Effect of Flow Velocity on the Performance of the Savonius Hydrokinetic Turbine

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1 Introduction

1.1 Preface

Compared to the wind power, small hydropower generation units gathered much attention due to its predictable power development and high power density. It can generate the electrical power directly by installing in the natural path of the water streams, without use of massive structure of dam. It can be used as a standalone power generation unit in the remote location where the water stream is naturally available, as shown in Fig. 1. The Savonius turbine is one of the hydrokinetic turbines, predominantly a drag force-driven type of turbines. In spite of its low power coefficient, their starting characteristic is quite good.

1.2 Status of Global Research and the Aim of the Present Work

River current turbines, which operate at lesser depths, are necessarily smaller, and their rated output rarely exceeds 400 kW, even in very strong currents of 4.5 m/s. [1]. The hydrokinetic turbine installed by Hydro-Québec is in the experimental and pre-commercialization stage. In September 2010, a first industrial prototype was connected to the Hydro-Québec grid. The hydro turbine was submerged in the Fleuve Saint Laurent (St. Lawrence River) near the old port of Montréal, with a planned

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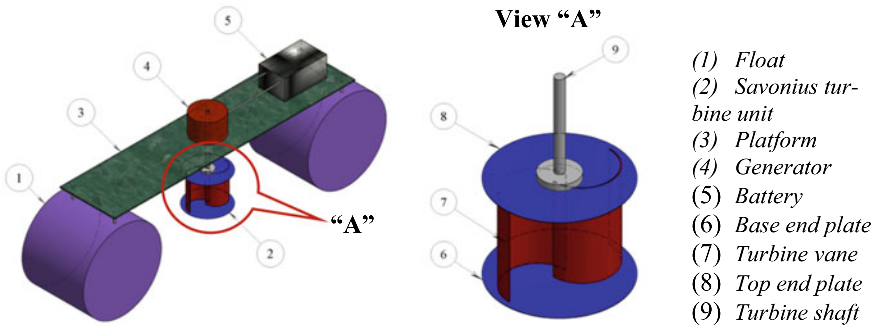


Fig. 1 Standalone power generation using Savonius turbine

capacity of 100 kW. It fed electricity into the Hydro-Québec grid from 2010 to 2013 [1].

Patel et al. [2] carried out an in-depth experimental investigation to find the effect of gap between the two vanes (overlap ratio) and height of Savonius turbine (aspect ratio). They concluded that the overlap ratio nearly 0.11 provides best performance with minimum aspect ratio of 1.8. The turbine shows higher power coefficient if the experiments are carried out in narrow canal. The in-depth methodology for the performance correction is explained by Patel et al. [3] for Savonius turbine. The theoretical calculations for prediction of the performance of the Savonius turbine are given by Patel et al. [4] based on impulse-momentum principal and stagnation pressure.

Based on the literature review, it is observed that the effect of flow velocity, vane thickness, and various vane shapes on the performance of the Savonius turbine for hydrodynamic application is still not investigated extensively. In the present investigation, it is targeted to analyze the effect of flow velocity on the performance of the Savonius turbine with CFD simulation.

2 Conceptual Discussion

The fundamental concept by which torque generates by water flow on the Savonius turbine runner is shown in Fig. 2. The drag force developed by the concave surface, *Fd (Adv),* of advancing vane is quite high compared to the drag force, *Fd (Ret)*, generated by the convex surface of the retarding vane. Power development is depending on (a) momentum change of the water passing from the vane surfaces and (b) pressure difference between upstream and downstream side of the vane. Two cases can be considered at this stage, (i) high incident water flow velocity and (ii) slow incident water flow velocity.

If the incidence of water flow velocity is relatively high, the pressure difference between upstream and downstream of the vane will increase. It will enhance the

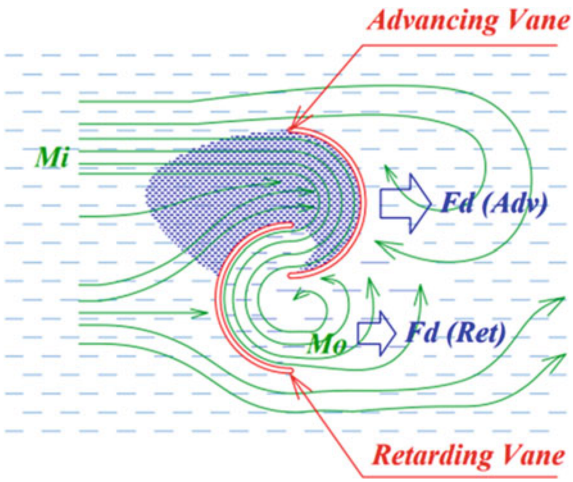


Fig. 2 Conceptual flows over Savonius turbine vane

momentum change *(Mo-Mi)* of the water passing from the gap of the vanes, as pressure condition at the downstream of the retarding vane is comparatively low. Subsequently positive torque due to momentum change of water from advancing vane increases.

If the incidence water flow velocity is relatively less, the mass flow rate of water from the vane gap reduces due to the water backflow from the downstream side of the water, toward retarding vane. Subsequently, the momentum change (*Mo-Mi*) of water passing from the gap decrease, and it may adversely affect the performance of the turbine.

To validate the considered concept, it is decided to study the effect of water velocity on the performance of the Savonius turbine by CFD simulations. Also, the study is further extended to find the cutoff velocity to provide best performance from the turbine.

3 Numerical Simulation

The numerical study was carried out to check the validity of specific turbulence models in the computational fluid dynamics (CFD). In the present investigation, the pressure-based, transient, absolute, planner with viscous turbulent K-ω SST two-equation models are selected. The boundary conditions used for the investigations are shown in Fig. 3. The mesh was prepared using triangular elements and 15 inflation layers with growth rate of 1.15. The average aspect ratio, orthogonal quality, and skewness of the used elements are 32, 0.36, and 0.76, respectively.

The grid-independent and domain optimization study are also carried out before investigation of the flow velocity effect. The conditions used for grid-independent study are inlet velocity equal to 0.85 m/s, diameter of rotor equal to 0.22 m, rotational

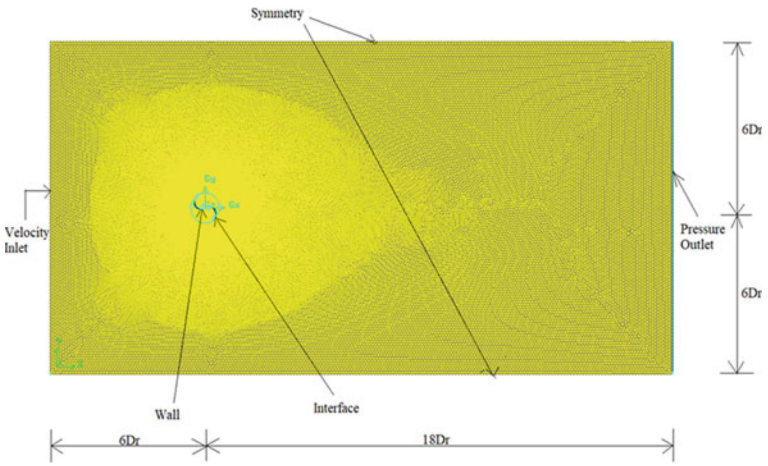


Fig. 3 Boundary conditions and domain used in the present investigation

velocity equal to 9.35 m/s, TSR equal to 1.1 also constant rectangular domain 12D × 24D. Graph of C_p versus total grid size is shown in Fig. 4, which says that after total grid size as 90,962 C_p value remains constant. C_p value is 0.334. All the numerical simulations are done using total grid size higher than 90,962.

The size of the study domain also affects the results obtained by the simulation. Hence, domain optimization study is carried out with free stream velocity $V = 0.85$ m/s, diameter of rotor $D_r = 0.2$ m, thickness of blade $t = 5$ mm, eccentricity

Fig. 4 Effect of the number of elements on C_p

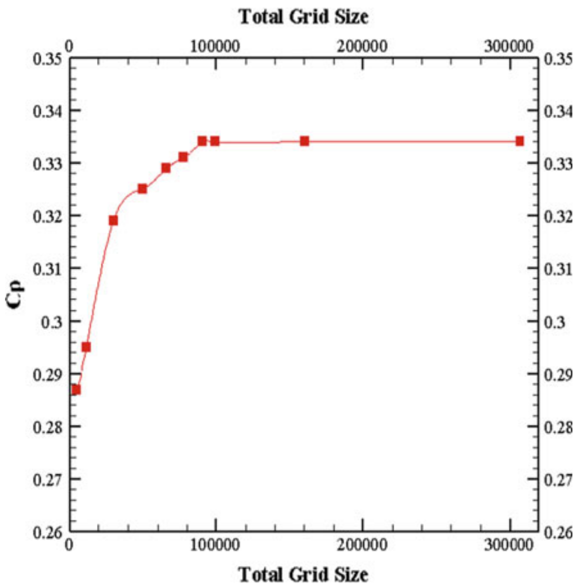
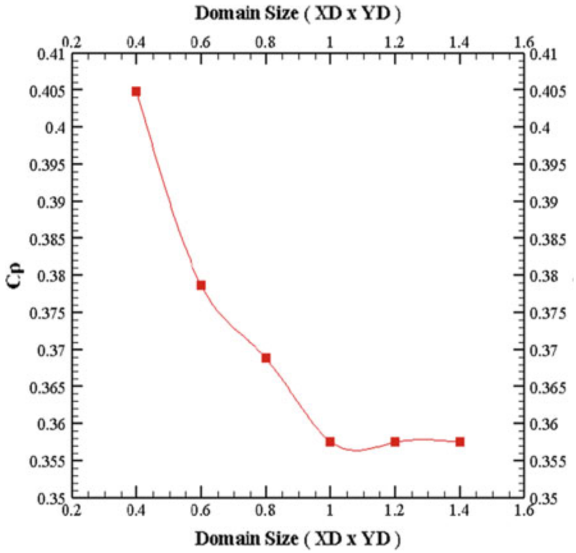


Fig. 5 Effect of domain size on the obtained value of C_p



$e = 0.01$, and TSR = 1.0. Domain study is done in the multiplication of the basic domain size 12D × 24D. From Fig. 5, the graph of the C_p versus TSR with the multiplication factor of 1 [means (12D × 24D)] the result indicates nearly constant value of C_p. So, it can be assumed that for the higher multiplication it will remain same. Therefore, domain size can be taken as 12D × 24D.

The average value of the coefficient of moment (C_m), after reaching steady-state variation in C_m was obtained from simulation. The variation of C_m with different flow time is shown in Fig. 6

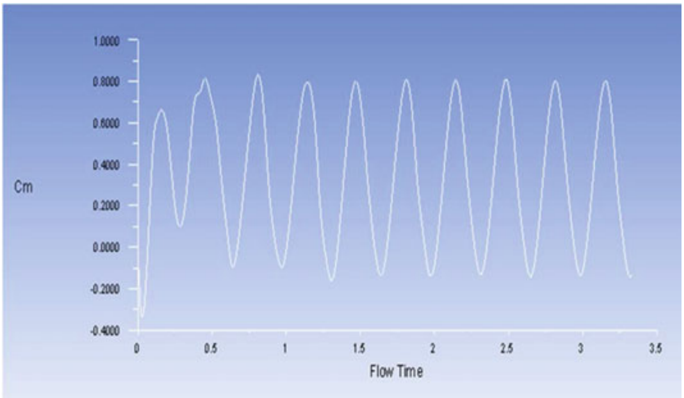


Fig. 6 Variation of C_m during analysis

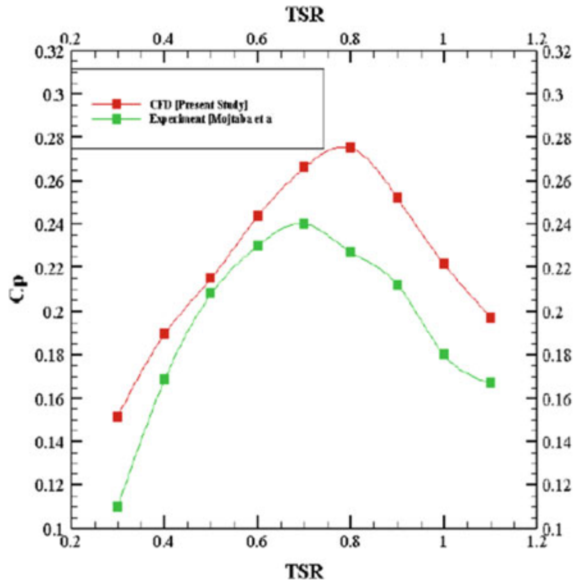


Fig. 7 Validation of the methodology used in the present investigation

For the validation of the considered methodology, the grid size and domain size are kept as 110,652 elements and (2.4 m × 4.8 m), respectively. Also, boundary conditions were velocity inlet at left edge with 6 m/s of wind, pressure outlet at right edge, top, and bottom was taken as symmetry. The blade radius r was 0.0585 m, the endplate diameter D was 0.23 m, and the eccentricity is 0.023 m. The results obtained from the present simulation are quite matching with the experimental results available in the published literature [5]. The close matching and same variation trend validate the adopted methodology used in the present investigation (Fig. 7).

4 Effect of Flow Velocity

4.1 Investigated Parameters

The numerical analysis is carried out for the conventional Savonius rotor. The diameter of the blade (D) is 0.1 m, the gap between the vane is 0.01 m, diameter of rotor is 0.2 m, and thickness of the blade is 5 mm. The simulations are carried out for different values of velocity of water (0.5, 0.6, 0.7, 0.8, 1.0, 1.4, 2.0, 2.5, 3.0, 3.5 m/s) keeping the remaining parameters as constant. Here, simulations were done for tip speed ratio values 0.4–1.2. So, rotational velocity value is between 3.6 and 10.2 rad/s.

4.2 Results and Discussion

The pressure and velocity contour obtained with the present investigations are shown in Figs. 8 and 9, respectively.

The upstream side pressure is comparatively higher than that of downstream side of the rotor. This higher pressure is generated due to stagnation of the flow due to resistance offered by the turbine rotor.

The downstream side velocity is also quite low compared to the upstream side of the rotor. It is due to the utilization of the kinetic energy to generate mechanical power, which decreases the velocity of the flow at downstream side.

The simulations are carried out for different velocity of water in the range of 0.5–3.5 m/s. The variation of C_p for different TSR obtained from the present investigation is shown in Fig. 10. The result indicates that the C_p value increases as the velocity of flow increases. Also, the value of Cp_{max} appears at higher TSR as flow velocity increases.

Fig. 8 Pressure contour

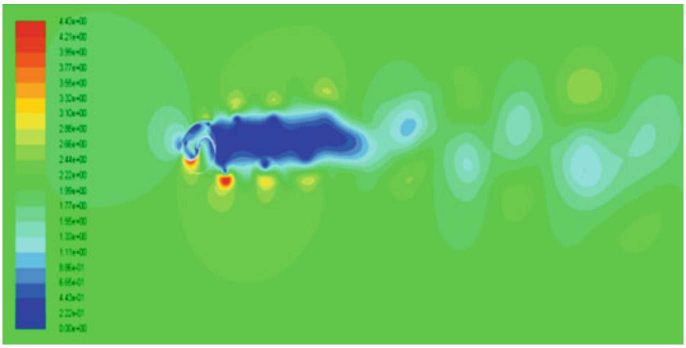


Fig. 9 Velocity contour

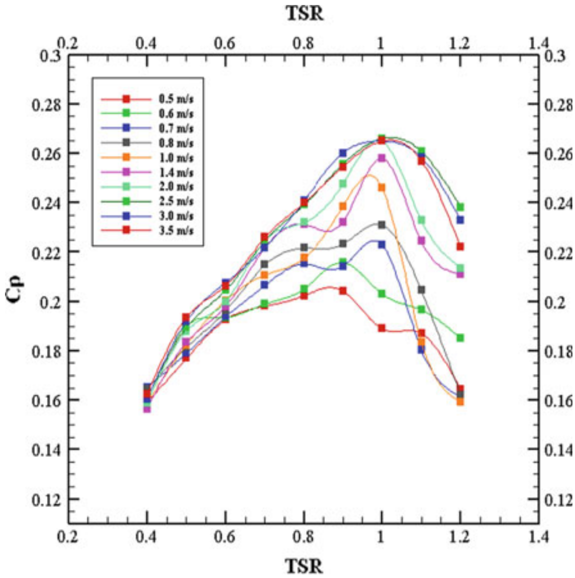


Fig. 10 Effect of flow velocity on the performance of the turbine

5 Conclusion

To obtain the optimum value of the velocity of water, the graph of Cp_{max} obtained at different velocity is drawn and shown in Fig. 11. From the results, it can be concluded

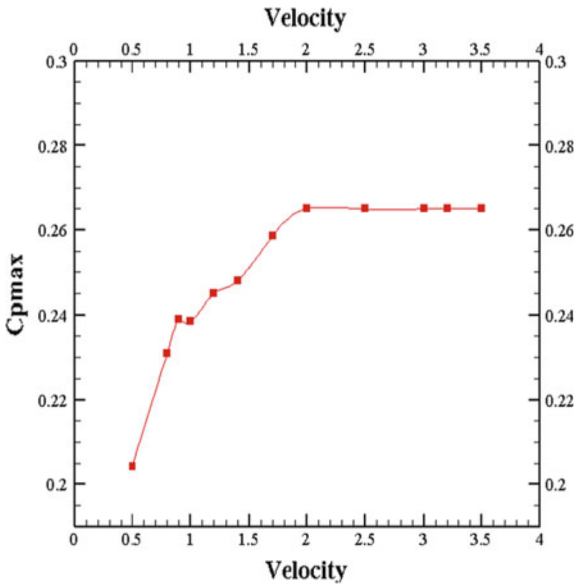


Fig. 11 Obtaining critical velocity for optimum performance of the turbine

that the coefficient of power becomes constant from velocity value 2.0 m/s. Hence, to get optimum performance from the turbine, minimum 2 m/s velocity is required for the considered design of the turbine. It is to note here, the power output will be continuing to increase flow velocity rise even beyond 2 m/s. However, the coefficient of power becomes nearly stagnant beyond flow velocity of 2 m/s.

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