Performance Analysis of Multi Rotating Savonius Turbines for Exhaust Air Energy Recovery Through CFD Simulations

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Abstract: Wind energy generation in adverse conditions has always been a challenge for the energy industry, especially at low wind speeds. Vertical axis wind turbines (VAWT) are promising solutions for low and turbulent wind conditions; however, their low efficiency has been a problem for their application and requires further research initiatives. In this research, a multi-Savonius VAWT arrangement is tested to use an unnatural wind source, a cooling tower exhaust, to improve the turbine performance. The analysis method was done by a series of two-dimensional computational fluid dynamics (CFD) simulations using Ansys FLUENT. A grid sensitivity test was performed, and the numerical model was validated against published numerical and experimental results. Test cases with single and multi-turbine configurations at six different tip speed ratios were then performed to calculate the turbine performance. The multi-turbine configuration improved the power coefficient by up to 19.29%. The multi- turbine arrangement has proven to be an effective means of increasing the performance of the Savonius turbine for the exhaust energy recovery system.

1. Introduction

Exhaust air energy recovery systems allow the low wind urban areas to utilize the uniformly exhausted air from the exhaust systems. The vertical axis wind turbines (VAWT) proposed for such applications have very low efficiency due to their inefficient design and functions in the exhaust systems. Several studies have been carried out in the last decades to improve VAWTs, mostly for free-flow wind conditions [1, 2]. These include optimizing the wind turbine by modifying the blade shape, strength, pitch, blade profile and configurations.

Another method of performance enhancement is the use of flow directing or augmentation mechanisms using deflectors, guide vanes or tail vanes, stators, curtains, diffusers, casings, ducts, shrouds, nozzles, shrouds and guide vanes [3, 4]. Flow augmentation aims to reduce negative torque, direct the flow to a better angle of attack, or increase wind speed. One or more mechanisms can be used together.

Positive coupling of multiple wind turbines arranged in close proximity to each other is another method of increasing efficiency. When multiple wind turbines are located in close proximity to each other, as is the case in wind farms, horizontal axis wind turbines (HAWT) experience aerodynamic interference, resulting in performance degradation, as well as greater aerodynamic loads and shorter life [4]. However, research has proven that VAWTs have mutually beneficial or positive coupling when placed adjacent to each other, especially when the turbines are counter-rotating inwards synchronously [5, 6]. The presence of an adjacent turbine causes an increase in the wind speed near the blades and
surrounding vortices, resulting in an increase in the power coefficient [6]. The larger pressure gradient over the blades positively affects the power extraction.

2. Methodology

2.1. Data Processing and Important Parameters

The data processing of this study has been carried out using equations 1-3 for the coefficient of power ($C_p$), coefficient of moment ($C_m$), and tip speed ratio (TSR).

\[
TSR = \frac{\omega R}{V} \quad (1)
\]

\[
C_m = \frac{T}{\frac{1}{2} \rho AV^2} \quad (2)
\]

\[
C_p = \frac{p}{\frac{1}{2} \rho AV^3} = TSR \cdot C_m \quad (3)
\]

where ‘$V$’ is the free stream velocity, ‘$\omega$’ is the angular velocity of the blade, ‘$R$’ is the turbine radius, ‘$T$’ is the time-averaged turbine torque, ‘$p$’ is the density of air, and ‘$A$’ is the wind incident area.

2.2. Simulation and Model Parameters

Two dimensional CFD simulations were performed to solve URANS equations in the finite volume based ANSYS Fluent. The Shear Stress Transport (SST) k-$\omega$ turbulence model was used, as been used satisfactorily by Youssef et al. [4]. The sliding mesh technique has been used for turbine rotation and interaction with adjacent cells.

The turbine selected for this study is a conventional Savonius turbine with a semicircular blade profile. Dimensions were set similar to the traditional Savonius turbine in Roy and Ducoin’s [7] study. The measurements used for both the turbine and cooling tower [8] are listed in Table 1 and 2, respectively.

| Table 1. Turbine dimensions                  | Table 2. Cooling tower dimensions [8] |
|---------------------------------------------|--------------------------------------|
| Dimension                                  | Value (m)                            |
|---------------------------------------------|--------------------------------------|
| Rotor diameter, D                          | 0.21                                 |
| Rotor radius, R                            | 0.105                                |
| Rotor chord length, c                       | 0.1135                               |
| Rotor height, H                            | 0.21                                 |
| Aspect ratio, AR                           | 1                                    |
| Rotor blade thickness, t                    | 0.00063                              |
| Overlap gap width, c                        | 0.01703                              |

| Dimension                                  | Value (m)                            |
|---------------------------------------------|--------------------------------------|
| Cooling tower height, H                     | 1.175                                |
| Throat height, H_t                         | 0.881                                |
| Base diameter, D_b                         | 0.910                                |
| Throat diameter, D_t                        | 0.525                                |
| Outlet diameter, D_o                       | 0.570                                |

Figure 1 shows the computational domain, consisting of fluid and rotating fields. Full domain dimensions are 12D by 20D, following the recommendations by Mohamed et al. [9]. The rotating domain is 1.5D, placed at 1.5D downstream of the inlets. Two inlets, namely the cooling tower inlet and the ambient inlet, are set to velocity inlet boundary conditions; and the outlet is set to pressure outlet boundary conditions. The full boundary condition settings are shown in Table 3.

| Table 3. Computational boundary conditions |
|--------------------------------------------|
| Boundary                                    | Condition        | Value          |
|---------------------------------------------|------------------|----------------|
| Cooling tower outlet inlet                  | Velocity inlet   | 3.7 m/s        |
| Ambient inlet                               | Velocity inlet   | 2 m/s          |
| Outlet                                      | Pressure outlet  | 0 Pa           |
| Side edges                                  | Stationary wall  | No-slip        |
| Turbine profile                             | Moving wall      | No-slip        |
Figure 2 shows the mesh used for the computation domain, which is refined through mesh independence study to ensure the proper grid for the study. An inflation layer is also applied to the blade profile to capture boundary layer characteristics adequately. Rotational speeds of 94.6 to 567.4 RPM was assigned to the rotating zone to achieve tip speed ratios (TSR) from 0.2 to 1.2. The timestep sizes for turbine simulations were assigned to ensure accuracy and the Courant number close to 1. To ensure quasi-steady convergence, the simulations were conducted for eight full turbine rotations.

3. Model Validation

The computational model was first verified by conducting a grid independence test. The simulations were run at TSR = 1, with mesh density varied from coarse to fine in steps of roughly 10,000 cells. Following seven mesh refinements, a difference of less than 0.1% from the preceding step was obtained. A total of 83,046 cells were selected as the final mesh resolution. The results of $C_m$ against number of cells at each refinement are plotted in Figure 3. Further, a validation study was done by verifying simulations results of the conventional Savonius turbine (without cooling tower) against the results of Roy & Ducoin.
Roy and Ducoin [7]. The results show a good agreement. However, it is noted that the mesh type and software used in both the study are different. This may be attributed to the triangular cells used in the present study with ANSYS software whereas Roy and Ducoin [7] used polyhedral cells in StarCCM+ software.

4. Results and Discussion

The averaged $C_m$ of the single and multi turbine at varying TSR and corresponding $C_p$ are summarized in Table 4 below. The results indicate a maximum performance ($C_p$) of 0.3002 at a TSR of 1 for the single turbine under the exhaust-air system. For the multi turbine configuration, the maximum performance ($C_p$) of 0.3387 is obtained at a TSR of 1.

The percentage differences are also presented in Table 4. It indicates a maximum percentage increase of 19.29% at a TSR of 0.6. For maximum power extraction at TSR = 1, the $C_p$ value has been increased by 12.8%. On average, it has been observed that the multi turbine configuration increased performance by an average of 13.72% from that of a single turbine configuration. This power enhancement under exhaust-air conditions is found aligned with the 11% power enhancement achieved by Sun et al. [10] for multi-turbines under free stream air conditions.

| TSR  | Single turbine | Multi turbines | Percentage difference |
|------|---------------|----------------|-----------------------|
|      | $C_m$         | $C_p$          | $C_m$              | $C_p$ |                  |
| 0.2  | 0.5073        | 0.1015         | 0.5718              | 0.1144| 12.72             |
| 0.4  | 0.4953        | 0.1981         | 0.5395              | 0.2158| 8.92              |
| 0.6  | 0.4078        | 0.2447         | 0.4865              | 0.2919| **19.29**         |
| 0.8  | 0.3564        | 0.2851         | 0.3974              | 0.3179| 11.50             |
| 1    | **0.3002**    | **0.3002**     | **0.3387**          | **0.3387**| 12.80             |
| 1.2  | 0.2167        | 0.2600         | 0.2538              | 0.3045| 17.12             |

The $C_m$ and $C_p$ curves with respect to variable TSR are presented in Figures 5 and 6. It is observed that the turbine performance increased in the multi turbine configuration for all the TSRs. However, the improvement is less apparent at lower TSRs. The improvement is evident with the increase of TSR. This attributes to higher rotational speed (increased TSR) causes an increase in the resultant wind speed near the blades and surrounding vortices, enhancing the power coefficient.

The velocity and pressure contours for the single and multi turbines are presented side by side in Figure 7. In the velocity contour, the single turbine’s wake is asymmetrical, as expected for the Savonius (drag-based) turbine. The advancing blade is a more significant obstruction compared to the returning blade. Shigetomi et al. [11] reported this flow characteristic as a Magnus effect created by the turbine’s vertical axis rotation.
Whereas, the multi turbines were arranged symmetrically. As such, the flow characteristics around both turbines are too. In the velocity contour, both turbines’ wake is near perfectly mirrored. Also, a larger and wider wake is observed compared to that of the single turbine. This indicates turbines experience more drag which is favourable for Savonius turbine performance under the exhaust systems.

![Velocity contours and pressure contours](image)

**Figure 7.** Velocity contours and pressure contours

Also, in the velocity contour of the multi turbines, a higher maximum velocity at the tip of the advancing blade is observed. This is attributed to stronger vortex shedding. The wind velocity at the upwind of the multi turbines is found greater than that upwind of the single turbine due to turbine interactions. In between the turbines, the velocity is much greater than the freestream. Flow is accelerated when it passes between the turbines. Probe measurement for the velocity at the centre indicates a velocity of 6.2697 m/s compared to 4.9234 m/s for the single turbine case. The advancing blades in this region redirect that flow more efficiently.

In the pressure contours of the multi turbines, higher pressure on the concave side and lower pressure region on the convex side of the advancing blade of both the turbines are observed. The higher pressure difference results in better performance. There is also a lower negative pressure generated on the concave side of the returning blade. Also, a more significant negative pressure behind the turbines is observed for multi-turbine configuration, enhancing the turbine torque positively.
5. Conclusions

A two-dimensional unsteady CFD study has been presented in this research to investigate single and multi-turbine configuration performance to recover the mechanical energy from the discharged air in an exhaust-air system. The Savonious design of vertical axis wind turbines was selected for this study. The simulations were conducted at various tip speed ratios to observe the turbine performance and study its moment coefficient and power coefficient characteristics. The Savonious turbine was selected for this study. The mesh independence study and computational model has been performed to ensure the accuracy of the results. The key observations are listed below:

- Savonious turbines can be a viable option for energy recovery in exhaust systems. The single turbine configuration achieved a maximum $C_p$ of 0.3002 at a TSR of 1. In contrast, the multi turbines achieved a maximum $C_p$ of 0.3387, a 12.80% increase from the single turbine case.
- For the multi-turbine configuration, the $C_p$ increased by 19.29% at TSR of 0.6, and on average, increased by 13.72% from that of a single turbine configuration.
- The counter-rotating multi-turbine arrangement has accelerated the flow impacting the advancing blade for both turbines, enhancing the system's overall performance.

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