Numerical Investigation of flow instabilities in Speed No-Load operation of a Bulb turbine

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Abstract. When operating in Speed-No-Load (SNL) turbines function at their synchronous rotating speed with a fraction of discharge at best efficiency point, ready for electrical grid connection. The flow in SNL condition is characterized by high swirling component leading to complex flow structures inside the runner, which generate wide-band pressure fluctuations. When operating in SNL condition, high specific speed turbines, such as axial turbines, can be damaged by significant pressure oscillations attributed to rotating vortex phenomena, alike rotating stall. However, understanding the fundamental flow dynamics behind the phenomena in SNL condition remains challenging due to the stochastic nature of flow. The present research highlights unsteady flow characteristics and flow instabilities in SNL condition of a bulb turbine by URANS simulations based on scale-resolving techniques. The simulation results provide evidence that the development of a rotating vortex array in precession with the runner is a major source of pressure oscillations and torque fluctuations in SNL operation of this bulb turbine.

1. Introduction
A speed-no-load (SNL) operation is a mandatory operating condition for hydraulic turbines when they are operated at their synchronous rotation speed but without power generation ready to be linked to the grid. The turbine's runner flow in the SNL operation is characterized by low discharge and high swirl leading to low-frequency high amplitude pressure fluctuations potentially leading to blade damage and more maintenance downtime[1, 2, 3, 4, 5].

For hydraulic turbines with high specific speed operated in SNL conditions, large pressure oscillations can occur in the runner channels. Pulpitel et al. [6] reported that those fluctuations are associated with columnar vortices extending from the vaneless space to the entrance of the draft tube. Houde et al. [5] studied the inception mechanism of those columnar vortices and demonstrated that their existence is unrelated to the runner blades themselves but are rather an instability of the flow at the exit of the guide vanes. The development of these vortices was mainly studied for axial turbines such as propeller and Kaplan turbines where they appear linked to shear layers at the distributor exit, recirculating regions around the hub and a change of flow orientation at the runner entrance. Their existence in bulb turbines, where the change of orientation of the flow at the runner entrance is minimal, is still questionable.

Therefore, this paper aims to investigate the flow dynamics in the SNL operation of a bulb turbine by numerical simulations through comparisons with experimental measurements carried out within the framework of the BulbT project.
2. BulbT project

The BulbT project is a collaborative project between industrial and academic partners started in 2011, focusing on documenting and analyzing hydrodynamics of bulb units around the best efficiency point (BEP) and during start-up and no-load operations. The project is centered on flow measurements on a model turbine installed on the Laboratory of Hydraulic Machines (LAMH) test rig in Université Laval. Within the BulbT project, a number of numerical simulations as well as experiments have been conducted to outline flow characteristics under off-design operation of bulb turbines [7, 8, 9, 10]. In particular, the experimental measurements including the SNL operation and start-up sequence were performed with on-board pressure measurements on the runner blades, revealing notable pressure fluctuations in the SNL operation [11, 12]. In the present study, the simulation is targeted at the SNL operation presented in [12], where the pressure fluctuations on the runner blade were analyzed but for which the origin could not be confirmed.

3. Simulation setups

The studied bulb turbine has 4 runner blades and 16 guide vanes. At BEP, the turbine features the specific speed $\nu = 0.50$ and the non-dimensional IEC factors $Q_{ED} = 0.76$ and $n_{ED} = 0.91$ [13]. In the SNL operation, the output power is close to zero with the guide vane opening and the discharge respectively at 47.2% and 52.9% of their values at BEP. The numerical simulations are based on unsteady RANS equations. The governing equations of mass conservation and momentum conservation are written as follows;

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{C}) = 0 \quad (1)$$

$$\frac{\partial \rho \vec{C}}{\partial t} + (\rho \vec{C} \cdot \nabla) \vec{C} = -\nabla p + \nabla \cdot (\bar{\tau} + \bar{\tau}_t) \quad (2)$$

where $\rho$, $\vec{C}$, $p$, $\bar{\tau}$, and $\bar{\tau}_t$ represent density, velocity, pressure, viscous stress tensor and turbulent stress tensor, respectively. The equations are discretized by the finite volume method using the commercial flow solver ANSYS CFX 19.1. The simulation domain includes guide vanes, runner,
and draft tube, as shown in Figure 1a. The total number of the nodes are $12.4 \times 10^6$ nodes (5.1 $\times 10^6$ in the guide vanes, 3.9 $\times 10^6$ in the runner, and 3.4 $\times 10^6$ in the draft tube, respectively). The mesh size stemmed from the numerous studies published on BulbT and yielded torque values and pressure fluctuations in good agreement with the experimental measurements as shown in Section 4. Figures 1b and 1c illustrate the location of pressure probe used in the numerical simulations to characterize the flow instability. Probe B1 and B2 correspond to the actual location of pressure sensors mounted on the runner blade and described in Coulaud et al [12]. The distance between Plane 01 at the inlet of the draft tube and the reference level $z_{ref}$ is $0.59 \times D_{ref}$.

The simulations are based on the SAS-SST turbulence modeling approach. This model provides results between classical eddy-viscosity based models (SST) and LES (Large Eddy Simulation) in the unsteady flow region by introducing a source term in the turbulent equation [14]. It has been shown that the model is used for various industrial applications with satisfactory results [15], including vortex phenomena at low discharge conditions of hydraulic turbines [16, 5, 17]. For boundary conditions, the velocity profile is imposed at the domain inlet and zero pressure is given at the domain outlet. For all the wall boundaries, the non-slip wall condition is applied. For the velocity profile imposed at the domain inlet, the time-averaged velocity from URANS simulations of the intake section is adopted. For the entire calculation domain, the value of the non-dimensional wall distance $y^+$ is in the range of $30 \leq y^+ \leq 300$.

![Figure 2. Pressure fluctuation at B1 on the blade suction side (a) and the frequency analysis result (b)](image)

![Figure 3. Pressure fluctuation at B2 on the blade pressure side (a) and the frequency analysis result (b)](image)
4. Pressure fluctuations and developed vortex structures

As reported in [12], dominant pressure fluctuations were observed on the runner blade in the SNL operation. In Figures 2 and 3, the time history of the pressure fluctuations at B1 and B2, and the frequency analysis result (PSD: power spectral density) are shown. Time and frequency are made non-dimensional by $t \times f_n$ and $f/f_n$ where $f_n$ is the rotational frequency of the turbine, respectively. The pressure signals in the experimental measurement at the same operating condition are presented together. Both amplitude and frequency of the fluctuation in the simulation show very good agreement with the experiments. In the frequency analysis result, two dominant peaks are observed at $f'_1 = 0.57$ and $f'_2 = 1.22$. The value of $f'_2$ is close to double of $f'_1$, suggesting that there are possibly twin vortex structures rotating in precession with the runner at a different angular speed.

In Figure 4a, instantaneous vortical structures are highlighted by iso-surfaces of Q-criterion and non-dimensional pressure. The non-dimensional pressure is calculated by $p^* = p/\rho E$, where $E$ is the specific energy of turbine. It is observed that low pressure regions are generated by the inter-blade vortex structures attached to the hub. In addition to these inter-blade vortices, two large low-pressure cores are observed at the runner exit across Plane 01. It appears that these low-pressure cores are also generated by vortical structures according to the distribution of Q-criterion, and these vortices are attached to the hub. The low pressure regions caused by these vortical structures can be clearly observed in the distribution of the non-dimensional pressure at Plane 01 shown in Figure 4b.

Since the hub-attached vortex arrays are in the precession with the runner, these vortices induce periodic pressure oscillations in the draft tube. The simulated pressure signals at the draft tube wall (P1 and P2, see Figure 4b for their locations) are presented in Figure 5a together with its frequency analysis result. In the time history of the pressure fluctuations, it can be observed that the pressure periodically fluctuates, and the two signals simulated at P1 and P2 feature an opposite-phase oscillation. In the frequency analysis result, the dominant peak is observed at the frequency related to the precession of the twin vortex structures $f_r = 0.77$. There is also the convection frequency $f_{\text{conv}} = 0.41$ (the rotation frequency of one vortex structure),

![Figure 4](image-url)

**Figure 4.** Development of vortical structures highlighted by iso-surfaces of Q-criterion and pressure (a) and the distribution of the non-dimensional pressure on Plane 01 at the draft tube inlet (b). The values of Q-criterion and non-dimensional pressure for iso-surfaces are $Q = 1.0 \times 10^5$ and $p^* = -0.5$. 
which corresponds to about half of the precession of the twin vortex structures. In Figure 6, the coherence of the signals at P1 and P2 and the phase of the signal at P1 with respect to P2 are presented. A high coherence is observed at the frequencies corresponding to the convection frequency $f_{\text{conv}}$ and the precession frequency $f_r$ (see Figure 6). The phase differences of the signals are respectively $\pi/2$ at $f_{\text{conv}}$ and $-\pi$ (same as $\pi$) at $f_r$, which validates the rotating mode of the signals. The phase of the convection component in the pressure signal at P1 has $\pi/2$ advance to the one at P2, suggesting that the vortical structures co-rotate with the runner (same direction). Taking into account the rotating direction of the vortex arrays, the estimated convection frequency and precession frequency in the rotating domain should correspond to $f'_{\text{conv}} = 0.59$ and $f'_r = 1.23$, which are sufficiently close to the observed frequencies $f'_1$ and $f'_2$ in the simulation result on the blade ($f'_{\text{conv}} = 0.57$ and $f'_r = 1.22$ in the frequency analysis result, respectively). Hence, the simulation results confirm that these vortex structures are responsible for the measured pressure fluctuations on the runner blades [12] as well as the draft tube.

5. Torque fluctuations
It is shown that the development of the vortex array across the runner induces the pressure fluctuation on the blade. These vortex structures can also induce the torque fluctuations on the runner blades. In Figure 7, the torque fluctuations on each of the four runner blades are presented together with the frequency analysis result of the torque fluctuation at one blade (blade #1). The torque is made non-dimensional by the mean value of the torque at BEP. It
can be observed that the torque fluctuates periodically on all blades with a phase shift. In the frequency domain, frequency peaks appear at $f_{\text{conv}}$ and $f_r$, which are the same as the result of pressure fluctuations on the blade.

In Figure 8, the phase-averaged signals of the torque fluctuation on blades #1 and #2 are presented. The total amplitude of the fluctuation reaches nearly 5% of the torque at BEP. Since there are two vortex structures located in 180-degree shifted locations and the runner has four blades, the torque fluctuations at the blades situated next each other have an opposite phase oscillation, and the blades situated in the opposite location have in-phase fluctuation. Hence, for the present case, the total torque associated with these vortex structures is almost canceled. However, even if the net torque output is relatively constant, the fluctuations on the individual runner blade may not be neglected depending on the combination of the number of vortices and blades. This suggests that it is crucial to understand the inception mechanisms of those vortex arrays to determine the number of vortices and ensure stable operations in the SNL condition.

6. Flow structure in SNL condition of a bulb turbine

In general, the flow at deep part load operation including SNL conditions is characterized by a large recirculating region associated with flow separations, even inside the runner blade channels [16]. For the case of an axial turbine, Houde et al. reported that the development of the vortex arrays is caused by the boundary layer separation on the head cover in the downstream of the guide vanes [5]. In Figure 9a, the distribution of the time-averaged axial velocity on the axial plane is presented. The variable is averaged over 20 runner revolutions. It can be observed that
Figure 9. Distributions of the time-averaged non-dimensional meridional velocity (a) and the time-averaged vorticity in the axial direction (b) on the axial plane

the flow is dominated by the notable recirculating region at the center of the draft tube, which starts from the hub region of the turbine. In the distribution of the time-averaged vorticity $\omega_z$ (in the axial direction) shown in Figure 9b, it can be observed that there are two major regions of high vorticity near the runner hub, which are generated by structures of the vortex array and inter-blade vortices, respectively. The onset of the recirculating flow from the hub corresponds to the region of the inter-blade vortex development, as previously observed in [16]. In the downstream of the inter-blade vortices, the limit of the recirculating region approaches the hub wall, but the flow appears to separate again from the hub wall, which is possibly caused by the boundary layer separation from the hub. This flow separation leads to a large recirculating zone at the center of the draft tube. It can be observed that the vortex array emanates from this flow separation region on the hub, and the strong vorticity region is developed outside the recirculating region where a strong shear layer is created.

7. Conclusion
In the present study, the origins of pressure fluctuations measured on a bulb turbine model operated in SNL conditions were investigated using numerical simulations. The simulation results confirm that the origins of those pressure fluctuations are linked with a vortex array attached to the runner hub. The precession frequency of these vortices is lower than the runner rotation frequency, inducing pressure fluctuations in both draft tube and runner domains. It is shown that the precession of these vortices generates the torque fluctuations on the blade, which cannot be negligible depending on the number of vortices. Moreover, the flow separation from the hub appears linked to the development of the vortex array. Although the development of these vortices may be similar to the vortex rope typically observed at part load operations, further investigations are underway to properly study the inception mechanisms of this vortex array and to identity the factors to define the number of vortices.

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