Flow field study in a bulb turbine runner using LDV and endoscopic S-PIV measurements

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Abstract. The flow in the inter-blade channels of a bulb turbine was measured using two different techniques. The first involved a classical laser Doppler velocimetry (LDV) setup whereas the second integrated endoscopic cameras to a stereoscopic particle image velocimetry (S-PIV) system. This paper presents results from both measurement campaigns and also provides some key conclusions based on the two datasets. Before getting into the thick of the data though, the technical aspect of both measurement configurations is addressed. A quick overview of the LDV setup is presented, but the main focus is on the novelties and challenges brought by the use of endoscopic cameras to achieve S-PIV measurements between the runner blades. Endoscopic PIV systems have already led to successful measurements of flow fields in a few studies concerning turbomachinery, especially in aerodynamics. However, to the author’s knowledge, the realisation of such measurements in a hydraulic turbine is a first. After this outline of the techniques used, the results and conclusions are shown. First, the influence of the guide vanes wakes on the runner flow is described. The size, localisation, strength and dissipation of those structures are inferred from the information coming from both measurement techniques. Then, a flow imbalance is assessed circumferentially. On another subject, the blade tip vortices are identified and characterized using the LDV data. The size, position and direction of rotation of those structures are all extracted from the measured flow field. Finally, the PIV data allows the identification of yet another vortex located near the suction side of the blades and originating from the corner between the leading edge and the hub when operating the bulb turbine at part load.

1. Introduction
The actual economic and environmental situation of many developed countries explains the recent enthusiasm to develop the low head hydroelectric market. As an example, in Canada the majority of the viable large hydro sites are already developed, but the available low head hydro sites represent a potential increase of 7 % of the actual country hydroelectric production [1]. Bulb turbines are the perfect candidates for many low head exploitation sites. This tendency motivates the research to improve the knowledge over the various flow phenomena occurring in this kind of turbines. This paper presents the investigation of the flow field between the runner blades of a model bulb turbine using both a Laser Doppler Velocimetry (LDV) system and an endoscopic Stereoscopic Particle Image Velocimetry (S-PIV) system.

Very few experimental studies have tackled the challenge of measuring the flow field in the rotating part of a turbomachine, especially in hydraulic turbines. Some recent investigations have been carried out in pumps [2] [3] and in compressors [4]. However, to the authors’ knowledge, the first flow
field measurements reported in the rotor of a hydraulic turbine are a result of the AxialIT project [5]. The resulting analyses provided information on the effect of operating conditions on the runner flow [5] and allowed identifying some secondary flow structures [6]. The measurements presented in [5] were however limited to a narrow region, about 15% of the inter-blade channel volume.

Following the current state of the art, the research project presented in this document aimed at obtaining a more complete portrait of the flow field inside the runner of a bulb turbine. Thereby, LDV measurements were set up to provide reliable data over a whole radius including near-shroud measurement. Endoscopic S-PIV measurements were carried out to investigate an axial-radial plane covering 62% of the inter-blade channel volume excluding the near-wall regions. The experimental setup and data processing relative to each measurement technique is first presented. Then the discussion focuses on four detected flow phenomena: guide vane wakes, flow imbalance, blade tip vortices and part-load blade leading edge vortex.

2. Experimental set-up

2.1. Test bench and bulb turbine model

The measurements were performed at Laval University Hydraulic Machines Laboratory (LAMH) within the framework of the Consortium on Hydraulic Machines involving hydraulic turbine manufacturers, electric utilities and Canadian government agencies [7]. LAMH's test rig consists of a hydraulic closed-loop that can accommodate axial and radial turbomachine models. The test rig is powered by a pump with a maximal flow rate of 1 m$^3$/s and a maximal head of 50 m. The turbine operates with a rotational speed up to 2000 rpm and a maximal net power output of 225 kW. Vacuum pumps are installed on the downstream tank to control the relative pressure in the test loop to perform cavitation studies. The test rig is set up to perform measurements according to the IEC 60193 standard.

The turbine model which is currently under investigation at the LAMH is a four bladed bulb turbine. A section view of the model is shown in figure 1. The inlet is composed of a bulb held with two piers (upper and lower) and of a distributor aligning 16 guide vanes. The first portion of the draft tube is conical and made of transparent PMMA to allow optical access to the runner and diffuser flow. The second portion is opaque and makes a transition from a circular to a rectangular section. The reference radius $R_{ref}$ is the shroud radius at $z = 0$.

![Figure 1. Left: Section view of the model bulb turbine. Right: Zoom on the runner.](image-url)
2.2. LDV measurement campaign

The LDV system allows measuring two velocity components with four-beams operating in backward scatter on-axis collection mode. There are two LDA (laser Doppler anemometer) inside the probe head and each LDA uses two laser beams at the same wavelength to measure one component of velocity. The probe diameter measures 27 mm with the front lens having a focal length of about 100 mm. The measurement volume of the crossing beams is estimated to be 0.05 × 0.9 mm in size and there are 14 fringes with 3.4 μm separation. A Bragg-cell shifting at 40 MHz is used to resolve directional ambiguity on a 5W Argon-ion laser. The particles used as a reflective medium are silver coated hollow glass spheres. Their diameter of 10 µm and density of 1.4 g/cm$^3$ allow them to accurately follow the flow fluctuations up to 12 kHz.

Figure 2 shows a schematisation of the LDV measurement setup. The view from downstream of the runner illustrates the linear guiding system, the LDV probe and the access window through which the laser beams pass to reach the measurement zone. Six similar setups were realised to measure the six azimuths indicated on the diagram (green dashed lines). As shown in the lateral view, those six azimuths are all in the same $z=0$ plane. Careful alignment of the probe with the measurement axis was realised with a plumb bob, a digital protractor and dedicated laser targets. This fine tuning led to a positioning accuracy of about ± 0.5 mm in all directions.

For each azimuth, 38 radial positions were investigated by acquiring 100 000 samples with an average datarate of 1 kHz at each coordinate. Each acquisition was tagged with the runner position obtained from the simultaneous reading of a digital encoder mounted on the turbine shaft. Thereby, the data were regrouped in 360 bins with 1° steps to calculate phase averages and turbulent fluctuation levels. Accuracy on the phase averaged velocities is estimated to ± 2.5 % of the averaged value and the precision on phase turbulent fluctuation levels is roughly ± 4 % of the calculated intensity.

![Figure 2](image)

**Figure 2.** Left: LDV measurement setup seen from downstream of the runner. Right: Measurement axes in LDV (green dashed lines) and measurement plane in PIV (green plane) as seen from above the runner.

2.3. PIV measurement campaign

The stereoscopic PIV system has two cameras equipped with CCD sensors of 1280 x 1024 pixels operated in dual-frame single exposure mode. They capture the light emitted by a 200 mJ Nd:YAG laser at a wavelength of 532 nm. The cameras are coupled with Karl Storz endoscopes 10-90-67 to deport their point of view inside the turbine and thus avoid any optical distortion caused by the discontinuous water/PMMA interface. The synchronisation of the laser and cameras is ensured by an IDT MotionPro X Timing Hub with a resolution smaller than 20 ns.
Figure 3 shows a schematisation of the PIV measurement setup. A guiding arm (not shown) redirects the laser beam to a probe equipped with a cylindrical lens to generate the 4 mm thick laser sheet. The illuminated plane coincides with the x-z plane and extends from the inlet to the outlet of the blades as depicted in figure 2. The endoscopic cameras are located on both sides of the light sheet in a forward scatter collection mode. One camera is positioned to look from upstream (US) of the runner and the other is looking from downstream (DS). These positions were carefully selected to avoid optical obstruction from the runner blades as much as possible. In compromise, the effective angle between the cameras was approximately 54° instead of the optimal 90° configuration.

A calibration was done prior to the measurements to map the pixel coordinate system of each camera to the coordinate system of the turbine. A two sided calibration target was used, each side presenting a pattern of white dots on a black background. The target was translated along the thickness of the illuminated region using a guiding mechanism and a precision micrometer (±2.5 µm). A total of nine images were acquired for each camera, for nine positions ranging from $y=-2$ mm to $y=2$ mm separated with increments of $\Delta y=0.5$ mm. The identification of the dots on the 18 images led to the determination of the parameters of a pinhole camera imaging model [8] [9]. This imaging model was used to reposition the gathered data on a known coordinate system and also to triangulate the velocity component normal to the measurement plane. The whole calibration methodology led to a positioning uncertainty of ±0.5 mm in all directions.

The PIV investigation was performed within one inter-blade channel: from the trailing edge of blade 1 to the trailing edge of blade 2. Since the channel covers an angle of 90°, images were acquired for 31 equally spaced runner positions (phases) resulting in a circumferential resolution of 3°. Synchronisation of the measurement devices with the runner was ensured by setting a variable delay (depending on the phase under investigation) between the acquisition and a once-per-revolution signal provided by an encoder mounted on the turbine shaft. A quantity of 1500 image pairs was collected for each phase to ensure statistical convergence of the phase locked statistics. The 31 phase-averaged velocity fields allow investigating 62% of the inter-blade channel volume as illustrated in figure 4.
The acquired images were first processed to remove background reflections by subtracting the corresponding phase averaged image. Then, the particles velocities seen from each camera were obtained with an adaptive correlation scheme using moving windows, sub-pixel interpolation, 50\% overlap and a final interrogation window size of $32^2$ pixels ($\approx 3.8^2 \text{ mm}^2$). A mask was applied on the obtained vector fields to discard the invalid data produced by the blade shadows, the wall reflections and the unlit areas. Next, the vectors had to pass three validation tests to filter any possible outliers. First, the ratio of the two highest peaks in their inter-correlation function had to be higher than 1.2. Second, the universal outlier detection [10] was applied with an interrogation size of $7^2$ vectors and a rejection criterion of 1.5. The third test was statistical and verified that any vector was less than $3\sigma$ away from the average. With the calibration model, the remaining vectors from each camera were finally transposed to the coordinate system of the turbine, thus leading to the instantaneous 2D3C vector fields containing 51x50 vectors. The average and standard deviation (i.e. turbulent fluctuation levels) were then calculated over each phase-locked sample. Accuracy on the averaged velocities is estimated to be $\pm 3.5$ \% of the averaged value and the precision on phase turbulent fluctuation levels is roughly $\pm 4$ \% of the calculated intensity.

2.4. Effects of intrusive endoscopes

The use of camera endoscopes offers a multitude of advantages at the cost of two major downsides: loss of image intensity and intrusiveness of the technique. The image quality issue was mostly countered by using a powerful laser system: the images reached an acceptable intensity level with an energy level of at least 30 mJ per pulse. The intrusiveness of the endoscopes presented a greater challenge. The operating points were obtained by imposing the same runner rotation speed and test stand pump rotation speed that were used during the LDV measurement campaign. The operating conditions were affected by the presence of endoscopes inside the turbine. The net head increased by about 3 to 4 \%, the flow rate decreased by approximately 0.7 \% and the efficiency went down by nearly 5 \%. Actually, the model performances were significantly affected according to IEC 60193 standards. Nevertheless, caution was taken to ensure that the measured region was not located in the endoscope wakes. Furthermore, the runner rotation speed was preserved and only a minor mass flow rate reduction occurred (0.7\%) compared to the velocity measurement accuracy (3.5\%). Therefore, locally, in the investigated inter-blade channel, the measured data can be considered representative of the flow field in the inter-blade channel for the targeted operating condition. This was confirmed with
a comparison between the two data sets gathered with the LDV non-intrusive campaign and the endoscopic S-PIV campaign.

2.5. Operating conditions
A total of eight operating regimes from part-load to full-load were investigated during the measurement campaigns. The turbine was operated with a constant runner blade angle of 30.2° and variable guide vane openings. The investigated operating conditions are shown in figure 5. This large database provides test-cases for validation of numerical simulations. This paper only focuses on specific flow features detected while analysing this database: guide vane wakes, flow imbalance, blade tip vortices at full-load and blade leading edge vortex at part-load. The complete velocity fields are not presented in this paper but can be found in [11].

3. Results and discussion

3.1. Guide vanes wakes
In the studied bulb turbine, the guide vanes are located upstream of the runner at a distance of approximately 1 to 2 guide vane chords depending on the opening. The guide vane wakes propagate downstream to the runner and impact the runner inter-blade flow. The wakes of two guide vanes cross the measurement area of both the LDV and PIV campaigns as illustrated on the left of figure 6. The averaged velocity field is locally disturbed. The averaged axial and circumferential velocity components show local magnitude deficits, the radial component presents a large gradient ($\partial C_r / \partial r$). The turbulent fluctuations increase significantly inside the guide vane wakes, about 1.5 to 2.5 times. As an example, figure 6 at right presents the axial turbulence intensity ($\sigma_{C_z}$) from LDV measurement and the radial velocity gradient ($\partial C_r / \partial r$) from PIV measurements for the same radius and runner position. Two guide vanes wakes are visible by the local maxima on $\sigma_{C_z}$ and the local minima on $\partial C_r / \partial r$. Both techniques agree well on the positions and widths of the phenomena.

From these observations, it appears that the wakes have an approximate width of 0.1 $R_{ref}$ and that their radial position varies according to two factors. Obviously, the guide vanes opening angle modifies their position, but so does the runner rotation. Indeed, due to the flow field in the inter-blade channel, the wakes are radially oscillating with every blade passages: they get closer to the hub as the suction side of a blade approaches then back away after being cut by the blade. Additionally, the PIV data indicates that wakes are present from the inlet plane to the outlet plane of the blades. Moreover, based on the value of the local minima of $\partial C_r / \partial r$, the wakes dissipate partially through the inter-blade channel: they are about 1.5 to 2 times less intense at the runner outlet than at the inlet.
3.2. Flow imbalance
Six azimuths were investigated using the LDV systems: 0°, 45°, 90°, 180°, 225° and 270° (figure 2). Based on the averaged velocity profiles, the mass flow rate was estimated for each azimuth and found to be roughly constant within ±1%. Nevertheless, small local differences were noticed on the velocity field at azimuth 45° and 225° compared to the four other azimuths. For example, in figure 7, the axial fluctuations are about 1.5 times larger. This difference is attributed to the wake of the two piers upstream to the guide vanes (see figure 1). However, it remains small on the measured fluctuations compared to the measurement accuracy and undetected on the averaged velocity fields. The velocity field inside the runner is then considered quasi-axisymmetric with 16 spatial oscillations due to the 16 guide vanes.

Figure 6. Left: Schematisation of the origin of the two measured guide vane wakes and their crossing with an LDV measurement axis (red dots) and the PIV measurement plane (intersections of the blue and green sheets). Right: Wake identification using axial turbulence intensity (LDV) and radial velocity derivative (PIV) over a radius located at z=0.

Figure 7. Axial fluctuation levels along a radius located at z=0 for the six investigated azimuths.
3.3. Blade tip vortices

In the present turbine model, with runner blade angle of 30.2°, the blade tip gap is 0.024 R_ref at the leading edge, 0.0008 R_ref at the blade rotation axis (Z = 0) and 0.006 R_ref at the blade trailing edge. This small gap between the tip of the blades and the shroud is the source of a leak flow from the pressure side to the suction side which enrolls itself to generate the blade tip vortex. This phenomenon was described based on refined velocity field measurements in [2] for example where an axial water-jet pump was studied. This rotating structure is located near the blade suction side and the shroud. In the present LDV campaign, the measurement area was extended to less than 0.006 R_ref away from the shroud wall and blades and thus allows characterising this structure. Furthermore, a refined measurement grid was used in the region up to 0.1 R_ref from the wall to ensure a sufficient spatial resolution to capture this phenomenon.

Due to the orientation of the structure and the lack of axial gradient and radial component in the LDV data, the axial velocity profiles proved to be an appropriate parameter to analyse the blade tip vortex (i.e. no vorticity component could be estimated). The local average velocity profile was subtracted from the phase-averaged velocity profile to isolate secondary flow structures. This modified velocity \( \langle C_{z,\text{mod}} \rangle \) is presented in figure 8. The two graphs indicate the presence of two contra-rotating structures when operating the turbine at full-load: the blade tip vortex and a secondary vortex. Part-load operating regimes presented similar but less intense phenomena. At the lowest discharge tested, the structures were either inexistent or too weak to be measured.

![Blade tip vortices characteristics](image)

Table 1 reports the main characteristics identified by inspecting the isolated velocity profiles at full load. The two structures are roughly the same size and located at a similar distance from the shroud wall. However, the secondary vortex is further away from the blade and nearly four times weaker than the blade tip vortex (Table 1). The secondary vortex might originate from the snatching and enrolling of the wall boundary vorticity by the blade tip vortex.

| Vortex    | Axis/Wall Distance | Axis/Blade Distance | Diameter | Rotation around z | Intensity in terms of \( \Delta C_z \) |
|-----------|--------------------|---------------------|----------|------------------|-------------------------------------|
| Principal | 0,03 \( R_{\text{ref}} \)  | 0,035 \( R_{\text{ref}} \)  | 0,06 \( R_{\text{ref}} \) | Negative | 0,9 \( C_{\text{ref}} \) |
| Secondary | 0,04 \( R_{\text{ref}} \)  | 0,070 \( R_{\text{ref}} \)  | 0,07 \( R_{\text{ref}} \) | Positive | 0,25 \( C_{\text{ref}} \) |

3.4. Blade leading edge vortex at part-load

When the turbine operates at extreme part-load, the flow angle on the runner blade leading edge generates a rotating structure near the hub which is convected by the mean flow along the blade suction side. This phenomenon was observed in cavitating conditions as seen in the photographs of
In non-cavitating conditions, PIV measurements were used to detect this vortex at part-load operating condition.

![Figure 9](image-url)

**Figure 9.** Photographs of the cavitating leading edge vortex encountered at part load in the bulb turbine.

Three quantities were analyzed to identify and characterize the vortex. On one hand, the negative circumferential and axial vorticity components ($\omega_\theta$ and $\omega_z$) indicated that the structure is rotating in the same direction as the runner, as indicated by the arrows in figure 10 at left and similarly to the cavitating structure shown in figure 8. On the other hand, the Q criterion [12] was used to identify the shape of the phase averaged vortex with an isosurface of $Q=0.012$. However, the structure is close to the blade on the border of the measured area and only a partial circular section of it could be identified. Thus, an adapted technique was used to extrapolate the vortex centerline assuming a cylindrical shape of the vortex. First, the Q criterion isosurface was discretized on 45 different planes. Then, the centers of the arcs found on each plane were used to generate a cloud of points. Finally, a linear regression was applied to these points to fit a polynomial curve which describes the radial and axial positions ($r, z$) of the phase averaged vortex centerline as a function of the runner position ($\theta$). This curve is given in eq. (1) and represented in figure 10. The centerline trajectory qualitatively matches the cavitating structure and allows assuming that a similar vortex is present with and without cavitating conditions.

![Figure 10](image-url)

**Figure 10.** Left: Q criterion isosurface colored with isocontours of $\omega_\theta$. Right: curve representing the averaged vortex centerline.
4. Conclusion

The flow field between the runner blades of a bulb turbine was successfully measured using joint LDV and PIV measurements. The LDV campaign provided reliable and accurate data. The PIV campaign pushed the boundaries of this measurement technique a bit further by implementing endoscopic cameras to the system. This new feature allowed the recovery of the flow field on a whole axial-radial plane inside the inter-blade channel of the runner for the first time in the hydraulic turbine domain.

Many flow features have been identified in the combined data of both measurement techniques. This paper focused on four of them. First, the guide vane wakes were recognised in both measurements. Their influence is visible from the inlet plane to the outlet plane of the runner blades, but they dissipate to about 1.5 to 2 times their initial intensity. Second, a local flow imbalance was assessed on the turbulent fluctuations due to the presence of the piers. Third, the blade tip vortices present at full-load were characterised with the LDV data. This step led to the identification of a principal blade tip vortex and a contra-rotating secondary structure. And fourth, the blade leading edge vortex was measured at part-load regimes with the PIV system. Its position and direction of rotation were given by the observation of vorticity components and of Q criterion isosurfaces. Three of those four phenomena are illustrated in figure 11 alongside the measurement area: the vortices and the guide vanes wakes. It would be interesting to numerically simulate this flow and observe if each phenomenon can be accurately reproduced and if their effects on the performance forecasts are negligible or not.

Figure 11. Portrait of the flow phenomena occurring in the inter-blade channel of the bulb turbine runner and localisation of the measurement area and axis.

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Nomenclature

- $C_r$, $C_\theta$, $C_z$: Radial, circumferential and axial velocity components [m/s]
- $C_{\text{ref}}$: Reference velocity: $Q/\pi R_{\text{ref}}^2$ [m/s]
- $C_{z,\text{mod}}$: Axial velocity component after subtracting the local average velocity profile [m/s]
- $R_{\text{ref}}$: Reference dimension: shroud radius at $z=0$ [m]
- $\alpha$: Guide vanes opening angle [°]
- $\eta$: Hydraulic efficiency [%]
- $\sigma$: Phase standard deviation [m/s]

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