Case Study and Numerical Analysis of Vibration and Runner Cracks for the Lipno I Hydroelectric Project

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Abstract. The refurbishment of the Lipno I TG2 Francis turbine, situated on River Vltava, with maximum net head of 165 m and required operational range from 0 to 67MW of turbine power was performed in 2014. The new hydraulic design of the spiral case, distributor and runner was developed for this project. After about 1000 hours of operation the site inspection was performed and the cracks were found on 8 runner blades of 17 blades altogether. The all cracks were found near runner hub beginning from the trailing edge. The dimensions of the cracks were different with maximum length of 123 mm and minimum length of 3 mm. The runner was repaired and the intensive investigation was started to define the main cause of the cracks creation and to determine the measures for their elimination. This paper presents the program of this investigation which consists of static and dynamic blade strain measurement, CFD and FEM analysis, discusses the crack causes and overview the solution how to return the turbine successfully to operation.

1. Introduction

Lipno I Hydroelectric Project (HEP) lies in the south part of Czech Republic on the River Vltava. As a part of Vltava Cascade, which collects nine hydro power plants and dams along the River Vltava, Lipno I HEP is mainly used for a long-term runoff regulation to increase the minimum flow, limit flood peaks, and increase the generation at the other hydro power plants. Considering the area, Lipno I is the largest artificial lake in the Czech Republic with water surface of 48.7 km$^2$. It is 42 km long and 5 km wide at its widest point. The lake is set in the beautiful countryside of Šumava Mountains in the very close to the Šumava National Park and it is used for summer recreation, water sports, and effective fish-farming. The Lipno I HEP was built between 1953 and 1958 and power plant commenced operation in 1959. The power house with its equipment and auxiliary systems is located in the underground cavern in depth of 160 m (see Figure 1 and 2).

There are two sets of Francis turbines with maximum output 67 MW each and maximum net head 165 m. The water is brought to the turbines by two 160 m long steel penstocks and carried away by a 3.6 km long underground discharge tunnel. Francis turbines have ability to quickly increase their output to the maximum in 226 seconds, so they can influence the country’s power balance at any given time. Turbines are completely remote controlled from the Hydroelectric Power Station Control Centre at Štěchovice.
In 2012 an extensive overhaul of unit TG2 started. The scope of work of CKD Blansko ELI Consorcia (CKD Blansko Engineering and CKD Blansko Holding) was to supply the new Francis turbine with reference diameter 2150 mm, distributor, spiral case, upper part of the draft tube cone and turbine shaft. The turbine shaft was planned to be refurbished only, but due to the visible defects the shaft had to be supplied as new one. The auxiliary devices and high pressure governing systems are also completely new. The overhaul of TG2 was finished in 2014 by commissioning.

2. Runner Cracks

An unexpected issue occurred during the regular inspection day in February 2015. After 1067 hours of operation and 422 turbine start-ups several cracks were found at the runner blades. Linear cracks travelling from the trailing edge along the runner hub were observed (see Figure 3). Some of them were located on the suction side only and others went through to the pressure side. The length of cracks varied from 3 mm to 123 mm. Turbine was not allowed to be used for further operation, runner was dismounted, transported back to the factory and the intensive investigation started to define the main cause of the cracks.

Several samples were removed from the affected locations for the detailed material analysis. The first finding was that the material analysis found the defects in the solid material of the samples (see Figure 4). It must be mentioned that runner as such passed all regular acceptance criteria at
the workshop before (including ultrasonic and X-Ray inspection, etc.) but no defects were indicated. Otherwise, it was obvious that material defects which were found after the detailed material analysis could affect the fatigue stability of the blade-hub junction in a negative way. Runner was decided to be repaired so it underwent an extensive non-destructive inspection. An inspection was performed to check the existing situation and to find out the other possible defects. After that, all indications were repaired.

At the same time, three main phenomena were investigated to be the most likely cause of the runner cracks initialization.

- **Rotor-Stator Interaction (RSI):** Regarding to the fact that the new spiral case has a different number of guide vanes (20 vanes, if compared with the old one - 24 vanes) and the number of runner blades remained (17 blades), it was necessary to check rotor-stator interaction once again and in a greater detail.

- **Von Kármán Vortex:** The shapes of all trailing edges (TE) of the runner were checked again after runner’s arrival to the factory. It was concluded that their real shape could be sensitive to the excitation through Von Kármán Vortices.

- **Speed-No Load operation:** As the turbine underwent more than 400 start-ups and operated at speed-no load continuously for several days, it was necessary to check the blade excitation at start-up and speed-no load operation. This kind of operation creates intense stochastic flow with high dynamic amplitudes which is still hard to predict by numerical methods. The only way how to determine the real amplitudes and frequencies is on-site measurement.

3. CFD Analyses of the Turbine

The possible sources of the excitation causing the cracks initiation were analysed by CFD methods. The main goal was to find out the values of the excitation frequencies as well as their amplitudes and to make the comparison with the natural frequencies of the runner in order to determine the possible risks. The CFD simulations were performed for the model scale with runner diameter of 320 mm and turbine nominal speed 1000 min⁻¹. Then the results were scaled up for the prototype turbine.

3.1. Rotor-Stator Interaction (RSI)

The rotor-stator interaction was analysed as the one of possible causes of the cracks. The maximum power of the turbine was chosen to be the turbine operational point, at which the rotor-stator interaction is the most significant. The transient incompressible CFD analysis was performed in ANSYS CFX with the time step corresponding to the one degree of the runner rotation. The transient incompressible CFD analysis was performed in ANSYS CFX with the time step corresponding to the one degree of the runner rotation. Two-phase model water-water vapour was used for the analysis. Figure 5 shows the CFD model for the numerical simulation and the contours of static pressure distribution in the runner, distributor and vaneless area, which is situated between runner and guide vanes.

![Figure 5. CFD model of the turbine (left) and static pressure distribution in turbine (right)](image-url)
The RSI pressure pulsations were analysed for stationary parts (guide vanes) as well as for the relative space of rotating parts (runner blades). The influence of the pressure taps position was considered in the analysis. The static pressure was scanned in three positions in the stator as well as in the rotor (near hub, in the channel center and near shroud). The FFT analyses in the Figure 6 shows that the biggest pressure amplitudes are reached near shroud of the runner (distributor), where the distance between the runner blades and the guide vanes is the shortest. The dominant excitation frequencies \( f/f_n \) (multiples of speed frequency) correspond to the number of blades (17 runner blades, 20 guide vanes). The maximum values of the RSI pressure pulsation are similar for the stator as well as for the rotor pressure taps.

3.2. Von Kármán Vortex Street

The runner blade trailing edge (TE) was analysed in detail by CFD. The goal was to simulate the Kármán vortex street from the runner blade TE for different operational points and its influence on dynamic behaviour of the turbine. The CFD model was simplified to 2D coordinates for one transformed meridional stream-surface of the runner (see Figure 7). The solver Fluent with unsteady setup and Scale Adaptive Simulation (SAS) model of turbulence was used for the simulation. Von Kármán vortex street (contours of velocity) at area of trailing edge with original shape is shown in Figure 8. The inlet boundary conditions for unsteady simulation described above were exported from the steady state CFD analysis of the entire turbine.

![Figure 6. FFT analyses of pressure pulsations in vaneless area: Stator (left) / Rotor (right)](image)

![Figure 7. Analysed runner streamline](image)

![Figure 8. Von Kármán vortex street (original TE)](image)

The inspection of the original shape of the runner blade TE identified also some differences to the prescribed geometry. The results of CFD simulation confirmed pressure fluctuation as risky regarding to the blade vibration. The suggested modification of the blade TE considering the cavitation limits of the turbine is shown in this Figure 9. The modification of the runner blade trailing edge decreases the
vortex-induced pressure pulsations from 100% to 30%. Also the frequency of the pulsations is slightly shifted towards the higher values in case of TE modification (see Figure 10). This modification was applied to the prototype runner (by grinding) in order to decrease the risk of the blade damage due to the von Kármán vortex-induced vibrations. The more extensive modification of the runner blade TE was not practicable due to the hard cavitation conditions on site.

**Figure 9.** Original and modified shape of blade TE

**Figure 10.** FFT analysis comparison (right)

### 3.3. Speed-No Load Operation (SNL)

The turbine Lipno I HPP was operated from 0 MW to maximum turbine power 67 MW. According to experiences the turbine operation at SNL (0 MW) and at the low part load (< 10 MW) might be the reason of the runner blade cracks initiation. Many authors (e.g. [1] and [2]) describe rated speed with no load to be the unorganized flow in the runner which causes cavitating channel vortices and results in high amplitude pressure fluctuations of stochastic nature. They also mentioned that LES analysis is the only way how to simulate the stochastic nature with turbulence fluctuations. Even the unsteady URANS simulation cannot predict the dynamic loading with stochastic characteristics at SNL or low part load at all.

**Figure 11.** Flow in the runner at SNL operation: Model tests (left) and CFD analysis (right)

The transient CFD analysis of SNL operation in ANSYS CFX (with the setup similar to the RSI simulation, described above) was performed. The results confirmed the cavitating channel vortices in the runner. In the Figure 11 the comparison of the flow in the runner at SNL between the model tests and the CFD analyses can be seen. The visual similarity seems to be very good as the cavitating area in the runner is placed similarly. On the other hand, any pressure fluctuations of stochastic nature in
the CFD simulation were not found. Thus the conclusion that the transient CFD simulation of SNL using the standard models of turbulence does not catch the stochastic nature of the flow was confirmed. This conclusion is also stated in e.g. [1] and [2].

Stationary mean values describing the time averaged flow field can be predicted quite well by steady-state CFD simulation. The resulting distribution of static pressure can be applied for subsequent finite element analyses providing accurate static stress distributions.

4. FEM Analysis

Besides the CFD analyses the FEM analyses were performed to describe the behaviour of the system. The vibration of the complete rotor of the HPP Lipno 1 TG2 with the 17-blade runner was analysed considering water environment. The vibration properties were computed using the ANSYS software. The analysed dynamic system consisted of the complete rotor with the runner surrounded by an adequately limited region of fluid. The runner, geometrically modelled in great detail, was fixed to the flange (see Figure 12). The shaft was flexibly supported by three radial bearings modelled using COMBIN14 type elements with specified, experimentally determined stiffness and by one axial bearing modelled using COMBIN14 elements with specified, experimentally determined stiffness. The auxiliary alternator, starting engine, alternator, etc., constituting the other parts of the rotor, were modelled with respect to their inertial properties only (i.e. by concentrated masses). The fluid domain was modelled using approx. 1.5 million elements of FLUID221 type. The elements in contact with the runner had four degrees of freedom per node (three of them correspond to displacements and one corresponded to pressure), the other elements in the domain had one degree of freedom (pressure). The complete computation model of the runner – fluid system involved 2.2 million elements and 3 million nodes with 5.2 million degrees of freedom. The properties of steel parts and properties of fluid were defined by values of Young’s modulus $E$, Poisson’s ratio $\mu$, density $\rho$ and speed of sound $c$.

A plenty of calculations were performed in order to find in which operational point the blade suffered from dangerous excitation. It was necessary to evaluate the natural frequencies of the system in the water. It was very important to determine the eigen modes of the runner blades especially those eigen modes with stress peak in blade-hub junction (see Figure 13).

The results were used for determination of the strain gauge proper position on the prototype.

5. On Site Measurement

After runner reparation the unit TG2 was assembled again and an extensive testing started on site to determine the runner blade excitation at different operational points. Runner was equipped with the strain gauges, which were placed near the expected maximum stress position on the blade. Altogether three strain gauges were installed on three runner blades.
All wires with electric signal led through the existing runner holes towards the turbine shaft (see Figure 14) and were brought out through the service hole to the wireless transmitter which was tightened to the shaft (see Figure 15). It served for a data exchange with the data acquisition computer.

Figure 16 shows the Time-Stress recording from the strain gauges on blade 1, 2 and 3. Turbine operated from start-up through SNL, low load (2 MW and 4 MW), high load (50 MW) to shutdown and coast down while strain gauges measured the stress on the blades. Mention that the blade stresses at the start up, SNL operation and the turbine shutdown are significantly higher than at the other operational points (low load, high load, coast down). Dominant frequency of this high intensity vibration was measured to be 280 Hz and it is common for all these operations. Comparing with the FEM modal analysis results the frequency around 280 Hz could be related to the 3\textsuperscript{rd} eigen mode of runner blade.

As Lipno I HEP is able to operate in condense mode, it is equipped with the compressors, pressured air tanks and related piping. The idea was to test the effect of pressured air on the runner blade vibration. Figure 17 shows the Time-Stress recording from the strain gauges on the same blades during the air admission test at SNL (the strain gauge on the blade 3 was not working any more).
For air admission tests the existing piping was used and pressured air was brought downstream of the runner into the draft tube cone. Short time (approx. 4 sec.) air admission was only allowed by control system, but it was sufficient enough for the measurement. The influence of air admission was checked three times in this run.

It was found out that the pressured air affects positively the blade excitation and decreases the blade stress in the measured locations significantly.

After that the influence of air admission was checked for start-up, low load operation and for the shutdown process. All measurements led to the finding that the most dangerous operation and the most likely cause of the runner blade cracks is long term operation at SNL. Hence, the suitable solution had to be found to return the turbine back to the operation.

6. Conclusions
An intensive investigation found the cause of the runner cracks of Lipno I HEP. The other suggested phenomena (RSI and excitation through von Kármán vortices) did not match the conclusions of the testing on site. The frequencies and amplitudes of RSI were measured to be at the lower bound of the excitation range and they did not influence the fatigue live of the runner significantly. Runner blade trailing edge was modified in order to minimize possible excitation through the von Kármán vortices. No excitation was measured after the modification. Operation at speed-no load as well as the turbine start-up and shutdown process led to the significantly higher stress amplitudes on the runner blade. Initialization of runner cracks could be also supported by the small and hardly detectable defects in the original solid material. Air admission test through existing piping into the draft tube decreased the amplitudes distinctly.

The new piping for separate air admission into the draft tube cone was installed. The new algorithm for air admission and air regulation at problematic operation was added to the control system.

For the future, it is planned to optimize the turbine start-ups and shutdowns by the replacement of the governing system in 2016. Existing control system does not allow performing this kind of optimization.

Formerly, the water turbines were mostly used for power delivery only, so they operated around the best efficiency point. On the other hand, these days the water turbines are also widely used for grid services so they have to be able to operate in stand-by regime, SNL, or at very low loads. The flow in the runner is strongly non-uniform and stochastic at these operational points. Moreover, it is still challenge to predict the dynamic behaviour of the runner by using of the CFD software in reasonable time. Additionally, the increasing of the start-stop cycles is also dangerous for fatigue live of the turbine. As mentioned in [1] “one start-up operation is equivalent to years of operation under normal operating conditions” The only way how to avoid the damage of the turbine parts is sufficient stiffness. The possibilities of numerical prediction at SNL are still the object of research.
7. References

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