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Characterizing the influence of upstream obstacles on very low head water-turbine performance

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ABSTRACT
The effect of an upstream obstacle – a backward-facing step (BFS) – on a very low head (VLH) water-turbine model has been investigated. The motivation of this study was to quantify the interaction between the VLH turbine and BFS, an abstraction of typical obstacles present in existing channels. Power measurements were obtained for three configurations; the turbine in the absence of the BFS was first examined as a baseline test, followed by configurations with the BFS placed at distances of half a turbine diameter and one turbine diameter upstream. It was found that turbine efficiency is comparable between the case with no step upstream and when the step is placed at one diameter. However, the configuration at half a diameter exhibited an efficiency deficit of approximately 7%. Particle image velocimetry measurements were then used to quantify the flowfield associated with loss in efficiency from the no step to the half a diameter step configuration. It was determined that the losses could be attributed to a highly non-uniform inlet velocity profile, which results in unsteady loading across the turbine blades.

Keywords: Cavity flow; particle image velocimetry; separated flows; very low head water-turbine; vortex dynamics

1 Introduction

Very low head (VLH) and run-of-the-river hydropower technology has become more widely implemented over the past few years due to minimal environmental impact, as shown in the recent work of Müller and Kauppert (2004) and Senior et al. (2010). Although this technology represents one of the best current options for decentralized power generation, development has been hindered by significant implementation costs, of which civil work generally constitutes 40–50% of the total initial cost. Typically, the majority of civil work associated with a low head site is related to the structural modification to accommodate a reduction in the turbine runner diameter, which in turn minimizes maintenance costs. Conversely, this reduction in size of the turbine leads to higher project costs due to extensive inlet and outlet civil work required to transfer water from the inlet to the runner, and in recovering kinetic energy at the runner exit, as discussed in Fraser and Leclerc (2007).

In contrast, the VLH turbine is designed specifically for low head sites but requires relatively little civil work and is easy to install. The turbine consists of an 8-blade Kaplan runner, a fixed set of guide vanes composed of 18 wicket gates, and a permanent magnet variable speed generator directly coupled to the runner shaft. The turbine is double-regulated in that it has adjustable blade pitch and variable speed capabilities. Due to the large runner diameter, the flow speed through the turbine is slow, which reduces the complexity of the required civil structures on either end of the turbine. The corresponding small rotational speeds of the blades also make the turbines fish friendly. The VLH turbine was also designed to operate under variable head and flow rate conditions. Generator outputs range from 100 to 500 kW for head values ranging from 1.4 to 3 m and flows from 10 to 30 m$^3$/s; see Fraser and Leclerc (2007) for more details. The scaled turbine used in this study is modelled after a prototype of this technology installed in Millau, France, as shown in Fig. 1.

Installation of VLH turbines has taken place most prominently at sites that often contain weirs, which are used to control the depth in the upstream channel. These weirs vary greatly in cross-section, although the most commonly encountered geometry in potential low head sites is the ogee-crested weir, as shown in Fig. 2a. This basic geometry, however, often sees a natural
Figure 1  (a) The full-scale VLH water turbine installed in Millau, France, courtesy of MJ2 Technologies and (b) a rendering of the scaled water-turbine model used in this study

Figure 2  (a) Geometry of ogee-crested weir as commonly found in potential low head deployment sites and (b) ogee profile after sediment build-up upstream of the weir

geometric alteration over time due to sediment build-up upstream of the profile. This sediment build-up has the potential to extend the weir structure several metres upstream, as can be seen in Fig. 2b. As a result, the effective shape of the weir can be modelled as the canonical backward-facing step (BFS) geometry, of which the characteristic flow features can be seen in Fig. 3.

Over the past few decades, studies on the BFS problem have been directed towards the understanding of shear-layer characteristics and reattachment properties in the presence of strong adverse pressure gradients. After separation, the flow reattaches downstream and a recirculation bubble is formed beneath the shear layer. Within this bubble, the velocity field is dominated by the presence of flow reversals and stochastic vortical events; see, for instance, Williams and Baker (1997) and Rani et al. (2007). The flow phenomenon responsible for the generation of these vortices in the shear layer is the Kelvin–Helmholtz (KH) instability. For three-dimensional channels with side-walls, the pressure field and size of the recirculation bubble are strongly influenced by the expansion ratio (ER), defined as the ratio of the downstream to upstream channel height \((h_1/(h_1 - s))\) in Fig. 3, as investigated by Le et al. (1997). For small ER values (ER \(\leq 1.5\)), the pressure adjusts quickly, the shear layer has only mild curvature and the wall-normal boundary layer recovers rapidly. However, for larger ratios, the pressure field is more noticeably affected, as can be seen by the larger recirculation region and streamwise distance at which the flow reverts back.

Figure 3  Schematic of the key BFS flow features: the flow separates from the step edge and a recirculation region forms, bounded by the dividing streamline, before reattaching at a distance \(x_R\) downstream
to a conventional wall boundary layer. These phenomena are discussed in detail by Hanjalic and Jakirlic (1998). Also of relevance to such three-dimensional configurations is the spanwise aspect ratio, defined as the ratio of channel width to step height. The side-walls produce complex three-dimensional topologies through the generation of “wall-jet” vortices, which have been shown to influence the mean flow within the separated region by Williams and Baker (1997). For confined channels, Tylli et al. (2002) reported the formation of three-dimensional flow structures, which were found to intensify with increasing Reynolds number in laminar-flow regimes. The introduction of the turbine in close proximity to the BFS resembles a second canonical problem in fluid mechanics, namely cavity flow. Cavity flow possesses many similar attributes to the BFS, and has received similar attention over the years. The flow structure inside a square cavity is dominated by a main captive primary vortex, which is fed by the vortex formation due to the KH instability in the shear layer, as studied in detail by Prasad and Koseff (1989). The key geometries and flow features of the BFS and cavity problem with the VLH turbine placed downstream can be seen in Fig. 3; the turbine inlet face coupled with the upstream step wall forms a cavity-like flow. Also indicated in this figure is the turbine head, which represents the energy extracted by the turbine from the flow.

In this study, the hypothesis that the BFS negatively affects performance of the VLH water turbine has been investigated. For a common turbine orientation angle of 45°, configurations without and with the BFS at distances of half a turbine diameter, \(x/d = 0.5\), and one turbine diameter, \(x/d = 1\), were tested to quantify any loss in performance for each step configuration for an ER of 1.4. Subsequent characterization of the flow field was then performed to better understand how the step altered the inflow conditions to the turbine.

2 Experimental methods

This section outlines the water tunnel and associated methodology used for the power and particle image velocimetry (PIV) measurements, as well as the experimental nomenclature used throughout the document.

2.1 Water tunnel

The experiments were performed in a free-surface water tunnel at the University of Calgary, as shown in Fig. 4. Water is pumped to a main plenum and through four conditioning units consisting of one flow straightener and three fine screens, before being accelerated into the test section through a six-to-one ratio contraction. Turbulence intensity has been measured at approximately 0.3% of the freestream velocity at the inlet of the first test section. The second test section ends with a reservoir, which recirculates water back to the plenum through the axial pump. The test sections total a length of 4 m, with a cross-section approximately 0.45 m in height and 0.40 m in width. The turbine was placed approximately six turbine diameters downstream of the inlet to the first test section and was directly coupled to the power measurement set-up.

2.2 Power measurements

The power data were obtained through turbine torque and angular velocity measurements acquired over a range of flow rates and head levels. The measurement set-up, shown in Fig. 5, consisted of a torque transducer (T22/5NM, HBM) and an optical encoder (ITD69H00, Baumer). A friction brake was used to apply a load to the turbine shaft to measure torque for corresponding
changes in shaft rotational speed for the three flow rates tested. The origin was located at the bottom corner of the step, as shown in Fig. 5, with the x-, y- and z-axes representing the streamwise, wall-normal and spanwise directions, respectively. The x- and y-coordinates were normalized with the step height, s, and the z-coordinate with the channel width, c.

2.3 Particle image velocimetry

Time-resolved PIV measurements were obtained in the xy-plane near the step geometry at a distance of z/c = 0.25 from the side-wall. The experimental set-up consisted of a laser (532GMLH1W, Dragon lasers) of wavelength \( \lambda = 527 \) nm, a high-speed camera (Photron SA4) with resolution of 1024 \( \times \) 1024 pixels and a 28 mm Nikkor lens. The PIV system used in this study can be seen in the rendering in Fig. 6 for a sample measurement in the xy-plane. Note that PIV measurements were acquired in regions away from the centre hub to avoid erroneous out-of-plane correlations. Data were collected in regions of relatively uniform freestream velocity to gain a fundamental understanding of the bulk flowfield entering the turbine. To ensure statistical significance of the data, a minimum of 2000 image pairs were collected at 125 fps and post-processed using commercial multi-pass cross-correlation PIV software (DaVis7.2, LaVision). The water tunnel was seeded with 100\( \mu \)m-diameter silver-coated, hollow glass spheres. These spheres had a Stokes number of approximately \( 2.4 \times 10^{-3} \) and were therefore assumed to follow the fluid accelerations accurately.

2.4 Kinematic similarity

Both the Reynolds number and Froude number affect the performance of scale model water turbines. The two numbers cannot both be scaled simultaneously, so when scaling one must decide whether to maintain one over the other. There is a trade-off between free-surface effects and localized blade effects; variations in the Reynolds number will cause variations in the drag and lift produced by the hydrofoils. The Reynolds number is calculated using the turbine diameter rather than the blade chord length, and indicates a fully turbulent flow. The Reynolds number for the VLH turbine problem is dominated by the Froude number rather than the Reynolds number, because the effect of variation in the Reynolds number on drag and lift coefficients decreases as Reynolds number increases (Sheldahl and Klimas 1981). Assuming this reduced effect of the Reynolds number, linear momentum actuator disk theory can be used for open channel flows to show that power available to a turbine is scalable as a function of the Froude number only (Bell and Mehta 1988). Kinematic similarity between the scaled model and the full-scale prototype was therefore achieved by equating the Froude number (the ratio of velocity squared with gravitational acceleration and turbine head) between each configuration:

\[
\frac{U_p^2}{gH_p} = \frac{U_m^2}{gH_m}
\]

where \( U_{avg} \) is the average velocity entering the turbine, \( g \) is gravitational acceleration, \( H \) is the overall change in potential, and subscripts \( p \) and \( m \) represent the prototype and model, respectively. Equation (1) can be rearranged in terms of the turbine diameter, \( d \), such that

\[
\frac{U_p^2}{U_m^2} = \frac{H_p}{H_m} = \frac{d_p}{d_m}
\]

This ratio of heads and diameters can finally be expressed in terms of dimensionless head (\( H/d \)) for the prototype and model as follows:

\[
\frac{H_p}{d_p} = \frac{H_m}{d_m}
\]

where Eq. (3) was used to ensure appropriately-scaled inflow conditions between the prototype and model. Velocity and head values were scaled to a 4 m-diameter full-scale VLH turbine.
diameter of the scaled-turbine model was 0.27 m, corresponding to a scaling of approximately 1/17. The step was designed with a height of half the turbine diameter to represent conditions encountered in current deployment sites. The upstream velocities of the prototype were required to remain between 0.7–1 m s\(^{-1}\) to meet fish-friendliness standards. The model testing, therefore, had a flow rate range of 0.0125–0.018 m\(^3\) s\(^{-1}\).

2.5 Power curves

The power data obtained are presented here in the form of the standard flow rate quantity, \(Q_{11}\), versus efficiency, power and rotational speed. \(Q_{11}\) is a normalized flow rate ratio, and can be defined as follows, where \(Q\) is the measured flow rate:

\[
Q_{11} = \frac{Q}{d^2 H^{1/2}} \quad (4)
\]

In order to protect the associated intellectual property, the maximum efficiency in a given data set is offset to a value of 100%. The derivation for the efficiency of the VLH turbine is presented in the appendix. The torque and velocity measurements were used to calculate the actual power extracted by the turbine for a given set of operating conditions. This was then normalized by the theoretical power available to determine the efficiency for each set of experimental values.

3 Results and discussion

This section both quantitatively and qualitatively illustrates the differences between the no step and step turbine configurations by comparing results obtained from both the power and PIV measurements, respectively. The maximum uncertainty in the measured power data presented in this study was calculated to be approximately 1.5% relative to the measured power.

3.1 Power measurements

For all the power measurements, head values were obtained approximately five turbine diameters upstream and downstream of the installed turbine from the base of the water channel to minimize read-out errors due to free-surface fluctuations.

Optimum blade pitch angle

Three flow rates, approximately 10, 15 and 20 m\(^3\) s\(^{-1}\) in prototype scaling, were tested without the step in place to determine the blade pitch angle that would yield the highest efficiency. The blade pitch angle could be varied from 0° to 24° in 2° increments. Four angles (10°, 14°, 18° and 24°) were initially tested. The results of these tests showed that 14° produced the highest efficiency for each of the three flow rates. This trend is shown in Fig. 7 for the second highest flow rate, where the difference in efficiency between each blade pitch angle was most prominent. Therefore, all subsequent power measurements for the three turbine-step configurations were taken at this blade pitch angle since overall efficiency trends were deemed to be insensitive to the pitch angle.

Power and efficiency measurements

Important parameters in the characterization of turbine performance include the turbine speed, flow rate and overall head. In this study, turbine speed was matched between each of the three configurations, and three different turbine speeds were tested. Data are presented here for the second highest turbine speed, although similar trends were observed for the other two speeds as well.

When the turbine speed is held constant between the three obstacle configurations, the resulting head loss across the turbine and power generation are the same. This observation is shown in Fig. 8a and 8b, which shows the effect on power output for constant input values of turbine speed and overall head. Unlike the power output and overall head loss, the flow rate was not constant between the three cases. It was found that a higher flow rate was required for the \(x/d = 0.5\) configuration to produce the same amount of power when compared to the other two cases. This necessary increase in flow rate is manifested in the turbine efficiency, which is a function of only the flow rate for a constant overall change in head. The \(x/d = 0.5\) configuration experiences a drop in efficiency of approximately 7%, while the efficiencies of the no step and \(x/d = 1\) configurations are comparable. The effect on the flow rate and efficiency can be seen in Fig. 8c and 8d, respectively.

It is therefore hypothesized that for the \(x/d = 0.5\) configuration, there is a significant change in the velocity profile thus requiring a higher flow rate to compensate for this non-uniform inlet flow. This hypothesis was further investigated using PIV measurements directly upstream of the turbine.

3.2 PIV measurements

PIV measurements were obtained for the no step and \(x/d = 0.5\) configurations to quantify the velocity field directly upstream of
fields and instantaneous image sequences, all of which are normalized with the freestream velocity, $U_\infty$.

**Time-averaged velocity fields and profiles**

The velocity field upstream of the no step and $x/d = 0.5$ turbine configurations and corresponding topological differences between the two can be seen in Fig. 9. The no-step configuration is accompanied by a uniform (irrotational) velocity field approaching the turbine, while the $x/d = 0.5$ configuration is characterized by a separated shear layer, beneath which a primary eddy and other smaller flow structures dominate the flowfield.

The time-averaged flowfields are shown adjacent to the turbine to indicate the streamline topology for each configuration in Fig. 9. The red dots indicate the lowest inlet point of the guide vanes and the black lines represent hypothetical streamlines. For the $x/d = 0.5$ configuration, the position of the dividing streamline suggests that the turbine is subjected to an uneven inlet velocity distribution. This inlet condition was further evaluated through velocity profiles at the dotted black lines indicated in Figs. 9 and 10. The grey area indicates the region of the flow below the lowest point of the turbine inlet. The blue line represents the velocity profile entering the turbine for the no-step configuration, which shows a velocity distribution that decreases by less than 20% relative to the freestream velocity above the grey area. In contrast, the red line, representing the $x/d = 0.5$ configuration, shows a highly non-uniform velocity profile above the turbine and thus understand how the step adversely affects turbine performance. Information is presented in the form of time-averaged velocity fields, velocity profiles, rms-velocity fields and instantaneous image sequences.
the grey area for approximately a quarter of the overall profile entering the turbine.

The velocity profile of incoming fluid results in substantially different kinetic energy distributions between the two configurations. Less energy is directed towards power production for the $x/d = 0.5$ configuration for identical operating conditions, unless the overall flow rate is increased to compensate for this loss in energy. The rms-velocity and time-resolved data are used to investigate the temporal contributions to the loss in efficiency.

![Figure 10](image)

**Figure 10**  Velocity profiles for the no step and $x/d = 0.5$ configurations at a location in the measurement area closest to the turbine inlet. The grey area indicates the region of the flow below the lowest point of the turbine inlet.

Unsteady nature of flow

Low levels of velocity fluctuations were observed for the turbine configuration without the step corresponding to an overall turbulence intensity of approximately 3% relative to the freestream. Conversely, much higher rms-velocity values were observed in the vicinity of the primary eddy for the $x/d = 0.5$ configuration, also shown in Fig. 11. These levels of unsteadiness can be attributed to separate contributions from turbulence production at the downstream wall and meandering of the primary eddy and other smaller structures. This turbulence dissipates energy, which is extracted from kinetic energy in the freestream. The amount of energy available to the turbine is therefore reduced. With the BFS in place, the velocity (based on the rms-velocity data) is uniform for approximately two-thirds of the turbine inlet. The shear layer and separated region are subjected to velocity non-uniformities and small-scale turbulence, which alter the effective angle-of-incidence of the turbine blades near this region.

Further, instantaneous PIV realizations, shown in Fig. 12, provide an insight into the temporal dynamics of the primary eddy. At multiple instances over the duration of the data set, the primary eddy, indicated by the white dot, is convected downstream into the turbine. In the third frame, the vortex has almost completely convected downstream, and can no longer be seen in the field of view. The resulting vorticity, which is convected towards the turbine inlet, results in a non-uniform and unsteady loading distribution on the turbine blades.

![Figure 11](image)

**Figure 11**  Time-averaged rms-velocity field of the $x/d = 0.5$ configuration. Turbine placement is once again representative of its location during the measurements.

![Figure 12](image)

**Figure 12**  Three instantaneous snapshots depicting the convection of the primary eddy downstream towards the turbine over the time interval indicated. The centre of the primary eddy is indicated by the white dot. In the last snapshot, the primary eddy has been convected downstream and is no longer present in the field of view.
4 Conclusions

Power and flowfield measurements were used to characterize the effect of a BFS on the power generation and efficiency of a VLH water turbine. For the power measurements, three configurations with and without the step were investigated. Based on these measurements, a drop of approximately 7% in turbine efficiency was observed with the introduction of a BFS upstream of the turbine at a spacing of \( x/d = 0.5 \), while the no-step and \( x/d = 1 \) configurations has comparable efficiencies.

The flowfield measurements suggest that the significant loss in efficiency at a turbine-step spacing of \( x/d = 0.5 \) is due to a highly non-uniform velocity profile entering the turbine. For the no-step configuration, the velocity decreases by less than 20% relative to the freestream velocity, while for the \( x/d = 0.5 \) configuration, the reduction is more than 70%. This results in an unsteady and non-uniform velocity distribution across the turbine blades, resulting in the measured loss in efficiency for this configuration.

Negligible velocity fluctuations were observed for the configuration without the step, with an average uniform inflow turbulence intensity of approximately 3% relative to the freestream. The rms-velocity distribution for the \( x/d = 0.5 \) configuration, however, is characterized by a highly-unsteady region below the shear layer near the downstream wall and in proximity of the primary eddy. The unsteadiness in this region is primarily a result of the primary eddy and other smaller flow structures, which contain high fluctuating intensities. The dissipation of this turbulence reduces the amount of energy available to the turbine. Instantaneous PIV snapshots reveal the unsteady convection of the primary eddy downstream towards the turbine, which results in increased levels of unsteadiness acting on the turbine blades.

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Appendix

The derivation of the efficiency for the VLH water turbine is presented in detail below. Figure A1 illustrates the control volume chosen for the analysis and Fig. A2 defines the streamlines used in the derivation.

Bernoulli’s equation can be used to find the pressure drop across the turbine by evaluating the equation across streamlines on either side of the turbine. Application of Bernoulli’s equation from points 1 to 2 yields

\[
P_{atm} + \rho g(h_1 - y_1) + \frac{1}{2} \rho u_1^2 + \rho g y_1 = P_2 + \frac{1}{2} \rho u_2^2 + \rho g y_2 \tag{A1}
\]

which can be simplified to

\[
P_2 = P_{atm} + \frac{1}{2} \rho (u_1^2 - u_2^2) + \rho g (h_1 - y_2) \tag{A2}
\]

Applying Bernoulli’s across points 3 and 4 gives

\[
P_3 + \frac{1}{2} \rho u_3^2 + \rho g y_3 = P_{atm} + \rho g(h_4 - y_4) + \frac{1}{2} \rho u_4^2 + \rho g y_4 \tag{A3}
\]

which can similarly be simplified to

\[
P_3 = P_{atm} + \frac{1}{2} \rho (u_3^2 - u_4^2) + \rho g(h_4 - y_3) \tag{A4}
\]

Equations (A3) and (A5) can then be used to determine the pressure drop across the turbine:

\[
\Delta P = P_2 - P_3 = \left[ P_{atm} + \frac{1}{2} \rho (u_1^2 - u_2^2) + \rho g(h_1 - y_2) \right] - \left[ P_{atm} + \frac{1}{2} \rho (u_3^2 - u_4^2) + \rho g(h_4 - y_3) \right] \tag{A5}
\]
giving the final pressure drop expression:

\[ \Delta P = \frac{1}{2} \rho (u_1^2 - u_2^2) + \rho g (h_1 - h_4) \]  

(A6)

The power through the turbine is given by the pressure drop across the turbine times the flow rate through it. This yields the following expression for the actual power:

\[ P_{\text{actual}} = \Delta P A_2 u_2 = \left[ \frac{1}{2} \rho (u_1^2 - u_2^2) + \rho g (h_1 - h_4) \right] A_2 u_2 \]

\[ = \frac{\dot{m}}{2} (u_1^2 - u_2^2) + \dot{m} g (h_1 - h_4) \]  

(A7)

The theoretical power available to the turbine is given by the kinetic energy available in the oncoming fluid and the body force contribution of the fluid potential across the turbine. The ideal power can therefore be expressed as

\[ P_{\text{ideal}} = \int \frac{\rho}{2} \mathbf{u} \cdot \mathbf{u} \, dA_1 + \dot{m} g (h_1 - h_4) \]  

(A8)

which when simplified yields

\[ P_{\text{ideal}} = \frac{\rho}{2} u_1^3 A_1 + \dot{m} gh_1 = \frac{1}{2} \dot{m} u_1^2 + \dot{m} gh_1 \]  

(A9)

Finally, the overall efficiency can be expressed as the ratio of actual power to ideal power, giving the final equation for the efficiency of the VLH water turbine:

\[ \eta = \frac{P_{\text{actual}}}{P_{\text{ideal}}} = \frac{(1/2)(u_1^2 - u_2^2) + g(h_1 - h_4)}{(1/2)u_2^2 + g(h_1 - h_4)} \]  

(A10)

**Notation**

- \( Q \) = expansion ratio (m³s⁻¹)
- \( s \) = step height (m)
- \( u \) = velocities (ms⁻¹)
- \( U_{\text{avg}} \) = average velocity (ms⁻¹)
- \( U_\infty \) = freestream velocity (ms⁻¹)
- \( x \) = streamwise direction (–)
- \( y \) = wall-normal direction (–)
- \( z \) = spanwise direction (–)
- \( \lambda \) = wavelength (m)
- \( \rho \) = density (kgm⁻³)

**References**


