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Life time assessment and plant operation optimization based on geometry scan and strain gauge testing – START/STOP optimization

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1. ABSTRACT

With the expansion of renewable energies the energy market requires highly variable machines and fast changes of operating regimes, which means higher numbers of start-ups, shut-downs and load changes to the point of unit operation in formerly forbidden load conditions.
Consequently the changed load universe for hydro power plants has a non-negligible influence to the fatigue of the component. Besides of the reliable predictability of standard load conditions and their fatigue contribution, the impact of transient load conditions (Start-Up, Shut Down, Load Rejection) and off-design points like Speed-no-Load and Part Load is a challenging task. Therefore strain gauge tests are performed in order to measure the static and dynamic stresses especially during these non-predictable load conditions and to assess their fatigue impact.
This contribution describes the approach to optimize the machine operation regarding life time. It shows the necessary single steps from theoretical FEA calculations, to the definition of the correct strain gauge positions to the point of fatigue calculation. As an additional finding from recent measurements, the difference of the design and the real geometry was reviewed. Thus this procedure with measurement and data evaluation is not limited to new machines, also existing runners can be investigated and the unit operation can be optimized. Additionally this paper shows the possibility how to optimize transient load conditions like Start-Up or Shut Down and their influence to partial damages. Some results from recent runner strain gauge tests are shown, where different opening laws were tested and compared regarding partial damage.

2. INTRODUCTION

Basis of this paper is a life time assessment of an existing runner. Target was to verify the residual life time in spite of a power increase of the machine. The whole process to achieve reliable test results can be separated in a few main steps.

(1) CFD / FEA calculation of the nominal runner blade geometry
(2) Definition of strain gauge positions based on FEA results of the nominal geometry
(3) Scan of the runner blade geometry and of the defined strain gauge positions
(4) Strain gauge measurement
(5) Post processing of the scanned runner blade geometry
(6) CFD / FEA calculation based on the scanned runner blade geometry
(7) Comparison of calculation and measurement
(8) Life time calculation based on a representative load universe

With the above described process a general statement of the residual lifetime of the component can be given. Optimal unit operation points in means of partial damage can be identified and load conditions leading to a fatigue life reduction can be avoided.
As a result, the above described process gives the opportunity to improve the unit availability and to avoid or minimize load conditions with higher partial damages.

3. Theoretical background

3.1. Basics of Rotor-Stator-Interaction

Depending on the operating conditions and the head application range of the unit, different excitation phenomena are present in Francis turbines. At low outputs, a stochastic behavior with broad band frequency content is present, whereas at part load condition, the typical low frequency excitation caused by vortex rope phenomena related to the Rheingans frequency occurs. From part load to full load, a higher frequency component arising from Rotor-Stator Interaction (RSI), the so called Gate Passing Frequency (GPF) is dominating.

In the rotating reference frame of the runner, the interaction of runner blades and guide vanes induces pressure pulsations at the runner inlet area based on two different phenomena. On the one hand, the rotating observer passes the wakes of all guide vanes during a full rotation, and on the other hand, a pressure pulse is induced each time when a runner blade is passing a guide vane. Both phenomena lead to rotating pressure fields with distinct spin speeds (relative to the runner speed) and characteristic numbers k of diametrical node lines, see references [1-3]. Hence, the entire excitation pressure field is varying in space and time and may be described as a linear combination of rotating pressure mode shapes.

3.2. Prediction of runner dynamics

In order to assess the dynamic behavior of submerged components, the effect of the surrounding water flow must not be neglected. The fluid mass which is moving with the structure (added-mass effect), may reduce natural frequencies to less than 50% of the corresponding frequencies in air, and flow induced damping effects may become one or two orders of magnitude higher than structural or material damping.

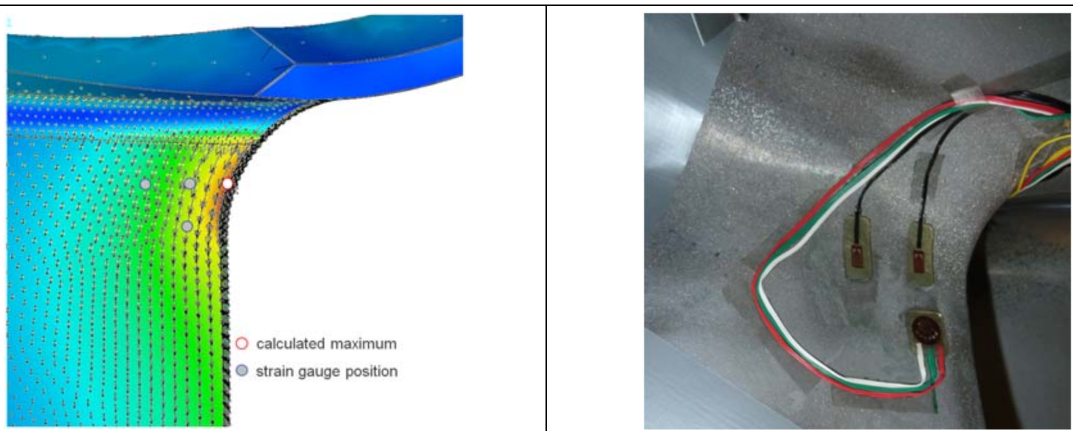
Additionally the Fluid-Structure Interaction (FSI) plays an important role for dynamic design optimizations of hydro turbines. Appropriate solution methods depending on the type of application are presented and discussed e.g. by Hübner et al. [4]. The dynamic runner response and dynamic stresses due to this RSI can be predicted reliably by means of harmonic response analyses of the entire runner in water using acoustic Fluid-Structure-Interaction (FSI), as shown by Seidel and Grosse [5].

4. METHOD

4.1. Definition of strain gauge position based on FEA

The strain can only be measured at several locations, therefore usually highest stressed areas of runner blades are chosen. In order to define these locations, a Finite Element analysis has to be performed prior to the test to investigate the most critical regions of the component. Typically these locations are at the connection from trailing edge to runner band and from trailing edge to runner crown. Within these calculations different load conditions are considered, at least a load case with very low or no output and a load case close to the optimum point. This represents the complete load range of the component and results in different calculated strain distributions. The orientation of the calculated main principle strain has to be considered, thus mainly unidirectional strain gauges are used for testing. If the orientation of the main principle strain is shifting, bi- or tri-directional strain gauges can be used.

As described the strain gauge positions are finally defined by the calculated strain distribution and the orientation of the main principle strains. The definition of the strain gauge locations within the Finite Element model enables the prediction of expected strains at proposed positions.



Strain distribution and orientation of main principle strains	Final strain gauge locations on the runner

Figure 1: Definition of strain gauge positions

4.2. Geometry scan and definition of strain gauge positions

A complete scan of runner blades can be performed, if measurements are done on old runners (with wear/cavitation), at rehabilitated / repaired runners or if differences between design and manufacturing should be validated. Geometrical differences will also have an influence on the calculated and measured strains. If calculated strains derived from the nominal geometry have to be compared with measured strains, there will be always a difference. For new built runners nowadays these differences should be a minor topic, as manufacturing requirements changed in the last years. Especially in the high loaded regions of a turbine only small manufacturing tolerances are applied.

The blade which is applied with strain gauges can be scanned with a laser scanner. The result of the scan is a point cloud with several million points. As it is not possible to execute a CFD / FEA calculation with the point cloud geometry, an intensive and time consuming post processing of the scanned data is necessary.

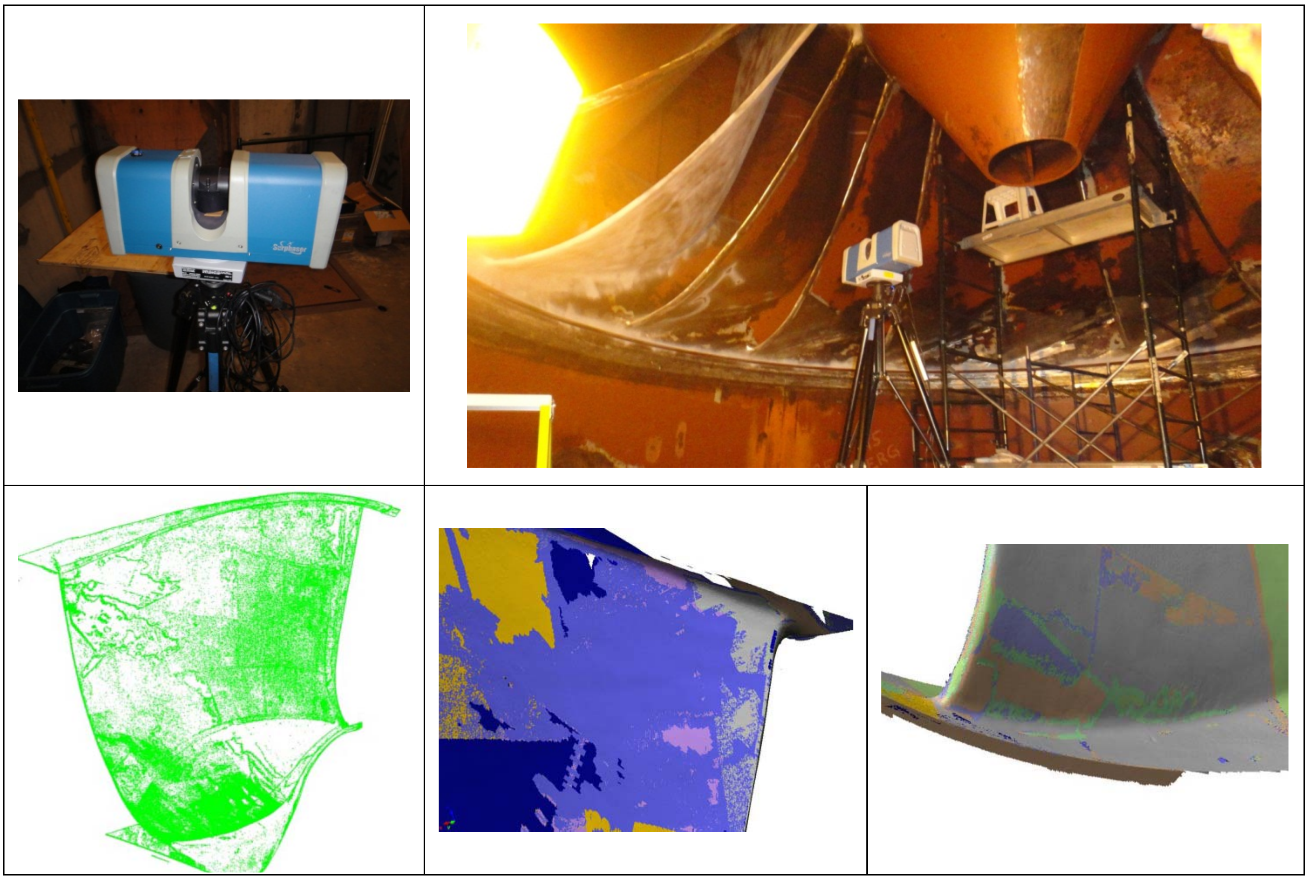


Figure 2: Geometry scan and point cloud (raw data) of scanned blade

Above figure is split into two parts. The upper one shows the scanner in the field and how and where the device is located. The lower part shows the geometry directly after the geometry scan (point cloud).

As already described before, a post processing of the point cloud has to be done in order to perform CFD and FEA calculations. The pictures below are showing the CAD geometry of the nominal and the scanned geometry. The nominal geometry (yellow), the blade 1 (grey) and blade 2 (red) are overlapped. Common fix point is the runner crown. It is clearly visible that there are differences especially in the outlet region.

			
Global view	Section view close to band position	Section view close to crown position	Section view at the middle of the blade

Figure 3: Comparison of nominal blade and scanned blade geometry

All the main steps from the Finite Element Analysis of the nominal geometry to the CAD-model of the scanned model is summarized in following sketch:

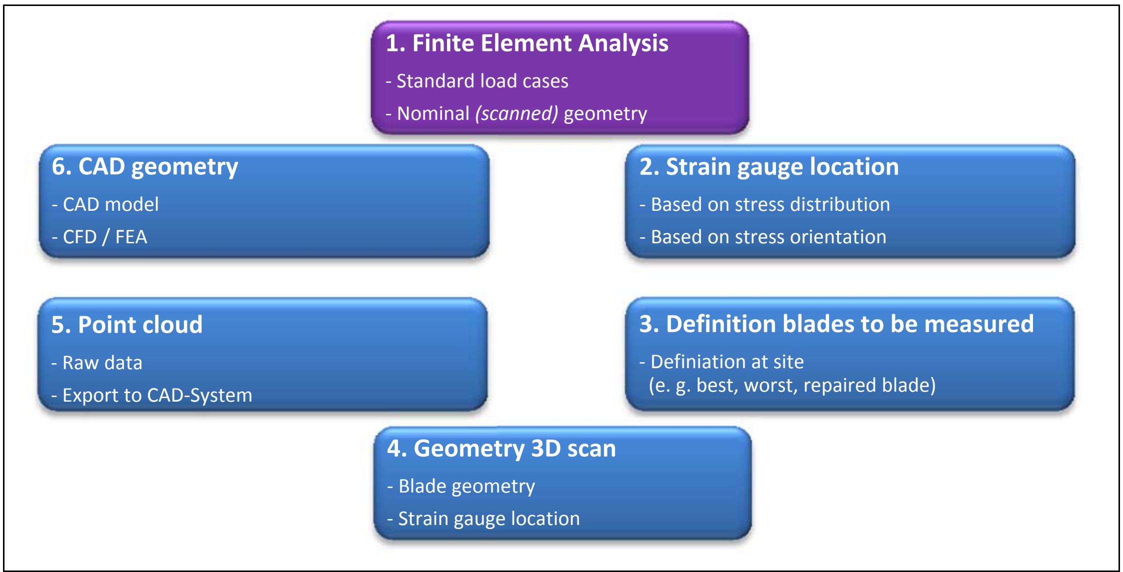


Figure 4: Action loop for geometry scan

4.3. Measurement of static and dynamic strains

A detailed technical description of strain measurements can be read in the paper "*Strain gauge measurements of rotating parts with telemetry*". [6]

Voith Hydro executes all strain measurements on rotating parts with telemetry systems. Several systems with analogue and digital data transmission are in use from one channel telemetry systems up to 32 channel systems.

Telemetry system	Quantity of channels	Bandwidth [Hz]	Typical application
Analog-System			
32CH-ANALOG	32	1.000	Francis Kaplan
16CH-ANALOG	16	2.000	Francis Pelton Generator pole connection
12CH-ANALOG	12	2.000	Generator pole connection
4CH-ANALOG	4	2.000	Generator air blades
1CH-ANALOG	1	2.000	Shaft torque
Digital-System			
30CH-DIGITAL	30 (40)	19.000	Francis

Table 1: Technical data of different telemetry systems

4.4. Evaluation of strain gauge data

During the test strain data for all typical load conditions of the unit like start-up, different stationary load conditions, shut down, load ramp and load rejections are recorded. This test sequence gives an overall overview of all possible load conditions the unit has to withstand during commercial operation.

The measurement of runner strains takes place in a very rough atmosphere and with the fact that the strain gauges are very sensitive, a detailed check of all time signals is a major task. For example, the shunt calibration of the strain gauges is performed in air, therefore a zero correction due to temperature drifts caused by cold water surrounding the runner must be considered.

A first and global evaluation for stationary load conditions is given by the mean and the characteristic peak-to-peak value; generally these characteristic values are plotted over unit output.

Based on these evaluations rough operation zones can be easily detected. Especially the part load zone, where the part load vortex is acting on the runner blades, is clearly visible in *Figure 5* (lower diagram).

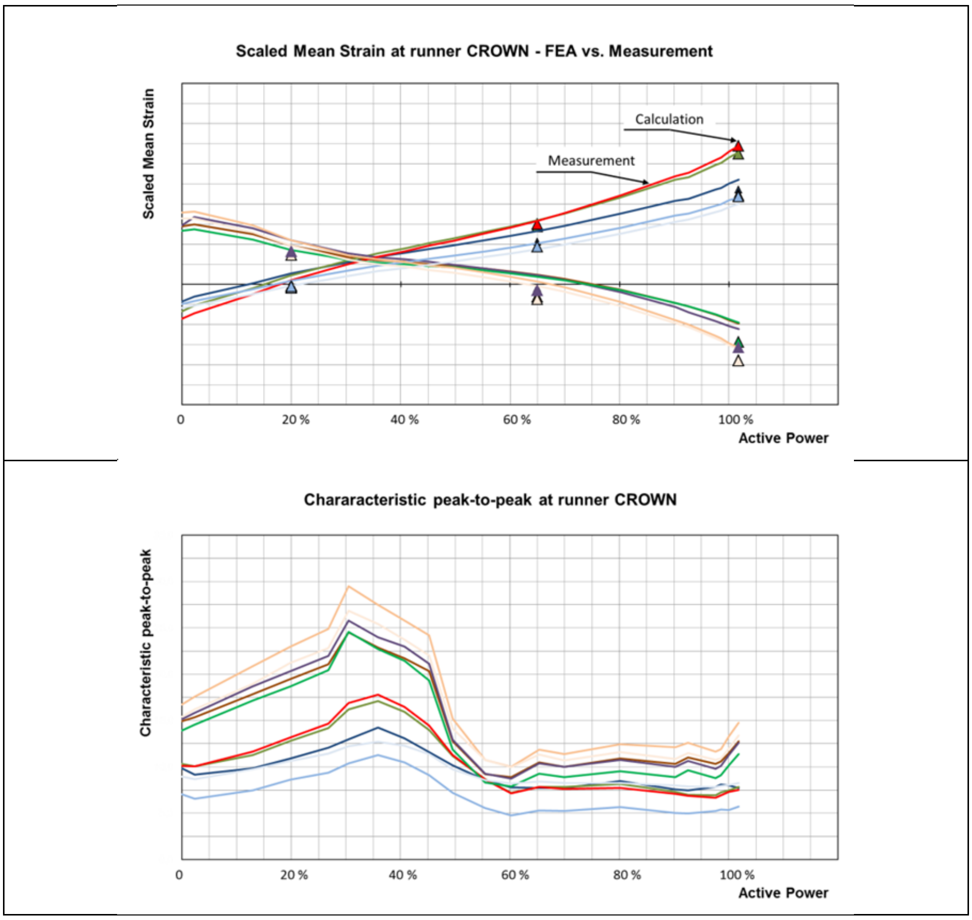


Figure 5: Comparison of measured and calculated mean strains | characteristic dynamic value

Besides of derived characteristic values also time signals for all transient load conditions are investigated.

As an example the time signal of a strain gauge at the transition trailing edge of the runner to crown for a start up from standstill condition to nominal speed is shown in following figure.

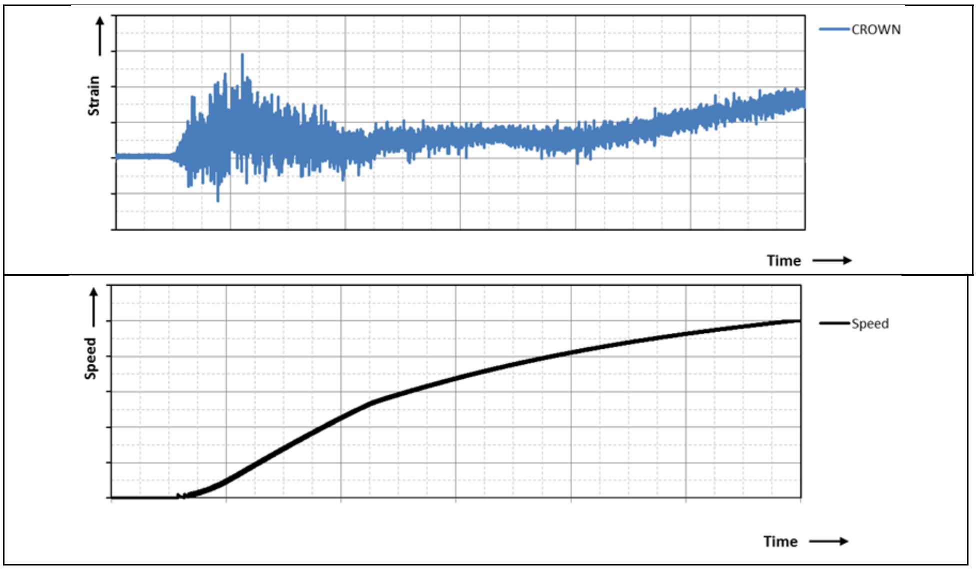


Figure 6: Strain time signal at runner crown for start-up

4.5. Fatigue Strength Assessment

4.5.1. Partial damage calculation

For the fatigue strength assessment a damage analysis is performed based on linear damage accumulation. Within this analysis partial fatigue damages are determined for representative time periods of steady state operation at different power outputs as well as for time signals of transient operating conditions. The total damage sum is obtained by accumulation of partial damages under consideration of a certain universe.

For the determination of partial damages for each time signal the scaling of the measured time signals (strains) by Young's modulus is done in order to obtain stresses at strain gauges. Next, the complex sequence of stresses is reduced to simple cyclic loading by rainflow counting technique. Based on a representative load universe and the consideration of the fatigue curve the linear damage accumulation is derived.

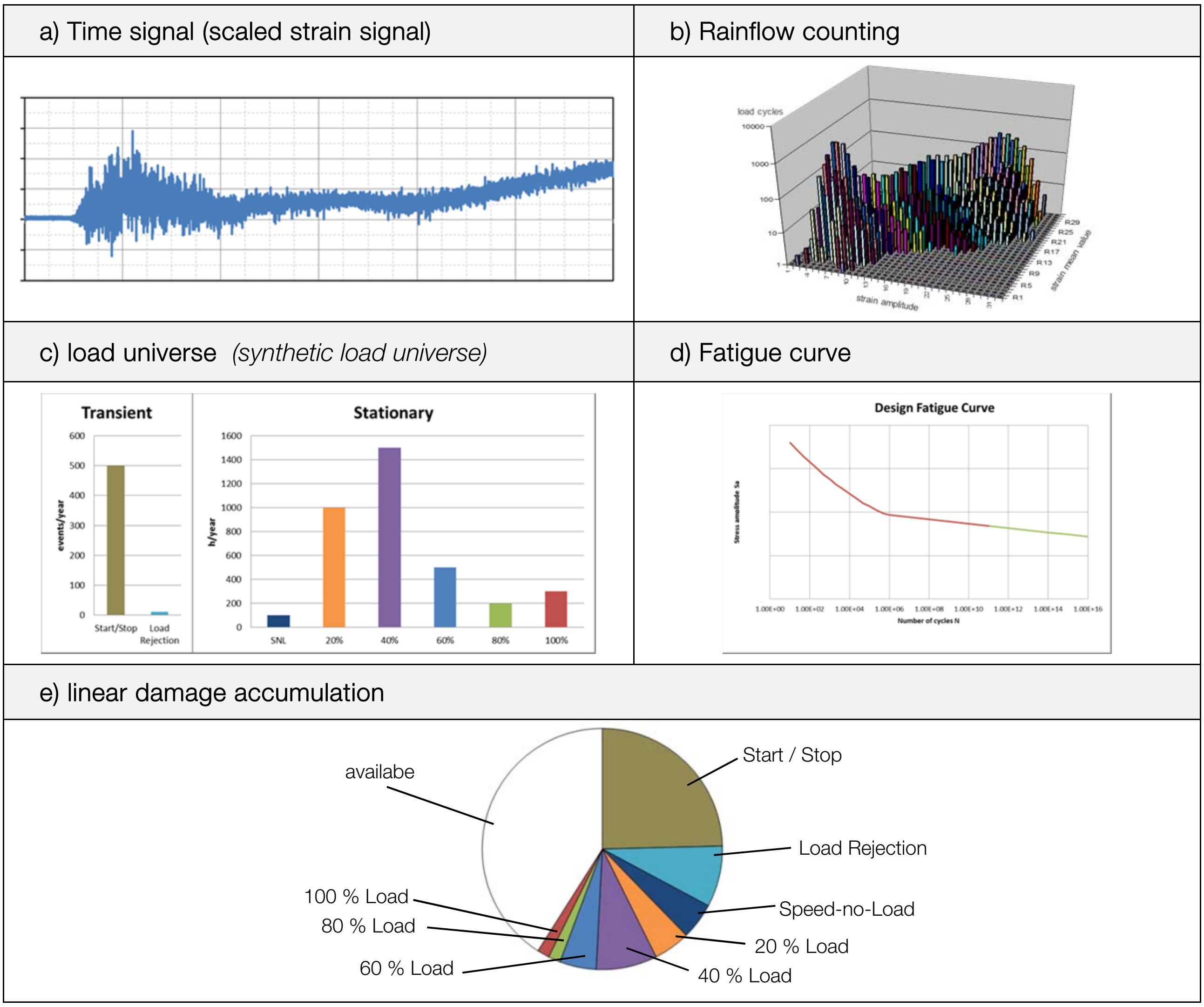


Figure 7: Work-steps for residual life time calculation

4.5.2. Start-up optimization

In order to reduce the partial damage contribution of the unit start up, the conditions (interaction of flow rate, speed, opening, and so on) being responsible for the highest damage contributions have to be identified. During post processing of different measured start-ups, the transient fatigue analysis is performed in order identify highest damage contributions, see [7]. By evaluating the time resolved damage accumulation together with synchronized governor data, improved settings for different start phases can be derived. With the given option to measure several start-ups, the fatigue optimized version can be determined, implemented and directly reject during the test.

An example of such an approach is given in following figure where the optimized start procedure is compared to a normal start. Maximum strain amplitudes are significantly reduced leading to minimized damage contributions.

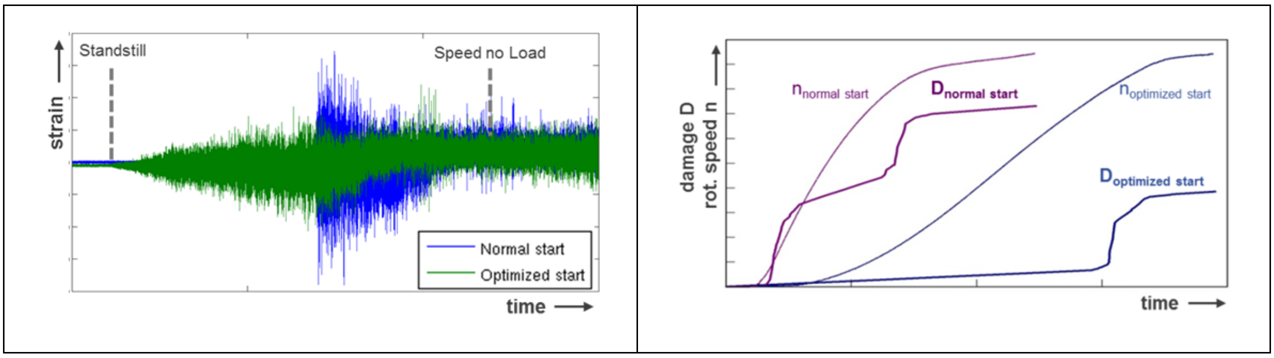


Figure 8: Time signals for normal and optimized start-up (left) and corresponding curves of speed an damage accumulation (right)

4.5.3. Plant operation optimization

The definition of the load condition based partial damages provides the possibility to easily detect main fatigue contributors. With the description as normalized partial damage, which shows the partial damage for 60 seconds of stationary and for one event of transient load condition, a more transparent comparison is given. Based on this approach the machine operation can be optimized regarding availability, output, and optimum operation with focus on life time.

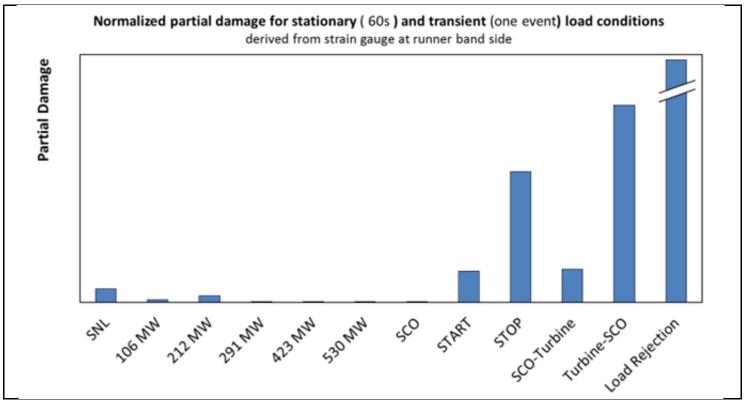


Figure 9: Partial damages for all measured load cases

In above figure normalized partial damages for all measured load cases are shown. Highest partial damage for this specific unit is caused by the load rejection and is dominating the overall damage sum. If this load case is not considered the other transient load cases are causing the highest partial damage.

The stationary load case with highest damage for the investigated runner is Speed-No-Load. Only with a few optimizations of the plant operation, a significant reduction of partial damage and a huge increase of fatigue life time of the runner could be achieved.

Following example should illustrate the possibility how to reduce partial damages without huge restrictions of the plant operation.

Example I: Load rejection is not considered
8299 hours of operation per year (every day 6h standstill)

Example II: Load rejection is not considered
8299 hours of operation per year (every day 6h standstill)
Instead of operating the machine in Speed-No-Load machine will be operated in SCO

	Load universe		Partial damage	
Load case description	Example I	Example II	Example I	Example II
Speed-no-Load	162 h	-	32,8 %	-
20 %	271 h	271 h	9,4 %	13,0 %
40 %	338 h	338 h	33,5 %	46,2 %
55 %	1300 h	1300 h	1,4 %	1,9 %
80 %	2100 h	2100 h	0,7 %	0,9 %
100 %	2628 h	2628 h	2,2 %	3,1 %
Condenser mode (SCO)	1500 h	1662 h	0,0 %	0,0 %
Starts	365 x	365 x	2,8 %	3,9 %
Stops	365 x	365 x	12,0 %	16,5 %
Turbine → SCO	90 x	180 x	0,8 %	2,1 %
SCO → Turbine	90 x	180 x	4,5 %	12,3 %

Table 2: Load universe and partial damage

Based on the calculated values described in above table, an increase of fatigue life of ~ 28 % is reached by Example II, where Speed-no-Load operation is replaced by turbine condenser mode.

5. DISCUSSION

With the requirements of the energy market, hydropower units are more often operated in the entire operating range. The demand of off-design operations and the increased amount of transient load conditions lead to the necessity to assess the fatigue life of these components.

As shown in this contribution, strain gauge measurements and the post processing of these data enables the possibility how to optimize the machine operation regarding life time.
All steps from preparation prior to the measurement, performing the test and post processing of recorded strain data are described. Appropriate methods to evaluate the fatigue contribution of start-up and individual operating conditions are explained and applied for optimization of operation sequences.
For more precise strain gauge location definition and for detection of geometry discontinuities the approach of a geometry scan is presented.

6. REFERENCES

[1] Tanaka H, 1990, Vibration behaviour and dynamic stress of runners of very high head reversible pump turbines, IAHR Symposium, Section U2, Belgrade.

[2] Franke G, Powell C, Seidel U, Koutnik J, Fisher RK, 2003, On pressure mode shapes arising from rotor-stator interactions, IAHR WG1 Meeting, Stuttgart.

[3] Fisher RK, Powell C, Franke G, Seidel U, Koutnik J, 2004, Contributions to the improved understanding of the dynamic behavior of pump turbines and use thereof in dynamic design, 22th IAHR Symposium on Hydraulic Machinery and Systems, Stockholm.

[4] Hübner B, Seidel U, Roth S, 2010, Application of fluid-structure coupling to predict the dynamic behavior of turbine components, 25th IAHR Symposium on Hydraulic Machinery and Systems, Timisoara.

[5] Seidel U, Grosse G, 2006, New approaches to simulate the dynamic behavior and dynamic stresses of Francis and pump turbine runners, IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems, Barcelona.

[6] Löfflad J, Eissner M, Graf B, 2012, Strain gauge measurements of rotating parts with telemetry, 9th international conference on hydraulic efficiency measurements, Trondheim.

[7] Weber W, Mende C, Koutnik J, 2013, Advanced fatigue analysis for transient operating conditions of Francis turbines, 5th IAHR International Workshop on Cavitation and Dynamic Problems in Hydraulic Machinery, Lausanne.