Static and dynamic stress analyses of the prototype high head Francis runner based on site measurement

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Abstract. More efforts are put on hydro-power to balance voltage and frequency within seconds for primary control in modern smart grids. This requires hydraulic turbines to run at off-design conditions, especially at low load or speed-no load. Besides, the tendency of increasing power output and decreasing weight of the turbine runners has also led to the high level vibration problem of the runners, especially high head Francis runners. Therefore, it is important to carry out the static and dynamic stress analyses of prototype high head Francis runners. This paper investigates the static and dynamic stresses on the prototype high head Francis runner based on site measurements and numerical simulations. The site measurements are performed with pressure transducers and strain gauges. Based on the measured results, computational fluid dynamics (CFD) simulations for the flow channel from stay vane to draft tube cone are performed. Static pressure distributions and dynamic pressure pulsations caused by rotor-stator interaction (RSI) are obtained under various operating conditions. With the CFD results, static and dynamic stresses on the runner at different operating points are calculated by means of the finite element method (FEM). The agreement between simulation and measurement is analysed with linear regression method, which indicates that the numerical result agrees well with that of measurement. Furthermore, the maximum static and dynamic stresses on the runner blade are obtained at various operating points. The relations of the maximum stresses and the power output are discussed in detail. The influences of the boundary conditions on the structural behaviour of the runner are also discussed.

1. Introduction

In modern electricity grids hydropower stations are increasingly used for primary control in order to balance voltage and frequency within seconds. This requires the turbines to run at off-design conditions, especially at low load or speed-no-load. With this and with the tendency of increasing power output of existing power stations, hydraulic turbine runners, especially high head Francis runners, are facing more challenges. Therefore, it is important to carefully carry out life time analysis based on the dynamic behaviour analysis of prototype high head Francis runners. Plenty of investigations are focusing on the dynamic behaviour of Francis turbine runners [1-6]. However, most of them are only numerical simulations without the validation by measurements. Some experimental model analyses were carried out on the Francis model runners in air and in water in order to obtain the natural frequencies and mode shapes [7-12]; however, rare experimental modal analysis was
performed on the installed prototype runners, especially in water [12]. Some numerical simulations were conducted with limited measurement data to predict the dynamic response and stresses of the prototype Francis runners [3-6]. However, few site measurements were performed to measure the stresses of the prototype Francis runners and rare measurement was carried out on prototype high head Francis runner. This paper presents and discusses a thorough experimental and numerical study of dynamic loads, static and dynamic stresses on a prototype high head Francis runner at various operating conditions.

2. Site measurement of the prototype Francis runner

2.1. Description of pressure sensors and strain gauges

The site measurement was performed on a high head Francis turbine with pressure transducers and strain gauges. The turbine has 28 guide vanes and the runner has 17 blades. The runner is designed for a rated head of 377 m. The nominal rotational speed is 375 rpm. The locations of the pressure transducers and strain gauges are shown in figure 1.

6 pressures transducers are located close to the trailing edge of the runner blade. P1, P2 and P3 are located at the pressure side of the blade, S1, S2 and S3 at the suction side. There are 14 strain gauge signals which are located close to the trailing edges on the runner blades. Some of the strain gauges (1R, 2R...) are installed pointing in the radial/ spanwise direction, while the others (1T, 2T...) are pointing in the tangential/streamwise direction. The strain gauges 1R, 1T, 2R, 2T, 3R, 3T are on the suction side of the blade near the band, while the strain gauges 11R, 11T, 12R, 12T, 13R, 13T are on the suction side of the blade near the crown. Strain gauges 14R and 4R, placed on the pressure side of the blade, are on the opposite side of the strain gauges 12R and 2R respectively. Strain gauges 15R and 5R are at the same location as 12R and 2R but on the neighbouring blade (see figure 1).

The measurement data were recorded with a sampling frequency of 1613 Hz. The overview of the measurement data (pressure and stress), lasting from start to shut down of the unit, is shown in figure 2.
Figure 2. Time domain signals of pressures transducers and stain gauges from start up to stand still.

From the measurement data, 12 steady operating conditions are selected for further analysis. The corresponding powers in percentage of these 12 operating points are listed in table 1.

Table 1. Selected operating points for analysis.

|          | OP0 | OP1 | OP2 | OP3 | OP4 | OP5 | OP6 | OP7 | OP8 | OP9 | OP10 | OP11 |
|----------|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|------|------|
| P/P\text{\textsubscript{rated}} | 0%  | 21% | 29% | 36% | 45% | 53% | 61% | 71% | 79% | 88% | 97%  | 106% |

2.2. Measured static and dynamic stress

After post-processing the measurement data, the static stresses of the strain gauges (3R, 3T, 13R, 13T) are shown in figure 3. It can be clearly seen that the static stresses for each strain gauge change with the power of unit. The static stresses in the spanwise (R) direction of the positions near the crown decrease with the increasing power, while the ones in the streamwise (T) direction slightly increase. The static stresses located close to the band show an opposite behaviour.
Dynamic stresses on the runner can be caused by different hydraulic excitations. But the most important hydraulic excitation on high head Francis runner is induced by rotor-stator interaction (RSI). RSI is generated by the relative motion between the rotating runner blades and the guide vanes in the spiral casing.

The exciting frequency on runner due to RSI depends upon the number of guide vanes and the rotational frequency. The guide vane passing frequency \( f_{GVp} \) acting on runner can be obtained by the following formula:

\[
f_{GVp} = Z_{GV} \times \left( \frac{n}{60} \right) \text{ Hz} = 28 \times (375/60) \text{ Hz} = 175 \text{ Hz}
\]

where \( Z_{GV} \) is the number of guide vanes, and \( n \) is the rotational speed in rpm.

The RSI induced dynamic stresses of a strain gauge at various operating conditions are shown in figure 4.

![Figure 3. Static stresses at various operating conditions. (Separated in spanwise and streamwise direction.)](image)

![Figure 4. Dynamic stresses at various operating conditions.](image)

The RSI induced dynamic stress has a large increase from OP4 to OP5, and RSI becomes the dominant hydraulic phenomenon up to overload (OP11). Between OP5 and OP11 a decrease of the dynamic stresses can be observed.
3. Numerical simulation of the prototype Francis runner

3.1. Numerical static stress analysis

The stress analyses of this Francis runner are executed using the finite element method (FEM). The pressure distributions for various load cases are based on corresponding computational fluid dynamics (CFD) analyses. The density of the steel runner is 7700 kg/m$^3$, Young’s modulus is 206 GPa and Poisson’s ratio is 0.3. The applied boundary conditions for the static stress at one load case are displayed in figure 5.

![Finite element model and boundary conditions](image)

**Figure 5.** Finite element model and boundary conditions for static stress analysis.

In order to compare the numerical results with the measurement ones, the static stresses on the strain gauge locations (see figure 1) are read out from the FE analyses. Figure 6 compares the static stresses of the simulation and measurement. It can be seen that the static stress for each strain gauge changes with the power.

![Comparison of static stresses at different locations](image)

**Figure 6.** Comparison of static stresses at different locations.

The agreement between numerical and experimental results is visualized using linear regression analyses (see figure 7). This is done for all operating points. One dot in the graph denotes one strain gauge. The horizontal axis represents the numerical result while the vertical one stands for the
measurement result. The slopes of the linear regression functions for all 12 operating points are close to 1, which indicate a good agreement between simulation and measurement.

![Figure 7. Comparison of static stresses at different loads.](image)

Based on the good agreement between measurement and simulation, the locations and maximum static von Mises stresses versus power are illustrated in figure 8.

![Figure 8. Locations and maximum static von Mises stresses at different loads.](image)

3.2. *Numerical modal analysis*

The modal analysis comprises of the derivation of the important natural frequencies of the runner with respect to RSI. The analysis is executed with the runner submerged in water; hence the added mass effect of the water is taken into account. The material properties of the runner have been introduced. The density of water is 1000 kgm$^{-3}$ and the speed of sound in water is 1200 ms$^{-1}$.

The numerical modal analysis is performed. The FE-model of the runner, surrounding water and the boundary conditions are shown on figure 9.
As mentioned before, the RSI induced exciting frequency over the runner is the guide vane passing frequency (175Hz). The most important nodal diameter (ND) mode of the runner excited by RSI can be derived from the following formula:

$$\text{ND} = m \cdot Z_B - m' \cdot Z_{GV} = 2 \cdot 17 - 1 \cdot 28 = 6$$

where $Z_B$ is the number of runner blades, $m$ and $m'$ are integers 1, 2, …

The 6ND mode shape and the corresponding natural frequency are shown in figure 10. The natural frequency of 6ND is very close to the guide vane passing frequency.

3.3. Numerical dynamic stress analysis

By means of the harmonic response analysis the dynamic stresses on the submerged runner caused by RSI are calculated. The dynamic pressure distribution is taken from the unsteady CFD analysis.
The calculated load cases are RSI dominated OPs for this runner. The FE model of the runner and the boundary conditions are shown on figure 11.

![Figure 11. Finite element model and boundary conditions for dynamic stress analysis.](image)

Since for this runner the exciting frequency is quite close to the natural frequency of 6ND, the dynamic stresses are very sensitive on the damping. Figure 4 shows a decrease of the dynamic stress of the stain gauge between OP5 and OP11. With linear regression analyses, the damping values of OP5 and OP11 were adjusted so that the calculated dynamic stresses, caused by RSI, match the measured ones (see figure 12). A higher damping is present for OP11 than OP5. The damping of OP11 is about twice of the one of OP5.

![Figure 12. Comparison of dynamic stresses at different loads.](image)

For the assessment the maximum dynamic von Mises stresses are illustrated in figure 13. The hot spots for OP05 and OP11 are at the same location, but the values are different. The calculated maximum dynamic stress of OP05, which has a lower damping, is double of the one of OP11.
4. Conclusions
Static and dynamic stresses caused by rotor-stator interaction (RSI), at various operating points of a high head Francis runner were obtained with site measurement and numerical simulation.

The calculated static stresses show a good agreement with the corresponding measurement values for all measured operating points. The static stress analysis shows that both the locations and values of the maximum static von Mises stress on the runner blade change with power.

The modal analysis shows that the natural frequency of 6ND is very close to the guide vane passing frequency.

The exciting force caused by RSI for this runner is dominant from middle load to overload. The damping values of OP5 and OP11 were adjusted in the harmonic response analysis so that the calculated dynamic stresses caused by RSI match the measured ones.

It could be demonstrated that the damping increases in this operating range. Hence a higher damping is present for overload (OP11) than middle load (OP05), and the dynamic stress at OP05 shows a higher value than the overload (OP11).

The tendencies of the maximum static and dynamic stresses are depending on the power of the unit.

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