Experimental and numerical investigations of a high-head pumped-storage power plant at speed no-load.

J Decaix¹, V Hasmatuchi¹, M Titzschkau², L Rapillard¹, P Manso³, F Avellan⁴ and C Münch-Alligné¹

¹ University of Applied Sciences and Arts Western Switzerland Valais, Route du Rawil 47, CH-1950 Sion, Switzerland
² Kraftwerke Oberhasli AG (KWO), Grimsel Hydro, Innertkirchen, Switzerland
³ Ecole Polytechnique Fédérale de Lausanne (EPFL), Plateforme de Constructions Hydrauliques (PL-LCH), CH-1015 Lausanne, Switzerland
⁴ Ecole Polytechnique Fédérale de Lausanne (EPFL), Laboratoire de Machines Hydrauliques (LMH), CH-1007 Lausanne, Switzerland
E-mail: jean.decaix@hevs.ch

Abstract.
Due to the increasing integration of new renewable energies, the electrical grid undergoes more frequent instabilities. Hydroelectric power plants, particularly pumped storage plants, are well suited for grid control. However, this objective requires extending the operating range of the machines and increasing the number of start-up, stand-by and shut-down procedures, which reduces the lifespan of the machines due to high mechanical stresses. The current study focuses on a pumped-storage power plant equipped with ternary groups. Recently, cracks on the runner blades of the Francis turbine have been observed without finding a clear explanation for their onset. During this period, the number of starts and stops per day has strongly increased. In order to better understand the origin of the fatigue of the turbine runner, external and on-board measurements along with CFD and FEM investigations have been performed. The on-board measurements provide evidence high mechanical stresses on the runner blades during the synchronization of the machine at speed no-load (SNL) operating condition. The frequency spectrum observed on the various signals suggests that the high stresses are linked to the excitation of one of the runner modes by some flow instabilities, which is supported by the CFD and FEM analyses.

1. Introduction
In the recent years due to the development and the increasing integration of renewable energy resources into the electricity grid, hydraulic turbines and pump-turbines are becoming key technical components for load shifting and frequency control of the grid. To act as grid stabilizer, the hydraulic machines have to extend their operating range [1], cope with frequent start-up, stand-by and shut down operations while improving its long-term availability [2–4]. However, such requirements leads to an increase in the dynamic loads undergone by the water turbines [5], which reduces the lifespan of the turbines. Consequently, a better knowledge of the influence of dynamic loads on the hydraulic machines is necessary.
Focusing on the full operating procedure of a prototype from water filling, start-up, synchronization of the generator with the grid, going to part load, then up to full load, back to shut-down and to the hydraulic system drain, the components of the hydraulic machine undergoes both easily predictable loading and less and/or even completely unpredictable loading states, and this without considering the transient modes. Therefore, experimental measurements on the prototype are often mandatory either to better predict the remaining residual lifetime of the machine [6] or to investigate the source of harsh conditions that may conduct, in the worst case, to cracks. One of the most active component of the hydraulic turbine is obviously the runner. Indeed, experimental measurements on the runner during the operation gives the most direct responses when conducting such investigations. With the actual technological possibilities, performing measurements in the rotating frame during operation becomes easier to put in operation. The are some examples in the literature of experimental measurements on reduced scale models using generally pressure sensors, accelerometers, or strain gauges [7, 8]. The advantage of laboratory tests comes from the ease to comfortably set-up the tests and to calibrate the sensors. On the other hand, at prototype scale, there is often more space to install the rotating equipment, but the communication with the exterior is more challenging or not possible. In recent decades several successful attempts have been performed on prototype by Gagnon et al. [9, 10] or by Egusquiza et al. [11].

Numerical tools are useful to investigate complex phenomena coupling fluid and structure in hydraulic machines [5, 12, 13]. Regarding the prediction of flow instabilities, Computational Fluid Dynamic (CFD) is a reliable tool to assess pressure fluctuations for instance due to Rotor-Stator Interaction or during speed-no load (SNL) conditions [14, 15].

In a complementary approach, Finite Element Method (FEM) is commonly used to provide modal and harmonic analyses of the runner surrounding by water in order to determine the modes and the eigenfrequencies of the structure [16] as well as the stresses induced by the fluid [15, 17]. This can be validated and/or completed with the information coming from the experimental measurements and CFD, as done by Eichhorn et al. [18].

The present paper focuses on both experimental [19] and numerical investigations of the phenomenon responsible for the appearance of the cracks on the runner blade of a ternary unit at a prototype scale, already investigated by Müller et al. [20]. First, the case study is briefly described. Then, the experimental and numerical set up are provided. Finally, the analysis of the measurements and the computations is carried out.

2. Case study
The case study is the Grimsel 2 pumped-storage power plant operated by KWO [21] and built between 1974 and 1980 in Switzerland. This power plant is equipped with four 90-MW ternary groups: a horizontal-axis motor/generator, a Francis turbine and a pump. The specific speed of the Francis turbine runner is \( \nu = 0.247 \). The number of guide vanes is \( z_g = 24 \) and the number of runner blades is \( z_b = 17 \). The speed and discharge factors of the turbine at the Best Efficiency Point (BEP) are respectively \( n_{ED} = 0.271 \) and \( Q_{ED} = 0.183 \). The unit 2 is considered for the experimental investigations. The main efforts are dedicated to the investigation of the turbine mode, since after several operation cycles, some cracks have been observed on the runner at the junction between the trailing edge of the blades and the hub. The phenomenon responsible for the development of the cracks is not yet clearly identified even if it seems related to the increase of the number of start-up, stand-by and shut-down procedures per day.

3. Experimental and Numerical set up
3.1. Experimental set up
The philosophy of the employed instrumentation to identify the possible harmful structural conditions for the turbine runner consists on synchronised measurements between the rotating
and the stationary frames of the machine. To this end, an autonomous digitizer from Gantner Instruments (able to acquire continuously synchronized signals at 10 kHz) has been installed in a sealed chamber into the nozzle of the turbine runner (see figure 1). The main components of the on-board measurements system are provided in table 1. Strain gauges placed on four impeller blades along with accelerometers and tachometers have been connected to the acquisition system. The rotating frame instrumentation has been supplied with power by two LiPo batteries connected in parallel. Apart from the operation during the rotation of the runner, once the machine stopped the system had to be accessible in terms of power switch on/off, programming and recovery of data as well as to recharge the batteries.

![Image of onboard measurements system](image_url)

**Figure 1.** Onboard measurements system along with the strain gauges installed in the runner.

<table>
<thead>
<tr>
<th>Component</th>
<th>Features/connected instrumentation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Power supply</strong></td>
<td>2x SwayTronic 6S1P LiPo batteries - capacity: 21000 mAh - voltage: 22.2 VDC 1x Power supply protection electronics - protection against deep discharge - protection against discharge between batteries</td>
</tr>
<tr>
<td><strong>Acquisition system and sensors</strong></td>
<td>1x Gantner Q.Station T - 10-30 VDC power supply - RS232 &amp; Ethernet - 2x 16GB USB memory sticks 1x Q.brixx A109 - 2x Sick IM18-10BNS-NC1 inductive tachometers 1x Q.brixx A101 - 2x Wilcoxon 726T IEPE single-axis accelerometers 2x Q.brixx A107 - 8x HBM 1-LY11-3/350 strain gauges</td>
</tr>
<tr>
<td><strong>Sealed connectors</strong></td>
<td>5x Water proofed MacArtney SubConn connectors - 1x Ethernet connector - 1x connector for batteries tension monitoring - 1x connector for power switch on/off and batteries recharging - 2x connector for strain gauges</td>
</tr>
</tbody>
</table>

The stationary frame instrumentation consists mainly on two synchronised National Instruments digitizers connected to several monitoring sensors placed on both the turbine and the pump of the GR2 unit (see figure 2 and table 2). A specific interface developed under Labview environment has been used to drive the continuous acquisition using a 10 kHz rate for
Figure 2. Stationary frame experimental instrumentation setup.

Table 2. Main components of the stationary frame instrumentation.

<table>
<thead>
<tr>
<th>Component</th>
<th>Connected instrumentation</th>
</tr>
</thead>
</table>
| National Instruments PXIe-1073 1x NI PXIe4497 | - 1x Kistler tri-axis IEPE accelerometer (turbine)  
- 1x Kistler single-axial IEPE accelerometer (turbine)  
- 1x Wilcoxon 726T IEPE single-axis accelerometer (pump)  
- 1x Kistler single-axial IEPE accelerometer (pump)  
- 1x GRAS IEPE microphone  
1x NI PXIe6612 | - 1x Sick optical PNP tachometer |
| National Instruments cDAQ9174 1x NI 9203 | - 1x Manfred relative pressure sensor (inlet of spiral casing)  
- 1x Endress Hauser absolute pressure sensor (outlet of draft tube)  
- 1x Endress Hauser absolute pressure sensor (atmospheric pressure)  
- 1x Baumer temperature sensor (water temperature)  
- 1x Balluff single-axis inclinometer (turbine guide vanes)  
- 1x Fluxus ultrasonic flowmeter (turbine upstream pipe)  
1x NI 9229 | - 2x Bruel & Kjaer bearing eddycurrent proximity sensors (turbine shaft bearing) |
| SCADA system | Recovered variables  
- Electrical power  
- Turbine & Pump discharge  
- Guide vanes opening  
- Turbine upstream & downstream static pressure  
- Upper & Lower dam water levels |
| Existing control/monitoring autonomous system of the machine | Recovered variables  
- Guide vanes opening  
- Runner rotational speed  
- Turbine net head  
- Turbine discharge  
- Turbine active power |

all connected sensors. Three pressures sensors, one flowmeter, one tachometer, two proximity sensors, one microphone and several accelerometers have been used to capture the source of the instabilities in the same time with the rotating frame instrumentation. It has been shown in Botero et al. [22] that hydrodynamic instabilities can be also detected from outside using only non-intrusive instrumentation, such as accelerometers or microphones. In the end, the final
purpose is to succeed in identifying the sources of instabilities (possible easily detectable with the on-board measurements) while using only non-intrusive measurements in the stationary frame. This approach should in the end provide a smart diagnostic tool useful for other machines suffering of similar problems. Finally, the synchronisation between the on-board and the stationary frame systems has been ensured by hammer impacts on the structure of the turbine registered simultaneously by the accelerometers connected on each acquisition system. The synchronisation with the data coming from the SCADA and from the existing control/monitoring acquisition system of the power plant is based on the identification of different events such as the runner start-up and the opening of the guide vanes.

3.2. Numerical set up
FEM is used to perform a modal analysis of the runner using the modal analysis system available in the Ansys Workbench 17.2. The FEM is carried out for the runner at rest surrounded by water, which allows focusing on the influence of the added mass, due to the presence of water, on the modes and eigenfrequencies. The computational domain is shown on figure 3 (right). A fixed support condition is set at the junction between the runner and the shaft (see figure 3 center). The runner and the surrounding water volume are meshed with approximately 300'000 tetrahedral elements (see figure 3 right).

Figure 3. FEM computational domain (left), location of the fixed support boundary condition (center) and mesh of the runner (right).

CFD simulations of the flow inside the turbine have been performed using the Ansys® CFX v17.2 software. The flow in the turbine is computed for the SNL operating condition, for which the guide vane opening angle is set to 2 degrees. The computational domain takes into account the whole turbine from the spiral case to the draft tube (see figure 4 left). The domain is meshed with hexahedrons. The total number of nodes is closed to 15 million with 2.8 millions of nodes in the runner and 4 million of nodes in the guide vanes. The mass flow rate is imposed at the inlet of the spiral case whereas an opening pressure condition is set at the outlet of the draft tube. The interfaces between a stationary domain and a rotating domain are computed using a transient rotor/stator algorithm. The flow is computed by solving the Unsteady Reynolds-Averaged Navier-Stokes (URANS) equations coupled with the Shear Stress Transport (SST) model to calculate the turbulent eddy viscosity. The computed physical time corresponds to 12 runner revolutions with a time step smaller than 1 degree of revolution per time step.

4. Analysis
4.1. Measurements
The prototype measurements campaign was focused mainly on the operation of the turbine during start-up, nominal condition as well as full operating range for a given testing head condition, including the deep part load operation, interesting in terms of operation flexibility. The start-up in pump mode has been also addressed. Finally, the signals of the 2 strain gauges installed on one of the blade of the turbine runner close to the hub and to the shroud (aligned with the blade trailing edge) along with the signal of one accelerometer during the start-up procedure are provided in figure 5.
Chronologically, the turbine filling, the turbine start-up, followed by the synchronization of the generator with the electrical network, then the stable part load operation and finally the turbine shut down procedure and the draining can be easily identified on the fluctuations of the strain as well as on the vibrations. As already stated in the articles of Gagnon et al. [23] and Coutu & Chamberland-Lauzon [3], the maximum amplitude of strain (extremely high compared with the values at part load operation) occurs at SNL operating point during the synchronization of the generator with the network as well as during the shutdown phase. Indeed, theses harsh operating conditions that accelerate the shortening of the runners remaining lifetime are encountered each time the turbine is started and stopped for several tens of seconds (up to few minutes in the worst case). Regarding the frequency of these fluctuations, despite the fact that they are very close to the first harmonic of the blade passing frequency, the observed value is the response of one of the runner eigenmodes. The amplitude of fluctuations for this
Figure 6. Identification of the main harmful structural loading fluctuation of the turbine runner blades during the SNL operation in the signals of the on-board accelerometer and of the non-intrusive instruments: accelerometer, microphone and proximity sensor.

strain direction is larger at the shroud than at the hub blade side. Moreover, a sub-synchronous frequency (compared to the runner rotational frequency) is observed in the variation of the strain fluctuation amplitude.

Then, in figure 6, considering the same SNL operating point, one may state that the same predominant oscillation \( f/f_n = 48.9 \) is observed not only on the signal of the accelerometer installed in the runner, but also in the signal of the accelerometer installed on the turbine casing as well as in the ones coming from the microphone and from the proximity sensor. This result is actually interesting in the way that the final purpose of this study is to establish a diagnosis protocol based only on a simplified instrumentation set (basically non-intrusive) to identify harsh operating conditions on different hydropower units in order to avoid them.

4.2. Numerical results
Based on the work of Tanaka [24], the resonance of the runner can occur if:

\[
mz_b - nz_g = k
\]

with \( m \) and \( k \) two natural relative integers and \( n \) a natural integer. This relation is verified for \( m = 3 \), \( n = 2 \) and \( k = 3 \). Therefore, the excitation source due to the Rotor-Stator Interaction (RSI) should have \( k = 3 \) nodal diameters (ND) and the frequency of the excitation should be \( 48 \ f_n \) in the rotating frame and \( 51 \ f_n \) in the stationary frame, with \( f_n \) the runner frequency.

The FEM modal analysis shows three bending modes with:
- 3 ND at a dimensionless eigenfrequency of \( f/f_n = 38.5 \) (see figure 7 left),
- 4 ND at a dimensionless eigenfrequency of \( f/f_n = 47 \) (see figure 7 center),
- 5 ND at a dimensionless eigenfrequency of \( f/f_n = 49 \) (see figure 7 right).

Compared to the work of Tanaka, the bending mode with 3 ND should not be excited due to the RSI since the dimensionless frequency of the bending mode \( f/f_n = 38.5 \) does not match the dimensionless frequency of the RSI \( f/f_n = 48 \). The frequency of the bending modes with 4 and 5 nodal diameters are closed to the one observed on the measurement.

In table 3, the time-averaged \( n_{ED} \) et \( Q_{ED} \) at SNL operating point predicted by the CFD is compared with the measured values. The overestimation of the speed and discharge factors by the CFD is less than 4\%, which is rather good for a simulation at an off-designed operating point.
Figure 7. Total displacement for the bending modes with from left to right: 3 ND and $f/f_n = 38.5$; 4 ND and $f/f_n = 47$; 5 ND and $f/f_n = 49$. Colour: blue, low displacement and red, large displacement. FEM.

Table 3. Comparison of the time-averaged $n_{ED}$ and $Q_{ED}$ measured and computed. SNL operating point.

<table>
<thead>
<tr>
<th></th>
<th>Measurement</th>
<th>Computation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n_{ED}$</td>
<td>0.270</td>
<td>0.280</td>
</tr>
<tr>
<td>$Q_{ED}$</td>
<td>0.0170</td>
<td>0.0176</td>
</tr>
</tbody>
</table>

Figure 8. Frequency spectra of the dimensionless head (left) and torque (right). SNL operating point. CFD.

The frequency spectra of the dimensionless head and torque\(^1\) are displayed on figure 8. On both spectra, the blade passage frequency $f/f_n = 17$ is captured even if regarding the head several slightly higher frequencies are also captured. On the contrary, the guide vane passage frequency $f/f_n = 24$ is not clearly observed despite the fact that a frequency is observed at $f/f_n = 25$. Focusing on the head, a frequency at $f/f_n = 39$ is present, whereas on the torque, frequencies around $f/f_n = 43$ are put in evidence. None of them shows the excitation frequency observed by the measurements.

Figure 9 displays the pressure on the hub and the trailing edge of a runner blade and also an iso-surface of the Q-criterion. A high pressure zone close to the trailing edge is clearly visible. This zone is surrounded by a large spanwise vortex and a smaller streamwise vortex located at the junction between the hub and the blade. This zone is the one where the cracks on the blade

\(^1\) The torque predicted by the CFD is the friction torque dissipated by the turbine at the SNL operating point.
Figure 9. Downstream view of the runner (left) and pressure contours on the hub and the runner blade (blue, low pressure and red, high pressure) and iso-surface of the Q-criterion at the trailing edge of a runner blade (right). SNL operating point. CFD.

Figure 10. Frequency spectra of the dimensionless pressure at two different locations P1 (left) et P2 (right) on the blade (see figure 9). SNL operating point. CFD.

are observed.

The pressure spectra of two probes P1 and P2 located at the trailing edge of a blade close to the hub (see figure 9) are displayed on figure 10. The probes P1 exhibits the guide vane passage frequency \( f/f_n = 24 \) and a frequency at \( f/f_n = 48 \) close to the experimental one but with a small amplitude. On the contrary, the probes P2 located in the streamwise vortex does not clearly show any specific frequency.

5. Conclusion

External and on-board measurements of the turbine of the Grimsel 2 ternary unit have been carried out for the whole range of the operating conditions from the start-up to the shut down of the turbine mode.

CFD and FEM analyses of the turbine have been performed to investigate the SNL operating conditions and to understand the origin of the frequency observed on the measurements. Up to now, RSI does not seem to be directly responsible for the cracks observed at the trailing edge of the blades. At the moment the most plausible explanation for the high strain amplitudes observed seems rather linked to the dynamic of the vortices located at the trailing edge of the runner blade, which could create enough pressure fluctuations in order to excite some of the bending modes of the runner.
In order to further investigate these phenomena and clarify the sources of instability and turbine wear, additional CFD investigations using advanced RANS models such as the SST-SAS model or hybrid RANS-LES models should be carried out in order to improve the resolution of the pressure fluctuations and the contents of the frequency spectrum close to the trailing edge. Then, the CFD pressure should be used in order to carried out additional FEM analyses.

Acknowledgments
This project is part of the research program FLEXSTOR (flexible solutions for hydropower storage plants in changing context) of the Swiss Competence Centre for Energy Research Supply of Electricity (SCCER-SoE, Phase II), with co-funding by the Swiss Commission for Technology and Innovation (grant CTI - 17902.3 PFIW-IW) and by Kraftwerke Oberhasli AG.

References
[21] Schlunegger H and Thöni A 2013 Hydro 2013 (Innsbruck, Austria)