Experiences with the hydraulic design of the high specific speed Francis turbine

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Abstract. The high specific speed Francis turbine is still suitable alternative for refurbishment of older hydro power plants with lower heads and worse cavitation conditions. In the paper the design process of such kind of turbine together with the results comparison of homological model tests performed in hydraulic laboratory of ČKD Blansko Engineering is introduced. The turbine runner was designed using the optimization algorithm and considering the high specific speed hydraulic profile. It means that hydraulic profiles of the spiral case, the distributor and the draft tube were used from a Kaplan turbine. The optimization was done as the automatic cycle and was based on a simplex optimization method as well as on a genetic algorithm. The number of blades is shown as the parameter which changes the resulting specific speed of the turbine between \( n_s = 425 \) to \( 455 \) together with the cavitation characteristics. Minimizing of cavitation on the blade surface as well as on the inlet edge of the runner blade was taken into account during the design process. The results of CFD analyses as well as the model tests are mentioned in the paper.

1. Introduction

In case of any modernization the customer has to consider the investment return very properly. In the field of water turbines the annual production of energy is one of the key criteria that can indicate how the modernization will be profitable.

At the beginning of 20th century a lots of small hydropower plants were built worldwide. They were often erected in the sites with low heads \( (H) \) and high discharges \( (Q) \) which means higher specific speed \( (n_s) \) of turbines. This paper deals with the area around \( n_s = 400 \) to 500 (see Fig. 1).

Equation (1) below describes how the specific speed is computed (where \( n \) is nominal speed).

\[
n_s = 3.65 \times n \times \frac{Q^{0.5}}{H^{0.75}}
\]

Although this area is also covered by Kaplan turbines, the small hydropower plants were early equipped with Francis turbines. Even when they produced energy their hydraulic efficiency was low in most cases. Hydraulic design for such specific speed of Francis turbine must be very sophisticated which was not possible that time.
In view of produced energy the small hydropower plant is a large (and expensive) construction thus the potential way of modernization must be considered very carefully. In general, there are two options how to approach to the modernization of small hydropower plant. The first option is to change the Francis turbine for the Kaplan (or propeller) turbine. This solution can be quite extensive in relation to construction work and amount of the parts that are needed to be changed. It is also necessary to consider the time requirements and investment return for such reconstruction as well as to check the cavitation limits of hydropower plant very properly.

The second option of modernization is to retain as much original parts of the turbine as possible and (at best) to replace a runner only for a newly designed one. Even this way of modernization can bring efficiency increase by up to 5 % and performance by up to 30 % if compared with original turbine subjected to the same cavitation guarantees. Higher annual production can be also supported by designing of the runner for maximum possible discharge at given cavitation limit. It is obvious that the hydraulic design of that kind of high specific speed Francis turbine must respect series of hydraulic and geometrical restrictions therefore it can be suitable to find the final hydraulic shape of the blade by using of an optimization method.

2. Hydraulic Design

An optimization method serves to optimize certain properties of a system which are described by parameters. In most technical applications these methods are used for minimization of an objective function (described below) which can be accompanied by some constraints. It is important to note that the results that are obtained by using of an optimization depend strongly on the selected optimization method and on an objective function. In water turbine design there are lots of parts that can be optimized (e.g. Skotak [5], [6]). The purpose of this paper is to find hydraulic shape of the high specific speed runner in terms of high hydraulic efficiency and good cavitation behavior.

2.1. Computational Model

Unlike the “common” hydraulic design of the water turbines which takes into account the design of all parts of the turbine such as spiral case, stay and guide vanes, runner and draft tube, the approach to the hydraulic design of this turbine was slightly different.

Previous project with Kaplan turbine with similar specific speed was considered to be the base geometry. The next goal of hydraulic design was to use all main parts of this turbine and retain them for design of a totally new Francis type runner (see Fig. 2). Some parts had to be modified, added or removed from the turbine, of course, but all of this gave a good opportunity to simulate real conditions of refurbishment. Meaning that the main dimensions and properties such as distributor height, maximum possible runner diameter etc. were already given and determined by existing Kaplan turbine's parts. All of that decreased the degrees of freedom during the design; on the other hand, there was a chance to compare the main parameters of the different types of turbines working at similar $n_s$.  

Figure 1. High specific speed area of Francis turbines.
2.2. Optimization

The runner blade and the runner meridional shape were optimized regarding to their hydraulic shape. In order to find suitable shape of the runner blade 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. 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.

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. Finding of extreme is done using modification of simplex. Great advantage of this method is very fast convergence.

2.3. Parameterization of the Blade

Blade was described by several parameters using commercial software ANSYS BladeGen v.14. The number of the parameters depends on which optimization method is used.

The first method used the user-defined equation for the span-wise distribution of the runner shape. It means that only the blade shape on the hub and on the shroud of the runner was directly defined. Total count of the parameters was 26. This method was mainly coupled with the genetic optimization algorithm regarding to less count of the parameters. For the second method the equation was not
considered. Instead of that, the design using the count of streamlines between the hub and the shroud was applied. Total count of the streamlines was 13 whereas the 4 of them were directly changeable and the rest was interpolated. Total count of the parameters was 36 and this method was mainly used for the final shape optimization using the simplex method. Bezier curves served for the definition of the blade shape.

2.4. CFD Model for Optimization
The blade profile created in BladeGen 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 v.14 and it consisted of three main parts (see Fig. 3). 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 at the runner outlet was the third part. 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 320 mm and it was analyzed by k-epsilon steady state analysis which was the part of the optimization cycle.

2.5. Optimization cycle
The optimization was controlled by the in-house software created in VisualStudio.NET software. This utility controlled all optimization steps according to Fig. 4. 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. For this case, one cycle of the optimization lasted about 15 minutes. The runner blade was mostly optimized regarding to optimum operation point and maximum power output of the turbine.

The method of genetic algorithm (DE) with the population of 10 individuals was applied in phase one and around 700 blade modifications were calculated. After that, the final “brushing up” of the geometry was done using Simplex Nelder-Mead algorithm with next 300 variants in phase two.

Note that the objective function is very important for optimization. It can affect the results of the optimization significantly as its value is a casting vote in which way the blade modification will proceed.

Equation (2) defines the objective function which was considered for the blade optimization. It consists of several terms \( f \) and their particular weighed factors \( w \) and they are finally summarized 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. (Refer to Obrovsky [1] for more information about objective function and its terms).

Figure 4. The optimization scheme.
3. Verification in Additional CFD Models

3.1. Cavitation Analysis
The same computational model taken from optimization process was used for prediction of cavitation behavior. Two-phase steady state model was chosen for the calculation. The cavitation analysis was done for the optimum point and the maximum power output of the turbine.

3.2. Unit analysis
The model of the entire turbine was used for the verification of the optimized runner. This model allows creating the preliminary hill chart of the turbine unit. The CFD model consisted of three main components (see Fig. 5). The spiral case with 12 stay vanes and 24 guide vanes; the runner (which was taken from the optimization procedure) and the draft tube. For this configuration the CFD model of turbine was comprised of 2850k elements.

About five operational points were calculated for each guide vane opening. Altogether six guide vane openings were computed to check the whole operational area of the turbine.

4. Model tests
Turbine parameters, predicted by CFD model, are necessary to be checked by physical model (see Fig. 6). The turbine model was constructed in CKD Blansko Engineering, a.s. and tested in its hydraulic laboratory in accordance with IEC 60193. Model tests were performed to check the turbine behavior and to obtain the turbine characteristics such as performance, cavitation and other important data (pressure pulsations, guide vane torque etc.).

Two variants of the turbine runner were tested; the first runner with 13 blades and the second one with 11 blades. The number of the blades was the only difference between each other whereas the blade profile was the same for both and was given by the CFD optimization mentioned above. The model tests were carried out at the rotational speed of 1000 – 1100 rpm and the performance characteristics were recalculated on Reynolds number $Re = 7 \times 10^6$.

The assembly of the spiral case with the 13-blade runner viewed from the draft tube cone in the hydraulic laboratory is seen in Fig. 7.
5. Conclusions
The high specific speed Francis turbine design has been carried out with the following development work flow:

1. Automatic optimization of the runner blade by using in-house optimization software coupled with commercial CFD software;
2. CFD analysis due to the cavitation behavior prediction;
3. CFD analysis of the entire turbine for the preliminary verification;
4. Testing of the physical model of the turbine in hydraulic laboratory of CKD Blansko Engineering, a.s. for the final verification.

The model testing in hydraulic laboratory confirmed the CFD optimization via the in-house software to be a powerful and useful tool for turbine design. It has to be mentioned here that the position of the best efficiency point (BEP) of the turbine with the 13-blade runner is not identical if compared with the second 11-blade one but the efficiency value of the BEP is approximately equal for both the runners. The BEP for the turbine with 11-blade runner is shifted towards lower heads and higher discharges which is mainly given by bigger blade channel clearance of the 11-blade runner. The 13-blade runner is thus more suitable for lower discharges (see Fig. 8) and lower Thoma number (see Fig. 9). The 9-blade runner was analyzed by CFD only and you can observe even greater shift towards higher discharges and also lower value of efficiency in the BEP if compared with 13-blade and 11-blade runner (see Fig. 8). Simultaneously the cavitation behavior of the 9-blade runner is worst in comparison with all of runners and cavitation curve is shifted towards higher Thoma number and lower efficiency (see Fig. 9).

Relative efficiency charts in Fig.8 and Fig.9 are created by meaning of CFD 13-blade runner 100%.

![Figure 7. The assembly of the spiral case with the runner.](image)

![Figure 8. Efficiency comparison CFD vs. Model tests (optimum net heads of the turbines).](image)
Unit discharge $Q_{11}$ is given by equation (3):

$$Q_{11} = \frac{Q}{D_{REF} \cdot \sqrt{H}}$$

(3)

Where $Q$ is discharge in $[\text{m}^3\text{s}^{-1}]$, $H$ is net head in $[\text{m}]$, $D_{REF}$ is runner diameter acc. IEC60193 in $[\text{m}]$

Turbine efficiency $\eta$ is given by equation (4):

$$\eta = \frac{T \cdot \omega}{Q \cdot H \cdot \rho \cdot g}$$

(4)

Where $T$ is runner torque in $[\text{Nm}]$, $\omega$ is angular velocity in $[\text{rads}^{-1}]$, $\rho$ is water density in $[\text{kgm}^{-3}]$, $g$ is gravity acceleration in $[\text{ms}^{-2}]$

![Figure 9. Cavitation coefficient comparison CFD vs. Model tests (optimum of the turbines).](image)

Cavitation coefficient (Thoma number) $\sigma$ is given by equation (5):

$$\sigma = \frac{NPSH}{H}$$

(5)

Where $NPSH$ is Net Positive Suction Head in $[\text{m}]$, $H$ is net head in $[\text{m}]$

As mentioned the cavitation behavior of all the runners is different. The comparison of the 13-blade with the 11-blade runner for similar operation point near maximum discharge is exposed in Fig. 10. The cavitation area at the leading edge of the blades is observed in CFD analysis. Notice the area size of the inlet cavitation which increases with decreasing number of blades. On the other hand the model tests did not confirm the size and intensity of the inlet cavitation.

In comparison with Kaplan turbine working at similar $n$, the newly developed Francis turbine is able to reach the efficiency value at the BEP not worse than 0.4 % (at model). More over the cavitation behavior comparing Thoma number at the BEP is better by approximately 30 % (for 11-blade runner).
Figure 10. Cavitation behaviour CFD vs. Model tests (identical operation points).
Finally, the described procedure of the hydraulic design points out the significance of aesthetic sensibility in case of the blade shape. Besides the main numeric results the visual shape is also important part of the hydraulic design. Each new developed blade shape actually expresses the personality of the designer who is satisfied with the numerical as well as the visual aspect of the final shape.

6. References
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