High flexibility in Francis turbine operation and design philosophy: A review

To cite this article: J O Kverno *et al* 2022 *IOP Conf. Ser.: Earth Environ. Sci.* **1037** 012011

High flexibility in Francis turbine operation and design philosophy: A review

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Abstract. This paper examines and discusses how various operation schemes and design philosophies can prolong the expected lifetime of a hydro turbine subjected to many start-stop cycles and high ramping rates. With the proliferation of renewable and non-dispatchable energy sources in Europe, a higher demand for flexible operation is put on the rest of the system. Given the short response time of hydro power, there is a huge potential for the Norwegian power sector to act as a stabiliser for the rest of mainland Europe. With a more flexible operation however, the turbines will experience a higher load variation which can lead to premature fatigue and failure of existing turbines. This research reviews the current startup and ramping schemes, as well as design aspects, which might affect and reduce the mechanical stresses experienced by the hydro turbine.

Keywords: Fatigue loading, Flexibility, Francis turbine

1. Introduction and Background

The 2030 climate & energy framework of the European Union states that by 2030, co_2 emissions should be reduced by 40% from the 1990 levels, and at least 32% of the energy production shall come from renewable sources [1].

With the increased use of wind turbines and photovoltaic panels, which are so called non-dispatchable energy sources, the demand for more flexible and varied operation of traditional energy production is needed in order to maintain a stable frequency on the power grid [2]. According to Weitemeyer et al [3], up to about 20% of the energy demand can be met by intermittent energy sources without any major issues with grid stability, meaning; to reach the 2030 goal, both in terms of an increased use of renewables and a cut in co_2 emissions, hydro power is the perfect source fill in the gap as already seen in the Nordic grid [4]. Hydro power is one of the more flexible sources of energy and can typically go from a cold start-up to full load within a few minutes [5]. Additionally, hydro turbines can rapidly change the power output depending on the requirements and thus can be operated in a fashion which helps to counteract the changes in the grid frequency and voltage, which is directly linked to the mismatch between the supply and demand of energy in the grid. This operation scheme does put an increased mechanical load on existing hydro turbines, as they traditionally were designed and constructed to operate in a more predictable and stable manner [6]. This can ultimately lead to premature fatigue damages and costly downtime [7]. In this paper, the existing research on increased fatigue loads due to flexible operation of Francis turbines will be reviewed. The work behind this paper is a part of a larger project, HydroFlex, a multidisciplinary research project working towards increasing the flexibility of hydro power while also mitigating the environmental impacts.

2. Flexibility and turbine fatigue

During a start-stop cycle a Francis turbine will experience fatigue loads due to pressure oscillations, leading to cyclic loads with high stress amplitudes. This was illustrated when a Francis runner cracked after only 1067 working hours, and 422 startups [8, 9]. After the incident, a methodology developed to estimate the runner fatigue lifetime showed that the cracking could not have happened unless the turbine was operated with many start-stop cycles. Another power plant, with five identical high specific speed units developed visible cracking on three of the units after a mere 700 to 1500 hours of operation in a load peaking manner, leading to several start/stop cycles per day [10]. However, it has been demonstrated [11] that just by altering the guide vane opening scheme during a startup procedure, the overall cyclic loading on the runner can be drastically reduced due to a lower amount of high frequency pressure oscillation in the flow. This show that the expected lifespan of the runner can be extended, compared to a unit started using a non optimised startup scheme. Another thing to consider is the load variation during operation, and how much it will affect the runner [12, 13]. Introducing variable speed operation can reduce pressure pulsations in the turbine, thus reducing the fatigue loads [14, 15, 16]. During operation of traditional Francis turbines, there are several types of flow disturbances and oscillations which causes mechanical stress and fatigue on the unit. Traditional turbines are designed to operate at a narrow range close to the design point [6], and moving away from this best efficiency point (BEP) introduces different kinds of phenomena, depending on the load. During startup, the angle of the incoming flow and the angle of the leading edge of the runner blade will have a large mismatch, leading to flow separation and the generation of vortices travelling down the blade channels, leading to stress on the runner blades. These interblade vortices are typical for low part loads as well (< BEP). At part load, a rotational component of the flow appears in the draft tube while the total volumetric flow rate is low, causing a vortex breakdown and reversed flow in the centre of the draft tube. Above $\sim 50\%$ of BEP, and if the pressure in the draft tube is low enough, cavitation might appear in a helix shaped vortex, and this vortex will rotate around the zone of reversed flow in the centre with a rotational frequency of $f_R \approx \frac{1}{3} f_n$. Due to the low pressure of this vortex rope there will be a rotating pressure field in the cross section of the draft tube, and at the elbow of the draft tube this asymmetrical pressure field might lead to a plunging flow, leading to cyclic mechanical loading on the turbine [17]. When operating at full load (> BEP), there will also be a rotational component in the draft tube, albeit in the opposite direction of the turbine rotation. The volumetric flow is also high, meaning that a cavitated vortex in the centre of the draft tube will remain axisymmetric, as seen in figure 1.

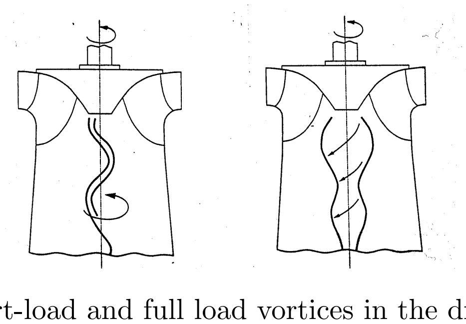
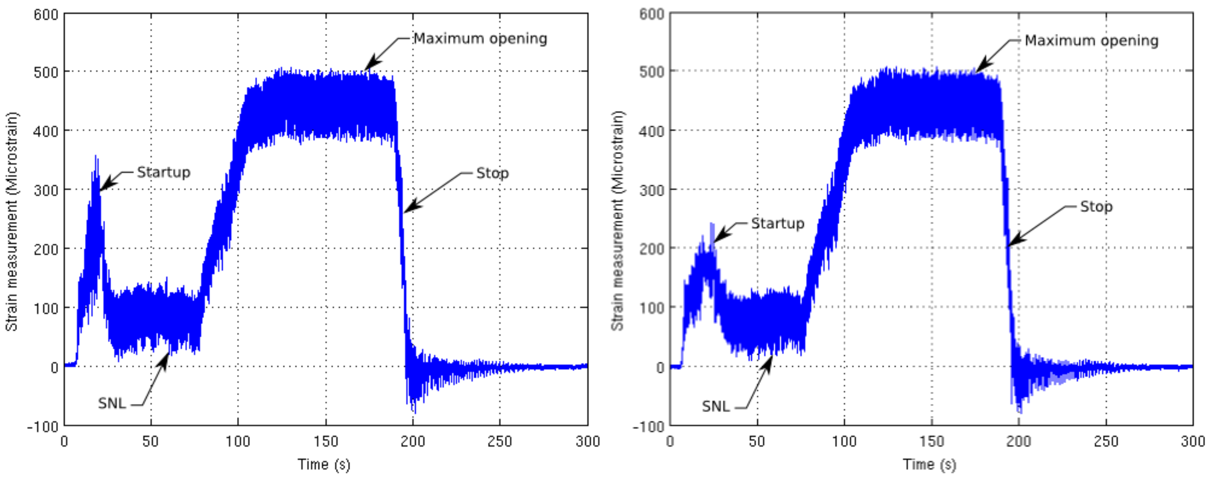


Figure 1: Part-load and full load vortices in the draft tube [18].

Francis turbines will also experience a phenomena called rotor-stator interaction (RSI). The cause of this is the throttling effect a runner blade has as it passes in front of the flow exiting the guide vane channels, leading to a higher pressure as the velocities drop [19]. The end result is that a rotating pressure field appears in the vaneless space with a frequency as a function of the rotational velocity and the number of runner blades (stationary frame of reference) or guide vanes (rotating frame of reference). If the frequency of the RSI or a harmonic of this frequency coincide with the resonant frequency of the turbine, fatigue damage can appear, in the matter of hours even in some cases [20]. If variable speed operation is introduced the frequency of the RSI is no longer fixed but rather move with the rotational speed, meaning that extra care needs to be taken to avoid getting in to resonance at certain operating conditions or speeds of rotation, and the resonant frequencies of the unit in question might have to be mapped out during commissioning so a complete and safe range of operation schemes can be made.

3. Current research

The end goal of the PhD work behind this paper is to perform model tests on a Francis turbine to validate numerical simulations of startup conditions. One key to reducing the fatigue load during a startup sequence, is to optimise the procedure with this in mind. Gagnon et al. [11] did an investigation on how the startup scheme affects the expected lifetime of a Francis runner. They combined theoretical models to estimate crack growth with experimental data from strain gauge measurements on a prototype runner at Beauharnois power plant to get a better understanding of how two different startup schemes alter the runners expected lifetime, as seen in figure 2. The main variable that was changed during startup was the opening degree of the guide vanes. The results showed that just by reducing the opening from 40-50% to just above 30%, the crack growth rate was drastically reduced.



(a) Startup scheme 1 (b) Startup scheme 2

Figure 2: Comparison of strain measurements between the two types of startup schemes showing the reduced strain experienced during startup with the more gentle startup scheme, Gagnon et al. fig. 6 [11].

Another factor which plays in to the fatigue damages to a turbine is how the unit is operated. A turbine providing base load to the grid will tend to operate close to the BEP and in general have few start-stop cycles and thus little to no reduction to the expected lifespan. However, a unit operated for grid stabilisation tends to have more frequent startups, and more operation time spent further from the BEP, as Seidel et al [22] demonstrated. Through experimental

IOP Conf. Series: Earth and Environmental Science 1037 (2022) 012011 doi:10.1088/1755-1315/1037/1/012011

measurements on an operating prototype turbine, model turbine experiments and advanced numerical simulations a better understanding of the different factors which inflict damage to Francis turbines was gained. The results showed that a turbine operating in a base load scenario experiences less than 15% of the relative damage, compared to a grid stabilisation unit (figure 3). The main contributors to the damage was the startup, speed no load (SNL) and low part load. One key finding was that to operate the turbines more flexibly the operation sequences should be optimised, which has not typically been done before.

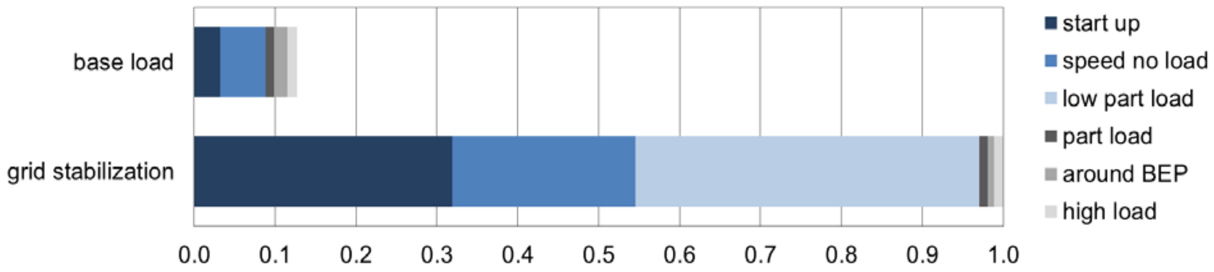


Figure 3: Comparison of the relative damage inflicted by the two operating schemes, Seidel et al. fig. 1 [22].

Halvar Bjørndal et al [21] demonstrated however that the rated head, or specific speed, of a turbine also plays a role when evaluating the main causes of fatigue. Several different Norwegian hydro power plants had been investigated, and it was found that while higher specific speed machines were mostly experiencing failures and blade cracking due to SNL and low part load conditions, meaning that higher part loads and up to full load (above BEP) were unproblematic. Low specific speed turbines however were more susceptible to other phenomena which causes cyclic loading as well, namely RSI. The measured RSI induced blade stress frequencies were also higher than the ones observed from startup and SNL on high specific speed units. The author also concludes that the operator/plant owner should be more specific about how the turbine is to be operated when they put out a tender.

Chirag Trivedi et al. performed a series of pressure pulsation measurements on a model Francis runner [23] and two different prototypes [24]. The main focus of the model experiment was to investigate the flow behaviour during SNL conditions utilising pressure transducers in both the stationary and rotating domain. Both transient and steady state conditions were investigated with the latter at the design speed of the turbine. The results showed an increasing pressure pulsation amplitude in the vaneless space as the wicket gate opened, and at SNL the amplitudes were nearly twice that which occurs during normal operation. An increase in pressure and strain has also been demonstrated numerically on different turbine designs [25, 26]. On the two prototypes pressure was only measured from the draft tube walls, close to the runner outlet [24]. The prototypes were of similar specific speed but different orientation, one vertical- and one horizontal axis turbine. The authors found that both during the initial phase of wicket gate opening and during generator synchronisation the pressure pulsations reached amplitudes of 3,5% of the head, or 2,8 times greater than at BEP. The pulsations were stochastic in nature, i.e. no sign of a vortex rope, and remained even after synchronisation. There were also considerable pulsations during transients with amplitudes of up to 1,6 times that of steady state operation, especially when operating further away from BEP. The orientation of the axis, and thus draft tube, also seemed to influence the behaviour and the type of pressure pulsations. The dominating frequency in the measurements by Chirag Trivedi et al. seemed to be the RSI, and the same can be seen in measurements performed at EPFL in 2004 on a low specific speed model Francis turbine and the associated numerical simulations done by Mélissa Fortin et al. [27].

IOP Conf. Series: Earth and Environmental Science 1037 (2022) 012011 doi:10.1088/1755-1315/1037/1/012011

Resonance in a turbine can cause a failure in a matter of hours, but accurately predicting the effects of pulsations and fatigue without empirical data from the turbine in question has previously not been done, leading to several cases of runners failing. However, as Petter Østby et al [28] demonstrated, important parameters such as the damping ratio can be calculated. With a more plausible damping ratio the numerical simulations of the fluid structure interaction becomes more accurate, and when the numerical simulations of three different high head Francis turbines were compared with experimental data of both on-board pressure and strain gauges, there were an agreement between the results. As expected the pressures had the smallest uncertainty and best agreement, while the dynamic stresses deviated a bit more. In Østby's case, the main focus was on the RSI, but it should still be applicable to other operating conditions as well.

4. Discussion

Flexible turbine operation implies a more frequent and/or rapid change in turbine operation, and since both far off-design conditions and transients have been shown to cause extra fatigue loading on turbines, a shorter lifetime or more frequent maintenance of existing units will probably be the end result [29]. Predicting the exact extent of fatigue damage on any particular unit given a set operation scheme is challenging, especially at deeper part loads due to the stochastic nature of the dynamic loading. Differences in fabrication and weld quality will also play a huge role in determining the mean time before failure (MTBF) [10]. Larger scale CFD simulations [30] and fluid-structure interaction (FSI) calculations [31] can be done to make more accurate predictions, but these are usually computationally demanding and not feasible in a commercial setting. Most Francis turbines are designed for a specific operating condition, and with a high efficiency as a relatively high priority. The best efficiency point (BEP) is typically at around 80% of full load, and as a consequence the conditions during start-up are further away from the optimum than what they would have been if the BEP was at a lower load. This can be better illustrated when looking at the velocity triangles at the inlet of the runner. As figure 4 illustrates, the mismatch

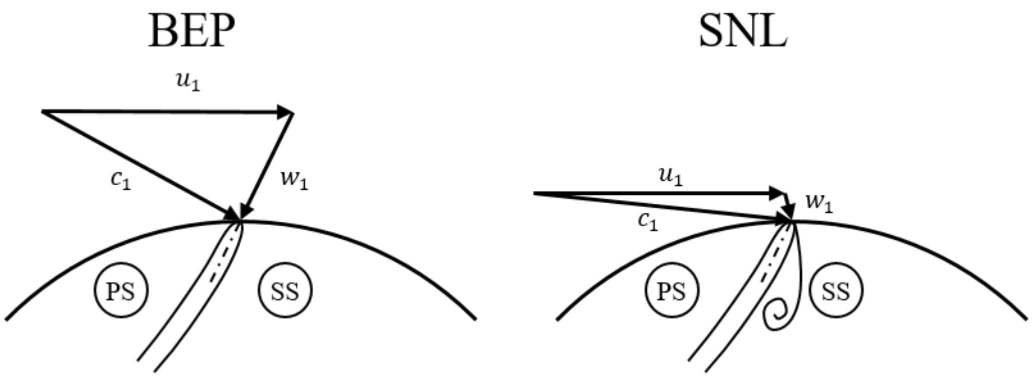


Figure 4: Inlet velocity triangles at BEP and SNL

between the relative flow angle and blade angle causes a flow separation and vortices on the suction side of the blade, as also seen in simulations [25]. These flow instabilities might then propagate through the blade channel towards the trailing edge of the blades, which is typically the location where premature fatigue and failure occurs [8, 10, 11, 20]. One possible solution to the problem of flow separation at the leading edge during SNL is to increase the angle of the runner blade to closer match the angle of the incoming flow. One drawback to this approach is that the BEP is moved to a higher head than what might have been originally intended.

However, this might actually be beneficial for flexible operation, as the turbine might be better suited for a wider operating range with lower part loads. If variable speed operation is included as well the overall efficiency might be increased a bit, and the pressure pulsations reduced. Another possible solution to the problems of flow separation at SNL is to use an aerofoil profile more suited for higher angles of attack.

If we were to consider the Euler turbine equation and what happens at SNL, or runaway in general, the picture might not seem be the same as previously discussed.

$$g \cdot H_n \cdot \eta_h = u_1 \cdot c_{u1} - u_2 \cdot c_{u2} \tag{1}$$

With the naive assumption that the hydraulic efficiency at runaway is zero, or close to zero, the rotational component at the inlet of the turbine can be calculated from

$$c_{u1} = \frac{u_2 \cdot c_{u2}}{u_1} \tag{2}$$

For the example case of a high head Francis turbine, the rotational component is quite drastically reduced from the BEP, contradicting the first assumed behaviour. However, this would also mean that the energy leaving the runner at the outlet is equal to the energy entering at the inlet, which obviously isn't the case. In reality the energy at the inlet is the difference between the available energy from the head and the energy at the outlet of the runner, or

$$u_1 \cdot c_{u1} = g \cdot H_n - u_2 \cdot c_{u2} \tag{3}$$

Meaning that there is no simple way to get the velocity components at SNL. This is probably complicated with secondary flows and channel blockage not being accounted for with the Euler equation as well. A more comprehensive model by Zhang [32] captures these effects better through the combination of the Euler turbine equation and the energy laws, and using some assumptions regarding the hydraulic efficiency and shaft torque. The model seems to make more accurate predictions regarding the speed ratio ($\frac{n_R}{n_N}$), even compared to what is typically given by literature. By assuming a speed ratio of 1, which is the case at SNL, the GV angle and flow rate can be found, provided that the radial geometry of the blade is known.

5. Conclusion and further work

The main goal of the research in HydroFlex is to identify the locations most susceptible to fatigue damages during startup and high ramping, and this is to be done through a combination of numerical and experimental measurement campaign on a model Francis turbine at the Waterpower Laboratory at NTNU. A new model turbine is to be designed and manufactured through the HydroFlex project and its partners. The main goal of the measurement campaign on the model turbine is to acquire the required measurements and sensor installations on board the runner. The results from the experiment shall be used to validate numerical simulations of the fluid structure interactions. To get validation data for the CFD analysis a set of pressure sensors throughout the turbine and test rig can be used to gather data. Additionally, there is a possibility of getting PIV measurements in the vaneless space to get a better understanding of the flow field and velocities during the startup. For the validation of the material stress analysis (FEM), strain gauges will be utilised. To simplify the calibration of these sensors, the trailing edge of the runner blades will have a straight shape, as seen in figure 5. Additionally, to obtain a higher signal to noise ratio the runner blades will be made as thin as possible.

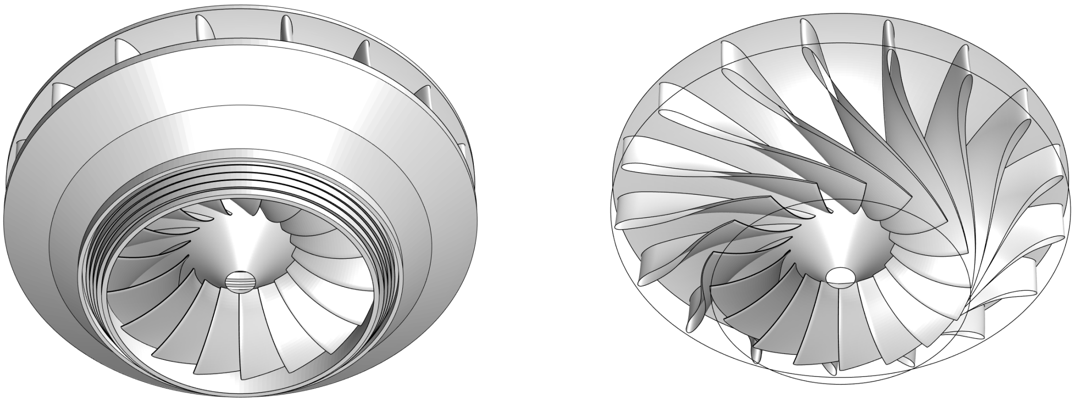


Figure 5: HydroFlex model runner design example with 17 runner blades and a straight trailing edge. The simple blade geometry can also be seen.

Acknowledgments

This work has been done under the HydroFlex project which has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement No 764011. As a part of HydroFlex, the research on the mechanical and hydraulic aspects of turbines are conducted in work package 3 with the partners the Norwegian University of Science and Technology (NTNU), EDR Medeso, Rainpower, Ss. Cyril and Methodius University in Skopje, and Luleå University of Technology.

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