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SPECIFIC SPEED EFFECT ON FRANCIS RUNNER RELIABILITY UNDER VARIOUS OPERATING CONDITIONS

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Abstract. *Energy market deregulation and arrival of new players, such as solar and wind turbines, led to an increasing demand for flexible operation of hydraulic turbines. Instead of continuous close to peak operation, it is nowadays not uncommon to see turbines being operated over the whole range, with many start/stops, extensive low load operation, synchronous condenser mode and power/frequency regulation. This new way to operate the units, however, does not come without cost on the machine life expectancy due to the increased number of high and low amplitude cycles introduced in the operation of the unit. To assess machine reliability, it therefore be-comes critical for the owner to understand the real effects of these dynamic phenomena.*

The purpose of this paper is to show how dynamic phenomena occurring at various operating conditions may affect the lifetime expectancy of different specific speed Francis runners. Runner blade strain gage and pressure site measurements, performed at various locations, and correlated to Computational Fluid Dynamics (CFD) results and structural Finite Element Analysis (FEA) using Fluid Structure Interaction (FSI) techniques, are used to discuss these dynamic phenomena.

Keywords: *Francis, runner, reliability.*

INTRODUCTION

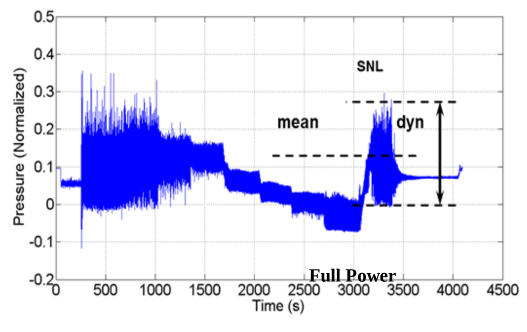
Not so long ago, hydraulic turbines were mainly used as base load machines, with operating conditions varying essentially due to the water head cycling over the year with seasonal rainfalls. Deregulation, followed by the increasing use of fluctuating sources of energy, such as wind and solar, modified the usage of hydropower plants from base load supplier to delivering regulating power and ancillary services to electrical grid [1].

Although very well suited to contribute to grid stability, changing hydropower plant main purpose from base load power provider to delivering balancing power resulted in more operating time in zones of poor weighted efficiency, high pressure fluctuations, significant cavitation, mechanical vibrations, etc. In addition to these rough steady state modes, transients such as start/stops and power swings, also became more common. All these dynamic phenomena do not occur without affecting the lifetime expectancy of runners [2] and should be taken into account during hydraulic development where, with the modern runner design tools, every hydraulic performance characteristic is always pushed to the maximum [3,4].

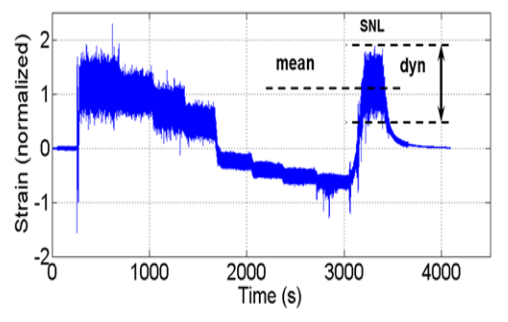
This paper will discuss the dynamic forces occurring on Francis runners under various steady and transient operating conditions and will look at how the relative amplitudes of these dynamic excitations may vary with the specific speed of the machines. The mechanical consequences on runner reliability will also be addressed.

1 OPERATING CONDITION CHARACTERISTICS

The dynamic phenomena occurring on Francis runners vary as the operating conditions change. Figure 1 shows measurements obtained from transducers installed on a blade of a medium head machine with nED = 0.32. Not surprisingly, there seems to be some relation between the pressures on the blade, in this case measured on the suction side mid span between the crown and the band, and the stress, in this case measured at an outlet corner of the blade [5]. However, the resulting stress on the blade has to be taken as an integrator of all the pressures on the runner. Therefore, considering one pressure sensor is not sufficient to predict what will be the stress pattern on the blade even if some correlation may be expected in this case. Various operating conditions will now be examined in more detail.



a) Pressure transducer time signal

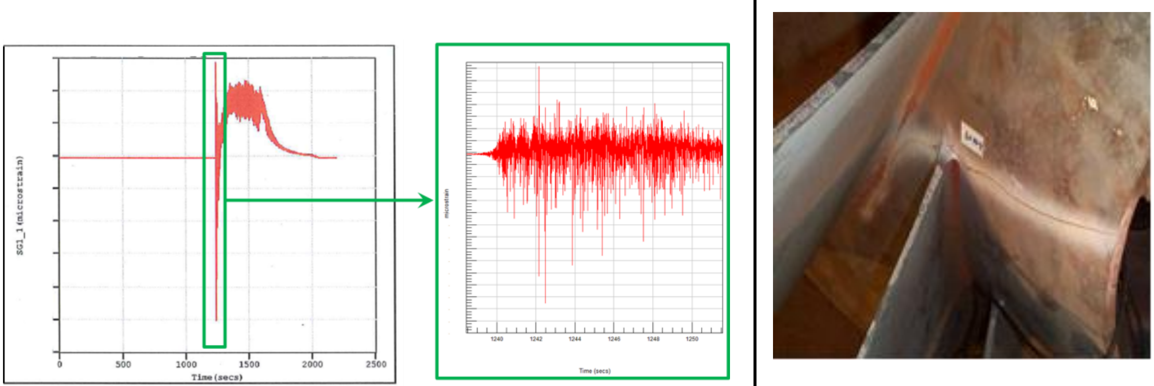


b) Strain gauge time signal

Figure 1. Runner transducer measurement sequences: Standstill, start-up, SNL, power increase to full power, SNL, shut down

1.1 Start-ups

Start-up is a highly transient condition where the runner goes from a completely stopped condition to steady state synchronous Speed No Load (SNL) condition. Figure 2a) presents a start-up event from strain gauge measurement including a zoom on the first part. The stress is highly stochastic and the processing of such a signal requires statistical methods to calculate the effect on runner lifetime [4]. Predicting this stress is still a challenge, but with a large set of experimental data, it is possible for the turbine designer to assess the dynamic stress fluctuations during a start-up based on statistics of some hydraulic and mechanical design parameters. In most cases, it is possible to design runners robust enough for aggressive peaking operation and capable of sustaining extended number of start-ups without special start-up sequence procedures [4] and avoid cracking issues such as the one presented in figure 2b) [6]. Start-up sequence optimization is however possible for existing machines [7].



a) Strain gauge start-up signal

b) Cracking due to multiple Start-ups

Figure 2. Runner start-up event

1.2 Speed no load / low load

SNL and low load operations are steady state operating conditions characterized by highly stochastic loads [2,5]. At speed-no-load the turbine is operated at synchronous speed without electricity production typically in stand-by ready to be connected to the grid. This operation can occur over extended periods of time. For these conditions, the stress is typically highly dynamic and can become quite high in relation to static mean stress as shown in figure 3 even if, as reported in [8], the contribution of speed-no-load static stress to fatigue life of Francis runners can also be significant.

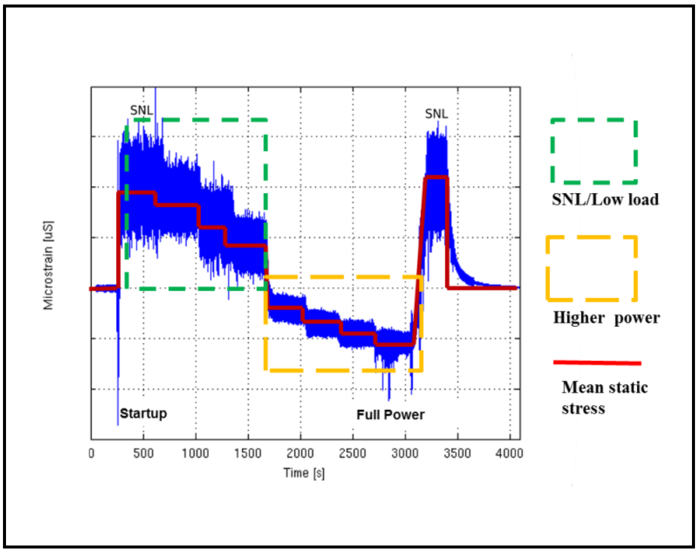


Figure 3. Runner blade strain gauge measurements

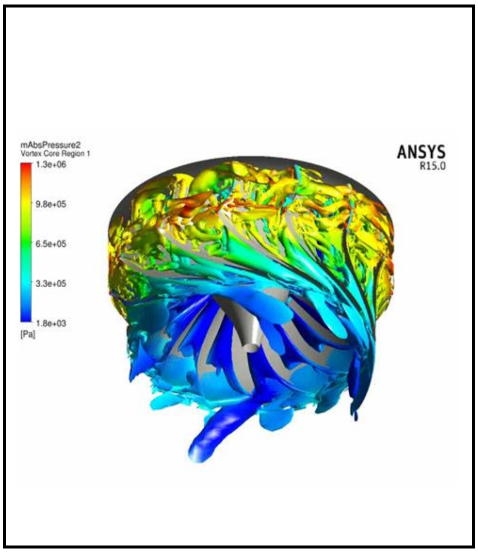


Figure 4. CFD No load simulation

At SNL, the flow passes through the turbine without power generation, but with non-negligible flow rate, and therefore all the potential energy in the flow has to be dissipated somehow. This takes place through a mechanism where the runner channels are partially pumping, thus generating large scale unsteady vortex structures which, by their nature, break down into smaller and smaller vortices until energy dissipation occurs at the smallest scales. A CFD based method for the prediction of the unsteady stochastic pressure loads has been developed and compared to unsteady pressure measurements on prototype Francis runners. CFD results are presented in figure 4. A Finite Element Method (FEM) based approach is being used for the conversion of the stochastic pressure loads into stochastic dynamic stresses, the results of which have been compared to strain gauge measurements on the same prototype in [5].

The stochastic stress field can then be processed to calculate the damage on the runner. Because of potential high stochastic stress amplitudes, SNL and low load operation may be some of the most damaging continuous operating conditions for a hydraulic turbine when compared to other conditions. Some optimization of operating modes may be possible [2,4], but sometimes, these stochastic stresses on the blades can become a constraint in hydraulic design and the blades may often require to be stiffened up at design stage for added robustness.

1.3 Part load

At part load the flow rate is lower that what is ideal for the runner. Because the runner rotation is imposed, this rotation is imparted on the flow passing through the runner resulting in excess swirl at the runner outlet with a large aspect of the swirl character being that of solid body rotation. The helical part load vortex rope is the result of vortex breakdown. The rope frequency is typically in the range of 0.2 to 0.4 times the runner rotation frequency. An example of the vortex rope as seen in laboratory tests and the equivalent in a CFD prediction are shown in figure 5 [9].

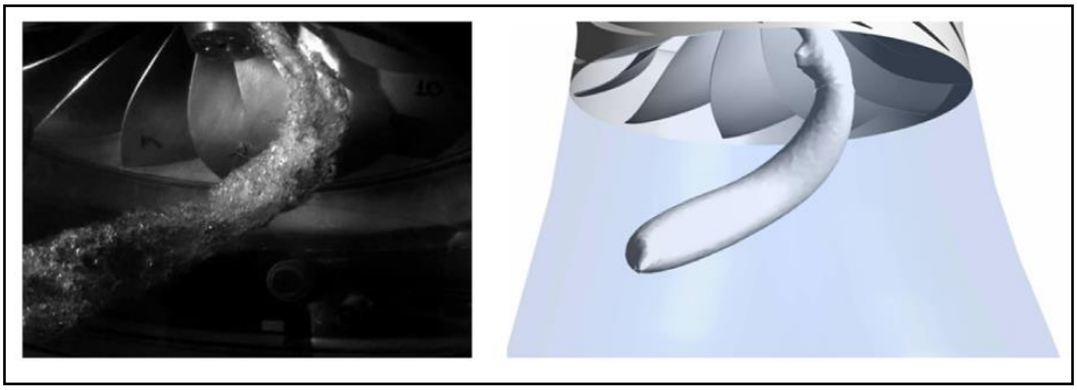


Figure 5. Helical draft tube vortex rope in laboratory visualized by cavitating core (left) and predicted by CFD visualized by means of pressure iso-surface calculated with Ansys-CFX [9]

The part load vortex rope can result in unacceptable pressure pulsations and power swings if its natural frequency coincides with a system natural frequency and the synchronous part of the pressure pulsation is sufficiently large. These effects, however, affect mainly the stationary components. Even if some cases have been reported [10], the part load rope does not generally exert large dynamic stresses on the turbine runner and the rope effect can hardly be seen on runner usage calculation factors [11]. The reason for such situation is illustrated on figure 6 where typical runner strain gauge rainflow diagrams for the same length of time are presented at the same scale for part load and SNL. On figure 6a) dynamic stress of the rope is visible with dynamic range higher than the low general noise. However, both the dynamic range and the number of cycles of this low frequency phenomenon are lower than the dynamic stress ranges and numbers of cycles of SNL stochastic stresses (figure 6b).The resulting cumulated damage on the runner by the rope phenomenon ends up being small. On the other hand, at SNL, the much higher number of cycles and higher dynamic stress range is generating significantly more damage on the runner than part load operation.

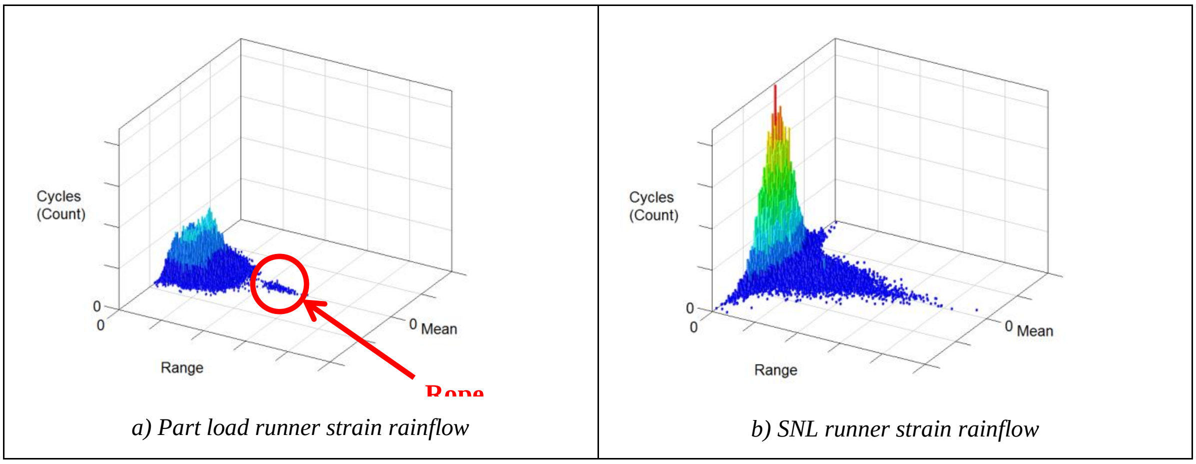


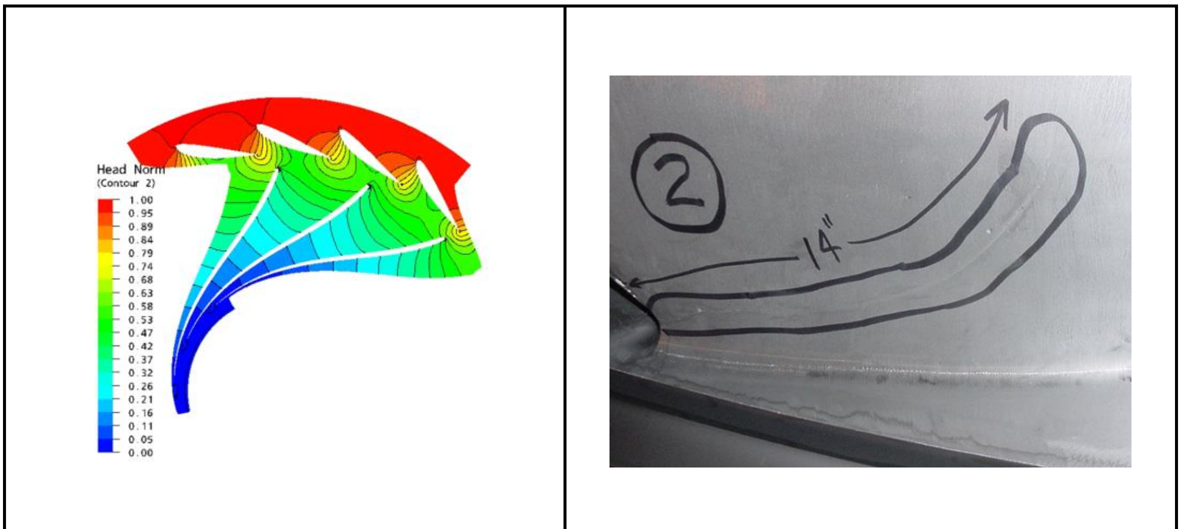
Figure 6. Runner strain gauge rainflow diagrams

1.4 Peak / full load operation

If we exclude possible instabilities related to full vortex, which has the tendency to self-excite when its intensity increases above a certain level [9], the full load operation is normally the most stable operating zone of the turbine. The efficiency and cavitation characteristics are normally the best and one would expect everything to run smoothly. In reality, the flow being well organized, this is the operating zone where high frequency steady state phenomena are the most likely to occur. The best known are trailing edge von Karman vortices and guide vane-runner rotor-stator interactions.

Vortex shedding can potentially occur on all trailing edges of hydraulic profiles depending on approach flow, Reynolds number and trailing edge shape. When it occurs in hydraulic turbines, it can often be heard, and has therefore been referred to as “singing turbines”. The general concept of vortex shedding including the Strouhal number as well as an example of the shedding phenomenon from a model hydrofoil and a prediction with CFD are well explained in [9]. Runner blades are at risk of vibrations due to von Karman vortex shedding but this can easily be avoided by means of suitable trailing edge modification. A general modification works well because the relative flow velocity range at the trailing edge is limited.

The other steady state and highly periodic phenomena occurring at full load are Rotor-Stator Interactions (RSI) between the guide vane and the runner blades. The hydraulic effect leading to the unsteadiness is primarily a potential flow interaction between the non-uniform flow distribution at the outlet of the guide vanes and the rotating runner blades passing through this flow. Since the flow field in the radial space between the guide vanes and runner is non-uniform circumferentially, both the static pressure and the velocity (velocity magnitude and flow angle) vary circumferentially. This results in pressure variations on the runner at guide vane passing frequency as shown in figure 7a). These pressure variations produce periodic stress variations on the blade, as shown in figure 4 in the higher power zone. RSI was the first unsteady phenomenon that could be analysed with reasonable computational resources and accuracy [12]. When strong RSI or resonance at the guide vane passing frequency occurs, rapid high cycle fatigue failure can be the consequence as shown in figure 7b) [13]. Due to the significance of potential consequences if RSI stresses surpass acceptable limits, unsteady CFD analysis of RSI are nowadays systematically performed at the design process.



a) RSI Pressures *b) Blade cracking due to RSI*

Figure 7. Rotor stator interaction between blades and guide vanes in Francis turbine

2 DYNAMIC EFFECTS SENSITIVITY TO SPECIFIC SPEED

Knowing the dynamic phenomenon occurring on Francis runners, it would be interesting to relate the amplitude of the dynamic excitations to some hydraulic characteristics of the machines. This section opens the discussion on this interesting challenge.

2.1 Start-up

Among the papers in the literature on start-up dynamic pressures, Nicolle [12] has carried out one of the most complete CFD and FEM analysis. The acceleration of the runner during the start-up and the evolution of the mean stress were predicted with good agreement to in-situ measurements for two different start-up sequences. The prediction of the dynamic stresses was not presented but the analysis of the dynamic strain gauge measurements for the same start-up sequences in [7] shows the impact of the start-up procedure on the damage occurring to the runner. Therefore, the start-up dynamic hydraulic excitation is closely related to the start-up sequence and the resulting stresses are linked to runner blade flexibility.

For new runners, it is possible to design them robust enough for aggressive peaking operation and to be capable of sustaining extended number of start-ups without special start-up sequence procedures [4]. But optimizing start-up sequence may be an interesting option to solve cracking issues of existing runners with aggressive operating scheme, as discussed in [15].

2.2 SNL/low load stochastic forces

As mentioned above, at SNL, the flow passes through the turbine without power generation, but with non-negligible flow rate. Most of the potential energy in the flow has to be dissipated somehow in the runner, since no power is produce at the generator. Figure 8 presents the maximum normalized stochastic stress amplitudes of different measured runners as a function of the specific speed NQE (according to the IEC definition [16]). Not surprisingly, except for few outsiders, a clear relationship appears showing that runners with higher unit flow (lower head machines) are more sensitive to stochastic hydraulic loads at SNL. Design expertise of the runner manufacturer, as described in [4,5], is essential to ensure reliability of runners running at low load for extended period of time, especially for high specific speed machines.

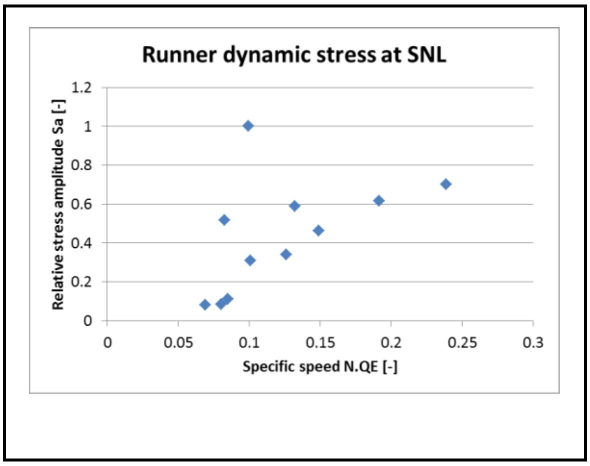


Figure 8. Runner dynamic stress at SNL

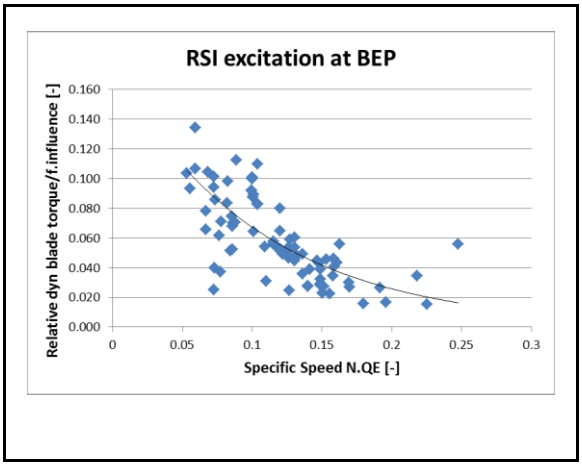


Figure 9. RSI dynamic blade torque

2.3 Part load rope

Part load rope non dimensional intensity has been thoroughly discussed in [17]. According to IEC 60193, the pressure fluctuation factor is defined as

$$\tilde{p}_E = \tilde{p}/_{\rho g H} \qquad (1)$$

Using an overall limit for the pressure fluctuation factor, this definition means that the low heads are always more critical that higher heads. This is not logical, since a given absolute pressure fluctuation is not more critical because the head is lower. Dörfler [17] proposes a dimensionless measure of draft tube pressure pulsations called pressure pulsation coefficient with which a “generic” curve may be defined as a reasonable limit for pressure pulsation acceptable values. This coefficient relates the pressure pulsation to the runner peripheral speed u2=πnD2 and allows to compare machines between each other.

$$Pressure\ pulsation\ coefficient = \tilde{p} \Big/ \left(\rho u_2^2 /_2\right) \qquad (2)$$

As discussed above, however, the relation between the rope intensity and significant runner accumulated damage is not clear.

2.4 RSI

While rotor-stator interaction (RSI) of guide vanes and runner blades occurs in all hydraulic turbines, only in medium to high head machines do the pressure fluctuations become significantly large with respect to stress levels. This is the result of the high velocities at the guide vane outlet and the small radial gap between the blade rows in these machines. A useful way of representing the effect of unsteadiness on the blades of a runner is the unsteady or “dynamic blade torque”. This is the torque an individual blade contributes to the shaft torque as a function of time. On some runners, it so happens that the peak-to-peak dynamic blade torque calculated can be as high as the static torque generating the power of the machine [13]. The dynamic blade torque amplitude depends on several influence factors, such as the phase angle from one channel to the next, and cannot be directly connected to the specific speed NQE of the machine. However, by normalizing the dynamic blade torque with these influence factors, a clear relationship comes out with the specific speed NQE as shown in fig 9: The increasing RSI effect for higher head machines becomes clear.

In hydraulic turbines, there are two aspects to rotor-stator interactions: the dynamic flow field, which we just discussed, and the response of the structure to this unsteady flow field which depends on factors such as the closeness of the RSI excitation frequency to natural frequency of the runner. The response of the structure has been discussed in detail previously [13, 18].

3 CONCLUSION

In this paper, we discussed how the dynamic phenomena occurring at various operating conditions may affect the lifetime expectancy of different specific speed Francis runners. Dynamic phenomena occurring at four transient and steady state operating conditions have been presented: start-ups, speed no load, part load and full load. For each of these operating conditions, a correlation between dynamic hydraulic forces and runner characteristics have been discussed: start-ups and low load stochastic forces, part load rope, Rotor-Stator Interactions (RSI) between runner blades and guide vanes at full load. In this context, it was shown that low specific speed machines (high head) are more sensitive to Rotor Stator Interactions (RSI) than high specific speed (low head) machines while the opposite occurs for low load operation.

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