Performance gain of a horizontal axis hydrokinetic turbine using shroud

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***Abstract*— Renewable energy is the alternative to commonly used environment polluting and declining fossil fuels. Kinetic energy of water currents is a highly demanded source of renewable energy and horizontal axis hydrokinetic turbines are one amongst the various devices used to extract this energy. However, the low power density of the technique is an important barrier to its commercialization. While the analytical Betz limit for extracting available kinetic power of water streams is 0.59, a practical power coefficient of 0.45 looks ideal. Although the Betz limit cannot be overcome, a shroud placed around the rotor could increases the C_P. This paper presents results of an experimental work done on two different shrouds to study how much a horizontal axis hydrokinetic turbine gains using shrouds.**

Keywords—hydrokinetic turbine; shroud; performance

I. Introduction

Horizontal axis hydrokinetic turbines are one amongst the various devices which can be used to extract the kinetic energy of water streams. They are meant to be placed in water streams of rivers or oceans and generate electricity in the order of 15 to 1,000 kW without making significant changes to the environment. In horizontal axis hydrokinetic turbines, like in its counterpart wind turbine technology, the turbine axis holding rotor blade is horizontal and the turbine output power, P, is a function of fluid density, ρ, area swept by the rotor blade, A, and flow velocity, V:

$$P = C_P \tfrac{1}{2} \rho A V^3 \qquad (1)$$

The performance coefficient, C_P, in (1) is a nonlinear function of tip speed ratio, TSR, which is the ratio of linear velocity of rotor blade tip to the water stream velocity:

$$TSR = R\,\omega / V \qquad (2)$$

Radius of the rotor blade in (2) is represented by R where its multiplication by rotor blade rotational speed ω gives the blade tip linear velocity. Maximum theoretical value of the power coefficient, C_P, for any kinetic turbine is 16/27 and is called Betz limit. Coefficient of performance, C_P, when plotted against tip speed ratio, TSR, is recognized as performance curve of a turbine. Performance curve for different types of kinetic turbines has a general shape of a dome and varies for different turbine topologies and blade pitching arrangements. A flat performance curve indicates that the turbine is capable of providing its maximum power over a wide range of rotor RPM while a turbine with a sharp curve experiences performance drop when rotor runs slightly faster or slower than the narrow optimum RPM range. An ideal rotor blade would maintain maximum power for nearly all tip speed ratios and reach the Betz limit asymptote in large tip speed ratios. For hydrokinetic turbines this is done by the power converter electronics which varies the imposed load to achieve the optimal rotation. Although renewable nature of hydrokinetic energy is attractive, the low power density, technical difficulties due to need for submersible gearbox and generator, in addition to potential threats hydrokinetic turbines might pose to marine species life currently make it economically challenging. Among all challenges facing hydrokinetic turbine technology the low power density of turbines seems to be the most important barrier to its commercial utilization. While the analytical Betz limit for extracting available kinetic power of water streams is 0.59, a practical power coefficient of 0.45 looks ideal. Wind turbines tried taking advantage of shroud surrounding the turbine rotor blade in order to improve overall performance. More practical considerations overcame the advantages of the shroud and the idea didn't become commercially operational in wind turbine industry. However, it seems that a shroud due to some beneficial aspects like nearly fixed flow direction, available only for hydrokinetic turbines, has the potential to serve the hydrokinetic turbine technology. A shroud reduces blade tip-loss and increases the water velocity through the turbine rotor which comes with a power of three in the output power (1).

Although wind turbine technology has been studied and employed to generate electricity worldwide, extracting energy using hydrokinetic turbines is a relatively new technology. Literature is rich of both experimental and theoretical works done on ducted and un-ducted wind turbines [1], however little works are reported on hydrokinetic shrouded turbines. M. J. Khan et al. in their review paper [2] discussed different concepts for harnessing kinetic energy of water streams such as river currents and ocean tides including turbine and non-turbine systems and provided a technology status review of the field. In their study of 76 different turbine systems it was found that attentions are mostly devoted to horizontal and vertical

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axis turbines with respectively 43% and 33% of application. Good classifications for hydrokinetic turbines and augmentation channel schemes are discussed. Different channel and diffuser shapes which can be used to enhance the output power of hydrokinetic turbines along with advantages and disadvantages of horizontal and vertical axis turbines are also presented. In another review arcticle [3], F. O. Rourke et al. concentrated on tidal energy extraction techniques. Starting with explaining origin of tides, tidal energy is then stated to have two components of potential and kinetic energies and consequently tidal power facilities are divided into two categories of tidal barrages to extract the potential energy and tidal current turbines to extract the kinetic energy of tidal streams. After reviewing current status of tidal barrage technology, paper focuses on the kinetic turbines providing a review on the technology status to the date and mentions some key challenges facing the technology development. Several technologies from vertical axis to different shrouded or open center horizontal axis turbines in design stages or in pilot operations are reviewed. Based on the existing operational devices the authors assume that horizontal axis turbines might be the optimum configuration for tidal current energy extraction. Numerical and experimental works have been done on the design of horizontal axis hydrokinetic turbine rotor blades. In their paper [4], W. M. J. Batten et al. numerically studied effect of some turbine rotor design parameters such as blade airfoil section and pitch angle on the performance of hydrokinetic turbines. Using XFoil code they studied stall and cavitation characteristics of a few airfoil sections. Blade element momentum theory (BEM) was used to develop a numerical method employed to study and compare performances of turbine rotors with two different airfoil sections and pitch angles. The authors in their other work [5] presented results of experiments carried out on a marine current turbine scale model (80 cm three bladed rotor) in a cavitation tunnel and a towing tank. Effect of yaw angle, immersion depths, and an adjacent rotor blade on the turbine performance and also possible locations of cavitation inception are studied in their paper. A few numbers of research can be found in the literature conducted on shrouded horizontal axis hydrokinetic turbines and this is while different inconsistent results have been reported in the existing works for the effect of shroud on turbine performance. C. J. Lawn in a numerical study using a one-dimensional theory modeling ducts as contractions or expansions, looked into performance augmentation of water turbines using shrouds [6]. Based on the results obtained an optimum shroud is said to be capable of making a power enhancement of 30% and it is stated that claims in the literature for greater power enhancement of 2 or 3 times is associated with non-optimum conditions of un-shrouded turbines. In another numerical study [7], D. L. F. Gaden and E. L. Bibeau looked into compensating the low power density of the technology by enhancing performance of horizontal axis hydrokinetic turbines using shroud. Commercial code ANSYS CFX is used to perform numerical simulations and turbine is modeled as a momentum source region. Shrouds consist of a cylindrical section followed by a diffuser were used and an optimum shroud was found to have 20° diffuser angle and 1.56 area ratio. The optimum shroud is reported to enhance power output of the turbine by a factor of 3.1. In another research F. Scherillo et al. performed numerical and experimental studies on horizontal axis hydrokinetic turbines using shroud [8]. Numerical simulations were done in ANSYS Fluent and experiments were carried out in a wind tunnel and a towing tank. A diffuser having airfoil cross section with 26° pitch angle and exit to throat area ratio of 1.71 was found to be the optimum shroud. Experiments were performed on the optimum shroud and also one with 23° pitch angle. Effect of turbine yaw angle on its performance is also studied and it is stated that turbine performance does not change for yaw angles of less than 10° when shroud is used. Optimum shroud was found to double turbine maximum power coefficient which is stated to be an amount of 7% when diffuser diameter rather than the rotor blade diameter is used to calculate C_P.

This paper presents results of an experimental study carried out on a scale model horizontal axis hydrokinetic turbine using shrouds. A horizontal axis scale model turbine along with two different shrouds one having a convergent-divergent section and the other having a straight diverging wall are designed and fabricated. Series of experiments have been done on the test facility in three different water velocities in the variable speed water tunnel at the University of Manitoba. Discussions are made to find is it practically better to use a rotor blade encased in a shroud or to use a larger size blade without shroud.

II. Experiments

A. *Turbine Scale Model*

Depicted in Fig. 1 is CAD drawing of the horizontal axis hydrokinetic turbine scale model designed and fabricated by the author for tests in water tunnel. Turbine consists of an 8 inch educational wind turbine rotor blade (KidWind Project Inc.) with flat bottom airfoil section, a bevel gearbox, a torque transducer, and a DC gear motor. The rotor blade is fixed to a horizontal shaft coupled to a bevel gearbox in order to change the shaft direction to come out of water. The vertical shaft after the bevel gearbox attaches to a torque meter which is itself coupled to a DC gear motor. Torque meter measures the torque exerted by the flow on the turbine shaft and also the rotational speed of the spinning rotor blade. Two different shrouds were designed, manufactured, and used to experimentally study the effect of shroud on performance of the scale model turbine. One is a three inch wide convergent-divergent shroud which is called shroud-1 in the paper. Shroud-1 was rapid prototyped from the CAD drawing and consists of a one inch nozzle shaped inlet with 27° convergence angle followed by a narrow throat and then a two inch wide diffuser shaped outlet having divergence angle of 25°. The other shroud which is referred to as shroud-2 in the paper is a two inch wide straight wall diffuser machined out of acrylic in machine shop with divergence angle of 25°. Turbine and shrouds are designed such that the blade is placed at the minimum cross sectional area of the shrouds which is the throat for shroud-1 and the inlet for shroud-2. The convergence and divergence angles for the shrouds along with the inlet and outlet area ratios are designed based on results of previous numerical works of D. L. F. Gaden and E. L. Bibeau [7] and various experimental results available in the literature [1].

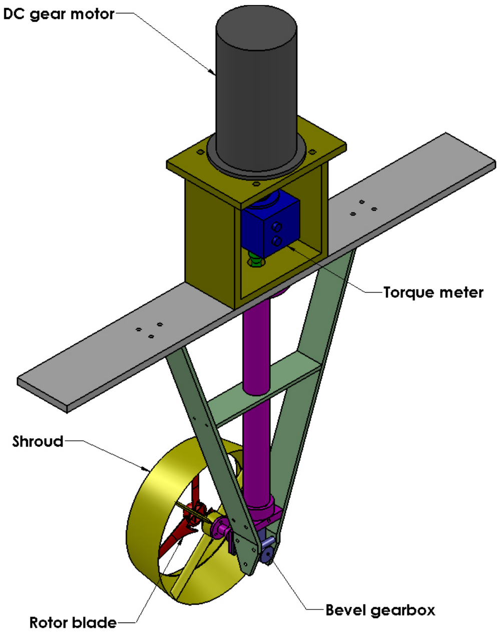

Fig. 1. CAD drawing of the scale model turbine.

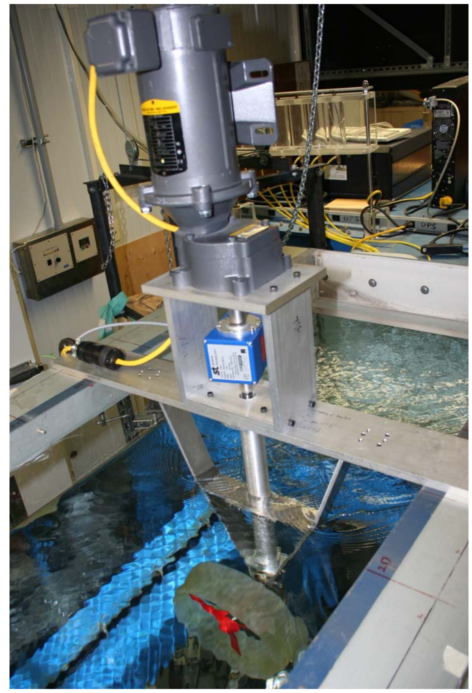

Fig. 2. Physical model of the scale model turbine in water tunnel.

B. *Water Tunnel Tests*

Tests were conducted on the turbine scale model immersed in the variable speed water tunnel at the University of Manitoba (Fig. 2). Torque exerted by the water flow on the turbine rotor was measured via the torque meter in several rotational speeds of the turbine rotor. For each set of experiment water flow was set to a certain speed and tunnel with the turbine in it was left for a while to make sure flow is steady. Since the flow is turbulent (flow regime will be discussed later) steady here means the time averaged flow velocity doesn't change with time. Using a speed controller the DC gear motor span turbine rotor in intervals of averagely 40 RPM starting from zero to a maximum and readings from the torque meter were recorded. Data at each RPM recorded with a sampling rate of 200 Hz after a few minutes of wait time to make sure flow around the rotating blade doesn't change significantly with time. Averaging the recorded data points generates a torque value at each TSR (one water velocity and one rotor speed). The maximum RPM to which experiments continued depends on the flow velocity and went as high as 800 RPM for the maximum flow velocity. Data acquisition was carried out with higher resolutions near the torque peak where changes occur rapidly and with lower resolutions elsewhere. The test procedure was done for three different flow velocities of 0.7, 0.9, and 1.1 m/s. Tests were carried out for the turbine with shroud-1 and shroud-2 the same as for the turbine with no shroud. Water velocity in the tunnel was precisely measured with an ADV (Acoustic Doppler Velocimetry) probe with a sampling rate of 200 Hz. The ADV probe was located at the inlet of the tunnel test section well upstream of the turbine location to make sure velocity measurement is not affected by flow disturbances made by the turbine. The probe also had 30 cm distance to the free surface and to the tunnel walls far enough for the probe to be in the wall boundary layer which was calculated to have a maximum thickness of about 2 cm. ADV gives instantaneous 3D velocity components of a single point in the flow with which all turbulence characteristics can be calculated. Turbulence intensity of water stream in the tunnel was found to be 3.5% in the flow direction for water velocity of 1.1 m/s.

III. RESULTS AND DISCUSSIONS

Shown in Fig. 3 to 5 are performance curves of the scale model turbine tested in water tunnel in three different flow velocities for three cases of turbine without any shrouds, turbine with shroud-1, and turbine with shroud-2. These cases are respectively referred to as Blade, Shroud-1, and Shroud-2 in the diagrams. Power coefficients are experimental torque values times the rotational speed divided by the maximum available hydrokinetic power in the stream. As can be seen in the diagrams C_P starts from zero, takes a maximum at TSR of about 4 and goes down to zero at TSR around 8. This means that the efficient design point for a horizontal axis hydrokinetic turbine is somewhere in the middle of the performance curve and that the power output of turbine does not always increase with increasing TSR. It is also apparent that using shrouds tends to push performance curve to right and slightly bigger tip speed ratios while lifting it and the more the curve is lifted the more the maximum C_P shifts to the right. This can be seen in the diagrams where the maximum C_P for Shroud-2 occurs in a larger TSR than the maximum C_P for Shroud-1 and likewise the maximum C_P for Shroud-1 occurs in a larger TSR than the maximum C_P for Blade. It is seen that power coefficients show the same trend of increase with respect to Blade when shroud-1 and shroud-2 are used in the three flow velocities. In all cases shroud-1 increases power output of turbine to some extent while shroud-2 has the greatest effect on it. It is read from the charts that the maximum power coefficients for Blade are 0.33, 0.38, and 0.40 in the flow velocities of 0.7, 0.9, and 1.1 m/s, respectively. These values are respectively 0.50, 0.54, and 0.55

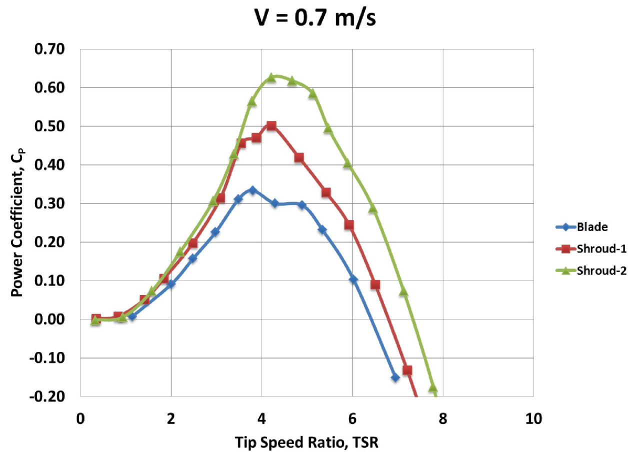

Fig. 3. Performance curves of the turbine in flow velocity of 0.7 m/s.

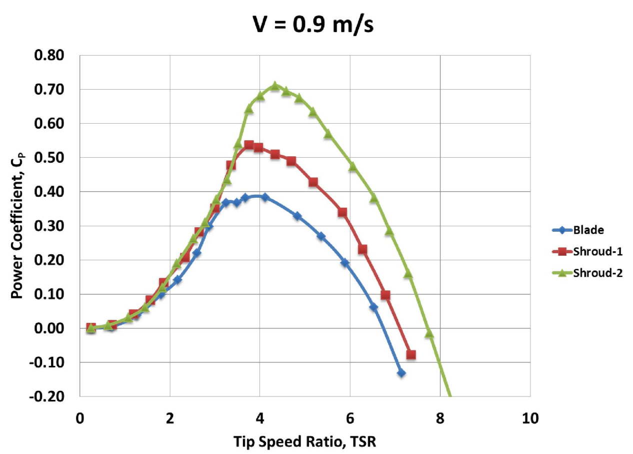

Fig. 4. Performance curves of the turbine in flow velocity of 0.9 m/s.

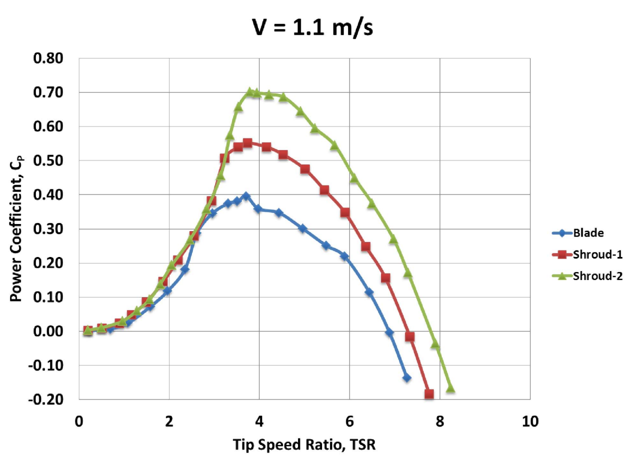

Fig. 5. Performance curves of the turbine in flow velocity of 1.1 m/s.

for Shroud-1 and, 0.63, 0.71, and 0.70 for Shroud-2. Shroud-1 could increase the maximum power coefficients by 52%, 42%, and 38% relative to Blade while shroud-2 could make relative increase of 91%, 87%, and 75% in the maximum power coefficients of Blade in flow velocities of 0.7, 0.9, and 1.1 m/s, respectively. This shows that both shroud-1 and shroud-2 have their greatest influences in lower flow velocities and as water velocity increases this enhancement decreases. This is clear when considering that shroud-1 and shroud-2 had respectively 14% and 16% more influences on the maximum power coefficients of the turbine in flow velocity of 0.7 m/s than in 1.1 m/s. It is observed that the turbine could not reach the Betz limit by utilizing shroud-1 where the maximum power coefficients in flow velocities of 0.7, 0.9, and 1.1 m/s are respectively 16%, 9%, and 7% less than the 0.59 Betz's coefficient. However, overcoming the Betz limit happened by employing shroud-2 where turbine could pass the limit by 6%, 20%, and 18% in the experiment flow velocities 0.7, 0.9, and 1.1 m/s, respectively. It should be noted that this surplus happens because the rotor blade diameter and not shroud-2 diameter is used to determine the power coefficients. It is obvious from the theory that the Betz limit cannot be overcome even with shrouded turbines considering the fact that in calculating power coefficients, shroud diameter should be used in computing the maximum available hydrokinetic power.

Shown in Fig. 6 is the turbine maximum power coefficients plotted against Reynolds number. Re is calculated using the rotor blade diameter and flow velocity. Obviously in all cases of Blade, Shroud-1, and Shroud-2 values of maximum C_P show the same trend of increase as Re increases, however rate of the growth decreases. Therefore it can be anticipated that there would be very little change (if not zero) in the C_P for higher Re which indicates that for instance the maximum achievable C_P using shroud-2 is 0.7 and no more performance enhancements would occur in higher Reynolds numbers. It is worth mentioning that the range of Re in the experiment is close to what is known as transition Reynolds number on a flat plate flow, however since highly swirling flow exists on the rotating rotor blade and behind it flow pattern is completely different from that of a flat plate and therefore flow is considered fully turbulent. Based on the scaling rules Reynolds number charts can be used for large scale tests on full scale prototypes when geometries and Reynolds numbers of scale model and prototype are the same.

An important question here is "what would be C_P of the turbine with shroud if shroud diameter rather than the rotor blade diameter is used in calculating power coefficients?" Since it was shroud-2 which had the greatest enhancement on the turbine performance, results of shroud-2 is used to answer this question. Recalculating the power coefficients of turbine with shroud-2 based on shroud-2 maximum diameter rather than the rotor blade diameter shows that there are very small differences in the performances of turbine with shroud-2 and turbine with no shroud. Ratios of the power coefficients of turbine with shroud-2 in the new calculation to the power coefficients of turbine with no shroud are 1.09, 1.08 and 1.01 in 0.7, 0.9, and 1.1 m/s flow velocities, respectively. This indicates that utilization of shroud-2 when its maximum diameter is used to calculate power coefficients adds only 9% at its best, which is in 0.7 m/s flow velocity, to the turbine performance. This enhancement is quite negligible and only 1% in 1.1 m/s flow velocity. This further implies that if a rotor blade having the same power characteristics of the turbine rotor

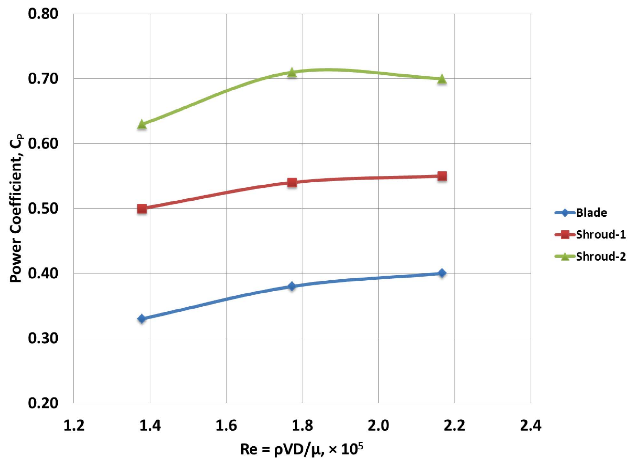

Fig. 6. Maximum power coefficients vs. Reynolds number.

blade but of the size of shroud-2 maximum diameter is solely used, turbine can generate as much power as it does when using shroud-2. This is important in industry side of view since a larger rotor blade compared to a small one with shroud uses less material. However selecting either of the approaches, smaller blade with shroud or larger blade alone, depends on several considerations such as fabrication, transportation, installation, holding structure, electricity generator location, maintenance, and the overall cost. Since from the power generating side of view an extended blade can always substitute for a blade and shroud, employing a shroud might not seem justifiable for just increasing turbine performance and its application would be reasonable only when its existence is structurally necessary such as for holding the coils and magnets of an electric generator in it or to make the turbine less susceptible to flow direction changes. These are considerations on the industry side while water depth, flow velocity, and marine life are considerations on the installation site side of view. Although utilizing larger blade without any shroud could be the superior choice from industry side of view, smaller blade with shroud seems to be the better option for sites with low speed flow, shallow water, and where marine life safety is of great concern. A turbine system with blade rotating inside a shroud introduces less life threat to marine species than a larger unprotected rotating blade for a shroud helps deviating fish from entering the blade.

IV. Conclusions

Effect of shroud on the performance of horizontal axis hydrokinetic turbines was studied experimentally. A scale model horizontal axis turbine along with two different shrouds were designed, fabricated, and experimentally tested to evaluate the influence augmentation of shroud has on the performance of horizontal axis hydrokinetic turbines. The scale model turbine along with a shroud having convergent-divergent profile named shroud-1 and also with a divergent section shroud called shroud-2 were tested in three different flow velocities in a water tunnel. It was showed experimentally that performance of the scale model turbine gains 52%, 42%, and 38% increase by adding shroud-1 in 0.7, 0.9, and 1.1 m/s flow velocities respectively. Power enhancement found to be 91%, 87%, and 75% for shroud-2 in the same flow velocities. This is 44% average increases in C_P by using shroud-1 and 84% average increase in C_P by utilizing shroud-2. Experimental results showed that C_P increases with Reynolds number however rate of the increase decreases with Re. Reason for overcoming the Betz limit was explained to be due to use of the rotor blade diameter rather than shroud diameter in computing the maximum hydrokinetic power of the water stream when calculating C_P. It was figured out that in flows faster than 1 m/s an extended blade generates the same amount of power as the scale model turbine with shroud-2 does when extended blade is of the same size of shroud-2 exit diameter. Discussion was made on industry and installation site considerations for either employing shroud or using a larger rotor blade. This study also provides consistent and reliable experimental results for the two shrouds which can be used as useful validating data for future theoretical and numerical works. The valuable data rare in the literature of the emerging hydrokinetic turbine technology which could help overcoming challenges facing the commercial utilization of the technology.

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