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A NUMERICAL ANALYSIS OF THE UNSTEADY FLOW ROUND A SAVONIUS TURBINE

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Abstract

The paper presents a numerical analysis of the transient flow round the two blades of the Savonius rotor. The main goal of the research is to examine the influence of unsteady effects on the turbine performance. Because of the intensity of the flow through the gap, controlled by the vortices forming in its vicinity, the performance of the rotor as a whole also depends on the flow structure between the blades. The definition of the transient efficiency is given and compared with the existing steady-state case. The obtained numerical results are compared with the experimental data available in literature.

Key words: Savonius Rotor, Transient Flow, Efficiency

INTRODUCTION

The Savonius rotor is a kind of vertical axis wind turbine invented by S. J. Savonius, a Finnish engineer, in 1922. In the past, this rotor did not found any larger interest in technical use. The Savonius turbine is a very simple design which consists of usually two separated blades rotated by 180 degrees with respect to each other. A distinguishable feature of this turbine is the gap between the blades, see Fig. 1.

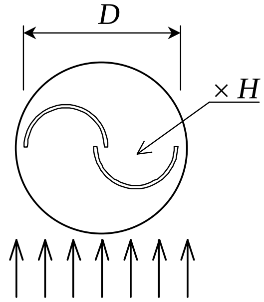

Figure 1: Diagram

There have been many directions of investigating the Savonius rotor construction through the years. The number of blades is one of the first characteristic features which is quite easy to change. The more blades, the greater stability and rigidity of the structure. Unfortunately, it is accompanied by remarkable decrease of efficiency, and the primary idea of just two blades has been proved to be the most efficient solution.

The second direction of investigation is search for a more efficient shape of blades. Works in this field are still in progress and the final solution has not been defined so far. There are still many configurations of rotor geometry to analyse and examine. The number of possible construction variants including such parameters as gap ratio and blade size is almost infinite. It gives a great chance for researchers to find an optimal shape and dimensions of the rotor and to popularise this construction for practical applications. The Savonius rotor is well know due to its simplicity and low cost of implementation. Whenever the efficiency is not an issue the Savonius rotor may be used. This advantage is rather important, especially for Polish investors due to high cost of other types of wind turbines available on Polish market.

DEFINITIONS

Torque coefficient

The most common definition of the torque coefficient is the following:

$$C_T = \frac{T}{\frac{1}{4}\rho U^2 D^2 H}, \tag{1}$$

which is valid for both the steady-state and transient flow. For the latter case we should use the time dependent torque $T(t)$ instead of T. This means that in real case we deal with the distribution of C_T as a function of the angular position of the rotor.

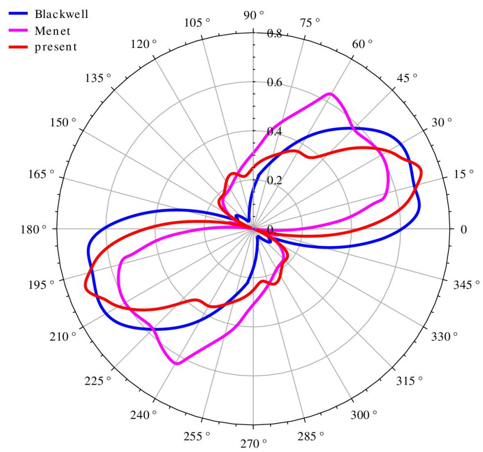

Figure 2: Torque coefficient distribution for a steady-state case

Figure 2 shows a comparison between the present calculations and other data available in the literature (Blackwell et al., 1977; Menet, 2004). All of them represent steady-state cases for different rotor configurations. Good agreement may be observed.

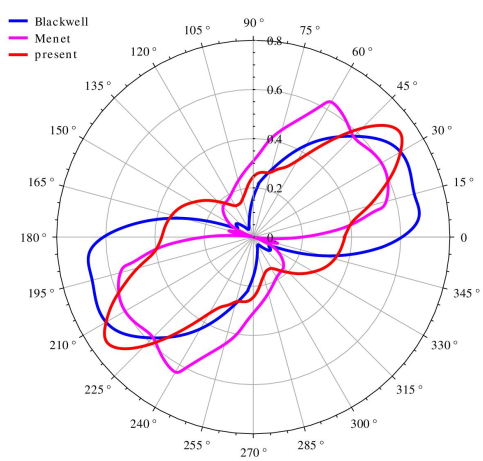

Figure 3: Steady-state and transient torque coefficient distribution

Figure 3 presents a comparison of the steady-state data vs. the present transient calculations. It has to be pointed out that most of the available data is obtained from steady-state calculations whereas the present calculations are fully transient. This means that figure 3 gives only quantitatively comparison.

Efficiency

A typical definition of the efficiency for the steady-state case takes under consideration the wind power $N_w = \dot{m} e_k = \rho U S 2^{-1} U^2$ and the power of the rotor $N = M\,\omega$. The wind power is treated here as the reference power. Combining the two above definitions we arrive at the following definition of the efficiency:

$$\eta_s = \frac{\omega T}{\rho \frac{U^3}{2} HD} . \tag{2}$$

In the above definition T stands for the torque, ω is the angular velocity, ρ represents the density, and U - the far field velocity. D is the diameter of the rotor, and H is the rotor's height. The steady-state efficiency is sometimes referred to as the power coefficient C_P.

For the transient case, which is typical for the Savonius rotor operation, we should consider the total energy of the wind $E_w = \int_t^{t+\Delta t} N_w \, dt = N_w \Delta t$ within the time interval Δt rather than the instantaneous power. The same concerns the energy of the rotor $E = \int_t^{t+\Delta t} N \, dt = \omega \int_t^{t+\Delta t} M(t)\; dt$. The transient efficiency may now be defined as

$$\eta_t = \frac{\omega \int_t^{t+\Delta t} T(t) dt}{\rho \frac{U^3}{2} HD\, \Delta \mathrm{t}} , \tag{3}$$

where Δt stands for the time of interest (e.g. half-turn). The above definition is directly related to the torque coefficient (1), see figure 3 for example. Combining the above definition and equation (1) we obtain

$$\eta_t = \frac{\omega D \int_t^{t+\Delta t} C_T(t) dt}{2U\, \Delta \mathrm{t}} . \tag{4}$$

The same for the steady-state case

$$\eta_s = \frac{\omega D C_T}{2U} . \tag{5}$$

Assuming that the time step $\Delta\tau$ of the transient CFD calculations is constant we can approximate the integral in equation (4) in the following way

$$\int_t^{t+\Delta t} T(t)\, dt \approx \sum_{i=1}^{n} T_i \, \Delta\tau = n\, \Delta\tau \frac{1}{n} \sum_{i=1}^{n} T_i \; = \Delta t\, \overline{T} , \tag{6}$$

where $\overline{T}$ represents the arithmetical average $\overline{T} = n^{-1} \sum_{i=1}^{n} T_i$ and the time of interest Δt (e.g. one or half-turn) is expressed as $\Delta t = n\, \Delta\tau$. This reasoning allows us to rewrite the definition (4) in a more useful version

$$\eta_t = \frac{\omega \overline{T}}{\rho \frac{\mathrm{U}^3}{2} H\, D} . \tag{7}$$

The total number of time steps is denoted here as n. The last definition (7) has an analogical form as the definition (2) that is valid for the steady-state case.

NUMERICAL CALCULATIONS

The numerical calculations were performed using the commercial CFD code CFX. The turbulent flow of air was treated as an incompressible medium. The turbulence was modelled by means of the standard two-equation turbulence model $k-\varepsilon$. The reason for this choice may be explained by a need of comparison with other calculations found elsewhere. The $k-\varepsilon$ model has now become a standard for this kind of calculations.

Grid

The both flow domains, i.e. the rotor and the ambient, were discretised separately. The rotor domain has an unstructured grid consisting of mostly tetrahedral elements, whilst the ambient flow domain is semi-structured. Table 1 show basic grid information. The total number of elements covering the flow area is about 1.2 million. There are also special elements around the blades to ensure that flow near a wall is properly resolved. The wall function approach was used to provide near wall boundary conditions for the mean flow.

Table 1: Grid information

	Number
Rotor nodes	266 716
Rotor elements	963 604
Ambient nodes	234 332
Ambient elements	253 934
Total nodes	501 048
Total elements	1 217 538

Figure 4 presents the discretised domains of the ambient with the rotor (on the left) and rotor itself (on the right). The closer the rotor domain, the denser the grid. The greed seed density was decreased at the blades to assure that the curvature of blades is preserved.

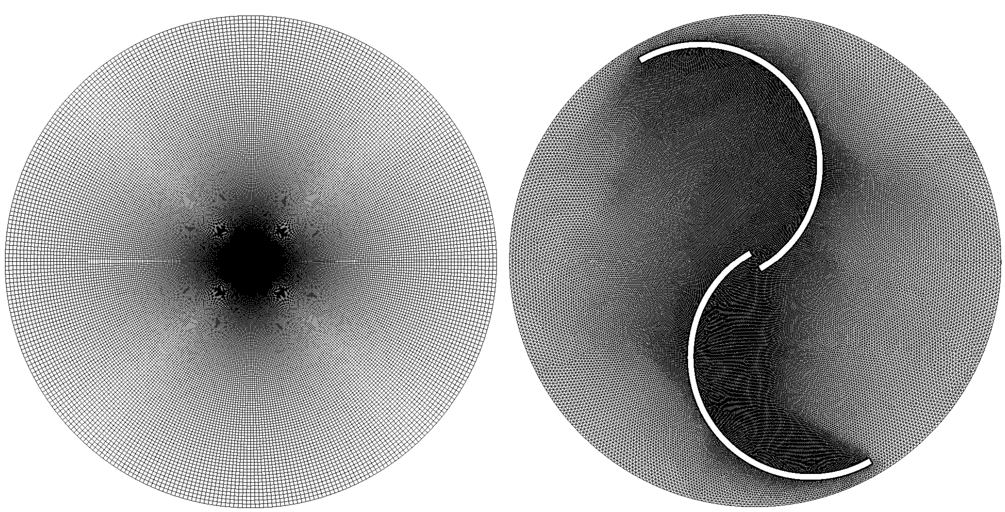

Figure 4: Grid

The quality of the grid near the blades may be inspected in terms of y^+ distribution which is shown in Figure 5. The maximal values of y^+ do not exceed 3 and the average value of y^+ does not exceed 1. Both values are presented as a function of revolution angle. The average value of y^+ is defined here as a surface average.

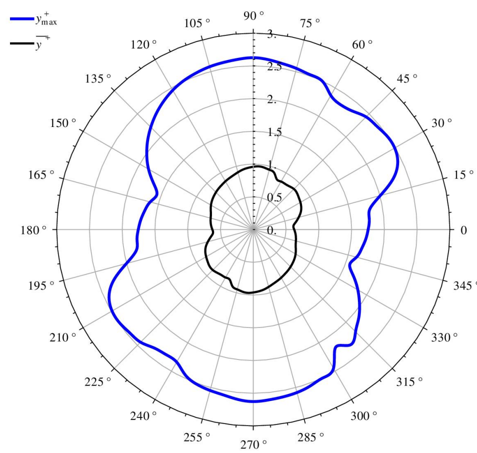

Figure 5: y^+ distribution

Boundary and geometrical conditions

The flow domain was divided into two parts: the rotating rotor and the steady ambient. Both part were merged by means of the domain interface of the so called `transient rotor-stator' type. The time step of the transient calculations corresponded to one degree of revolution. The far field velocity was chosen equal to $U = 8\,[m\,s^{-1}]$ (opening boundary condition), the angular velocity of the rotating domain (rotor) was $\omega = 2\pi\ [Hz]$ which corresponds to one turn per second, the time of interest was $\Delta t = 0.5\,[s]$ (i.e. half-turn). The diameter of the rotor was $D = 0.38\,[m]$ and its thickness was $H = 0.003\,[m]$. In the outer surfaces of the considered domain 'slice' the symmetry boundary conditions were applied. The two blades of the rotor were modelled as no slip wall in the rotating frame of reference.

SAMPLE RESULTS

Influence of the previous turns

Figure 6 shows the differences between subsequent turns on the transient values of the torque coefficient (1). Three full turns (or six half-turns) were taken under consideration. The figure shows that the first half-turn obviously cannot be considered as a fully developed flow. The second half-turn is almost correct and there are no visible differences between the third and higher half-turns.

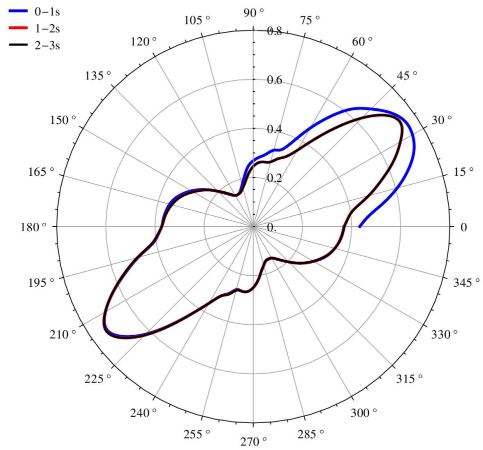

Figure 6: Torque coefficient distribution for subsequent turns

The values of the relative errors in % of torque coefficient are given in Figure 7. The highest values of error up to 30% are obtained for the first half-turn whilst for the second they do not exceed 2%. It is, therefore, enough to take under consideration only three half-turns.

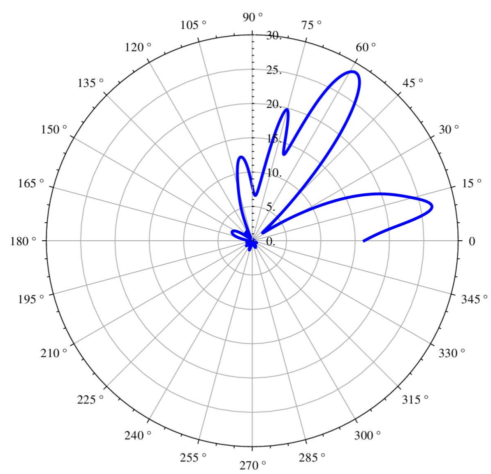

Figure 7: Torque coefficient error distribution for subsequent turns

Efficiency

The higher the torque coefficient, the more efficient the rotor. This is because the wind energy (and power) is kept constant. Figure 8 compares the efficiency obtained from steady-state and transient calculations. It may be observed that for the transient case we obtain higher efficiency $\eta_t = 5.22\%$ than for the steady-state case $\eta_s = 4.88\%$. These values may be easily validated by means of data available in the litarature (Le Gourières, 1980) where expected performance of the conventional Savonius rotor for $\lambda = 0.15$ is estimated to be $\approx 5 \div 6\%$. The rotor tip speed ratio λ is defined as

$$\lambda = \frac{\omega D}{2\mathrm{U}}. \tag{8}$$

What is also remarkable is that for the transient calculations there are no ‘negative’ values of torque, typical for steady-state assumptions (Menet, 2004), see figure 2.

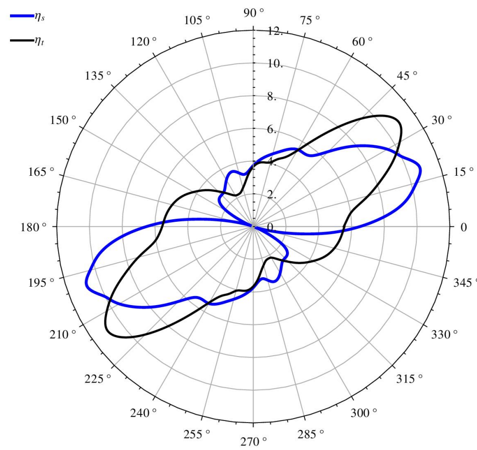

Figure 8: Steady-state and transient efficiency distribution

Vortex wake development

Figure 8 shows the velocity distribution round the rotor and in the wake that arises because of the rotor presence. Selected rotor positions are 0, 30, 60, 90, 120, and 150 degrees. The velocity field is presented in the stationary frame of reference. It has to be pointed out that this figure shows selected time steps of the transient solution and these are somewhat different in comparison with the series of steady-state solutions.

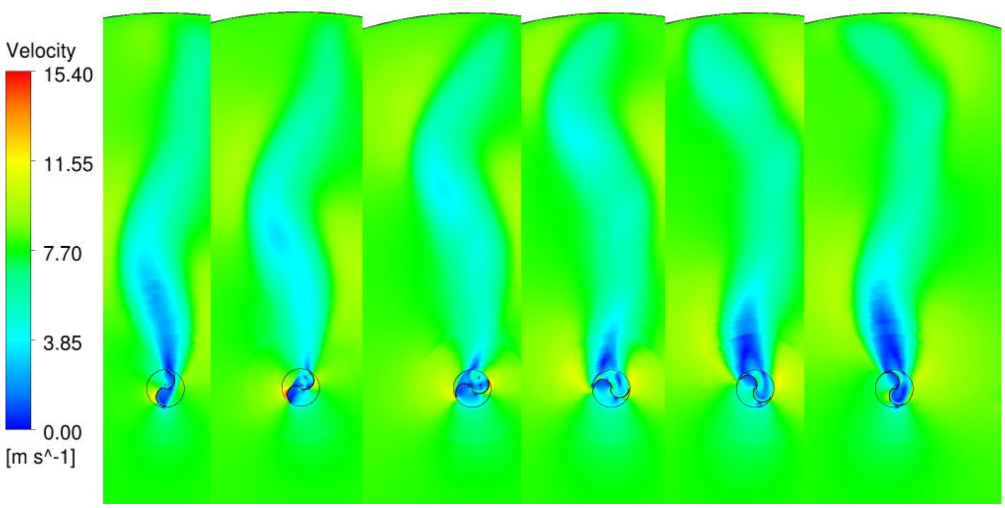

Figure 9: Velocity in stationary frame of reference distribution for selected rotor positions

CONCLUSIONS

1. The definition of the transient efficiency is shown and evaluated. This definition is more natural for the flow round a Savonius turbine, as this flow is transient by nature. The efficiency of the transient case is higher than that for the steady-state case, which suggests that turbine performance estimations based on steady-state or quasi transient analyses may underestimate its efficiency.
2. The calculated transient efficiency is in good agreement with prediction available in the literature.
3. There are no ‘negative’ values of torque in transient calculations which are often observed for steady-state solutions.

4. It is enough to take under consideration three half-turns to have the relative error of the torque coefficient smaller than 2%.

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