitch angle
l : Chord length
Z : Blade number

Fig.2 Cross-flow rotor and blades

4. Performance and Flow Condition of Cross-flow Wind Turbine with Symmetrical Casing

Figure 5 shows the power characteristics obtained by the experiment under the condition that the casing is inclined against the main flow by Θ=15°. The maximum power coefficient of the casing was C_{pmax}=0.17 and increased to 1.7 times compared to that of the rotor without the casing. Furthermore, the tip speed ratio at this point shifted to a higher tip speed ratio. It is considered that the power coefficient of the casing increased owing to the improvements of flow condition at the rotor inlet and outlet. Then, the internal flow condition at the inlet and outlet of the rotor was investigated by the experiment and numerical analysis. The power coefficient C_p varied according to the swept area and the power obtained by the wind increased with the increase of the swept area. Therefore, there was a discussion that the power obtained by the large rotor without the casing having the same swept area of our turbine, which included the rotor and casing should be compared to the results of this research. But we considered that the wind turbines with the same swept area of the large rotor and the small rotor with the casing were different in the point of view of safety and reliability for the citizen in urban district, where these kinds of small-sized wind turbines could be applied. Then, the area A was defined as a constant projected area of the rotor in this paper, however, the swept area increased by the casing and side boards.

Figure 6 shows static pressure distributions around the test wind turbine obtained by the numerical analysis. The casing was inclined against the main flow by Θ=15°. It was clarified from the previous report that the wind flowed in the opposite direction of the rotor rotation at the inlet of the rotor in the case without the casing[8]. Further, it was important to increase the incoming flow to increase the power of the cross-flow wind turbine. Therefore, the symmetrical casing was designed to block the flow which prevented the rotor from rotating, to increase the incoming flow in the rotor and to improve the inlet and outlet flow condition. The qualitative tendency of the inlet flow condition of the numerical results was similar to the experimental results, although the quantitative comparison was difficult because the numerical results was conducted under the two-dimensional condition to suppress the computational cost[8]. It was found in Fig.6 that the casing blocked the flow preventing the rotor rotation at the inlet of the rotor and there was a low static pressure region to increase the incoming flow beside the casing and downstream of the rotor. There were two regions at the inlet of the rotor. The one was the low pressure region near the inlet of the rotor. The other region was the high pressure region at the inlet of the nozzle of the casing. The low pressure region was the high velocity region and could give the angular momentum to a first stage of the rotor. On the other hand, the high pressure region could effect on the increase of the load at the second stage of the rotor. When the casing was inclined against the main flow, the external flow influenced on the outlet flow from the rotor and the incoming flow rates to the rotor decreased. Therefore, in order to suppress the influence of the external flow on the rotor outlet flow condition and increase the incoming flow, the side boards shown in Fig.3 were installed on the casing. Figure 7 shows the performance curves under the condition that the side boards were

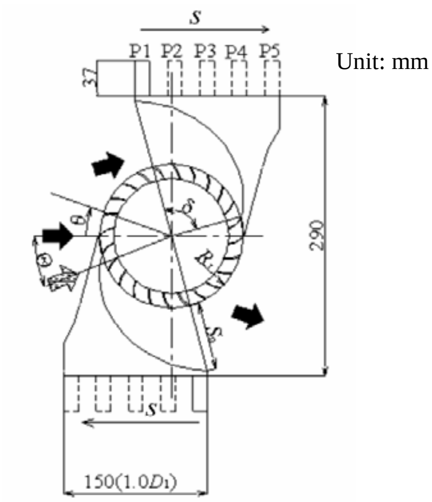


Fig.3 Cross-flow section of symmetrical casing with side boards

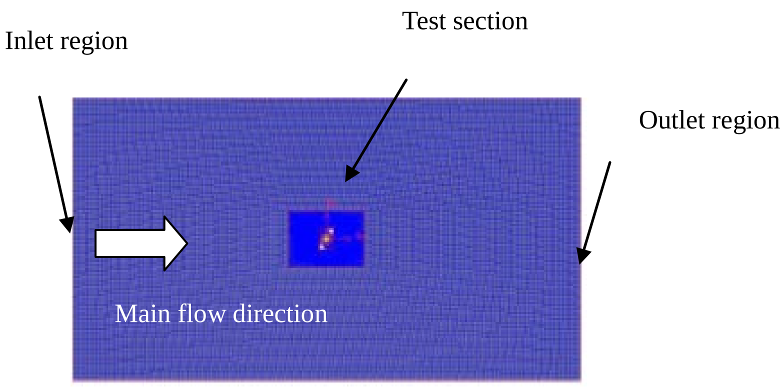


Fig.4 Numerical grids of computational domain

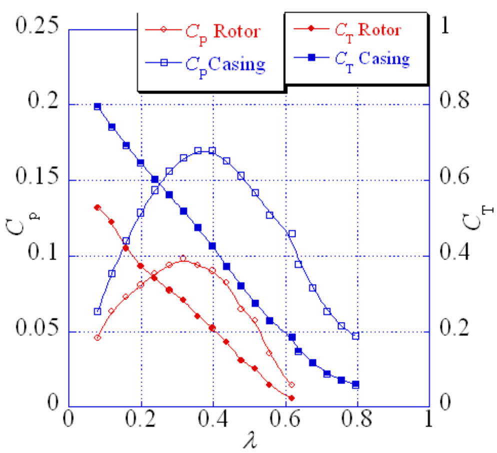


Fig.5 Performance curves

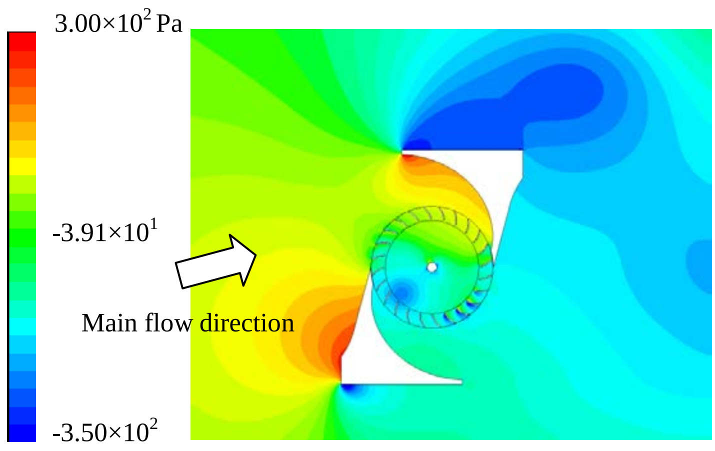


Fig.6 Static pressure distributions

installed at each position and the casing inclination angle was Θ=15°. The power coefficient in the case of $P1$ decreased compared to that of the casing without the side boards. Further, the power coefficients in the case of $P2$ and $P3$ were almost the same with that of casing. Therefore, the effects of the side boards on the performance in the case of $P1$, $P2$ and $P3$ were not confirmed. On the other hand, the power coefficients in the case of $P4$ and $P5$ increased compared to that of the casing and the torque coefficient in the case $P5$ was the highest in all cases. The maximum power at each side board position in the case with the inclination angle Θ=15° was shown in Fig.8. It could be found that the maximum side board position s=150mm ($P5$) was the best for the increase of the performance. It was thought at first that the side board position s=0 mm ($P1$) could be preferable for the performance increase because the side board at the outlet of the rotor could suppress the influence of the external flow on the rotor outlet flow condition. Therefore, the internal flow conditions were investigated by the experiment and numerical analysis to clarify the causes of the performance decrease of $P1$ and the performance increase of $P5$.

The circumferential distributions of the circumferential velocity V_t obtained by the experiment in the case with the inclination angle Θ=15° at the rotor inlet of the casing, $P1$ and $P5$ are shown in Fig.9. The circumferential velocity V_t in the case of $P5$ was the largest compared to those of casing and $P1$ in wide circumferential position. It was thought that the angular momentum given at the first stage of the rotor could be large in the case of $P5$ and the power coefficient increased. Figures 10(a),(b) and (c) show the static pressure distributions of the casing, $P1$ and $P5$ respectively at the inlet of the rotor. It could be found that the static pressure at the inlet of the casing in the case of $P1$ was large and the high pressure region spread in wide region near the side board at the inlet of the casing. This high pressure region would cause the decrease of the incoming flow rate and the decrease of the circumferential velocity in Fig.9. Therefore, the side board position of $P1$ at the inlet of the casing was not proper for the flow conditions at the inlet of the rotor.

The circumferential distributions of the absolute flow angle obtained by the experiment in the cases with the inclination angle Θ=15° at the rotor outlet of the casing, $P1$ and $P5$ are shown in Fig.11. It could be found in Fig.11 that the absolute flow angles in the cases of $P1$ and $P5$ were close to the best absolute flow angle α=-90° in the circumferential region 240° ≤ θ ≤ 260° and the losses downstream of the rotor were small in both cases. The effect of the side board at the outlet of the casing to suppress the influence of the external flow on the flow conditions at the outlet of the rotor could be observed. Therefore, the side board positions of $P1$ and $P5$ were suitable for suppressing the external flow influence on the outlet flow condition. However, the side board position of $P1$ was not proper for the flow condition at the rotor inlet and vice versa in the case of $P5$. As a result, the performance of $P5$ became better than that of $P1$.

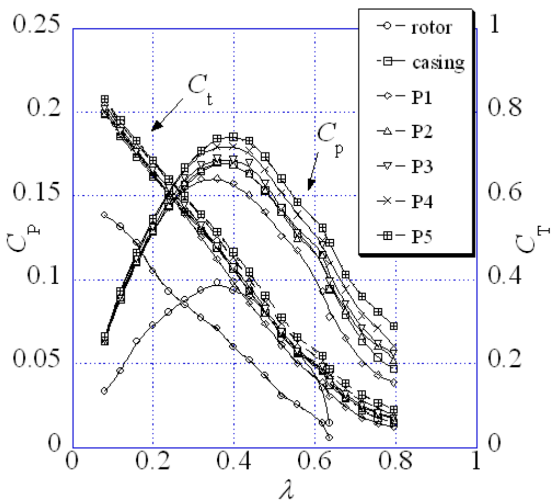


Fig.7 Performance curves at each side board position

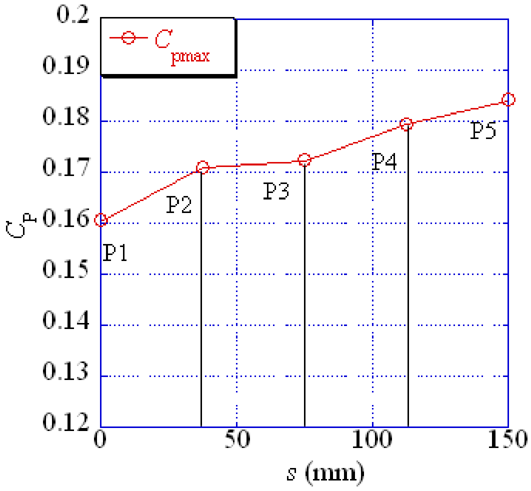


Fig.8 Maximum power at each side board position

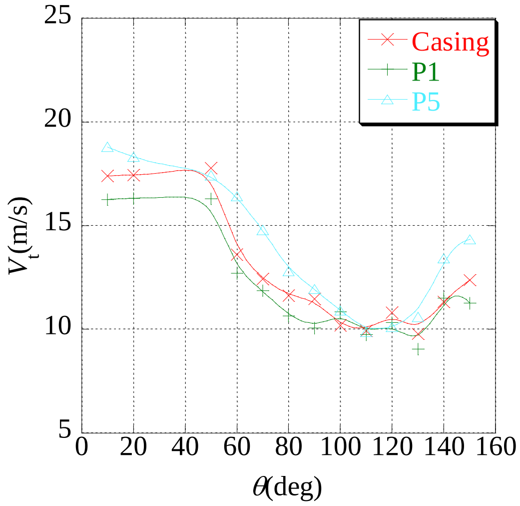
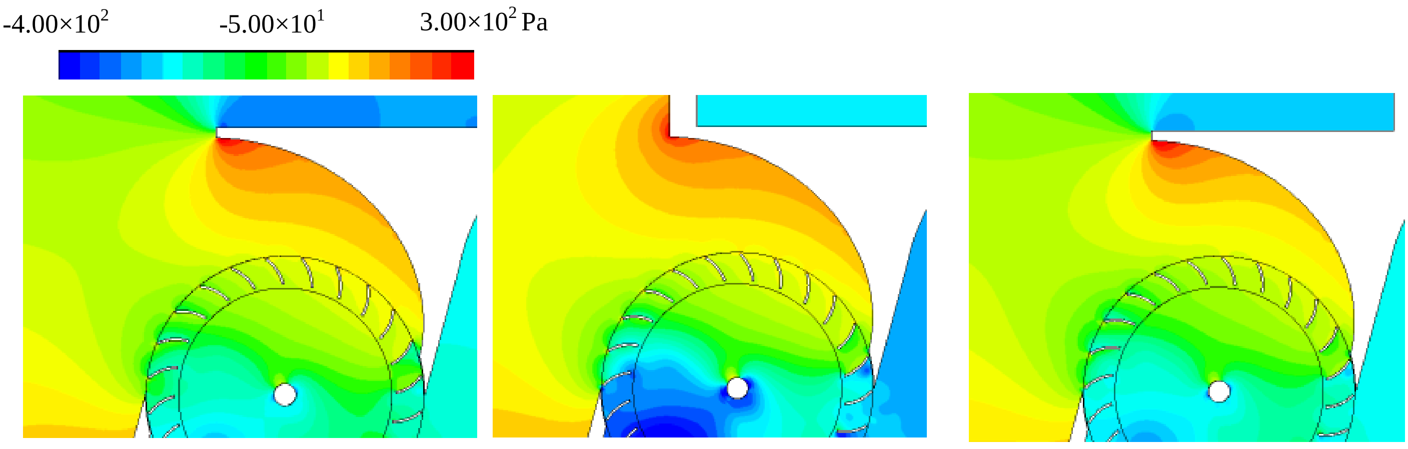


Fig.9 Circumferential velocity distributions at inlet of rotor



(a) Casing (b) P1 (c) P5

Fig.10 Static pressure distributions at inlet of casing

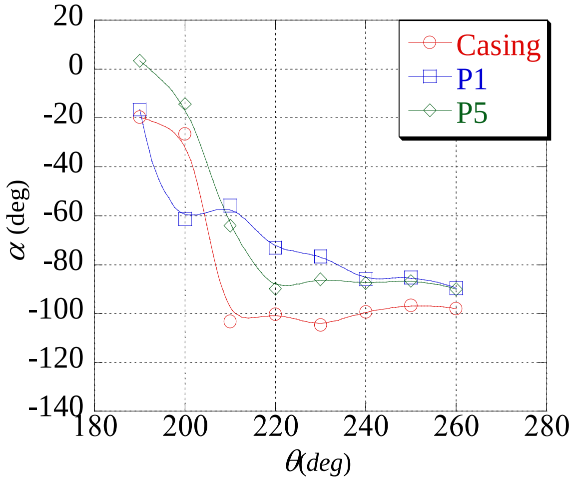


Fig.11 Circumferential distributions of absolute flow angle at rotor outlet

5. Conclusion

In order to improve the performance of cross-flow wind turbine, the symmetrical casing with the side board was proposed and the performance test was conducted. Furthermore, its flow condition was investigated by the experiment and numerical flow analysis. The following concluding remarks were obtained.

1. The power coefficients in the case of *P*4 and *P*5 increased compared to that of the casing and the torque coefficient in the case *P*5 was the highest in all cases. The circumferential velocity V_t in the case of *P*5 was the largest compared to that of casing and *P*1 in wide circumferential position. The angular momentum given at the first stage of the rotor could be large in the case of *P*5 and the power coefficient increased.
2. The static pressure at the inlet of the casing in the case of *P*1 was large and the high pressure region spread in wide region near the side board at the inlet of the casing. This high pressure region would cause the decrease of the incoming flow rate and the decrease of the circumferential velocity. Therefore, the side board position of *P*1 at the inlet of the casing was not proper for the flow conditions at the inlet of the rotor.
3. The absolute flow angles in the cases of *P*1 and *P*5 were close to the best absolute flow angle α=-90° in the circumferential region 240° $\leq\theta\leq$ 260° and the losses downstream of the rotor were small in both cases. Therefore, the side board positions of *P*1 and *P*5 were suitable for suppressing the external flow influence on the outlet flow condition. However, the side board position of *P*1 was not proper for the flow condition at the rotor inlet and vice versa in the case of P5. As a result, the performance of *P*5 became better than that of *P*1.

Nomenclature

A	Projected area of rotor=$H \times D$ [m^2]	U_∞	Main flow velocity [m/s]
C_P	Power coefficient=$\omega T/(\rho A U_\infty{}^3/2)$	V	Absolute velocity [m/s]
C_T	Torque coefficient=$T/(\rho A U_\infty{}^2 R/2)$	V_t	Circumferential component of absolute velocity [m/s]
C_{PS}	Pressure coefficient=$(P-P_\infty)/(\rho U_\infty{}^2/2)$		
D	Rotor diameter [m]	Z	Blade number
D_1	Rotor outer diameter [m]	α	Absolute flow angle [°]
D_2	Rotor inner diameter [m]	β_1	Blade inlet angle [°]
H	Rotor height [m]	β_2	Blade outlet angle [°]
l	Blade chord length [mm]	β’	Pitch angle [°]
P_s	Static pressure [Pa]	λ	Tip speed ratio =$R\omega/U_\infty$
P_∞	Pressure of main flow [Pa]	θ	Circumferential position [°]
R	Rotor radius [m]	Θ	Inclination angle [°]
Re	Reynolds number	ρ	Density of air [kg/m^3]
T	Torque [N·m]	ω	Angular velocity [rad/s]

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