Runner optimization for climate-related changes in the operation of hydro plants

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Climate change is increasingly impacting hydro plants, posing significant challenges such as greater sediment load, erosion, and cavitation issues. The runners of turbines, in particular, are exposed to increasing loads of various kinds. This paper demonstrates how customised runners, designed during refurbishment projects, can effectively mitigate these challenges and enhance plant performance. All the plants discussed are located in Türkiye, a region where the effects of climate change are becoming more pronounced.

Climate change is not only affecting the lives of humans, wildlife and vegetation, but is also having an increasing impact on the operation of hydro plants. Dry periods are becoming longer and more intense, and extreme weather events with flooding are more frequent. While hydro plants of all types were designed for somewhat even discharge decades ago, today they must frequently operate at low partial load and full load conditions. At the same time, operation in the medium load range is becoming increasingly rare. If plants are operated at larger reservoirs, an increase in head fluctuations can also be observed. Because of climatic changes, however, the sediment load is also increasing, which can lead to erosion problems in various powerplant components. In particular, the runners of turbines are exposed to ever-increasing loads of various kinds.

One possibility of adapting existing hydropower systems to changes in the operational management is to customize the design of the runners which have already reached the end of their service life. Successful refurbishment of hydropower plants is achieved by many years of experience in turbine design and numerical flow simulation. Moreover, it is also helpful if model-tested reference designs can be used.

Based on the example of three reference projects, this paper shows how customized runners can be designed to prevent damage and an increase the annual workload during refurbishment projects. All the powerplants described in the paper are in Türkiye, where the effects of climate change are sometimes very clearly evident.

1. Overview of the applied procedure

The customized new Francis runners presented in this paper were designed with the help of computational fluid dynamics (CFD). The prerequisite for a CFD-based refurbishment study is the existence of a 3D CAD model representing the overall turbine geometry. In the case of rather old turbines, the 3D model of the spiral case, including the stayvanes, the distributor with its guidevanes and the draft tube, can be re-modelled using the original 2D manufacturing and/or assembly drawings of the powerplant.

However, a realistic reconstruction of the runner is often only possible if a 3D surface scan is performed. The scan proves particularly difficult if the runner cannot be dismantled easily, the blades are heavily worn or damaged, the runner is very small, or the blade channels are very narrow (as is often the case with runners with the lowest specific speed). However, in the case of the projects presented in this study, 3D-remodelling based on available drawings and a 3D scan was possible without any restrictions.

To achieve a highly efficient design during a refurbishment project, a reliable CFD approach based on accurate and dense meshes representing the flow path from the inlet of the spiral case to the outlet of the draft tube is required. Furthermore, at least a section of the tailwater region is typically modelled by the research team to avoid any potential negative impact of the applied boundary conditions on the flow in the draft tube.

Fig. 1 shows an exemplary visualization of the modelling meshing and simulation processes used in the presented reference projects.

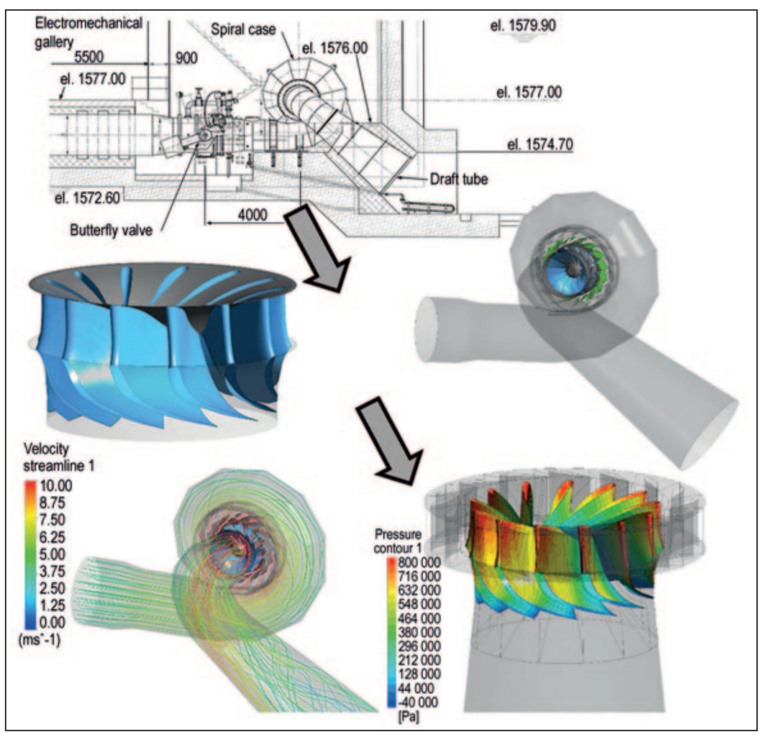


Fig. 1. Example of a visualization of the modelling, meshing and simulation processes.

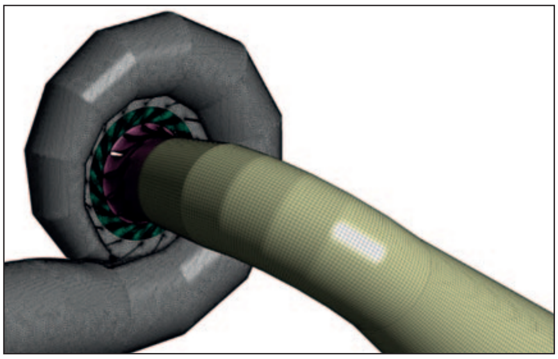


Fig. 2. Overall visualization of the meshes..

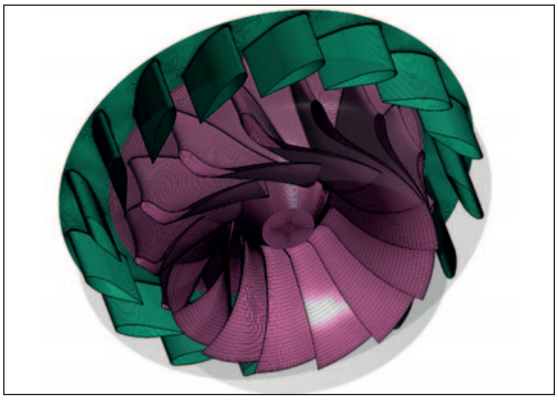


Fig. 3. Visualization of the meshes, distributor and runner.

2. Simulation approach

To perform the numerical simulation, including the generation of meshes, the creation of the simulation set up, the solution process, and the evaluation of results, the commercial software package ANSYS-CFX® was used. CFX® uses a finite volume-based discretization scheme up to second-order accuracy and is commonly used, among things, to simulate hydraulic fluid machinery.

2.1 Meshing and CFD model

For preparation of the numerical simulation, the turbine is typically divided into the following flow domains: 'spiral', 'distributor', 'runner', 'draft tube', and 'tailwater'. The CFD meshes were created using software packages ANSYS-Turbogrid© and ICEM-CFD© . Computational meshes with around 14.5 million nodes and around 20 million elements were used for the calculations performed. The averaged non-dimensional wall distance y+ achieved at the best efficiency point of the turbine accounts for $y+ \approx 25$. According to the authors' experience, further refining the meshes does not significantly improve the simulation results. Referring to reference project no. 2, Figs. 2 and 3 show a visualisation of the assembled CFD meshes.

The meshes were assembled to a simplified single-channel turbine model and a full 360° CFD model using ANSYS-CFX-Pre©. The single-channel model consisted of just one guidevane channel and one runner blade channel (connected with periodic boundary conditions) attached to the full draft tube and tailwater, and was used to optimise the runner designs. In contrast, the entire 360° CFD model contained all components of the turbine, from the inlet of the spiral case to the outlet of the tailwater block, and was used for the evaluation of the final turbine design and the simulation of the operation points required to create turbine hill charts.

The only permissible simplification concerns the runner side spaces, which are not taken into account typically to reduce the effort for the modelling and simulation. Instead, empirical approaches are applied to estimate losses because of disc friction and internal leakage passing the labyrinth region. However, there are some instances where the runner side spaces were modelled in the CFD process [Schiffer, Benigni and Jaberg, 2017[1]].

The STAGE-type interface model at the transition from the guidevanes to the runner, and the runner to the draft tube was used for steady-state and single-phase CFD simulations. Thus, circumferentially averaged velocity fields were applied at the corresponding interfaces. Furthermore, as a boundary condition, a total pressure level was defined at the inlet of the spiral case. In contrast, a constant static pressure level was set at the outlet of the tailwater. Consequently, the flow rate could be adjusted according to the opening of the guidevanes. For the wetted walls, which were treated as hydraulically smooth, the so-called automatic (or hybrid) wall functions were used. Thus, the software automatically switched between the wall function approach and the low Re approach, depending on the grid spacing near the wall.

As the turbulence model, the k-ω-based SST approach initially developed by Menter [1994[2]] was applied, as it has repeatedly proven to achieve the best solution for engineering applications in the field of hydraulic fluid machinery in terms of robustness, stability, and accuracy; see for example, Moessinger, Jester-Zuerker, Jung [2015[3]]. Furthermore, as the advection scheme, the 'High Resolution' method was chosen for the simulations presented in this paper. Moreover, the 'Automatic Timescale' option was used for the simulation process, extending over 750 iterations. With this approach, the root mean square residuals of pressure and velocity were kept below 5×10^{-5}, and convergent results for the turbine efficiency, discharge, and cavitation coefficient were achieved.

Table 1: Overview of nominal data

	Reference project No. 1	Reference project Nos. 2 and 3
Number of units:	3 vertical Francis turbines	3 horizontal Francis turbines
Net head H:	53 m – 73 m	108 m
Maximum discharge Q_{Max}:	19 m³/s – 21 m³/s	10.4 m³/s
Turbine speed n:	300 rpm	600 rpm
Turbine power P:	3 × 9 MW	3 × 10 MW
Specific speed $n_{q,BEP}$:	~ 50 rpm	~ 50 rpm
Runner diameter at outlet DS:	Ø 1750 mm	Ø 1035 mm
Number of runner blades z_{Ru}:	15	13
Number of guide vanes z_{GV}:	20	16

2.2 Evaluation of results

To evaluate turbine performance according to the IEC 60193 standard [1999[4]], the definition of the net turbine head H_{net} and the hydraulic efficiency $\eta_{hydr.}$ according to Eq. (1) and Eq. (2) was used. All quantities required for the calculations can be derived from the CFD results.

$$H = \frac{p_{Total-Inlet} - p_{Total-Outlet}}{\rho \cdot g} = \frac{\left(p_{Static-Inlet} + \frac{\rho}{2}\cdot\left(\frac{Q}{A_{Inlet}}\right)^2\right) - \left(p_{Static-Outlet,DT} + \frac{\rho}{2}\cdot\left(\frac{Q}{A_{Outlet,DT}}\right)^2\right)}{\rho \cdot g} \quad \text{... (1)}$$

$$\eta_{hydr.} = \frac{\text{energetic output}\,P_{out}}{\text{energetic input}\,P_{in}} = \frac{T_{Runner}\cdot\omega}{\rho\cdot g\cdot Q\cdot H_{net}} = \frac{\left(T_{Runner\,Blades}+T_{RunnerHub}+T_{Runner\,Shroud}\right)\cdot\frac{2\cdot\pi\cdot n}{60}}{\rho\cdot g\cdot Q\cdot H_{net}} \quad ...\,(2)$$

For a separate analysis of the performance of each component of the turbine, a head loss analysis was carried out. For this purpose, the total head difference between the inlet and outlet of each component is set in comparison to the net head. Thus, Eq. (3), the basis for a loss composition (see Fig. 5), can be applied.

$$\eta_{hydr.} = 1 - \frac{\sum H_{Loss}}{H_{net}} = 1 - \frac{H_{Loss-Spiral}+H_{Loss-Guide\,Vane}+H_{Loss-Runner}+H_{Loss-Draft\,tube}}{H_{net}} \quad ...\,(3)$$

With the help of an appropriate statistical evaluation method, it is possible to indicate cavitation susceptibility, although only single-phase CFD simulations were performed. Hence, the ‘σ-histogram method’ was applied to validate the cavitation performance. The results achieved with this method were cross-checked several times with experimental data gained from the model test rig. According to our experience, the σ-values calculated thereby correspond to the σ_0-values detected during a model test. For more details see Schiffer, Benigni and Jaberg, [2018[5]].

3. Selected case studies

Refurbishment projects with particularly impressive damage patterns were selected for this paper. The powerplants concerned are located at different sites in Türkiye and are equipped with vertical and horizontal Francis turbines. The nominal data of the turbines are summarized in Table 1.

3.1 Reference Project No. 1

Reference Project 1 is a hydro plant located approximately 350 km northeast of Ankara on the Yeşilırmak river, which runs into the Black Sea. The plant is equipped with three Francis units with a total capacity of $P = 3 \times 9$ MW and was commissioned in 1966. Since the plant began operation, frequent repair works have been required because of massive cavitation damage found in different locations on the runner blade and shroud ring. The damage is particularly pronounced on the suction side of the blade at the runner outlet (main blade region and blade root region on the shroud side), and on the suction side of the leading edge (on the hub and the shroud side).

After about 55 years of operation, it was decided that comprehensive refurbishment work should be undertaken. A causal analysis of cavitation damage in the original design was performed, which relied on an extensive CFD study as a basis for the customized optimisation of the runners. Because of the comparatively high head variation between one- and three-unit operation, it was required to focus the simulation and optimization on at least two different head levels ($H = 53$ m and $H = 73$ m).

With the help of the CFD results, the cavitation damage found on the original runner could be explained well. For this purpose, Fig. 4 was created. During the analysis, it turned out that several independent flow phenomena could explain the cavitation damage. While some damage resulted from high load operation at a comparably low head (three-unit operation), other damage arose from the part-load operation of just one unit. The pressure distribution calculated at full load

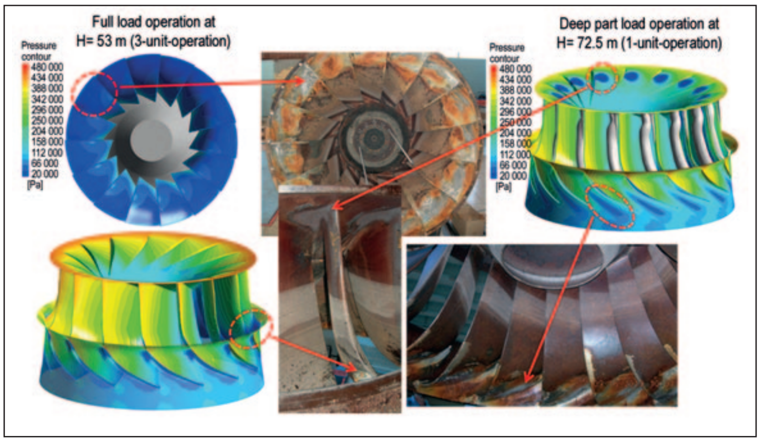


Fig. 4. Comparison of low-pressure zones calculated using CFD, and damage found on the original runner.

operation at $H = 53$ m (three-unit operation, see left pictures above) and the pressure distribution calculated at deep part load at $H = 73$ m (one-unit operation, see right pictures) show that the regions with the lowest pressure values correlate with the cavitation damages visualised on the photographs. Furthermore, the cavitating interblade vortices, which are typical for operations at deep part load, are visualized with the help of grey-coloured 3D ISO surfaces showing the lowest pressure in the runner.

Based on these findings, an optimized runner design was developed. The target was to shift the best efficiency point to a significantly lower discharge and to achieve an overall efficiency improvement, a more balanced pressure distribution on the runner blades and thus better cavitation behaviour.

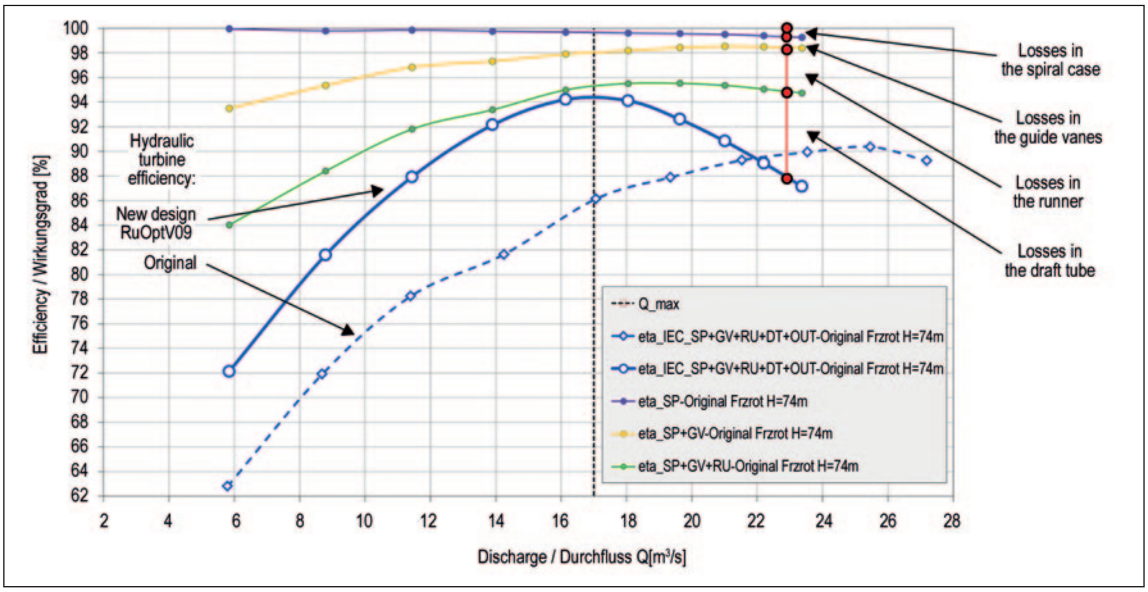
Referring to a head level of H =74 m, Fig. 5 depicts the total hydraulic turbine efficiency calculated with the original (see dashed blue curve) and optimized (solid blue curve) Francis runner by applying the full 360° CFD model. In addition, a loss composition showing the losses in the single components of the turbine was prepared. Obviously, the original turbine had been designed for too high a discharge. With the new runner design, the peak point of the efficiency curve was shifted from $Q \approx 25$ m³/s to $Q \approx 17$ m³/s and the peak efficiency was increased from $h_{hydr.,max} = 90.5$ per cent to $h_{hydr.,max} = 94.5$ per cent. Moreover, the part-load efficiency and part-load behaviour could be improved strongly in terms of vibrations. The losses occurring in the runner side spaces (because of disc friction and leakage) were considered with 0.75 percentage points of the power generated at the best efficiency point [IEC 2008[6]].

3.2 Reference Project No. 2

The second reference project refers to a powerplant located on the Çataksuyu river in southeast Türkiye, equipped with three horizontal Francis turbines. The nominal output of the turbines, which were commissioned 15 years ago, accounts for around 3×10 MW. Because of a comparably high sediment load, massive erosion damage was frequently found on the runner blades. Therefore, different types of coatingswere used for several years to postpone the repair period. A visualization of the damage found on the original runner is presented in Fig. 6. The different colours of the blades refer to the various coatings that were tested.

After the appearance of the first obvious damage, an elastic coating of Rilsan® polyamide was applied to the original runner; all the other runners that were initially tested were coated with tungsten carbide. Despite

Fig. 5.
Comparison of the calculated efficiency curves of the original and the optimized turbine.



the subsequent coating, the customer had to carry out extensive repair works every 2 to 2.5 years on average.

However, the erosion problems could not be solved by coating alone, and the situation was no longer acceptable for the operator. Thus, the customer wanted to investigate how to optimize lifetime of the runners.

Fig. 6.
Visualization of the different damage found on the original runner.

The only option was to design and manufacture new runners with significantly thicker runner blades and a slightly wider blade channel. For this purpose, a detailed analysis of the actual situation and subsequent geometrical optimization of the Francis runner was performed by applying CFD modelling.

Referring to a net head of H = 108 m, Fig. 7 shows the total hydraulic turbine efficiency calculated with the worn-out original (green curve) and optimized (orange curve) Francis runner by applying the full 360° CFD model. The corresponding cavitation coefficients (light-coloured curves) are also presented.

With the help of the new runner design, the efficiency curve was shifted to a higher discharge and finally matched well with the efficiency curve taken from the original hill chart. At the nominal head of H = 108 m, a hydraulic peak efficiency of almost 94 per cent can be found at around Q = 8.5 m^3/s. As with Reference Project No. 1, the losses occurring in the runner side spaces

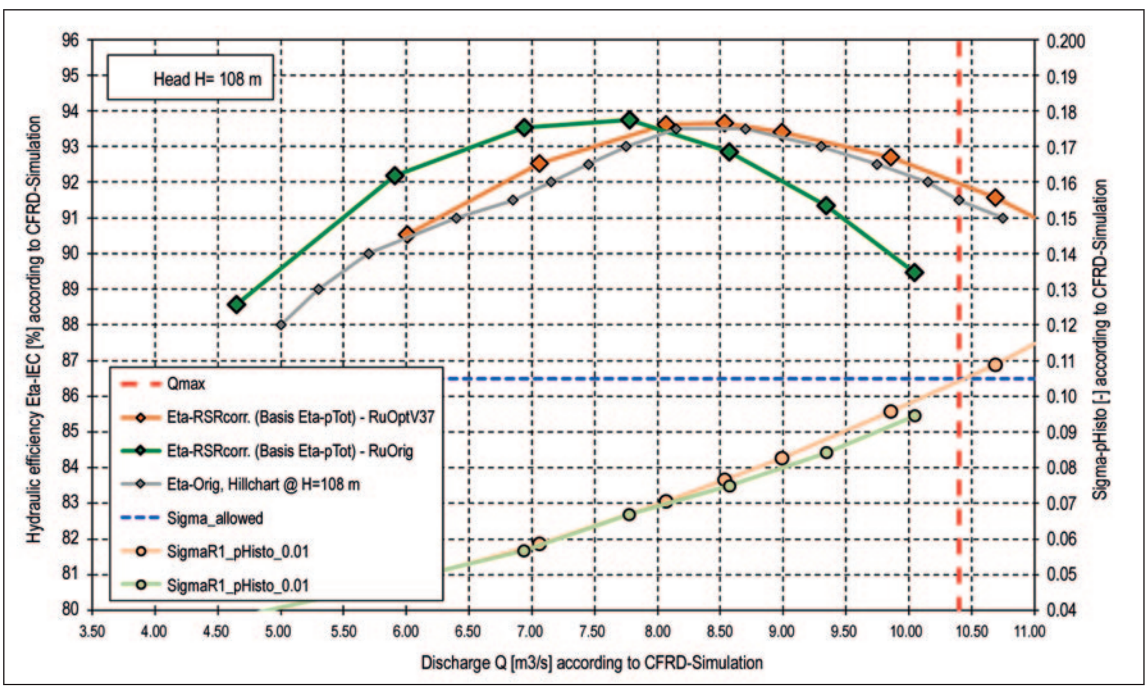


Fig. 7.
Comparison of the calculated efficiency and cavitation coefficient of the original and the optimized turbine.

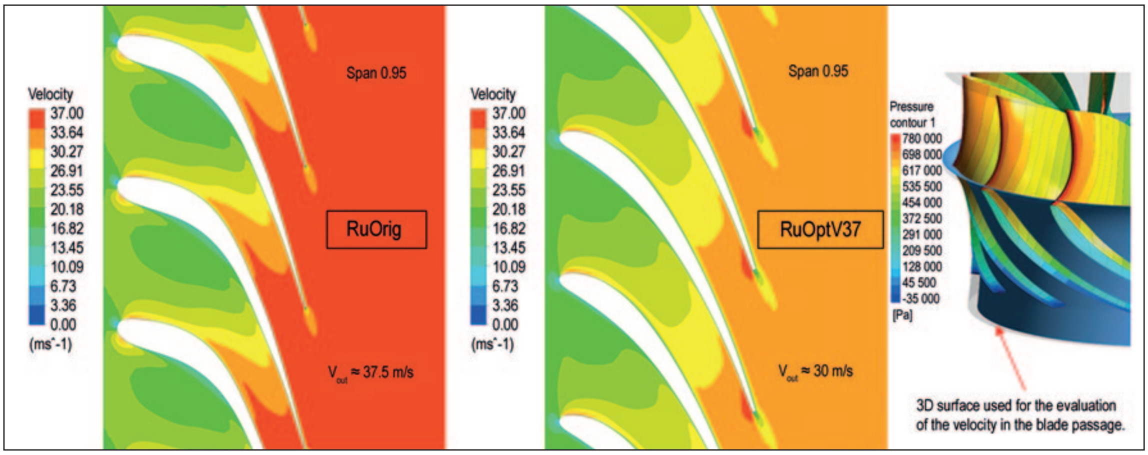


Fig. 8. Blade-to-blade view of the velocity in the blade passage at 95 per cent blade height at $Q \approx 10$ m³/s and $H = 108$ m (left view, original runner; and, middle view: optimized runner).

were considered according to IEC 62256 [2008[6]].

Furthermore, it was found that the cavitation characteristics of the new design are slightly above those of the original design. An explanation could be that the thickness of the blades had been significantly increased. However, the CFD results show that cavitation-safety is provided up to the maximum discharge of $Q = 10.5$ m³/s.

During the optimization process, special attention was paid to improving erosion resistance. Thus, the velocity distribution in the blade channel close to the shroud region of the runner was an important focus.

Referring to the same operation point ($Q \approx 10$ m³/s at $H = 108$ m), Fig. 8 presents a blade-to-blade view of the velocity in the blade passage at 95 per cent blade height. The associated evaluation surface is shown on the right (blue 3D surface). Especially in the shroud region, the blade channel of the original runner is much tighter than that of the new runner design RuOptV37. This results in a significantly higher flow velocity, negatively impacting erosive wear. Thus, the new runner design (with clearly lower velocity in the blade channel) is expected to be less sensitive in terms of material removal caused by erosion. Furthermore, the illustrations show that the blade thickness in the worn-out outlet region of the runner was significantly increased during the design process of the new Francis runner.

3.3 Reference Project No. 3

The third example refers to an additional idea realised for Reference Project No 2. In preparation for the optimization process, an analysis of the operating regime was carried out. It showed that the plant is frequently operated at high-load conditions, but also quite regularly at low-load conditions. Especially during the dry winter season, only one of the turbines operates with a shallow discharge. To solve this issue, it was decided to equip one of the three turbines with a new runner designed explicitly for operation at deep part load. By using the geometrical model of the runner optimized for standard operating conditions, blade angle distribution modifications were incorporated to shift the performance curve to lower discharge. During the optimization process, it became apparent that the runner blades needed to be significantly longer for a sufficient improvement in the part-load performance.

The results achieved during the optimization for part-load operation are summarized in Fig. 9. While the

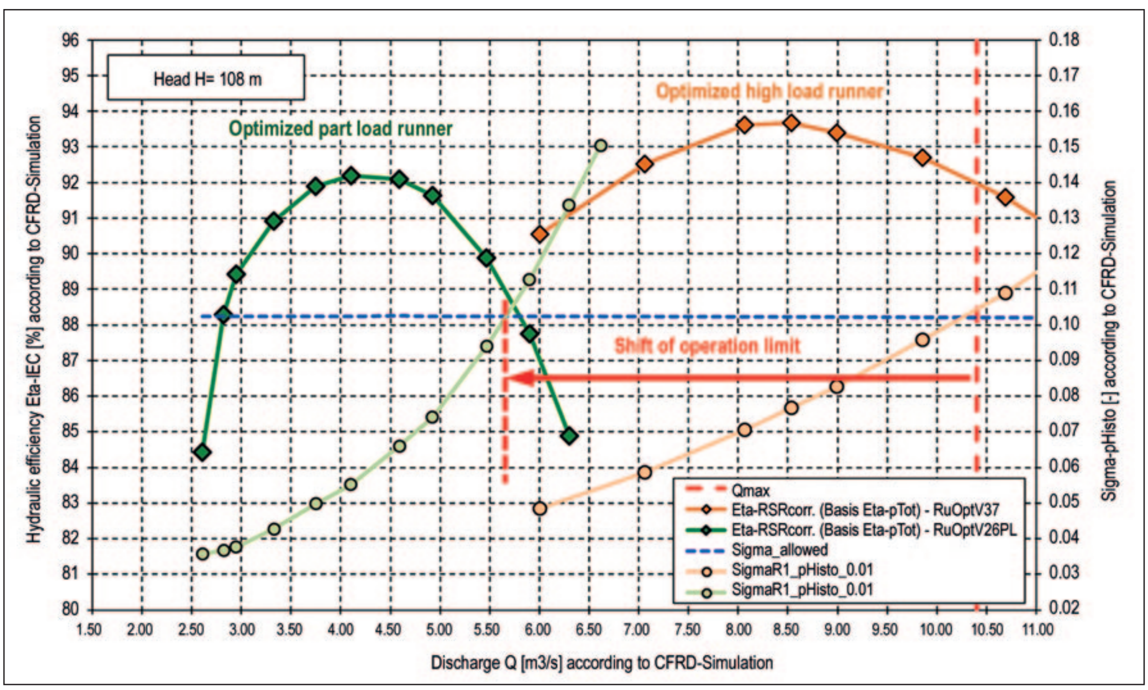


Fig 9. Comparison of efficiency and cavitation characteristics of the optimized high-load- and low-load runner.

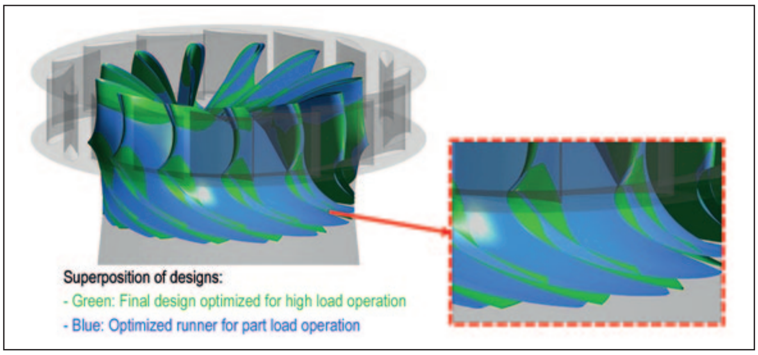


Fig 10. Superposition of the runner optimized for high load conditions and the one specifically designed for deep part-load operation.

orange-coloured curves refer to turbine efficiency (diamonds) and cavitation characteristics (circles) achieved with the optimized high-load runner, the green curves represent the results achieved over the course of optimisation of the part-load runner. While the operation limit was shifted from around $Q = 10.4$ m³/s to $Q = 5.6$ m³/s, the turbine efficiency at deep part load was improved enormously during the design process. At the same time, peak efficiency decreased from around 93.7 per cent to around 92.1 per cent, which can be explained by the fact that the blade channels of the part-load runner needed to be much longer and tighter.

Finally, Fig. 10 presents a superposition of the runner optimized for high-load conditions and the one specifically designed for deep part-load operation, where the required extension of the blades is clearly visible.

4. Summary and conclusions

During detailed CFD studies, the Francis turbines at two hydro plants in Türkiye were investigated as the basis for extensive refurbishment works. The original turbines had been affected by severe damagecaused by cavitation and erosion. The damage appeared in various places spread across the surface of the runner. The original 2D drawings and 3D surface scans of the runners were used to model and simulate the original turbine configurations over a wide range of operations. The results of the CFD studies could explain the cavitation damage. In the next step, improved runner designs were created to increase efficiency, shift the ideal operation range, and improve the cavitation behaviour. Moreover, pressure distribution on the new runners was also significantly improved. In the case of Reference Project No. 2, an additional runner design, optimized for operation at deep part load, was developed.

In the meantime, the runner designs described have been manufactured and installed, and the improved performance has been demonstrated at the plant. The refurbished units have operated without any trouble with regard to cavitation and erosion, noise emission or vibration for several years.

Fig. 11. View of the suction side of the runner after around 2800 h of operation. Left: original design; right: optimized design.

For Reference Project No. 2, Fig. 11 shows the original runner (left) and the optimized design (right). Both pictures refer to an operation time of around 2800 hours. The operational experience clearly proves the improved resistance to erosion.

With the new runner designs, which were finally flamespray-coated with tungsten carbide again, the turbines have been running since 2022. Apart from minor flaking of the coating, they remain in good condition. ◊

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