Numerical study of the flow dynamics at no-load operation for a high head Francis turbine at model scale

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Abstract. No-load operation is among the most damaging operating conditions. Since no energy is extracted by the turbine during this operation, the fluid energy is dissipated by highly energetic and turbulent flow phenomena causing high pressure and strain fluctuations on the turbine components. These highly turbulent flows are complex and, until now, poorly understood. This paper presents preliminary three-dimensional simulations of no-load turbulent flow within a high head Francis turbine at model scale using Computational Fluid Dynamics and the SAS-SST turbulence model. The numerical results are compared with unsteady experimental pressure data acquired on the runner blades of a model turbine. A preliminary study of the numerical results reveals some specific flow behaviors present at no-load operation.

1. Introduction
No-load operating conditions are among the most damaging a hydraulic turbine experiences since nearly all the energy transmitted to the machine is dissipated by highly turbulent flow phenomena [1,2]. This kind of operating condition may reduce the life expectancy of hydraulic turbines by generating high amplitude pressure fluctuations in all their components and, consequently, accentuating the effect of normal fatigue. At this time, the flow phenomena occurring at no-load conditions are poorly understood. Increased knowledge on the flow-induced strain field in the different turbine components at these operating conditions could lead to optimization of the turbine life expectancy and lower maintenance costs. Moreover, since a no-load condition is an intermediate step in most transient events, such as start-up and load rejection, a better understanding of this steady condition could help to understand and analyze complex transient flow dynamics.

This project is a preliminary step to a comparative study between two no-load conditions: the speed-no-load (SNL), where the runner rotates at the synchronization speed with no net power output, and the full-gate opening runaway condition, where the runner rotates at its fastest speed, under a given head, with no net power output. This paper is based on an experimental and numerical analysis of the flow dynamics in a high head Francis turbine at model scale. Some runner blades of the model turbine were instrumented with miniature piezoresistive pressure sensors. An analysis of the experimental pressure signals is performed in the frequency domain to characterize the dominant fluctuations. Numerical simulations based on single phase unsteady Reynolds averaged Navier-Stokes (URANS) equations were performed for the full-gate opening runaway condition only to get a deeper understanding of the flow dynamics.
dynamics. Experimental pressure measurements and engineering quantities, such as the head and the torque, are used for the validation.

In this paper, the objective was a preliminary identification of the main flow behavior at no-load condition using experimental data and numerical results. Also, the numerical setup, for the runaway condition, has been validated using the experimental pressure measurements. Since few studies in the literature have a validation level equivalent to the one presented here, this study brings a relevant contribution to the applicability of URANS simulations for highly turbulent flow prediction. The first phase of the project allows identifying some flow behavior in the runner and the draft tube at the runaway condition. The next step will be to compare the flow topology at SNL and runaway conditions, using numerical results.

2. Experimental data
The experimental study and the validation of the numerical simulations at runaway condition are achieved with pressure measurements on the blades of a model runner performed by the Laboratory for Hydraulic Machines of École Polytechnique Fédérale de Lausanne in 2004 for Andritz Hydro Canada Inc. (at that time, GE Energy Hydro).

As shown in figure 1, the high head Francis turbine model has a spiral casing with 19 stay vanes. The distributor is composed of 20 guide vanes and the runner has 15 blades. The draft tube is composed of a conical diffuser, a converging/diverging elbow and a trumpet diffuser. The turbine specific speed $N_{QE}$ is 0.1, as calculated following the IEC 60193 standard [3].

What is called speed-no-load (SNL) in this paper is, in reality, a quasi-no-load condition since 2% of the torque at the best efficiency point (BEP) is still produced. This operating point is characterized by $Q_{nD}/Q_{nD\text{ BEP}}$ of 0.11 and $E_{nD}/E_{nD\text{ BEP}}$ of 1. The full-gate opening runaway condition is a real no-load condition. The residual torque is equal to the mechanical losses. It is characterized by $Q_{nD}/Q_{nD\text{ BEP}}$ of 0.46 and $E_{nD}/E_{nD\text{ BEP}}$ of 0.33.

Pressure is measured with 21 miniature piezo resistive pressure sensors fitted on four blades (blades 1 and 8 pressure side and blades 2 and 9 suction side) covering two hydraulic channels with almost no geometry alteration. The pressure sensors have a 3 mm diameter. The experimental technique and the measuring instruments are detailed in [4]. The same runner as the one in [4] but with modified leading edges is used to perform the measurements. A relative calibration was performed on this modified runner mounted in the test rig filled with water. The static pressure on the blades was increased gradually up to 5m water column and the output voltages of the sensors were recorded during this operation. Only 21 of the original 32 pressure sensors were still functional. The resulting measurement error is around 0.5% for values above the vapor pressure.

The pressure measurements during runaway and SNL were performed at a sampling rate of 20 kHz during 3.28s. Five and three sets of data are available at runaway and SNL respectively. The averaged flow rate, energy, runner rotation speed and the static suction head $H_s$, which represents the distance in meters from a reference position to the tail water elevation, are available for each operating condition.

![Figure 1. Numerical domain for the high head Francis turbine.](image-url)
3. Numerical Strategy
The following methodology is applied to the full-gate opening runaway condition. The methodology for the SNL condition is currently under investigation.

The URANS equations are solved with the finite volume commercial solver ANSYS CFX 18.1. The Scale-Adaptive Simulation (SAS-SST) turbulence model is used for its capacity to resolve additional spectral content in unstable flows in comparison with a k-ε or a k-ω SST model, for example. The use of the von Karman length scale in the turbulence scale equation allows to reduce locally the level of eddy viscosity depending on the flow dynamics. The lower eddy viscosity level is limited by the spatial discretization. The chosen mesh resolution resulted in the use of wall functions to treat the near wall flow. This was done in the interest of reduced computational cost. Even if cavitation was apparent in the experiments, single phase simulations were performed for the preliminary simulations.

To solve the URANS equations, the time stepping algorithm uses an implicit second order backward Euler scheme. A blended advection scheme, referred to “high resolution”, by ANSYS CFX, is used for the momentum equations. This scheme changes the blend factor value $\beta$ to use a diffusive first order upwind differencing scheme ($\beta=0$) or a second order scheme ($\beta=1$) according to the mesh quality and flow dynamics. In this case, for the runner and the draft tube, the mean value of $\beta$, for each velocity components, is 0.89. For the runner, 18.84% of the nodes have a $\beta$ value lower than 0.75 for 17.60% in the draft tube. A value of $\beta=0.75$ is considered as a good compromise between robustness and precision [5]. The von Karman length scale allows capturing smaller structures but, in this case, the use of a high-resolution advection scheme reduces the blend factor in those structures. So, a more dissipative scheme is used in the small turbulent vortices and tends to dissipate them. For future simulations, a constant $\beta$ value will be imposed to avoid this dissipative effect. Since the literature about no-load conditions suggests a bad angle of attack at blade leading edges with strong stagnation points and a high swirling flow in the draft tube [6–10], Kato-Launder production limiter and curvature correction were activated.

Figure 1 shows the computational domain with an overview of the numerical strategy. To reproduce a complete interaction between the stationary and the rotating components, transient rotor-stator interfaces are used between the runner and the stationary parts.

![Distributor](image)

![Runner](image)

![Draft tube cone](image)

Figure 2. Mesh for the distributor, the runner and the draft tube cone at runaway condition.
A mass flow rate based on the experimental measurements is imposed at the inlet. A turbulent intensity of 5% is imposed, following Andritz’ general practice. A steady runner rotation speed is also imposed following the experimental measurements. An area-averaged static pressure calculated with the experimental suction head value $H_s$ is imposed at the outlet.

| Mesh characteristics for the casing (Cs), the distributor (Dist - including the stay vanes), the runner (Rn) and the draft tube (Dt) at runaway condition. |
|---|---|---|---|
| Number of nodes | 2 745 958 | 5 822 000 | 2 940 930 | 1 546 182 |
| Min angle | 13.9° | 34° | 25.315° | 21.42° |
| Max AR | 22.4047 | 15.1485 | 59.228 | 151 |
| Max exp. factor | 21 | 5 | 3 | 5 |
| Y+ max | 95 | 322 | 395.92 | 422 |
| Global mesh size | 13 440 600 nodes |

The table 1 summarizes the meshes characteristics in every subdomain. In each mesh, the minimum angle between element faces is 20°, the maximum element aspect ratio is lower than 151 and $y^+$ values are between 30 and 500, resulting in the use of wall functions. The only exception to these rules is the mesh of the spiral casing (excluding the stay vanes) made with Numeca Hexpress Hybrid, a software generating hexahedral dominant meshes. For this mesh, angles as low as 14° are tolerated since it does not affect the simulation convergence. For the stay vanes and the guide vanes, an unstructured mesh is generated with an Andritz in-house tool. Structured meshes are used for the runner and the draft tube. The software Numeca AutoGrid5 is used for the runner mesh and ANSYS ICEM 18.1 is used for the draft tube mesh. Figure 2 shows parts of the meshes used.

For the runaway simulation, a time step corresponding to 0.5° of runner rotation with six linear solver iterations is used. A maximum residual level below $10^{-5}$ for the draft tube and $10^{-4}$ for the distributor and the runner was targeted in both simulations. Except for local, negligible zones, this criterion is respected.

4. Results
The results analysis presented here is split in two parts: an analysis of the experimental pressure measurements for the runaway and the SNL cases in the frequency domain and preliminary numerical simulation results at runaway only.

4.1. Analysis of Pressure Measurements Acquired on the Runner Blades
Figure 3 shows the Fast Fourier Transform (FFT) analysis of the pressure signals acquired by the sensors S28 and S17, positioned on the blade 1 pressure side (PS) and blade 2 suction side (SS), at runaway and SNL conditions. The power spectral density function $G_{xx}$ is normalized with the energy level $E$ [J/kg]. The frequency $f$ is normalized with the runner rotation frequency $f_{rot}$. The values represent the average of the power spectral density function of each set of data for both operating conditions. The energy level in the frequency spectrum at runaway condition appears higher than at SNL. However, more frequencies stand out on the spectra at SNL. At SNL, the dominant fluctuation is the guide vane passing frequency, linked to the rotor stator interaction (RSI), in the rotating reference frame, $f/ f_{rot}=20$. The turbine studied here is known for its strong RSI [4], so it is not surprising to find this frequency dominating the power spectra of pressure signals acquired near the leading edge. At runaway condition, the RSI is clearly visible at blade 1 PS for sensor S28. Its energy level is similar to the one at SNL. A different tendency is visible on blade 2 SS. The literature at runaway condition, with larger guide vane opening than at SNL, suggests the presence of a stagnation point on the blade SS [8,9,11]. The location of this stagnation point is confirmed by the numerical results presented in the next section. It is interesting to note that the
RSI frequency disappears from the power spectra for sensor S17 positioned near the stagnation point. Other sensors on blade SS, near the stagnation point, at runaway condition, barely show this frequency.

| Inter-blade channel 1 | Runaway | SNL |
|-----------------------|---------|-----|
| Blade 1 PS S28        | ![Graph](image1) | ![Graph](image2) |
| Blade 2 SS S17        | ![Graph](image3) | ![Graph](image4) |

**Figure 3.** Power spectral density function $G_{xx}$ normalized with the energy $E$ as a function of the frequency normalized with the runner rotation frequency $f/f_{rot}$ at runaway and SNL for sensor S28 on blade 1 PS and sensor 17 on blade 2 SS.

| Coherence analysis | Runaway | SNL |
|--------------------|---------|-----|
| Blade 1 PS S28     | ![Graph](image5) | ![Graph](image6) |
| vs Blade 2 SS S17  | ![Graph](image7) | ![Graph](image8) |

**Figure 4.** Coherence analysis $C_{xy}$ as a function of the frequency normalized with the runner rotation frequency $f/f_{rot}$ between sensors S28 on blade 1 and sensor S17 on blade 2 at runaway and SNL conditions.

Figure 4 shows the coherence function $C_{xy}$ between two pressure signals from sensors in the same inter-blade channel but on two different blade sides (sensor S28 on the blade 1 PS and sensor S17 on
the blade 2 SS). The \( C_{xy} \) value is the average of the \( C_{xy} \) values for each set of data for both operating conditions. At SNL, a high correlation of \( C_{xy}=0.9999 \) is found for \( \frac{f}{f_{rot}}=20 \). This high correlation is found between every sensor signals near the leading edge at SNL, regardless of the blade side being analysed. At the runaway condition, even if the RSI frequency is not visible for the signal from sensor S17, the only significant correlated frequency between signals from sensors S17 and S28 is \( \frac{f}{f_{rot}}=20 \). However, the correlation level is not as strong as at SNL with \( C_{xy}=0.7316 \). The RSI effect is thus captured by the sensor S17 but does not stand out of the other frequencies, even if this turbine is sensitive to the RSI effect. The reason explaining this behaviour will be investigated in further studies.

At SNL, for sensor S17 on blade 2 SS, the frequency of \( \frac{f}{f_{rot}}=1 \) slightly stands out. As visible in figure 5, this frequency seems present in the power spectra of some sensors near the blade trailing edges (sensors S24 and S2) at SNL only. Pressure sensors in a fix reference frame, such as in the draft tube cone, do not detect the \( \frac{f}{f_{rot}}=1 \) frequency. Also, this frequency does not appear at runaway condition. Coherence analyses between pressure signals acquired at SNL and capturing the \( \frac{f}{f_{rot}}=1 \) frequency are presented in figure 6. A significant coherence level is only present between sensors on blade SS (for example, between sensors S1 and S12 on blade 9 and blade 2 SS). For some no-load or deep part load

| Inter-blade channel 1 – SNL only |
|----------------------------------|
| **Blade 1 – PS - S24**           |
| ![Power spectral density function Gxx normalized with the energy E as a function of the frequency normalized with the runner rotation frequency f/frot at SNL for sensor S24 on blade 1 PS and sensor 12 on blade 2 SS.](image1) |
| **Blade 2 - SS - S12**           |
| ![Power spectral density function Gxx normalized with the energy E as a function of the frequency normalized with the runner rotation frequency f/frot at SNL for sensor S24 on blade 1 PS and sensor 12 on blade 2 SS.](image2) |

| Coherence analysis – SNL only    |
|----------------------------------|
| **Blade 2 - SS - S12 vs Blade 1 - PS – S24** |
| ![Coherence analysis Cxx as a function of the frequency normalized with the runner rotation frequency f/frot between sensor S1 on blade 9 and sensor S12 on blade 2 and between sensor S12 on blade 2 and sensor S24 on blade 1 at SNL condition.](image3) |
| **Blade 9 - SS - S1 vs Blade 2 - SS – S12** |
| ![Coherence analysis Cxx as a function of the frequency normalized with the runner rotation frequency f/frot between sensor S1 on blade 9 and sensor S12 on blade 2 and between sensor S12 on blade 2 and sensor S24 on blade 1 at SNL condition.](image4) |

![Figure 5. Power spectral density function Gxx normalized with the energy E as a function of the frequency normalized with the runner rotation frequency f/frot at SNL for sensor S24 on blade 1 PS and sensor 12 on blade 2 SS.](image5)

![Figure 6. Coherence analysis Cxx as a function of the frequency normalized with the runner rotation frequency f/frot between sensor S1 on blade 9 and sensor S12 on blade 2 and between sensor S12 on blade 2 and sensor S24 on blade 1 at SNL condition.](image6)
conditions, a large backflow in the draft tube enters the runner by the blade suction side, causing vortices near the blade trailing edge [6–8,12]. In this case, an unbalanced backflow, because of the draft tube elbow, may affect the runner and induce a frequency of \( f/f_{rot} = 1 \) in the rotating reference frame. Since this backflow seems to evolve in the runner by the suction side of the blades, it would be normal to have a better coherence level between sensors on blade SS.

4.2. Preliminary CFD Results at Runaway Condition

A preliminary step to the numerical results analysis is to evaluate the accuracy of the predictions. Figure 7 compares the time-averaged pressure difference between numerical results (based on eight runner rotations) and the experimental measurements at runaway. A negative deviation represents an under-prediction by the simulation. The sensors chosen for the comparison never reached the water vapor pressure throughout the measurements.

The numerical runaway simulation over-predicts the head by only 1.79%. The numerical torque is near zero with a negligible difference to the experimental one. A volume representing 8.76% of the runner volume has an absolute pressure (\( P_{abs} \)) below the vapor pressure (\( P_{vap} \)) for the simulation at sigma plant. The numerical results from sensors on blade 8 PS, positioned near the blade leading edge, show a deviation to the experimental data lower or equal to 3%. Sensor S28 on blade 1 PS shows a larger deviation but still below 5%. This might be explained by the numerical pressure average on only 8 runner rotations, representing only 0.5s in real time.

![Figure 7](image_url)

**Figure 7.** Comparison between the pressures simulated and measured (numbers in boxes) for some sensor positions on the runner blade at runaway condition.

![Figure 8](image_url)

**Figure 8.** Absolute pressure field normalized by the head (left) and vector field for the runaway at a blade span of 0.5.
Figure 8 illustrates absolute pressure field (left) and the flow orientation with a vector field (right) near the leading edge at a blade span of 0.5 for the runaway condition. The stagnation point is clearly on the blade SS. The nearest sensor to the stagnation point is the sensor S4. It is positioned in a high-pressure-gradient zone. Consequently, small deviations between the experimental and the numerical position of the stagnation point could lead to large deviations of the numerical pressure prediction. Moreover, the experimental pressure is evaluated on a 3 mm diameter surface while the numerical pressure is extracted from a single point with a value interpolated on a computational grid. This difference of pressure evaluation method could also lead to different pressure values in a high pressure gradient zone. Those effects, and some linked to the difficulties of eddy viscosity based turbulence models in the vicinity of stagnation points, can explain the 10% deviation for the sensor S4. A good pressure prediction is made by sensor S17 on blade 2 SS with a deviation lower than 1.31%. Sensor S11 presents the worst pressure prediction. However, this sensor is next to a region with risk of cavitation as presented, in figure 7, by a blue surface of $P_{abs} < P_{vap}$ on blade 2 SS at a given time step. This could also explain the prediction deviation of sensor S32. Globally, the simulations of the runaway condition appear to predict the average pressure values with enough precision to warrant their use in exploring the large trend of the flow dynamic inside the runner.

![Figure 8](image_url)

**Figure 8.** Zones of $P_{abs} < P_{vap}$, illustrated in 3D by iso-surfaces at $P_{vap}$, in blue, (left) and in 2D by areas of $P_{abs} < P_{vap}$, in blue, (right).

In figure 9, iso-surfaces of $P_{abs} = P_{vap}$ show the location of regions having a risk of cavitation at a given time step. The size and position of low pressure regions are different for each blade channel. Their position is probably linked to inter-blade vortices evolving with time. It is interesting to note that, even if approximately 8% of the runner volume potentially cavitates, an accurate torque prediction has been obtained. Since the low pressure regions stay in the lower half of the runner blades, near the hub, their contribution to the overall torque is negligible. Moreover, as shown in figure 7, only sensor S11 is located near a zone subject to cavitation indicating that predictions at other sensor locations will only be affected indirectly by the use of single phase flow simulation. In follow-up simulations, cavitation will be included to evaluate its impact on the flow dynamics in the runner. However, Côté [6] found that considering the cavitation by using a homogeneous approach does not change the pressure fluctuation frequencies and amplitudes compared to the ones predicted by a single phase simulation for flow at a given no-load condition.

In the literature, Li et al [10] denotes a remarkably different flow topology for different no-load operating points for a given Francis turbine. One of the flow topology frequently reported in the literature has a backflow in the draft tube going up in the runner by the blade SS [6–8]. In some case, this backflow induce vortices near the blade trailing edges. The following sections expose a flow behavior slightly different for the full-gate opening runaway condition.

The backflow in the draft tube, at a given time step, is illustrated in figure 10 by a pink iso-surface of negative velocity following the z-axis. Its size is however nearly constant with time. At runaway condition, the backflow barely extends past the runner/draft tube (Rn-Dt) interface and do not enter the runner inter-blade channels. The backflow is surrounded by unorganized vortex structures. Those structures are highlighted with a Q-criterion.

![Figure 9](image_url)

**Figure 9.** Zones of $P_{abs} < P_{vap}$, illustrated in 3D by iso-surfaces at $P_{vap}$, in blue, (left) and in 2D by areas of $P_{abs} < P_{vap}$, in blue, (right).
Figure 10. Iso-surfaces of backflow regions, in pink, at runaway (left). Iso-surfaces of Q-criterion, in gray, (middle) and iso-surfaces of $P_{\text{abs}}=P_{\text{vap}}$, in green (right) around the backflow.

Figure 11. Pseudo-streamlines on a surface at runaway condition at a blade span of 0.5.

Figure 11 illustrates pseudo-streamlines on a surface at half of the blade span at runaway condition. Since the flow dynamics is highly three-dimensional, those 2D views cannot provide explanations on the inter-blade vortex origin. However, it highlights different flow behavior in the inter-blade channels. Some vortices are clearly linked to the flow separation at the blade leading edges. As mentioned previously, the backflow does not enter the runner, thus, no vortices are present near the blade trailing edges at the shown span location. Indeed, in the rotating reference frame, the flow at the inter-blade channel outlets seems relatively well aligned with the blades at runaway.

5. Conclusion and Ongoing Work

In this paper, two different no-load conditions are studied, on a model high head Francis turbine, with pressure measurements: a SNL condition, with some residual torque, and a full-gate opening runaway condition. Single-phase URANS simulations of the runaway condition are also performed. Local unsteady pressure and integral turbine performance measurements are used to validate the numerical strategy.

The FFT analyses at SNL show that the frequency linked to the RSI is the dominant fluctuation for signals from both blade sides, while generally a high frequency content is present. At runaway condition the RSI is visible on signals from blade PS but barely stands out of the power spectra for signals from blades SS. The reason for this behavior will be investigated in future studies. The numerical simulation of the runaway condition shows a good agreement with the experimental data. The torque prediction is
accurate even if approximately 8% of the runner volume is below vapor pressure. For this operating condition, the backflow barely enters the runner and is surrounded by unorganized vortices.

The second phase of the project will include the numerical results of simulations at SNL. A detailed comparative study between flow at runaway and SNL conditions will be undertaken. The complex flow configuration in the runner and the draft tube will be explored in more detail for both conditions.

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