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EXPERIMENTAL ANALYSIS OF A CYCLIC PITCH TURBINE

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ABSTRACT:

Cyclic pitch turbine is a drag based vertical axis fluid turbine which aims to optimize the drag forces that act on its blades. During operation, the turbine blades remain perpendicular to the flow in the drive stroke for maximum positive drag force and remain parallel to the flow in the recovery stroke for minimal negative drag. The blades pitch by 90 degrees between the two strokes using a dual cam mechanism. The blade pitching principle is inspired by bird's wing motion and oar blade motion in rowing. Multiple turbine prototypes have been built and have been tested in air and water for functionality as well as for quantifying the performance. Water tunnel tests are conducted to measure power output at different turbine loads and flow velocities. Experimental setup included turbine speed, position senor and also a torque sensor. Experiments were aimed at understanding the performance of the turbine at different conditions and also finding the best blade configuration for maximum efficiency. The results show the cyclic variation of speed, torque and power output and coefficient of performance (C_P) at different tip speed ratios (TSR). They show that the coefficient of performance is maximum which is 0.17 at around 0.55 tip speed ratio which is typical of drag based turbines and the value reduces with both increase and decrease in the value of TSR similar to conventional turbines.

INTRODUCTION:

Wind turbines and even tidal turbines have been a topic of discussion in past few decades and their importance has been increasing with each year with increase in concerns about global warming and its detrimental effects [1] on the earth's atmosphere. We have seen a proliferation of wind and tidal energy harvesting devices [2] and designs in the last two decades. The cyclic pitch turbine [3] is a fluid turbine that harnesses energy from fluid motion. It is a vertical axis turbine [4] with three flat plates as blades of the turbine equally spaced around the rotation axis of the turbine. The turbine has a dual cam mechanism at the hub which comprises of two concentric out of phase end cams and a pair of out of phase followers for each of the blades. This mechanism allows for the blades to be positioned vertically in fluid during the drive stroke and horizontally during the recovery stroke as shown in Figs. 1a and 1b. As a result, the blades experience maximum drag force during drive strokes at 90 degree angle of attack and avoid detrimental recovery stroke drag with zero angle of attack [5]. The mechanism also allows for the blades to passively pitch between the two strokes gradually without any need for actuators which minimizes the complexity and the cost of the turbine itself. And at the end of each cycle the blades are pitched back to their initial position and are ready to go through another cycle continuously capturing energy from the fluid motion.

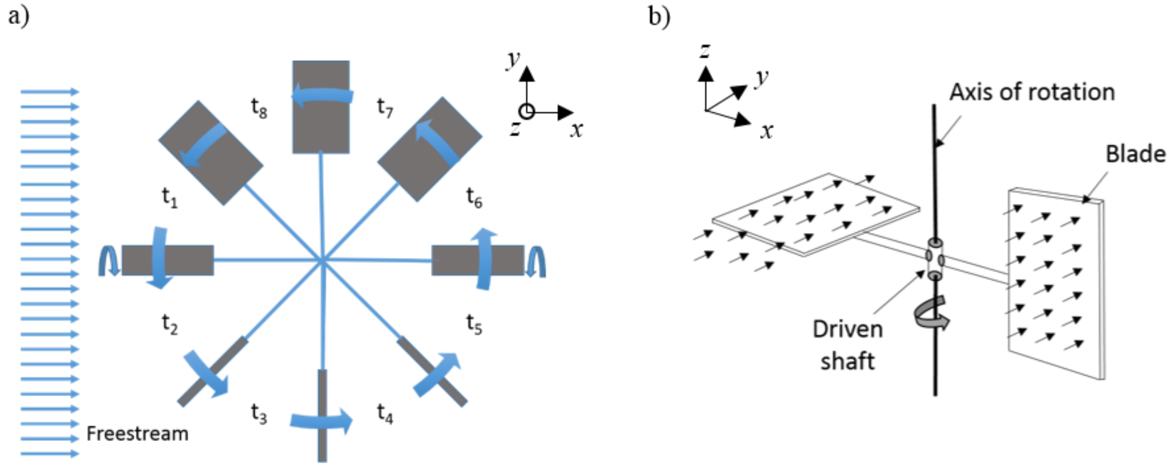

Fig. 1. (a) Top view sketch of flat plate orientations in a rotary cyclic pitching motion showing the flat plate perpendicular (vertical) to the freestream flow between t_2 and t_5 (drive stroke), and a parallel (horizontal) to the freestream flow between t_6 and t_1 (recovery stroke). The plate rotates from horizontal to vertical between t_1 and t_2 and from vertical to horizontal between t_5 and t_6; (b) 3D view showing one flat plate in the drive stroke and the other in the recovery stroke.

The cyclic pitch turbine is a drag based turbine and functions at tip speed ratios lower than one and hence rotates at very low rpm values. This also allows the turbine to have higher torque and function at low fluid velocities where vertical axis turbines might not be able to perform well. The turbine performs equally well in two directions (0 and 180 degree) which avoids a yawing mechanism as seen in horizontal axis turbines. This makes the turbine suitable for ebb and flow of the tidal currents. The turbine has additional advantages [6] as seen in other vertical axis turbines where the generator and the required gearing can be placed at the bottom of the tower making them accessible to repair and maintenance. Also vertical axis turbines are known to increase the energy density because of a rectangular cross section instead of circular blade sweep area.

PROTOTYPE:

The turbine prototypes are built using 3D printers and is designed to fit in the available water tunnel facility. It is a 3 bladed turbine with blades of 160x60 mm rectangular cross section. Two prototypes are built. The first one has a sliding cam follower mechanism and the second one shown in Fig. 2 has a roller mechanism which is expected to reduce friction in the mechanism.

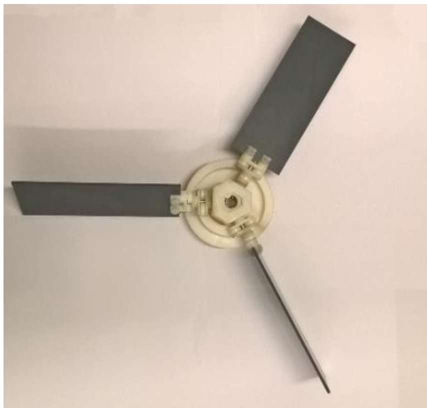

Fig. 2. Top view of the cyclic pitch turbine with roller cam follower mechanism showing the three rectangular profile blades.

EXPERIMENTAL SETUP:

Experiments are conducted in a 22 inch wide closed recirculating water tunnel shown in Fig. 3 where the pump speed can be adjusted to control the free stream velocity of the water. The water tunnel consists of two layers of honeycomb structures which straighten the flow and reduce turbulence. The water then travels through a section of gradual decrease in cross section area which increases the water velocity. The section after the converging cross section region is the test section. The turbine is suspended close to the mid portion of this long test section using suspended column vertically from the top. A 22 x 22 inch test section is used for the experiments.

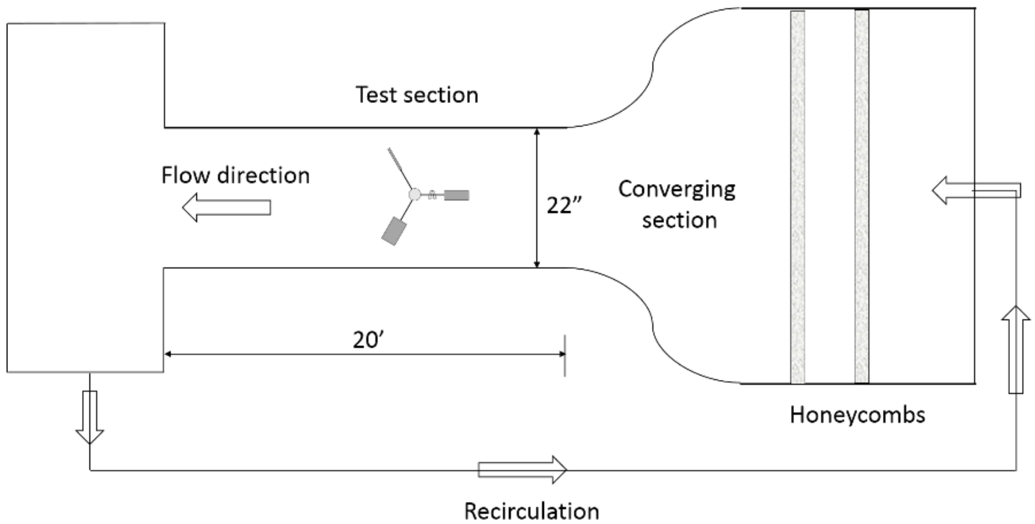

Fig. 3. Top view sketch of the water tunnel showing the different sections of the turbine, tunnel dimensions and the suspended turbine.

The free stream velocity of the fluid flowing in the water tunnel is calibrated using PIV technique for different pump settings and used in the calculations. The experimental setup to measure the rpm and power output from the turbine is described below.

The turbine is suspended from a vertical column and fixed using clamps to prevent rotation and other unnecessary motions. The drive shaft of the turbine is coupled to a steel shaft through a flexible coupling to allow for linear and angular misalignments between the two shafts. The metal shaft passes through a hollow optical rotary encoder with 1200 ticks which measure both the instantaneous rpm as well as the time taken for 1 complete rotation. This helps to observe the cyclic velocity variation of the turbine which is due to the changing relative velocity between the turbine blade and the fluid velocity at different azimuthal angles as well as the average rpm. The shaft on the top end is connected to a static torque sensor which can measure up to 1 N-m with an accuracy of 0.001 N-m. Since the torque sensor is of static type, a slip disk type clutch was machined and it lies in-between the torque sensor and the shaft. It consists of a rotating part fixed to the shaft and a stationary part fixed to the torque sensor and the torque gets transmitted through contact frictional forces between the two parts. A nut-bolt-spring systems lies below the slip disk clutch system to adjust the load on the turbine. As the nut is unscrewed from the bolt, it elongates and thus compresses the spring. Since the force applied by a compressed spring is directly proportional to compressed length, the traverse of the nut/screw is directly proportional to the force applied on the slip disc clutch system and hence the friction and torque transmitted from the shaft to the torque sensor. Thus we can adjust the load on the turbine. Shaft collars are clamped onto the shaft to prevent the parts from sliding up and down on the shaft.

EXPERIMENTS AND RESULTS:

Functionality test is conducted first on the first prototype both in air and water to successfully prove the ability of the turbine to harvest energy from both air and water flows. This test was performed without any measuring devices. The turbine starts to rotate as soon as air and water movement was turned on. In order to measure the performance of the turbine instruments are added to the setup to measure parameters like turbine velocity and power output from the turbine.

The torque output is calculated using Eq. (1) and it is proportional to the current generated and the motor constants. The power output is calculated using Eq. (2) and is proportional to the product of square of the current and the circuit resistance. These are used to observe the characteristics of the turbine without the rpm sensor by using a sensitive electric generator to produce power when the fluid rotates the turbine.

$$T = \emptyset K_T I \qquad (1)$$

$$P = I^2 R_h \qquad (2)$$

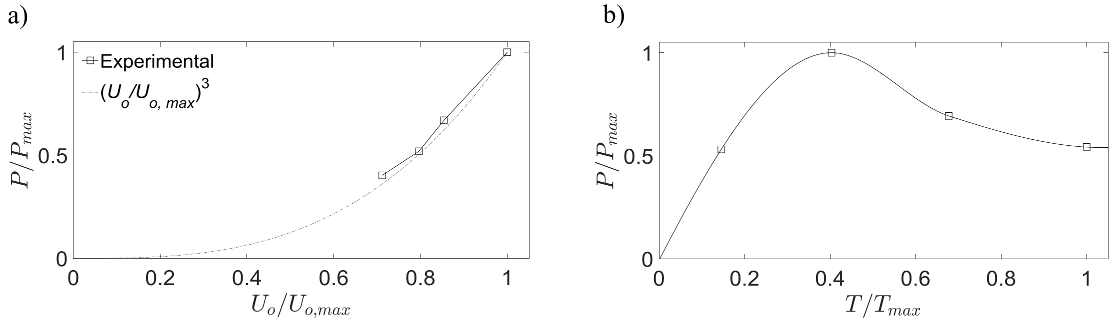

Fig. 4. Cyclic pitch turbine power output variation a) with flow velocity and b) applied torque on the turbine.

Figure 4a plots power output from the turbine for different free stream fluid velocities and it indicates that the power output increases monotonically and matches closely with cubic power of fluid velocity. Blockage factor, which tends to increase the local fluid flow velocity and hence the power output, is not considered in calculating the output power, but has been calculated to be around 3% based on the turbine blade's projected area. Figure 4b plots power output from the turbine for different magnitudes of load on the turbine applied through a rheostat. It shows that the turbine captures maximum energy from a fluid flow at an optimum load/torque value. The two plots shown in Figs 4a and 4b, show typical wind turbine characteristics. They also conform to the results obtained using theoretical formulation [7] based on drag forces on the turbine blades.

Static torque at various azimuthal angles for the two turbines one with bearings on it followers and one without are plotted in Fig. 5 and it is observed that static torque values and as a result, the static torque coefficient, C_{TS}, calculated using the Eq. (3) of the turbine has three peaks and three troughs due to the presence of the turbine three blade configuration. This peak and trough behavior is typical of any drag or vertical axis turbine where the angle of attack changes in a cyclic fashion with maximum torque being generated at once every rotation about the turbine axis. The static torque plot gives the starting torque of a turbine and it is shown in the plot that the its magnitude is positive at all azimuthal angle locations which indicates that the turbine is self-starting and starts to rotate irrespective of the turbine blade positions. The maximum starting torque is seen at three azimuth angles which are 0, 120 and 240 degrees which are where the one of the three blades is perpendicular to the flow. It must also be noted that these three angles of the turbine form the same configuration since each blade is 120 degrees from each of the adjacent blades and the three blades are equally spaced around the turbine axis. Equation (3) is the ratio of static torque, *T* produced by the turbine to the total available in the fluid flow for the given area, *A* of the turbine, and for a given fluid flow. Comparing the two plots Fig. 5a and 5b, it can be observed that adding bearings to the cam follower mechanism reduces friction and increase the torque generated by the turbine which directly relates to the power output and the efficiency of the turbine.

$$C_{TS} = \frac{4T}{\rho A D U^2} \qquad (3)$$

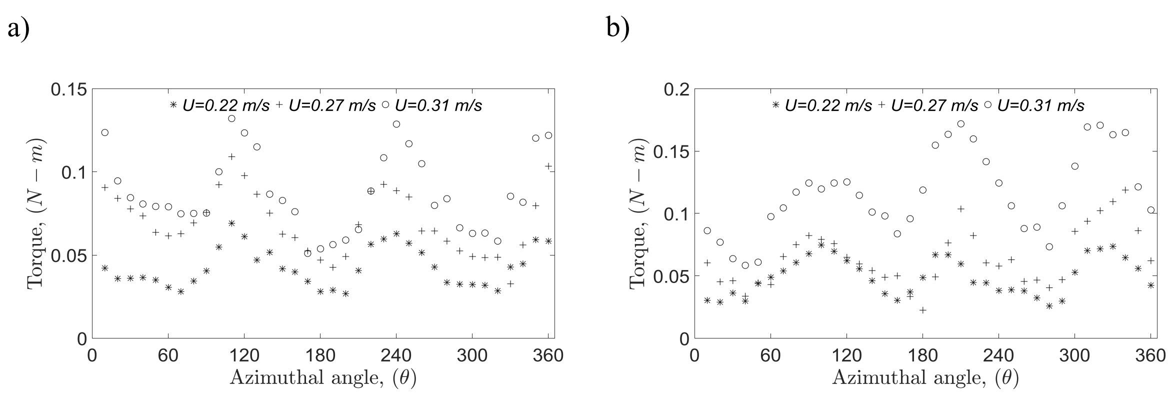

Fig. 5. Cyclic pitch turbine static torque variation with azimuthal angle for different flow velocities for a) sliding cam follower and b) roller cam follower mechanism.

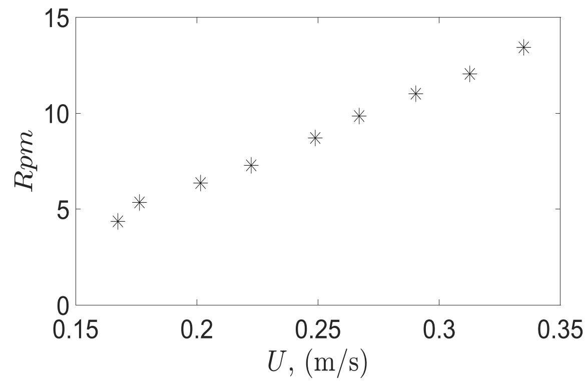

Fig. 6. Cyclic pitch turbine average free rpm output variation with different free stream velocities

The Cyclic pitch turbine was allowed to rotate without any load and the turbine speed was measured by varying the free stream velocity in steps. Figure (6) indicates that the rpm of the turbine under no load increases continuously and monotonically with increase in fluid velocity which is typical of any fluid turbine. Though the turbine is undergoing free rotations and does not generate any power, frictional torque still exits and consumes some power. A similar result is observed when a constant load acts on the turbine and the fluid free stream velocity is increased. From the plot, it can be calculated that the rpm increases at a constant rate of about 50 rpm per unit increase in fluid velocity at while undergoing free rotations.

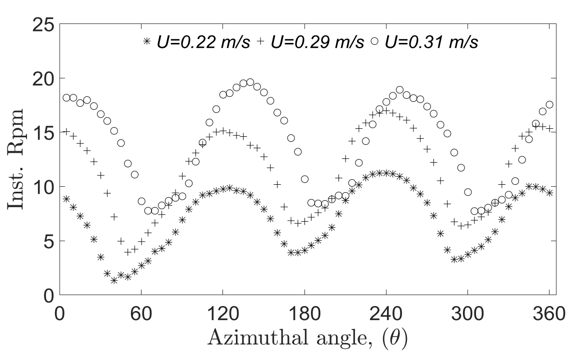

Fig. 7. Cyclic pitch turbine instantaneous rpm variation with different free stream velocities for small load values.

The turbine speed is measured in rotations per minute and is plotted above for the second prototype which has the bearings in the mechanism. The instantaneous rpm of the turbine is measure at various azimuthal locations for a period of time and averaged to obtain Fig. 7. The three bladed configuration is obvious from the plot because of the three peaks and they are located at azimuthal angles when each of the three blades is individually perpendicular to the fluid flow which are 0, 120 and 240 degrees. Also a steady increase in observed in the average rpm value as well as the instantaneous values of rpm at all azimuthal locations with increase in fluid velocity for similar values of coefficient of power. It can also be observed from the figure that as free stream velocity of the fluid increases the location of peaks and the troughs in the rpm values experience a slight shift to the right which means that turbine and the blades experiencing the drive stroke speed up slightly later in time.

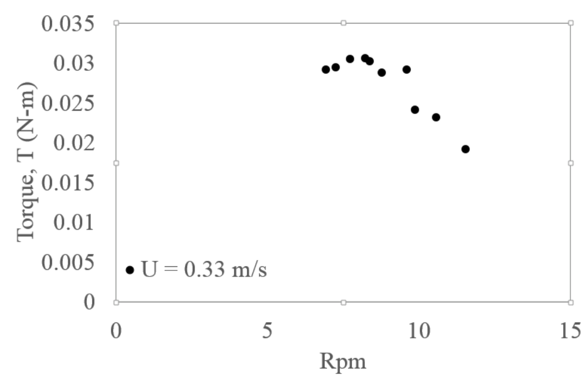

Fig. 8. Cyclic pitch turbine average torque output variation with turbine speed at a fixed free stream velocity.

The turbine torque output shown in Fig. (8) which shows that the turbine needs to run at an optimal rpm value in order to generate maximum torque. And the turbine speed depends on the torque load on the turbine. The turbine power output, P given by Eq. (4) needs to be optimized which is given by a non-dimensional parameter called coefficient of power, C_p given by Eq. (5) which is a function of tip speed ratio, TSR [8] given by Eq. (6). And the projected area of the turbine, A is given by Eq. (7).

$$P = T\omega \qquad (4)$$

$$C_P = \frac{2P}{\rho A {U_o}^3} \qquad (5)$$

$$\lambda = \omega R/U \qquad (6)$$

$$A = HR \qquad (7)$$

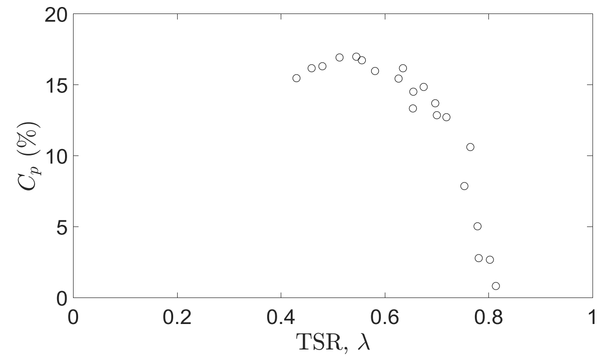

Fig. 9. Cyclic pitch turbine coefficient of power as a function of tip speed ratio for a rectangular blade profile.

The non-dimensional plot which measures turbine performance is shown in Fig. 9 for the cyclic pitch turbine and is the characteristic curve of the turbine. It is plotted between the coefficient of power and the tip speed ratio and it shows that a rectangular blade turbine performs at its maximum efficiency of 17% at a tip speed ratio of about 0.55 which is lower than 1 and typical of drag based turbines [9]. On either side of that optimal TSR, the efficiency drops quickly. Also it can be observed that the turbine generates almost zero power at about 0.85 TSR. It means that the total forces acting on the turbine at those and higher tip speed ratios sum up to zero and/or counteracted by the frictional forces and prevent the turbine from generating any useful power. Several experiments were conducted to observe the effect of using blade shapes other than rectangular but maintaining the same blade area which included trapezoidal with the shorter as well as the longer side facing towards the root.

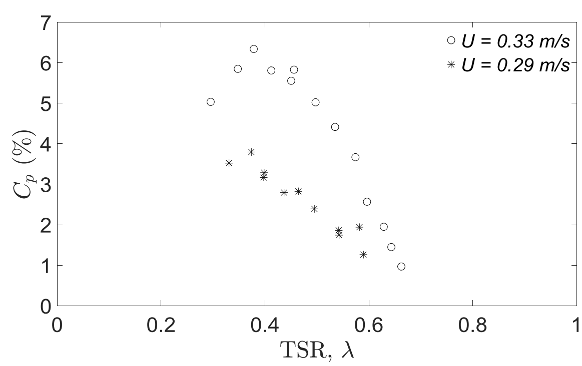

Fig. 10. Cyclic pitch turbine coefficient of power as a function of tip speed ratio for a trapezoidal blade profile with the same area.

Figure 10 shown above is the C_p curve for a trapezoidal blade with the shorter size of the trapezoid towards the root of the turbine. The length of the blade was kept constant. Comparing this plot with Fig. 9 shows that the trapezoidal blade produces a lower amount of power since the coefficient of power is much lower. Also a shift in the peak is observed and the C_p curve shifts a little to the left with optimal TSR value around 0.4. The other configuration with the shorter side of trapezoid facing the tip of the blade was also tested and results in lower coefficient of power values and is detrimental to power generation.

CONCLUSIONS:

The cyclic pitch turbine has been tested for functionality and is proved to be capable of generating energy from a fluid motion. Several tests are conducted on the turbine using the indicated experimental setup and results include static torque outputs, instantaneous rpm variation in a cycle, and turbine speed variation at zero load with variation in free stream velocity. The rpm of the turbine varies with the azimuth and has 3 peaks and 3 troughs indicating the 3 bladed configuration. Also the characteristic curve for the turbine is generated which shows that the turbine performs at its maximum efficiency of 17% at a tip speed ratio of around 0.55. The coefficient of power curve is compared to between a rectangular blade profile to that of a trapezoidal profile and the rectangular blades shown superior performance. Several more experiments in the water tunnel and wind tunnel are necessary for a comprehensive analysis of the turbine and to determine the optimal configuration.

NOMENCLATURE

A – Turbine swept area
C_P – Coefficient of power
C_{TS} – Coefficient of static torque
H – Width of blade
I – Mean motor current
K_T – Motor constant
n – Turbine speed in rpm
P – Mean power output in a cycle
R – Length of each turbine blade
R_h – Rheostat resistance
T – Torque output
U_o, U – Fluid free stream velocity
ρ – Density of fluid
λ – Tip speed ratio
θ – Azimuth angle
ω – Turbine angular velocity, 2πn/60
∅ – Magnetic flux of motor

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