To be published in Wasserwirtschaft,2019, no. 1, which should be cited to refer to this work.
DOI:https://www.doi.org/10.1007/s35147-019-0236-y

On-Board Measurements At A 100MW High-Head Francis Turbine

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Abstract: The Francis turbine runners of the four ternary groups of Grimsel 2 power plant in Switzerland showed fatigue cracks at the runner blades much earlier than expected. To find the source of these cracks, strain gauges and accelerometers were used to get information about the harmful stresses on the runner during operation. It is observed that the speed no-load operation during the start-up and the shutdown processes of the turbine impose large stresses on the runner blades.

1 Introduction

Modern sources of energy, like solar and wind, force hydropower machines to change their operational behaviour. Rapid changes in the availability of renewable energies lead to instabilities in the grid, which need to be stabilized by flexible machine operation (Lowys et al. [1]). This leads to a significant change in hydropower operations, e.g. to an increasing number of start-stop events (Seidel et al. [2], Robert et al. [3], Coutu and Chamberland-Lauzon [4]). The four ternary machine groups at the Grimsel 2 power plant have been designed under completely different circumstances in the late 1970's and are suffering from this change in operation. After only 35'000 hrs of operation all runners show fatigue cracks at the trailing edges close to the hub which needed to be repaired effortful. As a part of these repairs, the chamfer between the blade and the hub was increased from about 5 mm to 12 mm with local welding and heat treatment to reduce stresses. Apparently, this seems to work out fine but, since the source of the damage was not detected, this situation was not satisfying for the plant owner.
The turbines were running at smooth conditions most of the time. Since deep part load could be eliminated as a source of those cracks, it was assumed that either start-up, shutdown or some unknown phenomena lead to the cracks. From recent literature (Gagnon et al. [5] & [6]), it is assumed that the start-up process can be responsible for the observed kind of cracks. To get more information, extensive measurements have been performed on the Group 2 of the power plant.
The overall goal of this research is to advice the plant owner concerning the operation of the machines to increase the lifetime of the existing runners.

2 Background

The Grimsel 2 power plant was built between 1974 and 1980 and consists of four horizontal ternary groups, a Francis turbine at one shaft end, a pump impeller on the other side and a fix coupled motor/generator in the centre with a rated power of 100MVA and nominal speed of 750min^{-1} each. The maximum turbine flow rate is 25 m^3/s, the nominal head 400 mWC and the average annual production is 550 GWh. The Group 1 is equipped with a 100MW-frequency converter that is used for pump operation only (Schlunegger and Thöni [7]). The turbine runner, composed by 17

blades, has a specific speed of v = 0.247, whilst the governor is equipped by 24 guide vanes. The speed and discharge factors of the turbine at Best Efficiency Point (BEP) are n_{ED} = 0.271 and Q_{ED} = 0.183 respectively. The draft tube has a special design with a tube in the centre of the cone that goes up to the runner nose, the centre of the runner being inaccessible once the machine is mounted together.
For the pump mode, the group is started using the turbine. Therefore, each pump start means an additional turbine start-up and shutdown. During the pump operation the spiral casing of the turbine is drained and the runner is circulating in cooled air. This study focusses on the loads of the turbine runner during start-up, energy production and shutdown. The goal is to find out the operating point that leads to the increased fatigue of the material. Therefore, on-board measurements on the runner synchronised with non-intrusive measurements on the machine have been performed, as well as CFD simulations (see Decaix et al. [9]).

3 Experimental Setup

The purpose of the experimental setup is to identify the harmful operating conditions both on the runner and on the stationary parts of the machine. With the correlation of these synchronised results, a method should be developed in order to allow detecting harsh loads on different runners without performing expensive on-board measurements each time.

3.1 On-Board Instrumentation

The on-board instrumentation (listed in Table 1) consisted of eight strain gauges fixed on the turbine runner blades, two accelerometers and two tachymeters installed on the runner crown tip. The data acquisition system was mounted in a sealed chamber placed in the centre of the runner and supplied with energy by batteries. Once mounted into the turbine, the access to the system was only for charging and data transfer. Indeed, due to the special construction of the draft tube, there was no direct access to the acquisition system. The acquisition rate of the system was set to 10 kHz and the autonomous working time is of at least 12 hrs.

Component	Manufacturer	Description
Acquisition system	Gantner Instruments	3.1.1.1.1 1x Q.Station T 1 x Q.brixx A109 1 x Q.brixx A101 2 x Q.brixx A107
Strain gauges	HBM	8 x 1-LY11-3/350
Strain gauge glue	HBM	X60 2-K
Protective glue	Huntsman	Araldite 2022
Tachymeters	Sick	2 x IM18-10BNS-NC1
Accelerometers	Wilcoxon	2 x 726T IEPE
Batteries	SwayTronic	2 x 6S1PLiPo, 21Ah, 22.2V
Sealed connectors	MacArtney	5 x SubConn series

Table 1: Components of the on-board instrumentation.

Beforehand the chamber was tested under pressure since it had to withstand the maximum draft tube pressure of 15 bars. Four existing M20 threads were used to fix

the chamber to the runner properly even under harsh operating conditions like pump start-up or the imbalance imposed by the instrumentation inside of the chamber.
Moreover, the strain gauge application and protection was tested with over realistic harsh conditions under high pressure flow (Fig. 1) to ensure a minimum risk of losing them during the tests. The strain gauges and the cables were protected with a solid layer of glue that was grinded and polished to reduce as much as possible the influence to the flow. The accelerometers and the tachymeters were mounted directly inside the sealed chamber.

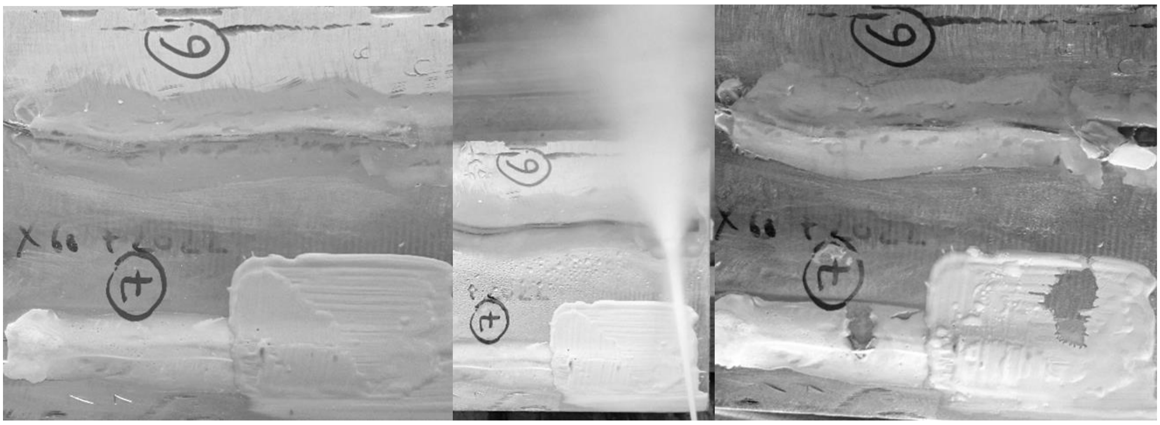

Fig. 1: Glue specimens, testing and results.

The Fig. 2 shows the setup of the different sensors on the runner. The strain gauges were applied at four blades at different angles close to the connection between hub/shroud and blade. During the installation one set of strain gauges at blade 4 got damaged. However, since the same direction of strain was measured at blade 17, this loss was acceptable.

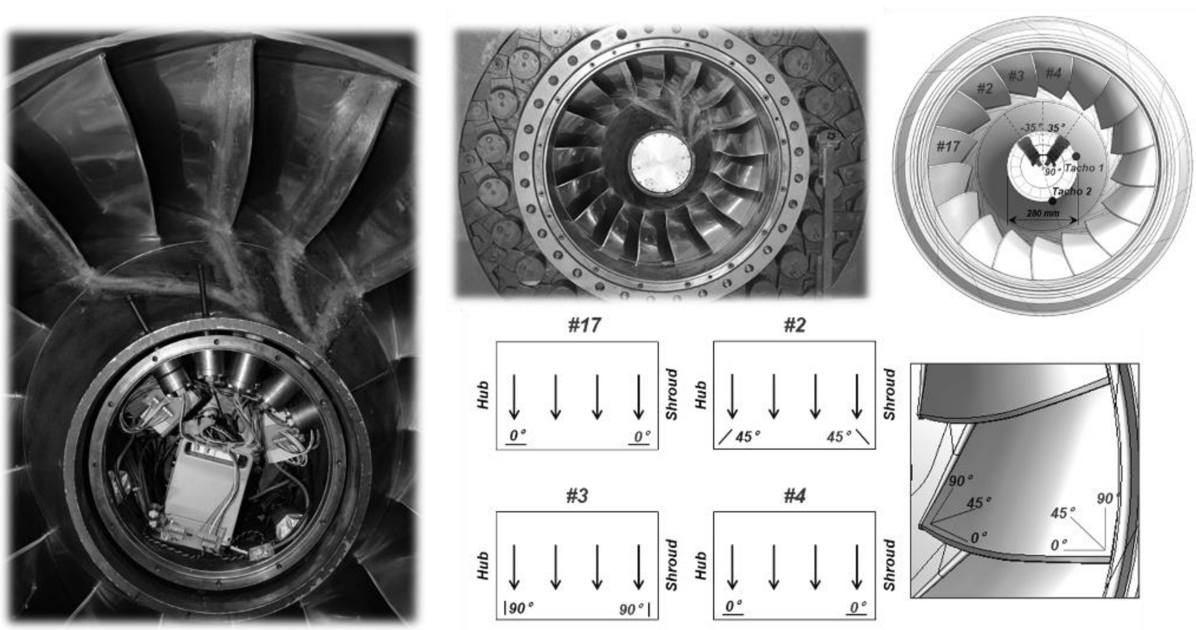

Fig. 2: Instrumentation setup on the runner, Hasmatuchi et al. [8].

3.2 Non-Intrusive Instrumentation

For the stationary frame, the existing instrumentation of the machine monitoring system was used and additionally a temporary instrumentation was installed at the turbine and the pump side of the group. The existing instrumentation includes vibration monitoring, flow rate of the machine group and the plant, upstream and downstream pressure, guide vanes position and electrical power (see the SCADA list of variables in Table 2). The data from the SCADA system is sampled with a maximum rate of 1Hz. The temporary instrumentation included several accelerometers fixed on the casing of the turbine and of the pump, a microphone at the turbine outlet, an optical tachymeter, an inclinometer at one guide vane, an ultrasonic flow meter close to the turbine spherical valve and additional pressure sensors upstream and downstream the turbine. The signal of these sensors were recorded with two synchronised acquisition systems at a frequency of 10 kHz.

Component	Manufacturer	Description
Acquisition systems	National Instruments	1x PXIe-1073 1x PXIe-4497 1x PXIe-6612 1x cDAQ-9174 1x NI 9203 1x NI 9229
Accelerometers (turbine casing)	Kistler	1x IEPE single-axis 1x IEPE tri-axis
Accelerometers (pump casing)	Kistler Wilcoxon	1x IEPE single-axis 1x 726T IEPE single-axis
Microphone (turbine draft tube)	Gras	1x IEPE microphone
Pressure taps (turbine upstream and downstream)	Manfred Endress Hauser	1x rel. pressure sensor 2x abs. pressure sensor
Thermometer (turbine inlet)	Baumer	1x water temp. sensor
Shaft displacement	Bruel & Kjaer	2x IN-081 proximeters
Inclinometer (turbine guide vanes opening)	Balluff	1x BSI000
Flowmeter (turbine discharge)	Flexim	1x FLUXUS F601
SCADA control/monitoring system		- discharge - electrical power - guide vanes opening - reservoir levels - surge tank levels - in- & outlet pressure

Table 2: Components of the stationary frame instrumentation.

The method of synchronization in between the different acquisition systems is sketched in Fig. 3. The synchronization between the different systems can be done easily by using several values like guide vane opening, rotational speed or pressure fluctuations that are sampled in both systems. To synchronise the on-board and the stationary frame systems, hammer impacts on the generator shaft, easily detectable especially by the accelerometers available in both systems, were used.

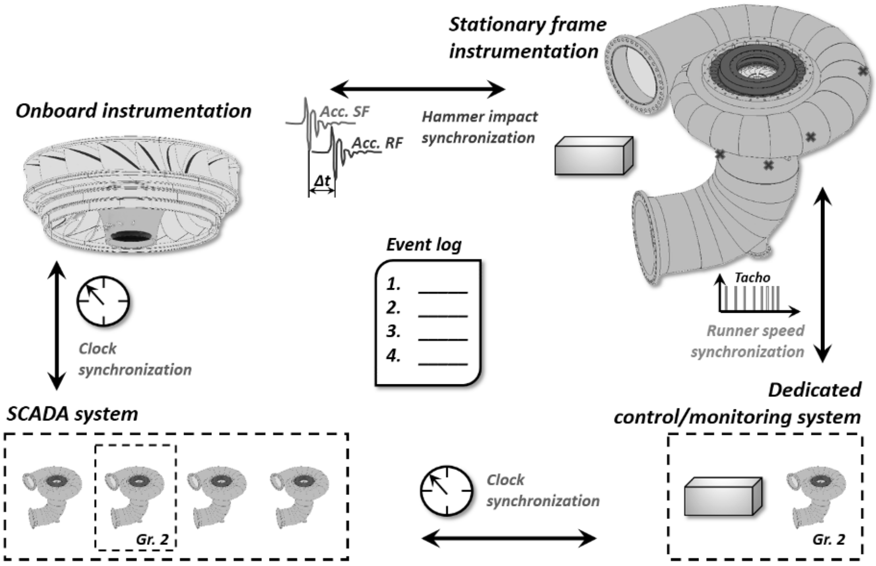

Fig. 3: Synchronization between the different acquisition systems, Hasmatuchi et al. [8].

4 Operational Setup

To have a chance to find out which operating point is responsible for the decreasing lifetime of the runner, all operating points that are permitted for standard operation were tested. This includes several start-ups of both turbine and pump as well as stable operating points throughout the whole operating range. Additionally, some potential operating points at deep part loads were tested. To minimize the risk of damage of the on-board equipment, no over-speed tests like emergency shutdown were performed. This load cases are happening rarely only once every few years, so this could be neglected as the main source for the damage to the runners.

Fig. 4 shows the history of load cycles during the testing campaign. Negative power means pump operation. Then, the several isolated positive peaks show the machine starts. Finally, the turbine load during the first period of tests includes two part-load to full-load and vice versa cycles performed in dynamic and in static conditions. In between the load tests, the machine was braked down electrically.

Furthermore, three different start-up paths were tested with the opening law of the guide vanes adapted as follows:

- Normal opening speed of $2^{\%}/_{s}$
- Modified opening speed of $1.5^{\%}/_{s}$
- Modified opening speed of $1^{\%}/_{s}$
- Modified opening speed with a broken law, first half of start opening with $1^{\%}/_{s}$ followed by a second half with $2^{\%}/_{s}$

For the synchronization phase of the generator and for the shutdown no changes were applied.

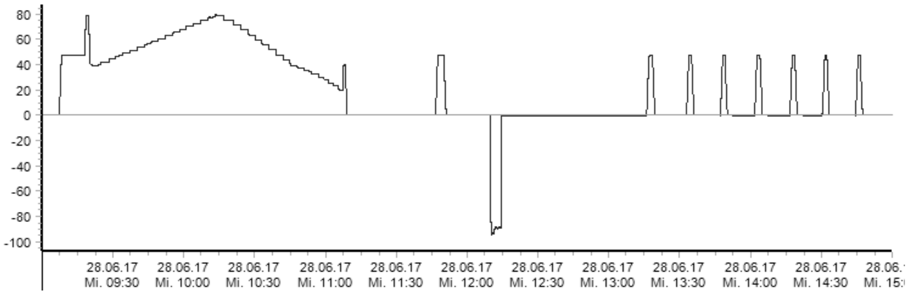

Fig. 4: History of electrical power [MW] during tests.

The given net head at the time of the tests was in between 368.5 mWC and 374 mWC, depending on the losses in the hydraulic system, and the maximum electrical output P_{max} was 79.31 MW. The other groups were running at steady conditions during the tests so that the hydraulic system was stable.

5 Results

5.1 Experimental Results

5.1.1 General Results

The results of the on-board measurements show clearly that there are significant differences in the strain behaviour during a complete cycle from start to standby load (47.5MW) and back to stop (see Fig. 5). On the basis of signal coming recorded from the strain gauge 17, hub, 0°, the load of the runner blades can be clearly separated in the following events:

- filling of the turbine (1)
- initial opening of the guide vanes and getting to nominal speed (2)
- synchronizing of the turbine (3)
- turbine operation (4)
- closing of the guide vanes for shutdown, passing very deep part loads (5)
- turn-off of the generator circuit breaker (6)
- draining of the spiral casing and run out (7)

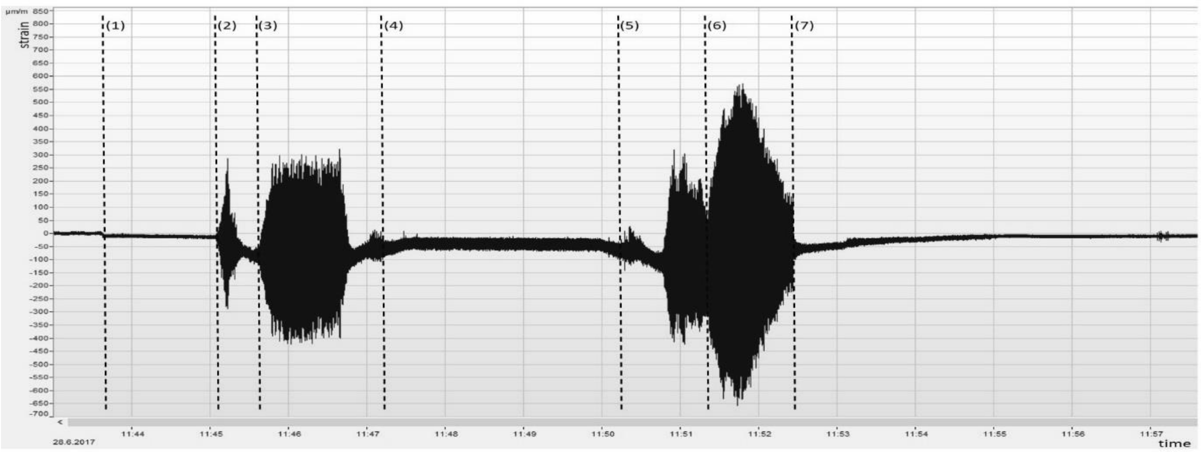

Fig. 5: History of strain fluctuations during one turbine start-to-stop cycle.

As expected according to recent published studies, the speed no load (SNL) operation phase (3) imposes high stresses on the runner. Depending on the frequency state of the grid, this process can take up to several minutes or even leads to a shutdown of the machine if the synchronization fails after a certain time. During operation, the stresses are getting smaller compared to the SNL operation. Passing the very deep part load during the shutdown phase creates a stress peak again. However, the most surprising is the fact that turning off the power circuit breaker (6) imposes the highest peaks of strain onto the blades. This could be observed at all strain gauges and happened during all shutdowns of the turbine. Compared to full load or deep part load, this peak is dominant, as shown in Fig. 6 and its amplitude being up to 4.5 times bigger than at full load.

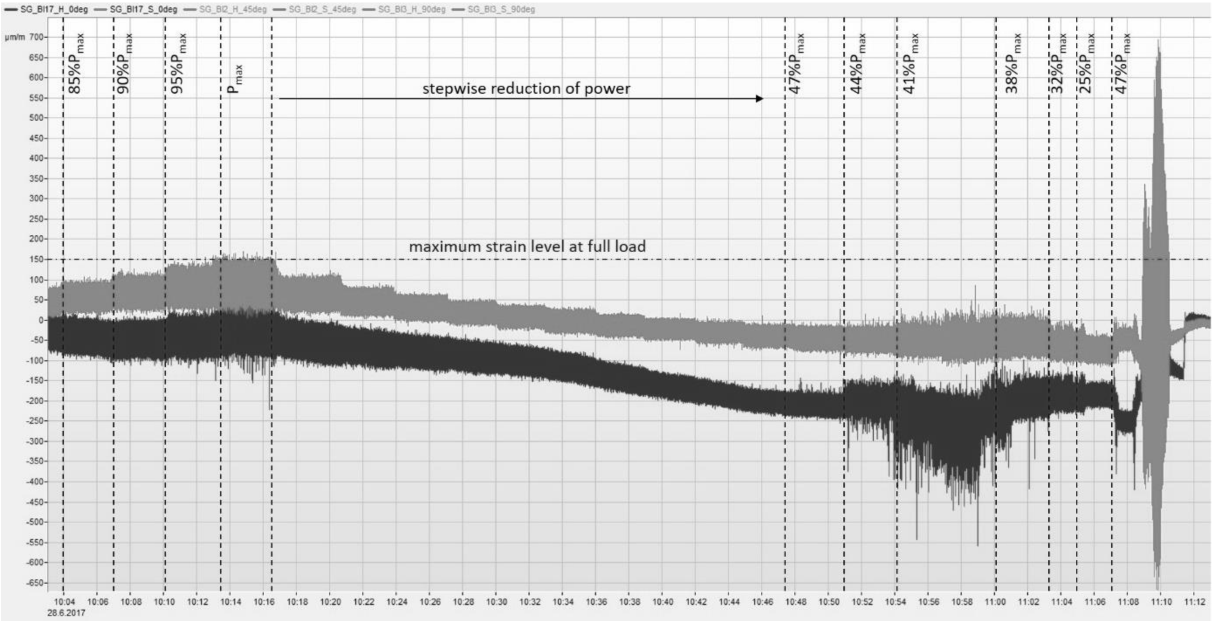

Fig. 6: Strain fluctuations at different loads, blade 17, hub, 0° (dark grey) and shroud, 0° (light grey).

It is also visible in Fig. 6 that there are higher amplitudes in some deep part load operating points in between 41% and 38%, corresponding to the points where the machine appears to be more noisy (cavitation inside the machine being noticed) than at all other loads except start and stop.

5.1.2 Frequency Analysis

The following section shows results from the strain gauge at the hub of blade two, 45° direction, in a complete start-stop-cycle. This corresponds to the region where the cracks used to happen. At stable part load operation, a closer look on the graphs in the top side of Fig. 7 reveals at least some speed dependant frequencies and the blade passing frequency. The first harmonic (12.5 Hz) dominates both the vibrations and the strain, whereas the blade passing frequency (12.5x24 = 300 Hz) is clearly visible as strain fluctuations. Additionally, a sub harmonic frequency is present during this operating point.

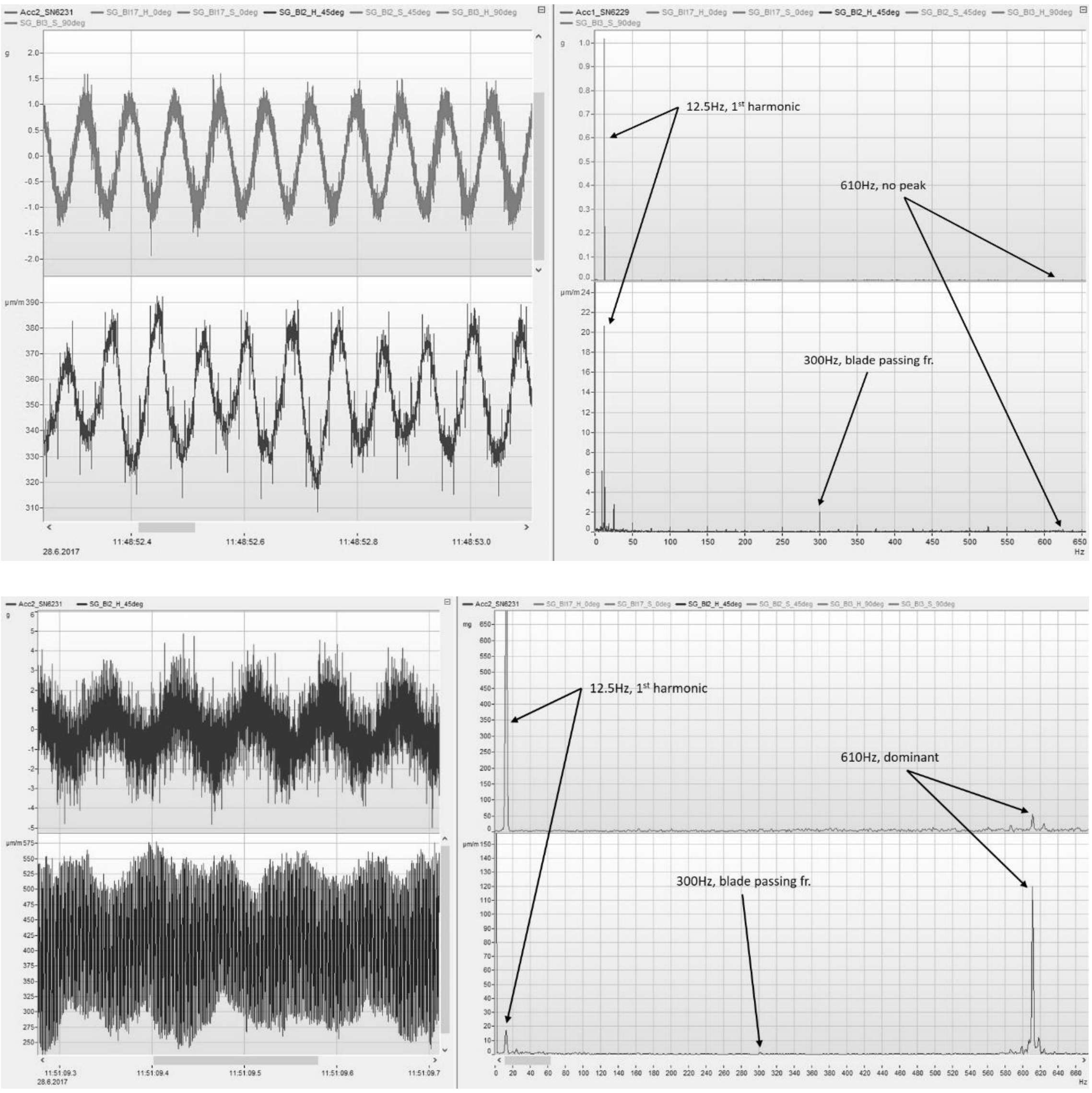

Fig. 7: Dominant frequencies during normal operation (top) and shutdown (bottom).

During the synchronization and shutdown phases, the behaviour changes completely and a new dominant peak at 610Hz appears (see the bottom graphs of Fig. 7). Compared to the values of the peaks during part load stable operation, this frequency imposes big additional load to the blade. Fig. 8 shows this effect during very deep part load before the guide vanes close completely. During the synchronization phase, the frequency behaviour looks very similar.

After the circuit breaker is turned off the strain fluctuations show the highest amplitude values. A closer look into the spectre shows two peaks in the region of 611Hz which diminish with decreasing speed. The second peak also adds a significant amount of strain which leads to the high values at this point. The source of this excitation is to be found in the machine control. Once the minimum power is reached the power circuit breaker turns off which influences the speed of the machine. This leads to a reaction of the turbine governor which is trying to keep speed by regulation the guide vanes and consequently keeps the turbine at SNL condition. This takes as long as the next step, shutdown of the erection, is completed (around several seconds). Only then, the guide vanes are closed completely.

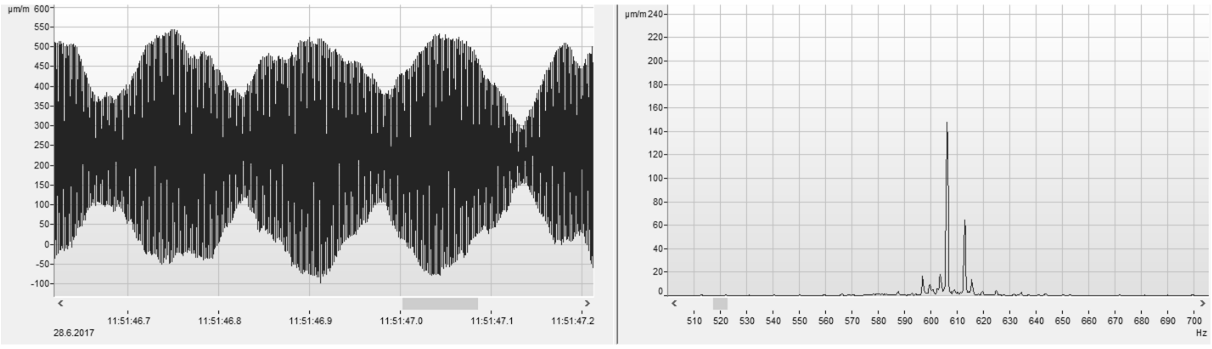

Fig. 8 : Strain fluctuations after the electrical shutdown.

The observed frequency of 611Hz compared to the nominal rotating frequency of 12.5Hz ($f/f_n = 48.9$) is very close to the first harmonic of the blade passing frequency and therefore shows the response of one of the runner Eigen modes (see the results presented by Decaix et al. [9]). Since this clearly dominating frequencies have been also detected by the external microphone, the proximity sensors and the accelerometers on the turbine casing, further non-intrusive measurements can be done by using this results as a foundation.

5.1.3 Variable Guide Vane Opening Speeds

The tested variations in opening speed of the guide vanes to reach nominal speed showed no improvements in the strain behaviour of the machine (see Fig. 9). Moreover, with the broken opening law, a structural excitation frequency in the cavern was triggered, being detected by the vibration monitoring system of the machine, but undetected on the runner blades.

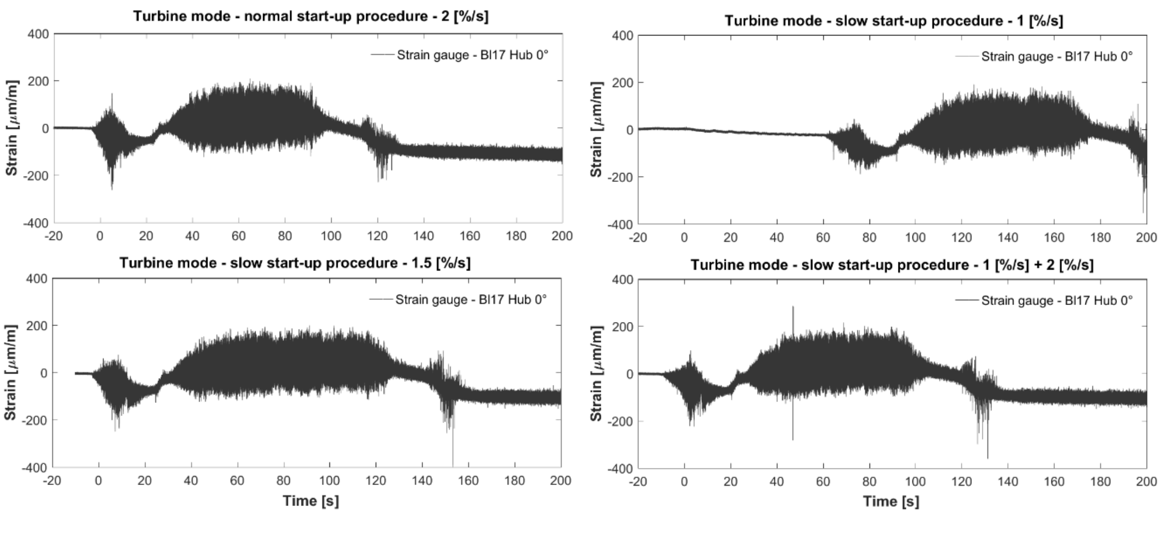

Fig. 9 : Strain fluctuations during normal and modified slower turbine start-ups.

6 Conclusion

The performed measurements show clearly that there are certain operating points that lead to high stresses at the Francis turbine during modern operating conditions. For the plant owner, it is obvious that reducing the number of start-stop cycles will increase the lifetime of the existing runners drastically. On the other hand, it was proofed that deeper part loads do not lead to harmful loads to the runner as expected from empirical observation. For the plant owner this means a wider field of exploration and though a reduced number of machine starts. At least one harmful point at shutdown can be easily eliminated, which is subjected to ongoing measurements.
The main frequencies found to be responsible for the ongoing fatigue process have been measured with the non-intrusive instrumentation. Further measurements at the other groups of the plant are planned to proof this theory and to improve the methodology of this measurements.
This means that such measurements are a valuable possibility to detect harmful frequencies. This gives a powerful tool to optimize the operation of such turbines.

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