Fatigue analyses of the prototype Francis runners based on site measurements and simulations

X Huang¹, J Chamberland-Lauzon², C Oram¹, A Klopfer¹, N Ruchonnet¹

¹ANDRITZ HYDRO AG. Hardstrasse 319, Zürich, 8021, SWITZERLAND
²ANDRITZ HYDRO. 6100, Aut. Transcanadienne Pointe Claire, H9R 1B9, CANADA.

E-mail: xingxing.huang@andritz.com, Joel.Chamberland-Lauzon@andritz.com, christian.oram@andritz.com, Andreas.Klopfer@andritz.com, Nicolas.Ruchonnet@andritz.com

Abstract. With the increasing development of solar power and wind power which give an unstable output to the electrical grid, hydropower is required to give a rapid and flexible compensation, and the hydraulic turbines have to operate at off-design conditions frequently. Prototype Francis runners suffer from strong vibrations induced by high pressure pulsations at part load, low part load, speed-no-load and during start-stops and load rejections. Fatigue and damage may be caused by the alternating stress on the runner blades. Therefore, it becomes increasingly important to carry out fatigue analysis and life time assessment of the prototype Francis runners, especially at off-design conditions. This paper presents the fatigue analyses of the prototype Francis runners based on the strain gauge site measurements and numerical simulations. In the case of low part load, speed-no-load and transient events, since the Francis runners are subjected to complex hydraulic loading, which shows a stochastic characteristic, the rainflow counting method is used to obtain the number of cycles for various dynamic amplitude ranges. From middle load to full load, pressure pulsations caused by Rotor-stator-Interaction become the dominant hydraulic excitation of the runners. Forced response analysis is performed to calculate the maximum dynamic stress. The agreement between numerical and experimental stresses is evaluated using linear regression method. Taking into account the effect of the static stress on the S-N curve, the Miner's rule, a linear cumulative fatigue damage theory, is employed to calculate the damage factors of the prototype Francis runners at various operating conditions. The relative damage factors of the runners at different operating points are compared and discussed in detail.

1. Introduction
The solar and wind power, as the new players in the energy market, give unstable output to the electrical grid. The smart grid requires the hydropower to give a rapid and flexible compensation by running the runners at off-design operating conditions and by an increased number of start and stops. The runners thence may suffer from strong vibrations induced by high pressure pulsations. These strong vibrations can lead to high dynamic stresses, decrease the design lifetime and therefore increase the operation cost of the unit. Fatigue damage and cracks caused by high level stresses are not
uncommon appearing on the prototype runners [1-4]. The owners of hydro power plants are eager to know the cost of operation of the runners for the whole operating range. Therefore, it becomes increasingly important to carry out fatigue analysis and life time assessment of the prototype Francis runners, especially at off-design operating conditions including start-stops.

A great quantity of work has been carried out on structural response and dynamic behaviour of Francis turbine runners [1-20]. However, only few of them performed stress analysis of the prototype runners based on sites measurements and numerical simulations [14-18]. And rare investigation was performed on fatigue analysis and life time assessment of the prototype Francis runners. This paper presents the fatigue analyses and the corresponding life time assessment of the prototype Francis runners based on the strain gauge site measurements and numerical simulations.

2. Site measurements
A large site measurement campaign was carried on several Francis runners. For each runner, strain gages were installed on the blades and strains were measured during transient and steady-state normal operating conditions.

In order to carry out a successful strain measurement on the prototype Francis runner, suitable positions for the strain gauges have to be selected. The stress hot spots on the runner blades at various operating conditions can be roughly predicted with stress calculations. Figure 1 shows the typical measured signal of one strain gauge installed on a prototype Francis runner. The static and dynamic stresses at different operating conditions can be obtained from the measured signals.

![Figure 1](image.png)

*Figure 1.* Time domain measured signals of a strain gauge.

3. Francis runner damage factor calculation

3.1. Overview of the damage calculation

Figure 2 shows an overview of the runner damage factor calculation, which needs the static and dynamic stress analyses.
3.2. Static stress analysis
The measured static stress is obtained by averaging the signal at each operating point and at each measurement locations. The corresponding numerical static stress can be calculated by finite element method (FEM) with the pressure distribution from computational fluid dynamics (CFD) simulation (figure 3). Static stress calculation is a standard procedure and was validated using strain gauges measurements of several Francis turbines [17,18]. The validation procedure is repeated for the runners studied in this paper, the agreement between the measurements and the simulations is found similar to previous work.

![Figure 3. Static stress distribution for a Francis runner blade.](image)

3.3. Rotor-stator interaction induced dynamic stress analysis
Pressure pulsations caused by rotor-stator-interaction (RSI), the relative motion between the rotating runner blades and the guide vanes, become the dominant hydraulic excitation of Francis runners from middle load to full load (zone III of figure 2). The RSI induced exciting frequency of Francis runners is the guide vane passing frequency and its harmonics. The guide vane passing frequency ($f_{GVp}$) with most of the excitation energy can be calculated by:

$$f_{GVp} = Z_{GV} \times \frac{n}{60} \text{ Hz}$$

where $Z_{GV}$ is the number of guide vanes, and $n$ is the rotational speed in RPM.
With the Fast Fourier transform (FFT) analysis the RSI induced dynamic stresses of the strain gauges at $f_{GVp}$ can be evaluated. With last decade developments on RSI dynamic stresses, the dynamic pressure CFD calculation and the harmonic response of the runner have become standard analysis in the industry. The finite element model of a runner used for harmonic response analysis is shown in figure 4.

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

**Figure 4.** Finite element model and boundary conditions for dynamic stress analysis.

The dynamic stress of the runner is shown in figure 5. With the same procedure the dynamic stresses of different runners at various operating conditions can be calculated.

![RSI induced dynamic stress distribution for a runner](image)

**Figure 5.** RSI induced dynamic stress distribution for a runner.

### 3.4. Stochastic loads induced dynamic stress analysis

At low part load, speed-no-load (SNL) and transient events (zone I of figure 2), the Francis runners are subjected to complex dynamic loading and random fluctuations dominate the strain gauges measurements. Some recent developments have shown promising results in calculating dynamic stresses under SNL [19].

From a pool of blade deformations obtained by standard calculations, the one having the best fit with strain gages measurement is selected. The peak stress amplitude is calculated by a geometrical extrapolation from the strain gauges to the peak stress location.
3.5. Combined dynamic stress analysis
For a given operation point (zone II of figure 2), the RSI stresses and the stochastic stresses are separated. For each component of the signal the hot spot stress time signal is calculated using a combination of the measurements and the simulation results. The resulting peak stress time signals are added together with the simplification that the stress direction is exactly the same.

3.6. Fatigue loading calculation
The extrapolated peak stress signals are composed of periodic and stochastic components that both contribute to fatigue of the material. The stress amplitudes of the cycles to be used for fatigue analysis are extracted using a rain-flow cycle count algorithm. The rain-flow cycle count algorithm is the standard method to evaluate fatigue cycles occurring with random loading. For simplicity, the static stresses obtained for the operating point is used as the mean value for all cycles.

The damage factors for the studied operating points are calculated using the Miner’s rule with the fatigue S-N curve for CA6NM stainless steel in water. The fatigue equivalent stress is calculated using the Goodman relationship. The total damage of an operating point is the sum of the damage of each counted cycles.

4. Fatigue analysis and life time assessment
Five prototype Francis runners with different speed factor ($n_{ED}$) are selected to carry out the fatigue analysis and expected lifetime assessment following the methodology described in section 3.

4.1. The studied turbines
The studied turbines cover the whole range of typical large Francis runners. A brief description of the turbines is presented in table 1. The rated power of the runners is described by three categories: Medium Power [0-200 MW], High Power [200-400 MW] and Very High Power [>400 MW]. The design type of the runners is also described by three categories. A standard designed turbine is a turbine that operates normally in the range of 50-100% of the maximum power. A wide operating range turbine can operate in a wider range of operating power and head than the standard one. A heavy-duty turbine is designed to work frequently at different operating points in the whole power and head range.

| Runner | $n_{ED}$ | Rated Power Category [MW] | Design type          |
|--------|----------|---------------------------|----------------------|
| A      | 0.4      | [200-400]                 | Standard             |
| B      | 0.4      | [>400]                    | Wide operating range |
| C      | 0.3      | [200-400]                 | Heavy-duty           |
| D      | 0.2      | [0-200]                   | Standard             |
| E      | 0.3      | [0-200]                   | Standard             |

4.2. Damage caused by periodic and stochastic stress
From SNL to 40% of maximum power, the stochastic stresses dominate the signals content. In these conditions the flow is turbulent and periodic structures are not developed. In the rope zone, the flow starts to be more organized and the stochastic pressure fluctuations reduce and periodic pressure fluctuations such as RSI or distributor unbalance may become more apparent. Despite the fact that the rope may generate strong and apparent pressure fluctuations, the effect on the runner is small. On the second half of figure 6 a) the flow becomes very well organized and the signal is dominated by RSI. As shown in figure 6 b) the stress cycles occurring are composed of a combination periodic and stochastic stresses.
a) Stresses function of operating point  

b) Rainflow at 45% of rated Power

**Figure 6.** Stress variation with operating condition

4.3. Comparison of the relative damage factors

The relative damage factors of all 5 Francis runners are calculated and demonstrated in figure 7. The damage factors of steady operating conditions including SNL are calculated for a continuous operation period of one day.

**Figure 7.** Relative damage factors at different operating conditions of five runners.
4.3.1 Relative damage factors of low load operation

For most runners, the dynamic stresses measured at low load operation are in the same order of magnitude as at speed-no-load. The operation in this zone for a reduced time period will not create significant damage to the runner. However, if the runner is operated for an extended period of time in the low load region, the lifetime of the runner may be consumed within the first years of operation of the runner unless the runner was designed for this specific type of operation as shown with runner B [20].

4.3.2 Relative damage factors of start-stops

Compared with other load cases the start-stops contribute a lot to the damage of each runner. The damage caused by one start-stop equals the damages of many hours, even many days of low load operation or SNL. One start-stop reduces the runner lifetime as much as many years of operation at full load.

The comparison of the relative damage factors between start-stop and SNL indicates that for the examined runners it is better for life consumption of the turbine to keep them running as spinning reserve for a few hours than to start many times per day. However, to run at SNL has a cost with the losses of water.

5. Conclusions

Fatigue analyses and life time assessments of five prototype Francis runners are carried out based on site measurements and numerical simulations.

The strain signals of the prototype runner measurements were analysed and compared with the FEM results. The static and dynamic stresses at various load cases are calculated based on suitable CFD and FEM analyses.

Using the Miner’s rule, the relative damage factors of the runners at different load cases are calculated and compared.

For all the studied runners, the start-stops contribute a lot to the damage of the runners. The damage factors of a few hours low load operation or SNL are less than that of one start-stop.

Francis runners which are designed for a wide operating range should have robust behaviour at low load. Good prediction of the damage factors is essential to assess the cost of operation of Francis runners during the design phase.

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