Optimization of Francis turbine start up procedure to extend runner lifetime

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Litostroj Power is performing a refurbishment of three low head Francis units, where distributor with mechanism is being replaced, other vital parts refurbished and runners reused after runner blades repair.

This article presents measurement of mechanical stress on runner blades trailing edge close to the runner crown of the unit two. Stress was measured with strain gauges using wireless WIFI connections from rotating parts and analysed during unit start-up sequence into mechanical run, unit transients, closing of main inlet valve under discharge and unit steady state operation at various levels of the loads from minimal to maximal power in the operating range. Aim of stress measurement on the runner blades was, to acquire data of average and peak to peak stress on the runner blades, its pulsations with frequency spectrum and to evaluate possibilities to decrease present stress during unit operation [1].

Prior to installation of measurement equipment, runner on unit two was checked for current status. Some of the runner blades were cracked and repaired in the past, while most of the repaired blades were already cracked from last repair, which was three months before our inspection. Cracks were repaired without dismantling of the runner, with neither additional machining, balancing nor heat treatment after welding repair. Origin of cracks was located at the junction of runner blades trailing edge with runner crown.

Analysed results had shown that, optimization of start-up sequence into mechanical run could have impact on runner blades stresses. It was concluded that Von Karman vortices downstream of runner blades trailing edge were not a factor in either case, due to chamfer on the trailing edge.

1. Possible reasons for cracks

Normally, maximal values of static stress are found at the trailing edge close to blade to crown intersection or blade to band respectively, where cracks, if present, are mostly recognized. For our project we choose exactly this spot for strain gauge installation.

Depending on the operating conditions and head application range, different excitation phenomena are presented in Francis turbines. Classification of the runner type may be defined by specific speed as follows

$$n_q = n \cdot \frac{\sqrt{Q}}{H^{3/4}} \tag{1}$$

With: n = shaft speed [rpm], Q = flow rate [m^3/s], H = head [m]

Cracks, in case of Francis turbines, may be attributed to the following three main important roots:

- Rotor – stator interaction (RSI) – 0
- Von Karman vortex phenomenon – 0
- Flow characteristics depending on operating conditions such as high load, best efficiency load, partial load, low load, speed-no-load and runaway speed; see 0.

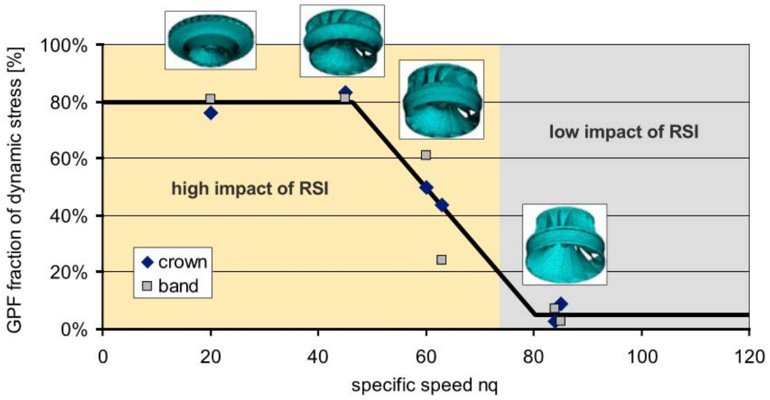
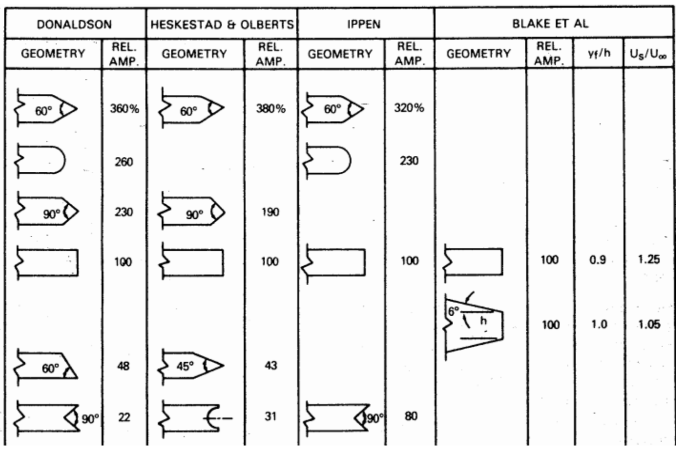


Fig 1: Rate of gate passing frequency (GPF) stress amplitude to overall stress versus speed factor n_q [4]

The results, summarized in 0, show that a resonance check with gate passing frequency excitation is recommended for higher head Francis runners. For low head runners with an n_q above approx. 80, RSI induced stresses are very low. Assuming a reasonable distance between the runner blade leading edge and the guide wicket gate trailing edge, RSI induced stresses will not contribute significantly to runner fatigue problems.



DONALDSON		HESKESTAD & OLBERTS		IPPEN		BLAKE ET AL			
GEOMETRY	REL. AMP.	GEOMETRY	REL. AMP.	GEOMETRY	REL. AMP.	GEOMETRY	REL. AMP.	y_f/h	U_s/U_∞
60°	360%	60°	380%	60°	320%				
	260				230				
90°	230	90°	190						
	100		100		100		100	0.9	1.25
						6° h	100	1.0	1.05
60°	48	45°	43						
90°	22		31	90°	80				

Fig 2: Trailing edge shapes, their relative sensitivities to singing and their parameters for scaling [7]

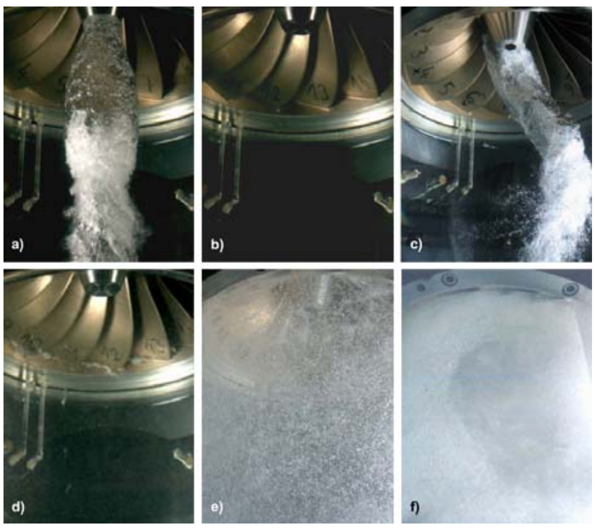


Fig 3: Typical flow patterns of a Francis turbine observed in model tests at the plant sigma:
a) high load, b) around BEP, c) part load), d) low part load, e) speed no load-SNL, f) runaway speed [1].

Each above presented cause for crack development on runner blades has typical frequency domain, which can be seen in stress measurement signal, in case that it is influential.

2. Strain gauge application

The strain gauge positions are derived from static Finite Element Method results calculations and occurrence of the cracks. Based on the stress distribution, for several calculated loads, the strain gauge positions and orientation were defined. The highest static and dynamic stresses on the Francis runner blades occur close to the trailing edge, at the transition to the crown and band side, see 0. Arrows indicate the orientation on the principal stress, the length of the arrows is magnitude based.

The foil uniaxial strain gauges, which were applied, have the surface covered with a special resin for waterproofing with integrated cable. The waterproof structure enables these gauges to serve for underwater measurement by being bonded to measuring object. Covering resin is flexible enough to enable easy bonding to the curved surfaces.

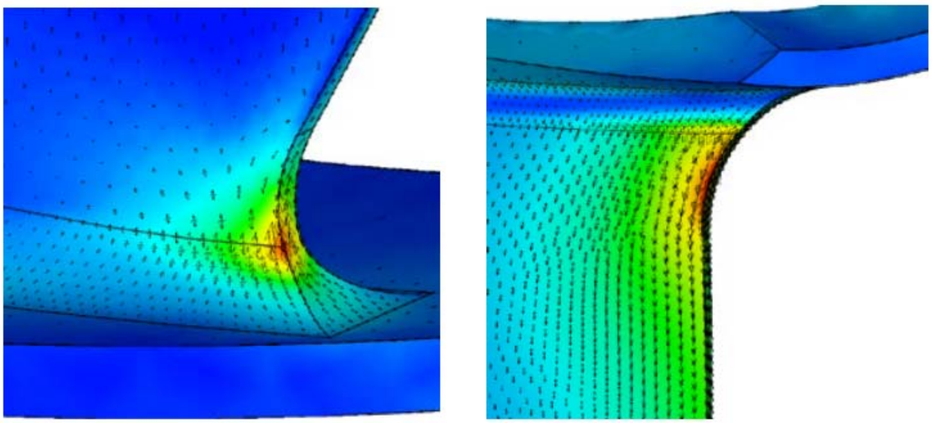


Fig 4: Strain distribution at trailing edge to band (left) and to crown (right)

The cables are first led from strain gauge along the runner trailing edge suction side to the crown and inside of the runner cone. At this location cables where protected with rubber hoses to enter the hollow shaft. Exposed part of the cables was additionally protected by a two-component epoxy resin, see 0, with excellent characteristics working under water and pressure and reinforced by one layer of the glass fibre strip.

For Francis turbine, which has a hollow shaft, it is relatively easy to realize connection of measuring cables from strain gauges to the top of the generator, where WIFI transmitter is installed, see 0.

Fig 5: Detail of the strain gauge and integrated cable protection

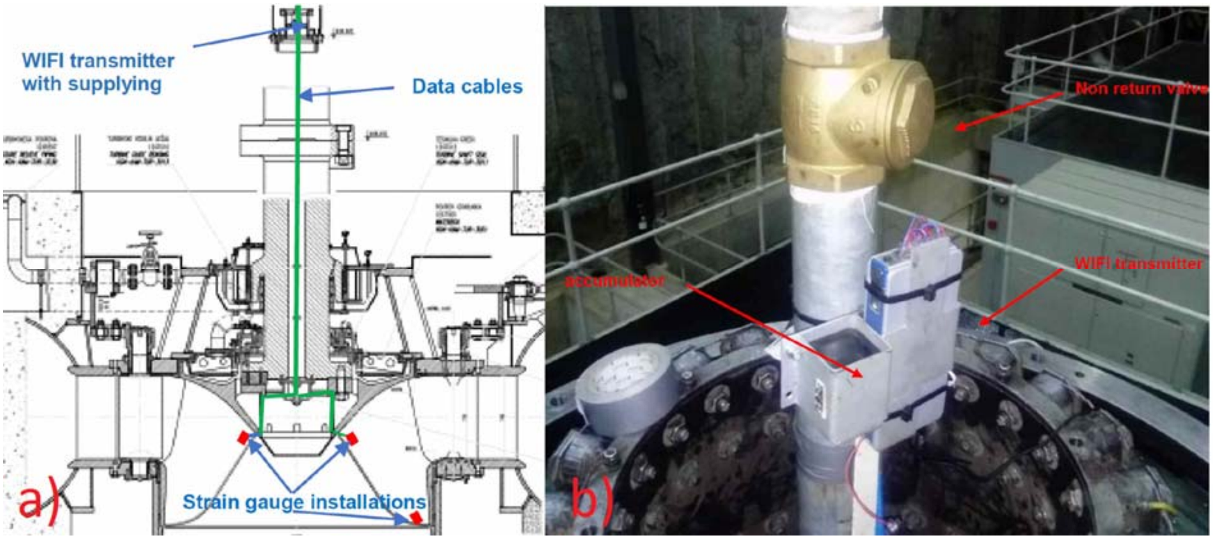


Fig 6: a) Air admission to the runner cone through hollow shaft and b) WIFI data transmission from rotating shaft to the data acquisition system

Fig 7: Runner blades instrumentation before protective epoxy installation (blade 4, 5, 6)

A photograph of a typical blade to crown/band strain gauge arrangement can be seen in 0. Strain gauges were attached in parallel to the blade trailing edges at both the blade to crown and blade to band intersections.

Strain gauges were placed on blades no, 4, 5 and 6 which were in different mechanical conditions; namely, some of the blades were being repaired by the onsite personnel:

- blade No. 4 – crack was repaired with welding; no visible damage of the weld was recognized (crown installation),
- blade No. 5 – repaired with welding and cracked again (crown installation),
- blade No. 6 – there is no crack, it seems to be as new (crown/band installation).

Blade no. 5, where crack was recognised, gave irrelevant small stress result values and was not analysed in continuation when the unit was synchronized. Dynamic stresses were consistently higher at the blade-to-crown trailing edge and although the results from the band were analysed, only those for blade-to crown intersection are presented, when the unit was synchronized.

3. Steady state operating measurements

For analysis of runner loading during steady state operation, stress variation around the average stress and its frequency domain are presented, see 0, 0 and 0.

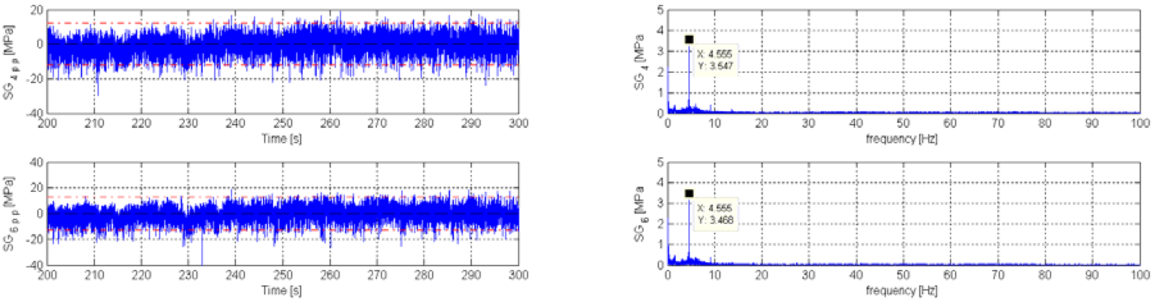


Fig 8: Stress variations and frequency spectrum – idle run without excitation

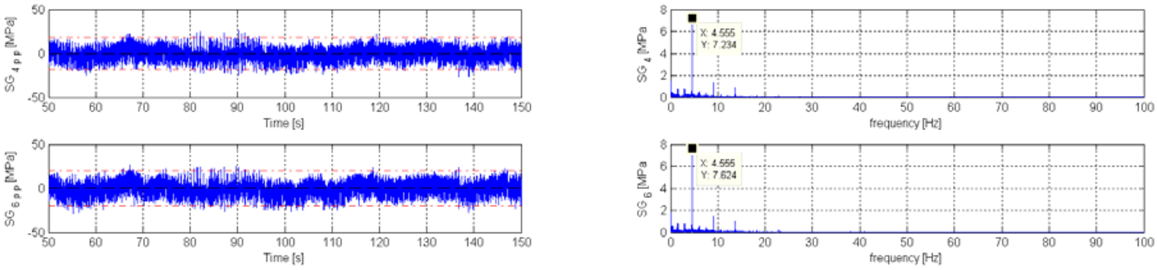


Fig 9: Stress variations and frequency spectrum – 3.7 MW

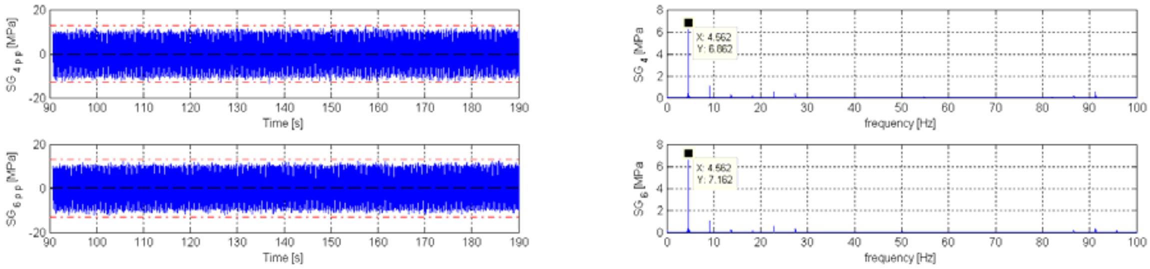


Fig 10: Stress variations and frequency spectrum - 25 MW

Average static stresses and pulsations over whole unit steady state operating range are presented in 0 and 0.

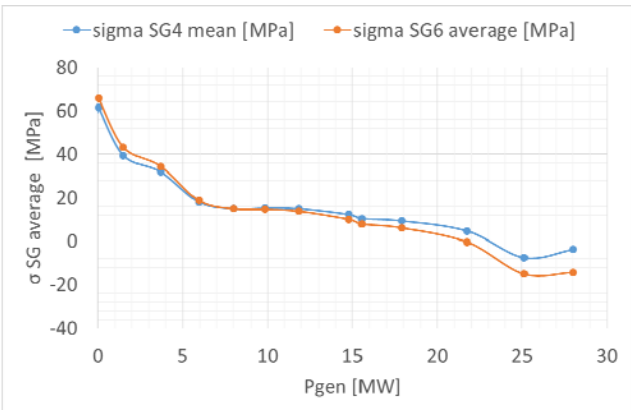


Fig 11: Averaged „static values" of stress at the blade-to-crown trailing edge intersection of blade No. 4 and 6

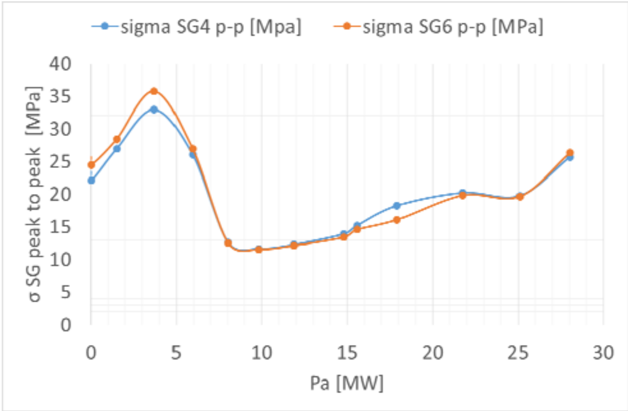


Fig 12: Dynamic" stress values at the blade-to-crown trailing intersection of blade No. 4 and 6

From the runner durability point of view, idle run is critical. At unit nominal speed "static" stress reached its maximal value – see 0.

Measured "dynamic stress", expressed as peak-to-peak value, see 0, is increasing from idle run to reach maximal value at part load Pgen = 3.7 MW. This stress is originated from vortex rope in the draft tube having frequency approximately 1.6 Hz which is 35 % of nominal speed – see 0.

Air admission bellow the runner, damps significantly pressure pulsations in the draft tube cone. Further the "dynamic stress" steeply descents with increasing power to the best efficiency point. Measurement results show very low stress amplitudes around the best efficiency point, see 0. Static stress at blades suction side is positive which means tensile stress direction.

The frequency component at rotational speed frequency, 4.5 Hz, is permanently recognized in the blade stress signal and it is dominant at all unit operating ranges.

4. Transient operating conditions

Strain gauge readings were taken during all normal transients, namely starting up, operational and emergency stopping and load rejections. For obvious reasons, the exceptional transients of runaway speed, load rejection from maximal power and mechanical over speed trip was not included due to runner mechanical integrity.

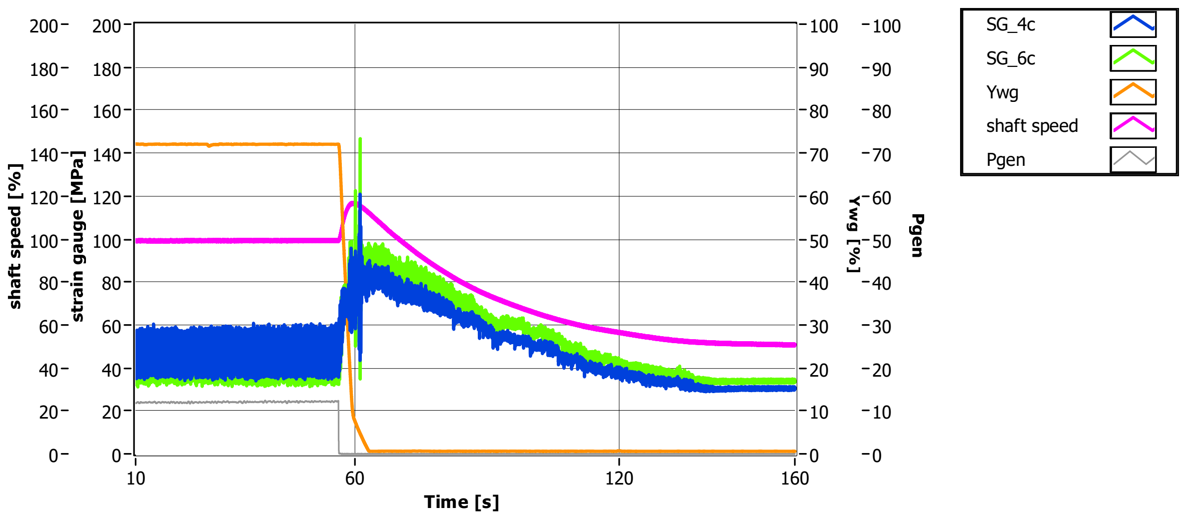


Fig 13: Emergency shut down from output Pgen = 24 MW (75 %)

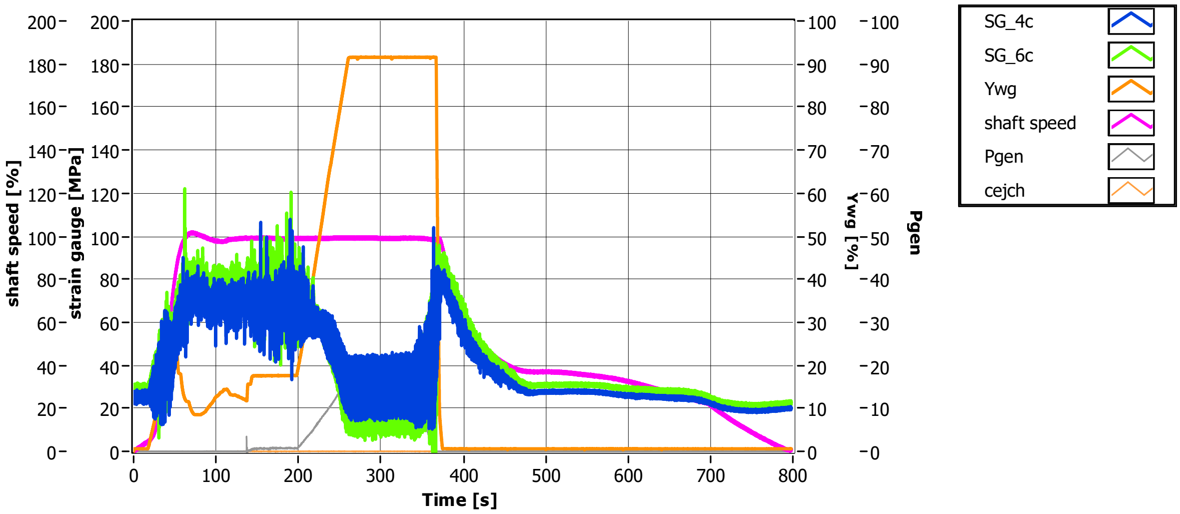


Fig 14: Closing of MIV to flow when output of Pgen = 30 MW was adjusted

The stresses during transients, were well below the maximum for infinite fatigue life for the runner material. Reduction of dynamic forces during transient operations would substantially reduce the propensity for runner cracking, even if some residual material defects.

There is no possibility of reducing the dynamic forces during load rejections. The guide vanes closure law is determined by the guaranteed pressure and speed rise values. However, load rejection is an infrequent occurrence and has very little cumulative effect on the runner fatigue life. It is evident that stresses during operational stopping are relatively smaller and maximum benefit could be gained from reducing the start-up stresses.

5. Impact of start-up scheme on Francis runner life expectancy

During start-up, the unit is controlled by the hydraulic governor. In the original design, the governor is programmed to open the guide vanes automatically at a set ramp to a value considerably higher than speed-no-load, to provide sufficient torque to overcome initial friction and to accelerate the unit, see 0 and 0.

After speed increase in proximity of nominal speed of rotation, the guide vanes were closed to just slightly higher value than the one required for speed-no-load operation and the control was transferred to the PID section of the governor for fine tuning to speed-no-load and eventual synchronizing.

Testing with different start-up ramps without and with support of air admission was further tested, see 0

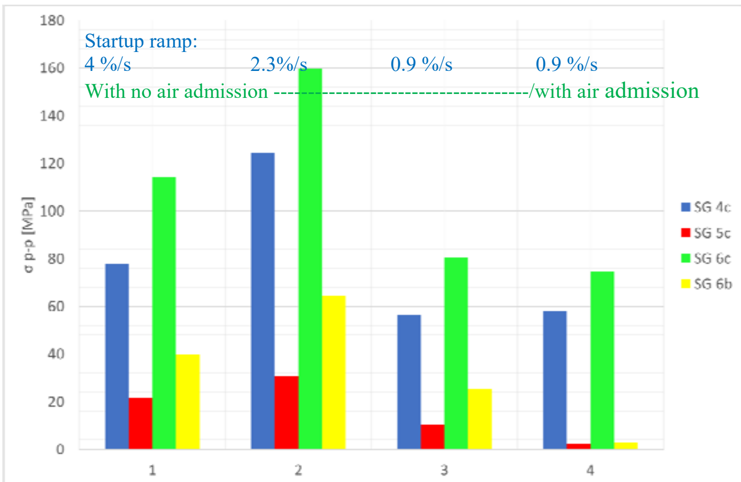


Fig 15: Maximal stress on the suction side of the blades No. 4, 5 and 6 (peak-to-peak)

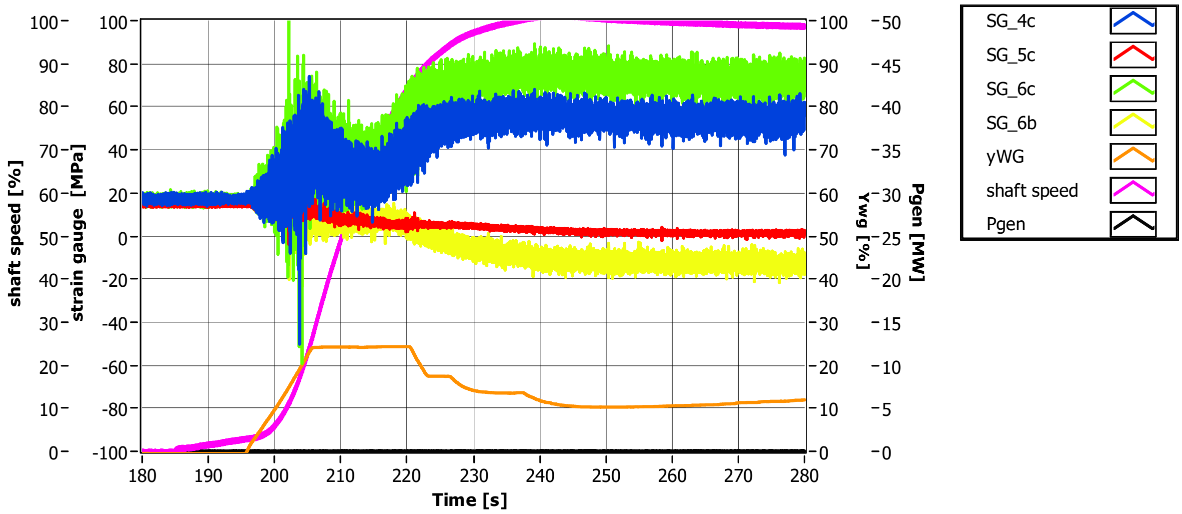


Fig 16: Measured stresses during initial start-up sequence with no air admission and start-up ramp of 2,3 %/s

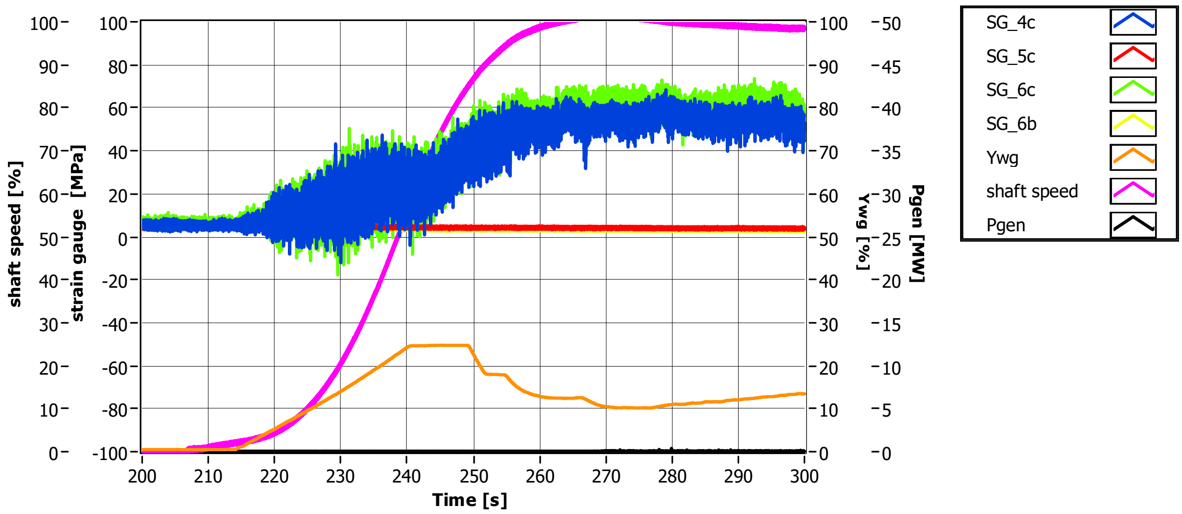


Fig 17: Measured stresses during modified start-up sequence with air admission and start-up ramp of 0,9 %/s

0 shows, that modification (slowing down) of the guide vanes opening ramp to 0.9 %/s lead to decreasing stress for 35 % at the blade-to-crown intersection and for 73 % at the blade-to-band intersection.

Modified start-up was applied into the governor. Hence, a combination of repairing the runner to remove weld defects, weld surface polishing and reprograming of the start-up routine, increase durability of the runner.

6. Conclusions and recommendations

During visual revision of the runner blades prior to start of tests, the cracks were recognized on five out of 15 runner blades, which were repaired repeatedly by site personnel. It was found out that welding procedure of the cracks was finished evidently without the technological process supported by any non-destructive inspection protocols.

Visual inspection of the weld surface and cracks reveals the origin of defects in the welds and subsequently their spreading due to local stress acting for relatively short period followed by long period of crack consolidation without any spreading.

To analyse the runner operating conditions, strain gauges were installed both on blades which had repaired crack and ones which had not. Dynamic stresses were consistently higher at the blade to crown trailing edge compared to blade to band trailing edge.

The stresses during steady-state running in the guaranteed operating range, were well below the maximum for infinite fatigue life for the runner material.

Although not typically considered for design purposes, high dynamic hydraulic forces during transients lead to cracking of low head Francis turbine runners, especially in units for load peaking and thus subjected to daily multiple start/stops.

Further testing had shown that the optimization of the start-up sequence could have a significant impact on the life expectancy. Slower guide vane opening start-up ramps were tested and successfully implemented into the governor.

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