Development of wide operating range runner for Francis turbine upgrading

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Abstract. The proposed paper describes the upgrading procedure of the six units installed at HPP Stejaru in Romania in the sixties of the last century. The power plant is furnished with two identical 55 MW units and four identical 28 MW units. The scope of the upgrading was same for all units, to install new runner and wicket gate assembly into the existing spiral case and draft tube. The paper is focused especially on the procedure of the hydraulic design of the new Francis type runner due to demanding request for unusually wide range of operation for such type of turbine. The most challenging was the condition of adoption of wicket gate height, which was insufficient regarding specific speed of the new runner. The first CFD computations confirmed that the standard design of the runner was not able to fulfill the expected parameters. Therefore, the automatic mathematical designer driven optimization process was applied. The results of the optimization iteration process were confirmed during development tests and finally approved by additional acceptance tests carried out at the neutral laboratory as was required in tender specification. The performance features of the new turbine were found as excellent especially applicable for wide range of operation heads and outputs. Even in the marginal operation there are not expected any troubles with the cavitation or excessive pressure pulsations and turbine vibrations. The manufacturing process of the new components is now in the progress.

1. Stejaru HPP refurbishment project
The Stejaru HPP, which belongs to Bistrita river hydroelectric development (see Figure 1.) and generates 435 GWh of electric power per year, was built between 1950 and 1960 as one of the largest hydropower plants in Romania at the time. The power plant has been commissioned during 1960-1961 and it is equipped with 6 Francis turbines, two identical 50 MW units and four identical 26.8 MW units (see Figure 2.). The pressure headrace system consists of headrace tunnel, surge chamber, valve chamber, two overground parallel penstocks, each connected to a manifold with branch for each unit. The scope of works consists of full refurbishment of all hydro power units having a total installed power of about 210 MW, as well as of
the whole assembly of mechanical, hydro and electric equipment within the hydro power plant. For an example, refurbishment of the hydro-mechanical equipment of the butterfly valve, the lifting and transportation equipment, upgrading of the electrical equipment and systems together with the external electrical substations of 220/110 kV, rehabilitation of the associated civil works to the hydropower plant, the automation system in order to bring them in line with the current level of technology and to extend their lifetime by 30 years. The refurbishment of the power plant will take five years, as the units will be replaced one by one, with the others operating. The Stejaru HPP is operated by Romanian Hidroelectrica, which own as much as 272 hydropower plants generating approximately 20,000 GWh of electric power per year which is 33 % of the total hydropower generated in Romania. You can see the general view of hydro power plant Stejaru in Figure 3.

2. Scope of the upgrading process
The scope of the upgrading was same for all units, to install new runner and wicket gate assembly into the existing spiral case and draft tube. The hydraulic design was focused especially on demanding request for unusually wide range of operation for such type of turbine. The range of new turbine net head varies according the number of turbine simultaneously in operation from minimal value 77 m up to maximal value 144 m, where the nominal value of net head is 126.5 m. Regarding the turbine flow operation, for bigger 55 MW unit the flow range is 14 – 47 m$^3$/s and for the smaller 27.4 MW unit the flow range is 7 – 23.5 m$^3$/s. The weighted efficiency area is very extensive and covers most of the operational area. The contract contains the table with 84 values of turbine efficiency inside the operational area used for weighted efficiency evaluation.

![Figure 2. View of the machine hall.](image2.png)

![Figure 3. General view of power plant Stejaru.](image3.png)

![Figure 4. Turbine layouts comparison.](image4.png)

Usually, in such type of hydro power plant rehabilitations, the guarantees are based on some existing type of Francis turbine with similar parameters. The guarantees of Litostroj Power were based on similar middle specific speed Francis turbine ($n_s\approx 200$) with some non-homological dimensions. As significant has appeared the difference of bigger spiral case together with the higher distributor of turbine designed by Litostroj Power CBE (see Figure 4.). The first CFD computations confirmed that the most challenging is the condition of guide vane height adoption, which is insufficient regarding specific speed of the new runner. The initial design of the runner was not able to fulfill expected parameters for such extensive operational area. Therefore, it was necessary to use the automatic mathematical designer driven optimization process, which was able to find new hydraulic shape of the runner blade capable to reach the guaranteed parameters of the Stejaru project.
3. New turbine hydraulic design

The improvement of Stejaru hydro power plant parameters did not include increasing of the maximum power output of each turbine unit only. The main focus was also paid on the flatness of the turbines characteristic. Our goal was to design the new guide vanes and the new runner to reach the guaranteed values of each unit which were very strict regarding the head as well as the discharge range. The main parameters of new turbines ($n_s \approx 200$) for Stejaru hydro power plant are listed in Table 1.

| Table 1. Table of main data of new turbines for Stejaru HPP. |
|-------------------------------------------------------------|
| Turbine power output [MW] | 55 | 27.4 |
| Number of units | 2 | 4 |
| Rotational speed [rpm] | 333.3 | 500 |
| Net Head range [m] | 79 – 143 | 77 – 144 |
| Discharge range [m$^3$/s] | 7 – 23.5 | 14 – 47 |
| Runner blades | 14 | 14 |
| Guide vanes | 18 | 18 |
| Reference diameter [mm] | 2330 | 1620 |

Due to unusually wide range of operation for such type of Francis turbine, it was necessary to optimize the guide vane and the runner blade regarding to their hydraulic shape. In order to find suitable shape of the runner blade for each unit, combination of two optimization methods was chosen. Each of them has different character of finding extreme but both of them minimize an objective function to find the goal. Global optimization method allowed exploring of space and determining the regions where the minimum could be found. To detect the finite value of extreme a local optimization method was used.

As global optimization method the Differential Evolution (DE) was used. It belongs to the class of genetic algorithms which use biology-inspired operations of mutation, crossover and selection on population to minimize an objective function (see Figure 5). It draws on Darwin’s phrase “survival of the fittest”. DE solves the optimization problem by evolving a population of several individuals in parametric space which are randomly generated between lower and upper bounds defined by the user. The process repeats until a stopping criterion is reached.

![Figure 5. Illustration of DE population.](image)

The simplex method belongs to the direct search class of local methods which do not require derivatives. The algorithm uses geometric figure called “simplex” consisting of ($n+1$) vertices in an n-dimensional space. For two variables ($n=2$), the simplex is a triangle and objective function $f$ is evaluated at each of its vertices $x_1$, $x_2$, $x_3$ (see Figure 6.). Finding of extreme is done using modification of simplex. Firstly, the vertices are arranged in descending order according to the objective function values. The worst vertex $x_{n+1}$ is discarded and replaced by a point with lower objective function value. This point is created by four operations namely reflection, expansion, contraction (inner and outer) and shrinkage operation. Big advantage of this method is very fast convergence.

![Figure 6. Simplex movements, $X$ centroid, $X_r$ reflection, $X_e$ expansion, $X_c$ inner contraction, $X_{c+}$ outer contraction, the coloured triangle−shrinkage operation.](image)
4. Optimization process

The runner blade was described by several parameters using commercial software Ansys. The number of the parameters depends on used optimization method. Total count of the parameters diverges from 26 to 36. Bezier curves are used for the definition of the blade shape. The created blade profile was meshed automatically and the mesh was connected to other pre-meshed geometry to create CFD model. This model was used for fully-automatic analysis in Ansys CFX and it was consisted of three main parts (see Figure 7). Inlet passage was constituted by the inlet volume with velocity profile prescription as an inlet boundary condition. The runner blade segment represented the second part and the draft tube was the third part with the opening boundary condition at the draft tube outlet. The inlet velocity profile was exported from the CFD analysis of entire spiral case including stay and guide vanes. It was composed of velocity components (radial, axial, circumferential) as well as of the turbulence components (turbulence kinetic energy, turbulence eddy dissipation). Every single part of the CFD model was connected to the next one by interface. The whole CFD model was created for runner suction diameter about 320 mm and it was analysed by k-epsilon steady state analysis which was the part of the optimization cycle.

The optimization itself was designed as multi-objective and was controlled by the in-house software. This utility controlled all optimization steps according to Figure 8. The optimization cycle starts either with some initialization geometry or with randomly generated initialization parameters. The whole fully-optimization cycle was composed of the mesh generation, CFD analysis, result analysis and objective function evaluation. Then the modification of the geometry was performed and the cycle was repeated. The runner blade was optimized regarding the flatness of the turbine characteristics and therefore several operational points were used in optimization cycle. At least the maximum output, the best efficiency point (BEP) and the partial load of the turbine were used as the optimised points. The method of genetic algorithm (DE) with the population of 15 individuals was applied in phase one and around 2000 blade modifications were calculated. After that, the final “brushing up” of the geometry was done using Simplex Nelder-Mead algorithm with next 700 variants in phase two.

Note that the objective function is very important. It can affect the results of the optimization process significantly as its value is a casting vote in which way the blade modification will proceed. Equation (1) defines the objective function which was considered for the blade optimization. It consists of several terms and their weighed factors which are finally sum of N operation points.

\[ f = \sum_{i=1}^{N} \left( w_H \cdot f_H + w_E \cdot f_E + w_K \cdot f_K + w_S \cdot f_S \right) \]  

Where subscript H deals with Head term, E deals with Efficiency term, K deals with Cavitation term and S deals with Swirl intensity term.
5. Hydraulic design of the runner

The hydraulic design of new runner has taken into account the issues which are related with Francis turbine for such wide operating range. The blade design covers the turbine operation for whole range of head as well as discharge. The leading edge (LE) shape together with the blade thickness distribution proved to be very important. Both above mentioned and other parameters were optimised to solve the issue of cavitation at maximum head together with the sufficiently flat characteristic at minimum head operation. Although 2 bigger 55 MW units were fitted by new almost conventional shaped blade design, the different combination of turbine parameters for 4 smaller 27.4 MW units (rotational speed, runner diameter) showed us the need to develop new unusual shaped runner blade, which helps to cover the wide operating area. The operating areas for both units are shown in Figure 9.

The non-conventional hydraulic design of small unit runner means, in this case, the S-shaped (vice versa) leading edge of the runner blade with the unusual blade thickness distribution while the trailing edge (TE) of the runner blade shows quite standard shape. This combination helps the runner to cover the wide operating area of the turbine without problems of blade cavitation and with very good behaviour, regarding the magnitude of pressure pulsation. The level of guaranteed efficiency was reached by very good flatness of turbine characteristics. The model runner for small unit is shown in Figure 10. The pressure distribution of conventional runner design for big units and non-conventional runner design for small units at nominal head near BEP is compared in Figure 11.

![Figure 9. Turbine units operating area.](image9)

![Figure 10. Model of S-shaped runner.](image10)

![Figure 11. Pressure distribution at conventional runner (left) and non-conventional runner (right).](image11)

In fact, the maximal efficiency level at BEP is approximately the same for both runners. The S-shaped runner shows better cavitation properties, better dynamic behaviour at deep partial load operation as well as the flatter turbine characteristic regarding the head and discharge range. Especially in case of the turbine operation at minimal head, the unusual runner design helps us to reach the sufficiently flat characteristic for small 27.4 MW unit.
6. Model tests
The results of the optimization iteration process were confirmed during development tests of both units and finally approved by additional acceptance tests carried out at the neutral laboratory in presence of customer as was required in tender specification (see Figure 12.).

![Figure 12. Model installed at CBE hydraulic laboratory (left) and at neutral laboratory (right).](image12)

The acceptance tests were successful for both turbine units and the performance features of the small turbine unit were found as excellent especially applicable for wide range of operation heads and outputs. Even in the marginal operation there are not expected any troubles with the cavitation or excessive pressure pulsations and turbine vibrations.

![Figure 13. Model tests results comparison – performance (left) and pressure pulsations (right).](image13)

As was mentioned, the most of parameters of new developed non-conventional runner was found as better in comparison with the conventional design. Especially the flatness of the turbine characteristic as well as lower pressure pulsations mainly at deep partial load (see Figure 13.). On the other hand, due to quite big improvement of the turbine characteristics flatness at low operating heads, the behaviour of the turbine at runaway speed has changed. The runaway speed for the new S-shaped runner is slightly higher in comparison with conventional design. The dependency of speed and discharge factor for runaway speed at sigma plant condition is shown in Figure 14.

![Figure 14. Diagram of runaway speed.](image14)
The flow comparison at the outlet of each turbine runner for several operating points at nominal head is shown in Figure 15 and frequency analysis of pressure pulsations at draft tube cone at same operating points are shown in Figure 16. At the first sight, the differences of the flow at sigma plant condition are not big. However, the evaluation of the pressure pulsations shows significantly lower amplitudes at turbine partial load and deep partial load operation in case of non-conventional runner (see Figure 13.). Regarding the turbine partial load operation (55% of $Q_{\text{max}}$), the lower amplitudes of pressure pulsations are associated with partially disintegrated vortex rope in the draft tube cone. The turbine deep partial load operation (30% of $Q_{\text{max}}$) shows significantly lower amplitudes for S-shaped runner blade (see Figure 16.) which is most probably related with better flow conditions at leading edge of the runner blade. This opinion is also supported by lower noise outgoing from the turbine vaneless area at this operating point during model tests. The operating regime at maximum turbine power output shows the similar behaviour at each runner outlet with no cavitation on runner blades.

Figure 15. Stejaru model tests - flow at runner outlet at sigma plant condition (no air admission).
The comparison of cavitation break curves from model tests for maximum turbine power output of small 27.4 MW unit as well as big 55 MW unit at nominal head operation is shown in Figure 1. The similarity of dependency of Thoma cavitation coefficient of both runners at this point is evident.

**Figure 16.** Stejaru model tests – frequency analysis of pressure pulsations at draft tube cone.

The comparison of cavitation break curves from model tests for maximum turbine power output of small 27.4 MW unit as well as big 55 MW unit at nominal head operation is shown in Figure 17. The similarity of dependency of Thoma cavitation coefficient of both runners at this point is evident.

**Figure 17.** Thoma cavitation coefficient for maximum turbine power output at nominal head.
7. Refurbishment progress
All turbine components are manufactured in Slovenia at Litostroj Power workshop, where the production of bigger 55MW as well as smaller 27.4 MW turbine units is performed (see Figure 18.). The hydro generators are produced by Romanian UCM Resita company.

![Prototype runner at Litostroj Power workshop (left) and works on site (right).](image)

Figure 18. Prototype runner at Litostroj Power workshop (left) and works on site (right).

8. Conclusions
The new hydraulic design of turbine runners for Stejaru HPP refurbishment has been developed. Two different runner designs were optimised to cover the wide operating area of each turbine unit. The conventional blade design was applied for two big 55 MW turbine units while for four small 27.4 MW turbine units the non-conventional blade design was developed. The reason of this solution was the difference between operating area of each unit, where the smaller units need better efficiency mainly at low heads. The new unusual S-shaped blade design helps to reach flat characteristic at low heads as well as the good cavitation and dynamic behaviour of the turbine at wide operating range. The guaranteed parameters for both turbine units were achieved by acceptance tests carried out at the neutral laboratory in customer presence.

9. References
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