Francis design and prediction technology for flexible operation

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Abstract. Traditionally, hydro power plants have been operated close to the best efficiency point, the more stable operating condition for which they have been designed. However, because of changes in the electricity market, many hydro power plant operators wish to operate or already operate their machines differently to fulfill new market needs. New operating conditions can include the whole 0 to 100% range of operation, numerous start/stops, extensive low load operation, synchronous condenser mode and power/frequency regulation. In order to design reliable 0-100% Francis™ runners for these new challenging operating scenarios, Andritz Hydro has developed various proprietary tools and design rules. These are used within Andritz Hydro to design mechanically robust Francis runners for the operating scenarios fulfilling customer’s specifications. Hydraulic development and mechanical verifications are done conjointly to converge toward an optimal solution. Francis runner dynamic stresses on the complete operating range from 0-100% load can be predicted and compare well with prototype strain measurements. In addition to operation under a wider operating range, the number of start-ups has also a significant impact on Francis runner fatigue life. In our paper, we will present details of the design considerations required for operating Francis turbines from 0 to 100% power.

1. Introduction

Until the last decade, hydro turbines were designed and operated as base load machines, operating close to the best efficiency point for extended periods of time. Deregulation of the energy market and an increase in highly variable power sources such as solar and wind created a need to provide additional power regulation and ancillary services to the grid. Hydro power has the capacity to efficiently provide timely response for power regulation as well as inertia, beneficial for the stability of the grid. Although very well suited to provide these services, the change of operation has created loads on some units that were not designed for such operation, leading to excessive vibration, stability issues or premature failure of the runner blades.

In response to these new design requirements, new FSI (Fluid-Structure Interaction) prediction tools must be developed in order to achieve designs that are suitable for the intended operation of the utilities [1] [2]. This paper presents the challenges of designing 0-100% Francis™ runners for today’s market.

2. FSI simulation technologies

While operating out of the traditional normal operating zones (<60% Popt), the flow angles on a single regulated turbine are not aligned with the blade angles, which leads to flow detachment, high vorticity and energy dissipation.
The basics in which flowing water dissipates energy is well explained with the Komolgorov turbulence cascade [4]. The larger eddies have the most energy, the smallest eddies account for the energy dissipation. As shown in figure 1, the turbulence induced stress – apparent from the dense and wide frequency content dominates the spectra in Deep Part Load (DPL) < 40% Pmax and Speed No Load (SNL). The rope is apparent in PL, at conditions around 60% of Pmax. However in PL, the broadband turbulence induced stresses have already significantly reduced compared to SNL and DPL conditions. Other operating points (> 60% Pmax) are dominated by more deterministic phenomena such as the interaction between the rotor and the stator, or the hydraulic unbalance at once per revolution. In the context of a fluid structure interaction problem, one must be able to resolve flow patterns down to a certain size that is still sufficiently large to provide relevant excitation to the structure. In practice this level of resolution is found through time step and mesh refinement sensitivity analysis and experimental results comparisons.

2.1. CFD modeling of turbulence
As presented by Nennemann et al. [5] and later by Morissette et al. [6], the SAS turbulence model is suitable to predict the low frequency content (<10f,ω) of the pressure fluctuations applied on a Francis runner blade operated in a no load condition. It is also discussed that the simulation would benefit from a mesh and time step refinement to better match the measured spectra. However, such increase in mesh size and time step refinement would increase the simulation time to a point where it is completely unpractical for the purpose of engineering design. It is obvious that the missing energy in the higher frequency range may lead to an underestimation of the predicted stresses. In facing this challenge of achieving conservative predictions, in a practical manner, some engineering workaround may be employed.

2.2. Water coupling
The Francis runner dynamic system is composed of a solid structure coupled to the surrounding water [7] [8]. The coupled system can be expressed as a simplified two dimensional matrix and sub-matrix
system, with the displacement as the degree of freedom for the structure and the pressure for the surrounding water, were $R$ is the coupling matrix between the fluid and the solid, $m$ and $k$ are submatrices. Forces acting on the system are expressed by the matrices $F$: $F_s$ for structural forces and $F_f$ for fluid forces.

\[
\begin{bmatrix}
  m_z & R \\
  R & m_f
\end{bmatrix}
\begin{bmatrix}
  \ddot{u} \\
  \ddot{p}
\end{bmatrix}
+\begin{bmatrix}
  k_z & R \\
  R & k_f
\end{bmatrix}
\begin{bmatrix}
  u \\
  p
\end{bmatrix}
=\begin{bmatrix}
  F_z \\
  F_f
\end{bmatrix}
\]

In the case of a highly disorganized flow with the presence of cavitation and small deformations of the structure, it is acceptable to neglect the effect of the vibration of the structure on the flow. The difference of the modes in still and flowing water is also neglected.

\[
\begin{bmatrix}
  m_z & 0 \\
  R & m_f
\end{bmatrix}
\begin{bmatrix}
  \ddot{u} \\
  \ddot{p}
\end{bmatrix}
+\begin{bmatrix}
  k_z & R \\
  0 & k_f
\end{bmatrix}
\begin{bmatrix}
  u \\
  p
\end{bmatrix}
=\begin{bmatrix}
  F_z \\
  F_f
\end{bmatrix}
\]

This coupled equations system can be solved using a FEM model of the runner, using acoustic elements for the fluid and coupling elements at the interface of the solid and the fluid. The coupling elements are responsible of applying the displacements from the solid elements to the acoustics elements and the pressure from the acoustic elements to the solid elements.

2.3. Simulation time
Morissette et al. [6] also presented a study of the influence of the simulation time on the maximum stress predictions. Since the modeled physics are highly stochastic, simulation time must be long enough in order to capture the largest stress variations occurring a few times per minute.

3. SNL and DPL dynamic stress predictions
A prediction tool was developed in order to predict the levels of dynamic stresses at SNL as presented by Nennemann et al. [5]. In the present paper, the validation of the model is extended to the data from nine prototype runners at various operating points.

3.1. Measurements
For each measured runner, the number of strain gauges and their locations on the runner were defined based on stress results of Finite Element Analysis (FEA). Uniaxial strain gauges were installed close to hot spot stress regions and aligned with the maximum principal stress direction. Multiple blades were instrumented, replicating the measurement position and allowing cross-validation of the measurements. Finally all measured runners had machined blades with very limited variation of the geometries between the blades.
3.2. Simulations
All simulations were performed with similar CFD setup. The meshing is performed automatically, mesh refinement size is set as a function of the prototype runner overall dimensions. Time stepping is defined at 2 degrees of rotation per time step. Simulations are performed at the measured head, setting and power.

The mechanical simulations are performed using ANSYS and an Andritz proprietary simulation software. A sector of the runner is modelled with second order elements and appropriate mesh refinement at the junction of the blade with the crown and the band and in the region of the strain gauges. The strain values are extracted at the exact location and orientation of the strain gauges.

3.3. Comparison metric
The values from the various strain gauges measured are compared to the simulated values using linear regression. The slope of the linear regression is then used to evaluate the quality of the prediction. In order to fully validate the model it must also be validated that the stress pattern can be correctly predicted which is ensured by a good correlation quality. The measured and simulated signals have sufficient duration to obtain statistical convergence as discussed in section 2.3.
An example of the comparison between measured data and simulation results is presented in figure 4. The selected case is runner 2 DPL from table 1 in section 3.4. Signals that are out of phase with the reference strain gauge on the same blade, at the moment of the largest cycle, are put negative to better represent the stress pattern.

![Figure 4. – Comparison between experimental and simulation stresses (black) linear regression, (red) interval for linear regression, (green) 95% confidence interval for data.](image)

3.4. Results

The simulation results from 8 runners are compared with measurements results in table 1, where \( Q_{ED} \) is the specific flow factor \( Q_{ED} = \frac{Q}{D^2 \sqrt{gH}} \). All 8 runners are non-homologous designs, therefore different results are expected even for turbines with similar characteristics. The validation is performed mainly for SNL conditions since it is the condition from which the most measurements are available. The comparison was also performed for some DPL conditions for runner 2 with satisfactory results. What is interesting with this validation point is that runner 2 recorded the highest DPL dynamic stress to SNL dynamic stress ratio \( \frac{\Delta \sigma_{dpl,dyn}}{\Delta \sigma_{snl,dyn}} \) among all measured units. For each runner, the simulation error is calculated by comparing the simulated and measured stress amplitude. In order to compare the significance of the various results, the measured stress amplitude normalized with the maximum stress amplitude measured among all runners is presented.

Runner 8 had a significant relative over-estimation of the dynamic stresses. It must be noted that in the case of these small size, high head runners the absolute dynamic stress is very low and the absolute error is comparable to the one from other runners. This outlier is not problematic for two reasons: the dynamic stresses are over-predicted preventing an under-designed runner, and even the over-estimated dynamic stresses are low preventing an oversized runner design as a result of these predictions.

The analysis of multiple runners is showing that the simulation tool developed is suitable for dynamic stress predictions for Francis runners operating at SNL. As for DPL, further validations are required for direct prediction of these conditions using simulations, which are currently carried out at Andritz Hydro. However, based on SNL prediction, it is possible to conservatively estimate the DPL stresses based on measured stress from similar prototype runners.
Table 1. – SNL and DPL dynamic stress prediction results.

| Turbine | Q_{ED} | Condition | Reference Diameter D_{ref} | Simulation prediction error compared to measurements [%] | Normalized stress amplitude [%Ref] |
|---------|--------|-----------|----------------------------|------------------------------------------------------|-----------------------------------|
| Runner 1 | 0.22  | SNL       | 4m<D_{ref}<6m              | -16.2                                                | 52                                |
| Runner 1 | 0.22  | SNL-2     | 4m<D_{ref}<6m              | 12.0                                                 | 52                                |
| Runner 2 | 0.36  | SNL       | 4m<D_{ref}<6m              | +3.0                                                 | 54                                |
| Runner 2 | 0.36  | DPL       | 4m<D_{ref}<6m              | -1.0                                                 | 86                                |
| Runner 3 | 0.26  | SNL       | 4m<D_{ref}<6m              | +4.0                                                 | 58                                |
| Runner 4 | 0.34  | SNL       | D_{ref}>6m                 | -2.1                                                 | 90                                |
| Runner 5 | 0.14  | SNL       | 2m<D_{ref}<4m              | -19.6                                                | 42                                |
| Runner 6 | 0.28  | SNL       | D_{ref}>6m                 | -2.0                                                 | 100                               |
| Runner 7 | 0.19  | SNL       | D_{ref}<2m                 | +8.2                                                 | 41                                |
| Runner 8 | 0.13  | SNL       | D_{ref}<2m                 | +329.0                                               | 10                                |

4. Start-up stress predictions

As shown in Chamberland-Lauzon et al. [9] start-ups are also contributors to the fatigue damage of Francis runners. As a matter of fact, for most Francis turbines the largest stress cycles may be seen during a start-up [10] [11].

The hydraulic design and mostly the mechanical design of the runner have influence on the amplitude of stresses that the runner will see. The opening sequence of the guide vanes has also a great influence on the dynamic stress amplitudes for a given machine.

Nowadays, with modern governor systems and high pressure injection in the thrust bearing pads it is quite easy to adjust the opening sequence of the start-up during the commissioning of a new unit. By doing so the transient stresses can be reduced by half, leading to a much longer runner life.

4.1. Start-up stress analytical model

An analytical model was developed to predict the dynamic stresses occurring for a given start-up sequence. The model was validated using strain gauge measurements of different start-up sequences on 5 of the 8 runners from table 1.

Once the model parameters are fitted, it is possible to predict the acceleration curve and the hydrodynamic intensity. The hydro-dynamic intensity is an indicator of the pressure fluctuations occurring in a given time frame of the start-up. Hereafter, it is assumed that the dynamic stresses are proportional to the pressure fluctuations, thus proportional to the hydro-dynamic intensity.
4.2. Relative optimisation results

The analytical model predicts the acceleration of the runner and the time evolution of the hydrodynamic intensity during start-ups. The comparison of the simulations and the measurements is shown in figure 5.

![Figure 5. – Relative start-up optimisation [ - baseline stress, - baseline hydro-dynamic intensity, - optimized stress, - optimized hydro-dynamic intensity].](image)

First, the runner acceleration curve was correctly predicted by the analytical model. The hydrodynamic intensity is proven to be an indicator of the stresses as it reaches its maximum at the same time as the runner stresses. The relative amplitude of the hydro-dynamic intensity is also proportional to the measured stresses when two cases are compared. This confirms that the intensity parameter is a good indicator to make a relative comparison between start-up sequences and their relative maximum stress amplitudes and associated fatigue damage.

4.3. Absolute transient stress predictions

Relative optimization can give positive results and reduce the risk of premature cracking, but an absolute assessment of the fatigue life cannot be performed. In the process of designing new Francis runners, the mechanical designer must evaluate the amplitude of such transient stresses in order to properly evaluate the expected lifetime of the new runner. Based on its numerous Francis runner strain gauge measurements, Andritz has developed an empirical rule to estimate conservative dynamic start-up stresses based on predicted hydro-dynamic intensity which is however beyond the scope of this paper.

5. Integration in the design process

Andritz Hydro uses an integrated design process between the hydraulic specialists and the mechanical engineers [1] [12]. The optimisation loops from the hydraulic design cannot be performed without verifying the mechanical integrity of the designed blade and then getting clear directions for optimizing the runner. At these validation steps, the blade and their associated pressure fields, for all relevant conditions, are passed to mechanical engineering for analysis. If the runner cannot be
accepted even with minor modifications it is returned for re-design. If the runner can be accepted and hydraulic performances are fulfilled, it is released for model testing or production.

![Diagram of the integrated runner design process.](image)

**Figure 6.** – Integrated runner design process.

To be integrated in this design process, simulation technologies must be developed so that a full calculation may be performed within a few days.

6. **Turbine Design**

Stationary components must also be designed to sustain the additional loads induced by off peak operation and frequent start and stop. This design consideration is subdivided in various aspects.

![Section view of a Francis unit.](image)

**Figure 7.** – Section view of a Francis unit.

6.1. **Stability and pressure fluctuations**

Rough zones are unavoidable even for runners designed for part load operation. Model testing can help in identifying the location of these possible rough zones but amplitude scalability is still a topic of development. Air injection ports may be foreseen in the head cover and in the bottom ring or discharge
ring in addition to the central air admission so that compressed air may be injected to damp pressure fluctuations.

6.2. Turbine Guide bearing
During rough zone operation radial static and dynamic loads are applied to the shaft line, these forces are supported by the turbine guide bearing. The guide bearing and its support are designed so that absolute displacement of the shaft line are limited, that bearing pressure is sufficiently low and that oil temperatures remain stable throughout operation. However, it must be expected as a normal behavior that the shaft vibrations and bearing vibrations be higher in DPL or PL than at the best efficiency point.

6.3. Fatigue of stationary components
The design of Francis runners for 0-100% operation draws a lot of attention because of the complexity of the operational loads in in part load conditions. One must not overlook the robustness requirement for stationary components and their sub-assemblies. As a matter of fact, the pressure fluctuations applied to the runner are, to some extent, also applied to the stationary components. This increase of pressure fluctuations upstream of the runner must be considered in the design of the head cover, the guide vanes and the operating mechanism. For PL conditions the increase of pressure fluctuations downstream of the runner must be considered in the design of the draft tube cone and its apparatus. To properly evaluate the fluctuating loads, the knowledge of a single pressure point is insufficient to capture the component pressure loadings. Instead the pressures should be measured, or calculated, on multiple points in order to evaluate not only the pressure fluctuations but also the synchronization of the fluctuations at different locations.

In cases of frequent starts and stops, fatigue of the component requires in-depth analysis. The frequent starting and stopping will have an effect on the design of the stay ring, head cover, guide vanes, operating mechanism, bottom ring and discharge ring and shaft seal. It should be expected that a machine designed for frequent start and stop be heavier but overall only marginally more expensive.

6.4. Refurbishment and operating conditions changes
For some projects, refurbishment is performed on a limited number of components. For example, it is typical to change the runner and guide vanes but to keep existing covers and operating mechanisms. The replacement of these components would not help in increasing the efficiency, the maximum power output or the operating range of the units, which usually drives the economics of a refurbishment project. Aging equipment was typically not designed for today’s operation that includes frequent starts and stops and off peak operation. It is advised to perform fatigue analysis of existing components using modern FEA tools prior to the start of the rehabilitation in order to reduce the risk of failure and forced outages.

7. Conclusion
The simulation of Francis runner dynamic loads for no load or deep part load conditions is a challenging task, requiring high end CFD simulations and high computational power. The challenges faced by the industry are shared by many researchers from utilities and manufacturers. The required numerical resolution in time and space cannot be reached solely by an SAS simulation within reasonable time. Instead, core knowledge of Fluid Structure Interaction and turbines are used to
develop a suitable model for dynamic stress prediction. The model is validated with measurements from 8 Francis runners at prototype scale.

The prediction of the dynamic stress during a start-up has to go a step further as the system must be resolved in a transient simulation. An analytical model was developed to optimize the opening sequence for the runner stresses and fatigue. It is demonstrated that this analytical model can help with relative stress optimization and that it can be a basis for absolute stress optimization. Turbine design considerations of all components must be looked at carefully for numerous start-up or 0-100%.

Finally, designing Francis runners and units for today’s market requires an integrated process that must cover many aspects and requires high end simulations tools validated by experience.

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