Efficient runner safety assessment during early design phase and root cause analysis

Q W Liang, S Lais, C Gentner and O Braun

ANDRITZ HYDRO LTD, Switzerland Hardstrasse 319/P.O. Box 2602, CH-8021 Zurich, Switzerland

E-mail: Quanwei.Liang@andritz-hydro.com

Abstract. Fatigue related problems in Francis turbines, especially high head Francis turbines, have been published several times in the last years. During operation the runner is exposed to various steady and unsteady hydraulic loads. Therefore the analysis of forced response of the runner structure requires a combined approach of fluid dynamics and structural dynamics.Due to the high complexity of the phenomena and due to the limitation of computer power, the numerical prediction was in the past too expensive and not feasible for the use as standard design tool. However, due to continuous improvement of the knowledge and the simulation tools such complex analysis has become part of the design procedure in ANDRITZ HYDRO. This article describes the application of most advanced analysis techniques in runner safety check (RSC), including steady state CFD analysis, transient CFD analysis considering rotor stator interaction (RSI), static FE analysis and modal analysis in water considering the added mass effect, in the early design phase. This procedure allows a very efficient interaction between the hydraulic designer and the mechanical designer during the design phase, such that a risk of failure can be detected and avoided in an early design stage.The RSC procedure can also be applied to a root cause analysis (RCA) both to find out the cause of failure and to quickly define a technical solution to meet the safety criteria. An efficient application to a RCA of cracks in a Francis runner is quoted in this article as an example. The results of the RCA are presented together with an efficient and inexpensive solution whose effectiveness could be proven again by applying the described RSC technics. It is shown that, with the RSC procedure developed and applied as standard procedure in ANDRITZ HYDRO such a failure is excluded in an early design phase. Moreover, the RSC procedure is compatible with different commercial and open source codes and can be easily adapted to apply for other types of turbines, such as pump turbines and Pelton runners.

1. Introduction

Hydraulic turbines are exposed to steady and unsteady excitation loads during the operation, which will result in stress concentration at various critical locations in the structure, depending on the design of blade and type of machine. Too high intensity of static stress or amplitude of dynamic stress can reduce the life time of the turbine significantly or result in cracks or damages in the turbine within a short operation period. In Francis turbines this kind of problems have been observed and published several times in the past. H. Brekke published in Hydro 2009 some fatigue related damages on two high head Francis runners (H=540 m) installed in Norway. He indicated that the cause was the high stresses induced by unfavorable outlet shape of runner blade.[1] A. Coutu published in HydroVision

2004 one case of runner blade cracks in SM3, which is also a high head Francis runner (H=330 m). The extensive measurement and numerical studies showed that the main reason was the too high dynamic stress induced by RSI.[2] E. Egusquiza recently reported one case of runner damage observed in a high head single stage reversible pump turbine (H=400 m). One piece of the runner broke away at the turbine inlet on crown close to blade suction side. It was concluded that the failure is probably due to high dynamic stress induced by the resonance with RSI.[3]

In order to minimize the risk of the failure in future operation, the safety of runner structure needs to be assessed during the early design phase. To assess the forced response of the runner structure a combined numerical approach of CFD and FEM is required, in terms of both static and dynamic analysis. Due to the high complexity of the phenomena and the high demand of computational resource, this type of coupled numerical prediction was in the past too expensive and time consuming, and was not feasible to be implemented into the design phase.

Nowadays, with continuous improvement of the knowledge and rapid growth of the IT technology, the runner safety assessment using such coupled fluid and structure analysis has become part of the standard design procedure of ANDRITZ HYDRO. A highly automatic tool has been developed, which allows a very efficient interaction between hydraulic designers and mechanical designers such that the runner safety issues can be detected and excluded during the design. It is even possible that the hydraulic designers themselves carry out the mechanical safety check for any of their intermediate designs to quickly get the feedback of their runner regarding safety issues. In case of need modification of the hydraulic design can be immediately performed so that the final release will meet not only the hydraulic requirements but also the safety criteria. In such way the interface between hydraulic and mechanical design is avoided. Moreover, considerable time and cost are saved.

In addition to the application during design phase, this established procedure can also be used for RCA, to find out the cause of the failure as well as search for a feasible technical solution in an efficient way.

The RSC procedure can also be easily extended and applied other geometries subject to similar interaction mechanism, such as pumps, pump turbines, Pelton runners, and wicket gates. Moreover, its compatibility is good so that it can couple different commercial and open source codes, both for CFD and for FEM.

This article will describe the general procedure of the safety assessment using RSC. One RCA case is used as an example and discussed in details.

2. Runner safety check (RSC)

As part of the standard procedure RSC applies the most advanced analysis techniques in design phase. Such procedure includes both CFD and FEM analysis by mapping the CFD derived static or dynamic pressure onto FE model as static load or excitation force respectively. Details of simulation will be discussed in the following chapters. From the FE analysis the response of runner structure is obtained and evaluated based on the safety criteria. Actions such as modifying the hydraulic design or changing mechanical design will be taken in case the criteria are not fulfilled. Criteria include static stress at critical locations, amplitude of dynamic stress, risk of resonance, etc. Iteration loops might be needed before all criteria are fulfilled and the runner design is released. Figure 1 describes such a procedure of RSC.

2.1. CFD analysis

CFD analysis includes steady state analysis and transient analysis considering RSI.

Steady state analysis produces the static absolute pressure distribution on all the wet surfaces of the runner. Absolute pressure is derived from CFD calculated pressure by applying the correction based on tail water level and draft tube efficiency. In order to produce the input for following FE analysis, steady state CFD analysis are normally performed for normal operation point, e.g. full load point, and exceptional operation points, e.g. runaway point, low part load.

Transient analysis produces the dynamic pressure distribution on all wet surfaces of the runner. By implementing a Fourier Transform routine into the CFD solver, the data in frequency domain in terms of amplitude and phase or real and imaginary parts are directly available in the CFD results for given frequencies. For the standard safety assessment, only the optimum operation point is considered. At this point, the dominant excitation appears at the wicket gate passing frequency, therefore only this single frequency is of interest for the following FE forced response analysis.

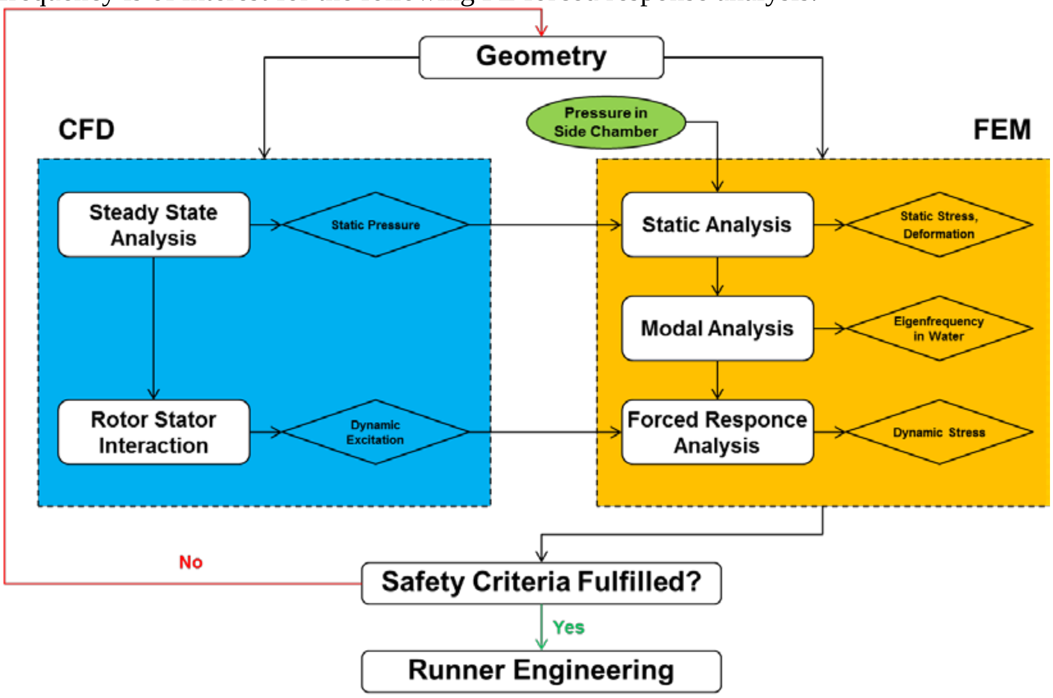

Figure 1. Runner safety check procedure

2.2. FEM analysis

FEM analysis includes static analysis, modal analysis in water and forced response analysis in water.

Static analysis takes into account the gravity due to runner weight, the centrifugal force due to rotation, the pressure force inside blade channel and the pressure force on external surfaces in runner side chamber. The inside pressure force is mapped from CFD static absolute pressure, while the outside pressure force is produced by axial thrust calculation. The distribution of static stress and radial deformation can be derived by the FE static analysis.

Modal analysis is carried out in water environment by coupling the structure elements with the acoustic fluid elements and the nearby boundaries, e.g. labyrinths seals, head cover, etc., are properly modelled, so that the added mass effect of the surrounding water and influence of nearby walls on the natural frequencies are considered. [4][5] The natural frequencies in a given range and the corresponding mode shapes are derived by the modal analysis.

Forced response analysis is performed by means of applying harmonic excitation at given frequencies. As mentioned before the interesting frequency here is the wicket gate passing frequency which is already known for a certain layout. The excitation forces are applied directly in frequency domain in terms of amplitude and phase or real and imaginary parts distributed on the blade channel surfaces, which are the output of the foregoing transient CFD simulation. An empirical constant damping value is given as input for the simulation. The water environment and nearby walls are also included. The amplitude and phase distribution on the blade channel surfaces can be derived by the forced response analysis.

2.3. Safety criteria

- Static stress
  Peak values of static stress at critical locations must be below the admissible values, depending on the locations and operation conditions. This check has to be done for both normal operation point and exceptional operation point.
- Radial deformation
  Maximum radial deformation in labyrinths must be below the admissible values, depending on the dimension of minimum gap and operation conditions. This check has to be done for both normal operation point and exceptional operation point.
- Resonance with RSI excitation
  The natural frequency of the critical mode has to deviate enough from the exciting frequency due to RSI. The critical mode is defined by the combination of vanes number of rotor and stator. [6]
- Resonance with von Karman excitation
  All the natural frequencies in the low frequency range, which contains most of the energy, must deviate enough from the possible von Karman excitation frequency range. The von Karman frequency can be derived by the blade trailing edge thickness, the relative velocity at blade trailing edge and the Strouhal number.
- Dynamic stress
  The peak value of the amplitudes of dynamic stress must be below the admissible value. This check is done only for the optimum point.

3. An example of root cause analysis (RCA)

Here shows an example of a RCA on a Francis splitter blade runner installed by a third party. After about 5000 hours of operation cracks have been found on 2 of 3 installed units in a hydro power plant. Main geometrical and operation parameters of the runner are listed in Table 1.

Table 1. List of runner parameters

Parameter	Nomenclature	Value	Unit
Maximum outlet diameter	D_2	1640	mm
Number of runner blade	Z_2	15+15	--
Number of wicket gate	Z_0	24	--
Maximum power	P_{max}	56.4	MW
Maximum head	H_{max}	247.4	m
Maximum discharge	Q_{max}	28	m^3/s
Rotational speed	N	428.6	rpm
Runaway speed	N_r	740	rpm

Cracks were found on totally 12 long blades of the 2 units while no cracks were found on the short blades (splitter blades). All cracks are located on the trailing edge close to the runner band surface. The distance of the cracks from the band surface varies between 25 and 45mm. One of the cracks is shown in Figure 2.

The crack pattern is very similar on the damaged blades which points to a systematic issue, like too high dynamic stresses. Usually, the highest static and dynamic stress on a radial turbine occurs on the trailing edge, close to the connection points to crown or band. Therefore, a change of the trailing edge geometry might show a significant impact on the stresses. RCA was performed using the RSC procedure to find out the cause of the cracks.

Figure 2. Crack observed on the long blade close to runner band

The static analysis shows that the maximum static stresses in long blade are much higher than the admissible values according to ANDRITZ HYDRO company standards for both normal operation point and runaway point. At the same time the maximum stresses in short blade are well below the admissible values under both operating conditions. In both cases the maximum stresses in long blade appear close to the connection between blade trailing edge and runner band, which coincide with the observed crack starting points. Figure 3 shows the distribution of static stress on long blade under normal and runaway operation conditions.

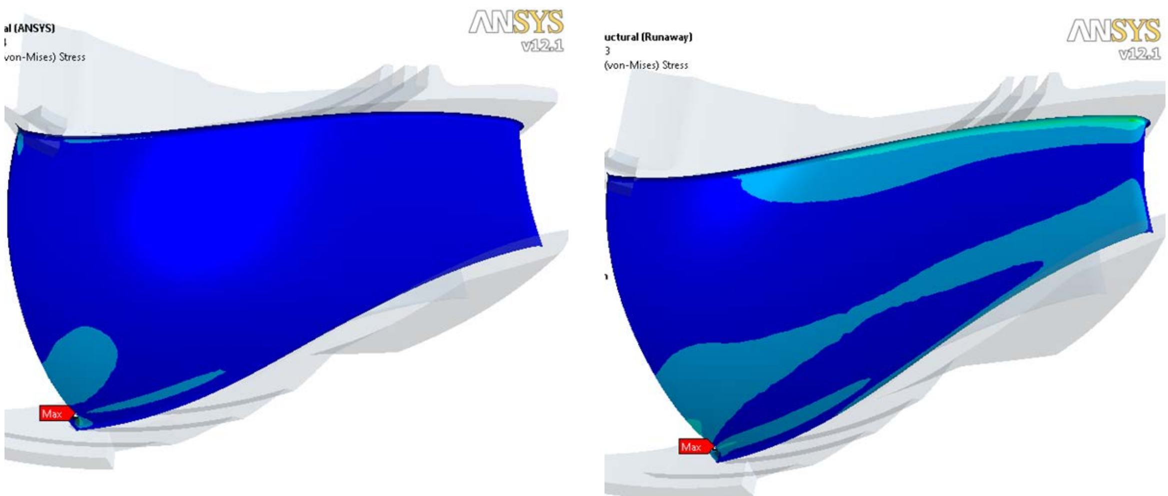

Figure 3. Distribution of static stress on long blade at normal operation point (left) and runaway point (right), location of maximum stress coincide with the crack start point. Please note that the scales of the colour schemes in these 2 pictures are different.

The natural frequencies of the runner prototype in water are revealed by modal analysis. For the most critical mode, which is the mode with 6 nodal diameters (ND6) determined by the number of runner blades and wicket gates, the frequency is 179.04 Hz. At the same time the excitation frequency from RSI is 171.44 Hz. The deviation between them is only 4.4% which is not enough according to the ANDRITZ HYDRO guideline. This indicates a high risk of resonance and check for dynamic stress is required. Figure 4 shows the numerical model and the resulted mode shape of ND6 mode.

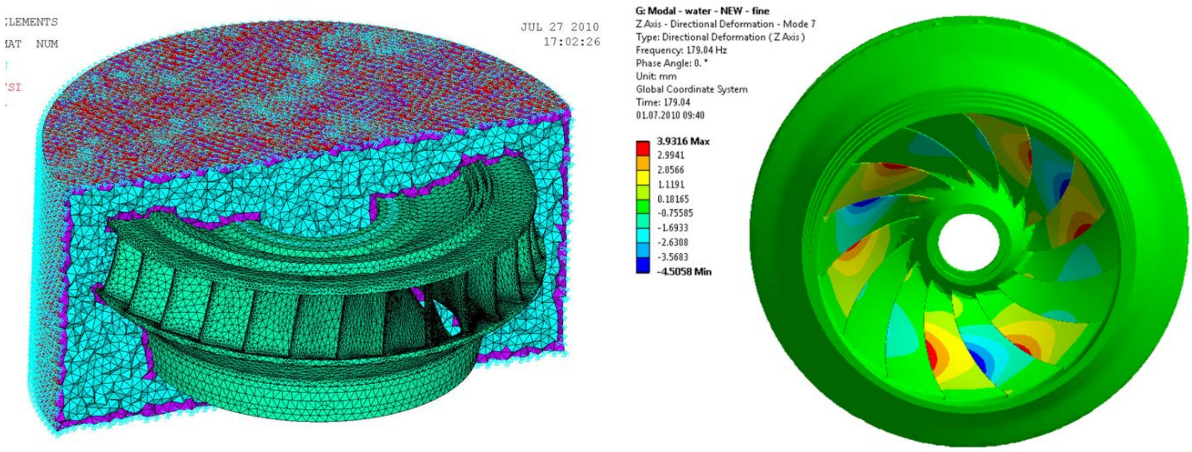

Figure 4. Finite element model of the runner in water environment (left) and mode shape of ND6 mode (right)

Dynamic stress was assessed by means of forced response analysis. Excitation forces at the RSI excitation frequency was obtained from transient CFD analysis. Dynamic torques[7] on long and short blades generated by pressure fluctuation are shown in Figure 5 (left), both in time domain and in frequency domain. Result shows that the maximum stress amplitude appears on the long blade close to the connection between blade trailing edge and runner band, which is again the same as the crack starting point, as shown in Figure 5 (right). The maximum value on long blade is much higher than the admissible value while on short blade the peak value is well within the admissible range.

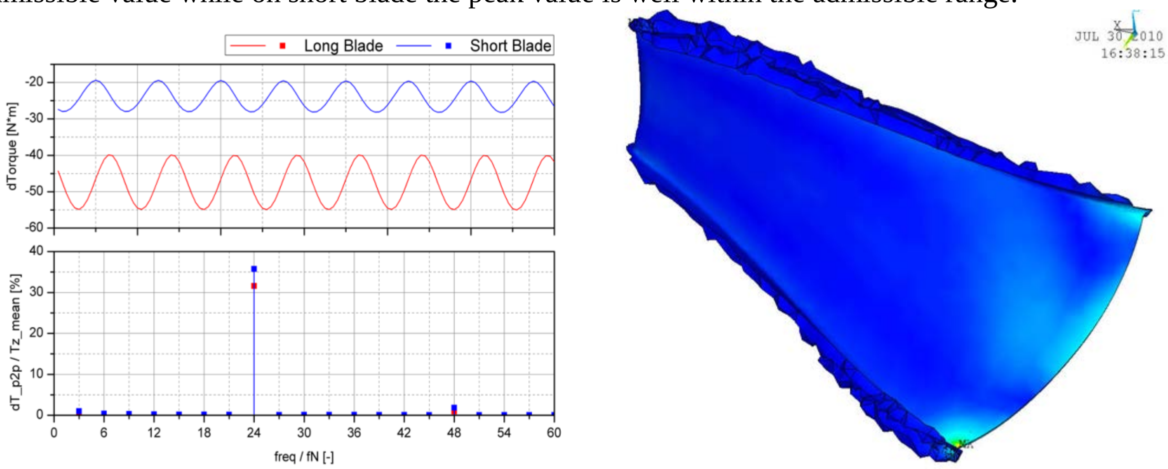

Figure 5. Dynamic torques on long and short blades (left) and the distribution of dynamic stress amplitude (right)

Based on the analysis results cutting back the long blade trailing edge was defined as one solution. Too big overhang, i.e. the curvature in flow direction, could be sensitive to both static and dynamic excitations. Therefore reduction of the overhang by means of cutting back can increase the stiffness of the runner blade hence reduce the local stress concentration and increase natural frequencies. Comparison of runner geometry before and after the cutting back is illustrated in Figure 6. The short blades kept unchanged.

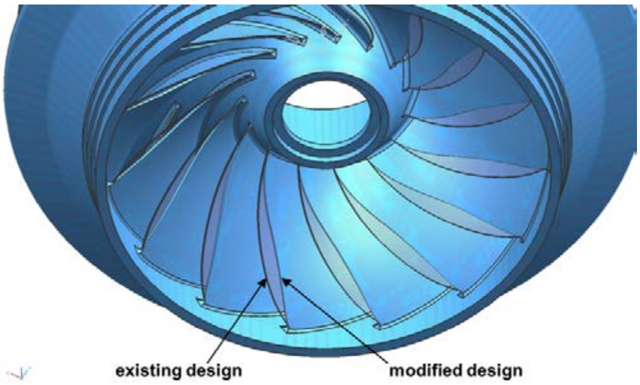

Figure 6. Comparison of existing runner and modified runner

The RSC procedure was applied to the modified runner as an examination of this solution. The results showed positive effect regarding all three criteria as mentioned before. Both static and dynamic stresses are reduced significantly and well below the admissible values. The natural frequency of ND6 mode in water is shifted up to 203.55 Hz so that the deviation from the RSI excitation frequency is 19%, which is large enough to exclude the resonance risk. Analysis results of the original runner and the modified runner regarding all criteria are summarized and compared in Figure 7.

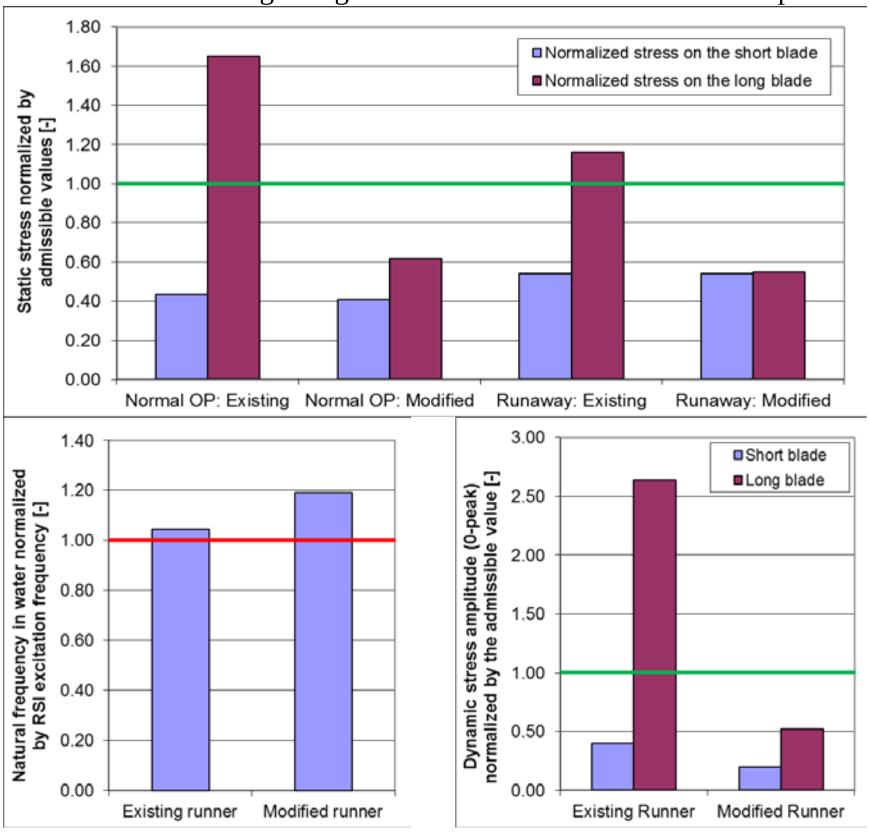

Figure 7. Effect of modification on trailing edge, in terms of maximum static stress (top, normalized by admissible values depending on operating conditions), natural frequency (down left, normalized by RSI excitation frequency) and maximum dynamic stress amplitude (down right, normalized by admissible value)

In addition to the influence on mechanical properties, the cutting back at trailing edge also impacts the hydraulic performance. This impact could be very significant and therefore must be carefully checked by means of CFD. The CFD results show the following:

- Peak efficiency level is slightly reduced by 0.1%.

- Peak efficiency position is shifted to higher discharge by 6%. By such shift the efficiency increases in high power output range while decreases in low power output range, as shown in Figure 8 (left). According to the operation record, hydraulic effects should be neutral or even lead to higher annual output.
- The modified runner is cavitation free at the most critical operation points.

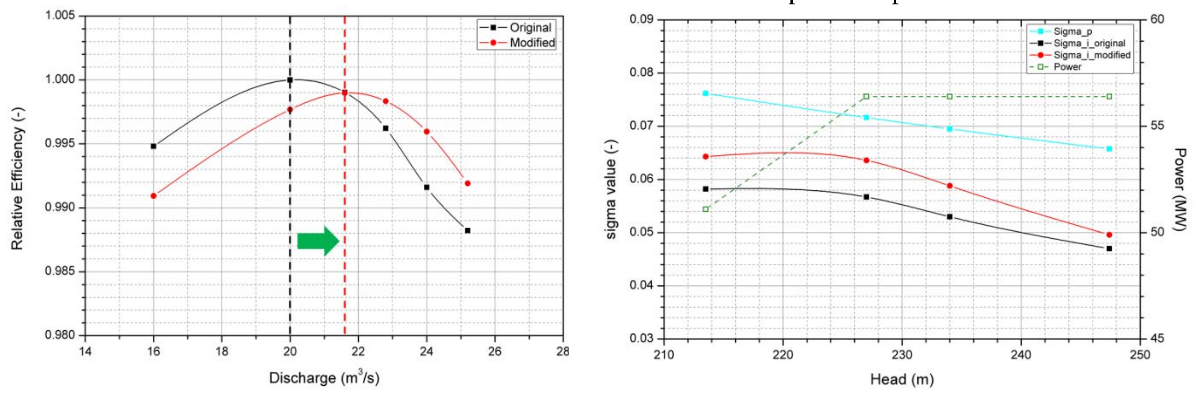

Figure 8. Impact of modification on efficiency (left, relative efficiency over discharge, peak position shifted to right side) and cavitation (right, sigma incipient lower than sigma plant indicates cavitation free)

So the conclusion is that such modification of the runner does not have significant impact on hydraulic performance in terms of efficiency, cavitation and power output. Therefore, it is considered as a feasible and sustainable solution for the problem of the existing runners, which will allow the operation of the units without restriction. This solution has been implemented and the units have been in operation for almost 2 years since the runners were repaired. No more failure has been reported by the customer.

4. Conclusions

The runner safety issues can be assessed very efficiently by applying the RSC procedure, which maps the excitation load from CFD to FEM and performs the various types of analysis in an automatic way. It has been implemented into the standard design procedure in ANDRITZ HYDRO so that safety risks can be detected and excluded during the early design phase. This tool is also useful for performing RCA. One example is quoted and discussed in this article.

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