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Original Paper (Invited)

Development of The New High Specific Speed Fixed Blade Turbine Runner

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Abstract

The paper concerns the description of the step by step development process of the new fixed blade runner called "Mixer" suitable for the uprating of the Francis turbines units installed at the older low head hydropower plants. In the paper the details of hydraulic and mechanical design are presented. Since the rotational speed of the new runner is significantly higher then the rotational speed of the original Francis one, the direct coupling of the turbine to the generator can be applied. The maximum efficiency at prescribed operational point was reached by the geometry optimization of two most important components. In the first step the optimization of the draft tube geometry was carried out. The condition for the draft tube geometry optimization was to design the new geometry of the draft tube within the original bad draft tube shape without any extensive civil works. The runner blade geometry optimization was carried out on the runner coupled with the draft tube domain. The blade geometry of the runner was optimized using automatic direct search optimization procedure. The method used for the objective function minimum search is a kind of the Nelder-Mead simplex method. The objective function concerns efficiency, required net head and cavitation features. After successful hydraulic design the modal and stress analysis was carried out on the prototype scale runner. The static pressure distribution from flow simulation was used as a load condition. The modal analysis in air and in water was carried out and the results were compared. The final runner was manufactured in model scale and it is going to be tested in hydraulic laboratory. Since the turbine with the fixed blade runner does not allow double regulation like in case of full Kaplan turbine, it can be profitably used mainly at power plants with smaller changes of operational conditions or in case with more units installed. The advantages are simple manufacturing, installation and therefore lower expenses and short delivery time for turbine uprating.

Keywords: fixed blade turbine, runner, draft tube, optimization, uprating, efficiency.

1. Introduction

The low head hydropower plants play the important role in the hydro energy market in many countries where the hydro potential of the huge rivers is not sufficient enough. Many of such power plants were built hundred years ago and today are partially out of the operation and they need a lot of maintenance. The upgrading of the units can solve the problems with reliability and insufficient turbine hydraulic performance, like low efficiency and consequently low power. In case of the low head power plants with low power outputs it is problem to find out the profitable solution. It is very important to carry out the upgrading process carefully in order to offer the best solution for the most suitable energy utilization.

2. Description of the original unit

The design of the original Francis turbine unit was based on the knowledge and possibilities of the engineers working hundred years ago. Today in case of the building of the new low head projects the units installed are usually of bulb or pit type turbines. The higher specific speed bulb turbines and straight draft tubes are much more suitable for unit installation. However, in years before 1913 when professor's Viktor Kaplan invented the Kaplan type turbine no other solution then Francis turbine could be applied. The Francis turbines generally have the lower specific speed, which results in necessity of large speed increaser usage in order to keep reasonable dimensions of the generator (Fig. 1). Such originally installed large right angle speed increasers are often causes of many problems, including fatigue cracks of the shafts. The efficiency of the original units is usually dropped down by hydraulically bad and shallow draft tube (Fig. 2). The maximum power is many times limited by maximum usable turbine flow.

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The next main characteristic of the old low head power plants is, that more (from 3 to 5) units are usually installed. The base plate is usually placed on the unstable soil and the extensive civil works can not be implemented according to the powerhouse static.

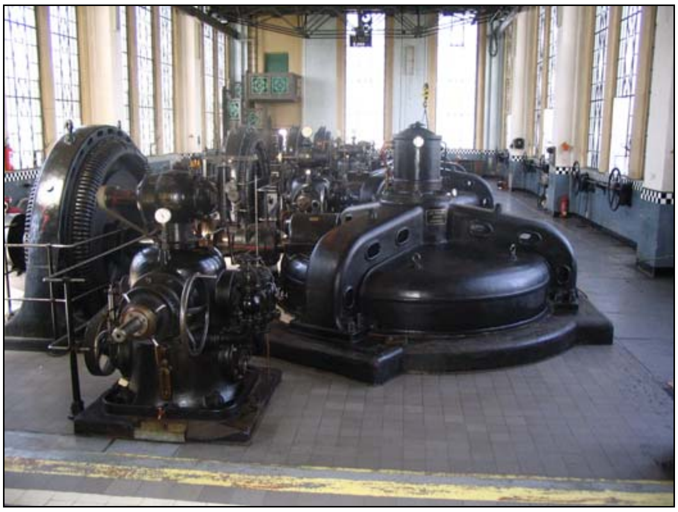


Fig. 1 Powerhouse - 4 original units in line

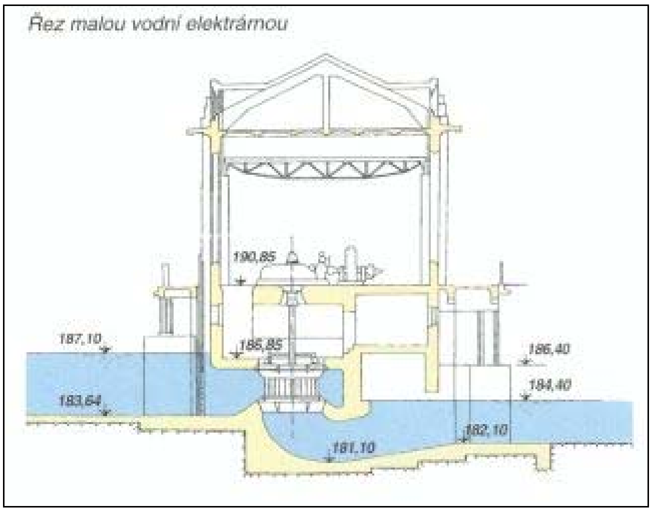


Fig. 2 Cross section of the original unit

3. Concept of the upgrade solution

The approach to the upgrading procedure is based on the assumption, that solution will be simple and profitable as much possible with relatively low expense. The replacement by similar Francis turbines does not bring so much benefit in increasing available power output. Also the necessity of the large speed increaser remains. The pit or bulb turbines can not be installed due to necessity of too extensive civil works, which can not be applied. The standard Kaplan turbine with higher specific speed allows increasing of the efficiency and power output and also increasing of the rotational speed. Obviously they are not optimized for such shallow draft tubes. Finally, in case of installation of more units in one powerhouse the double regulated turbines are not necessary and the expenses are consequently unnecessarily high. The most suitable solution should be the installation of the fixed blade so called propeller turbine runner. The original Francis turbine parts (Fig. 3) like runner, guide and stay vanes with lower and upper ring will be removed and replaced with new ones. The mechanical design of the new turbine is fulfilled as simple as possible (Fig. 4) based on requirement of reliability and maintenance free operation.

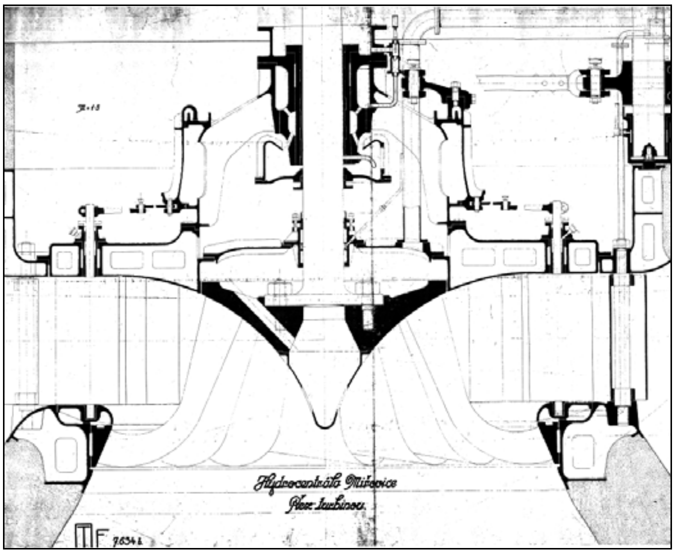


Fig. 3 Original Francis turbine runner

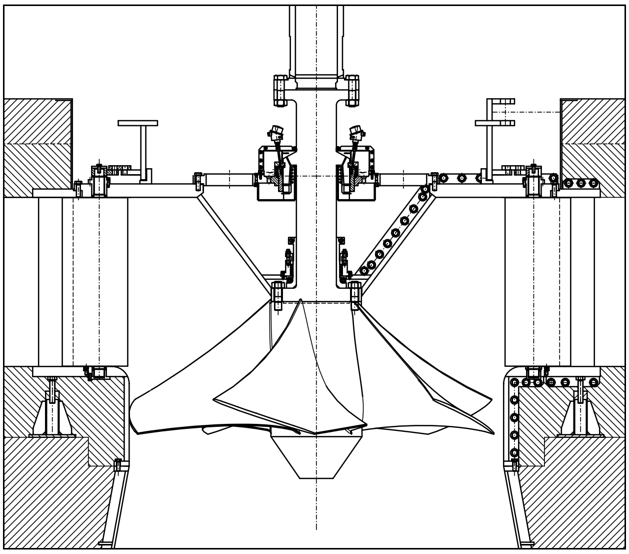


Fig. 4 New propeller runner

The original concrete spiral case dimensions satisfy the requirements for suitable hydraulic solution. The uniform flow at desired flow angle can be expected at the spiral case outlet. Therefore the original spiral case contours can be kept. The elevation of the new runner is little bit higher according to the maximization of the draft tube depth (Fig. 5). The cavitation conditions are afterwards little worse, but the cavitation safety factor is still sufficient enough. As was mentioned above, the draft tube geometry of the original turbine was not well designed. The effect of the bad draft tube performance is growing with the increasing turbine discharge. The new draft tube is built-in into the original draft tube outer contours without any necessity of demolition civil works.

The higher rotational speed of the new runner allows direct turbine-generator coupling. The size and consequently price of the new generator is becoming lower. The solution with direct coupling is sketched in Fig.6.

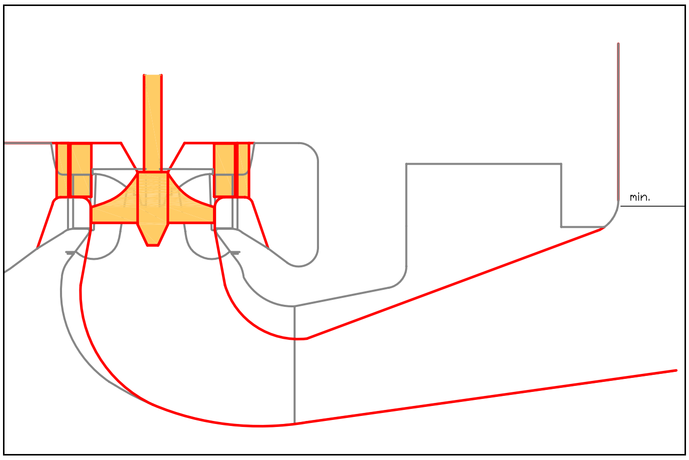


Fig. 5 Original and new geometry contours

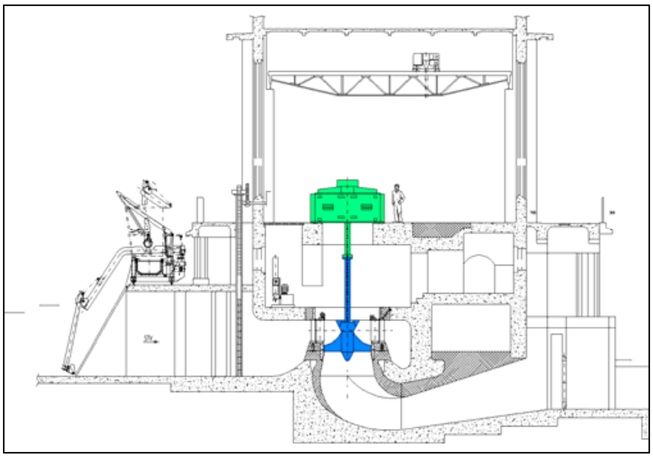
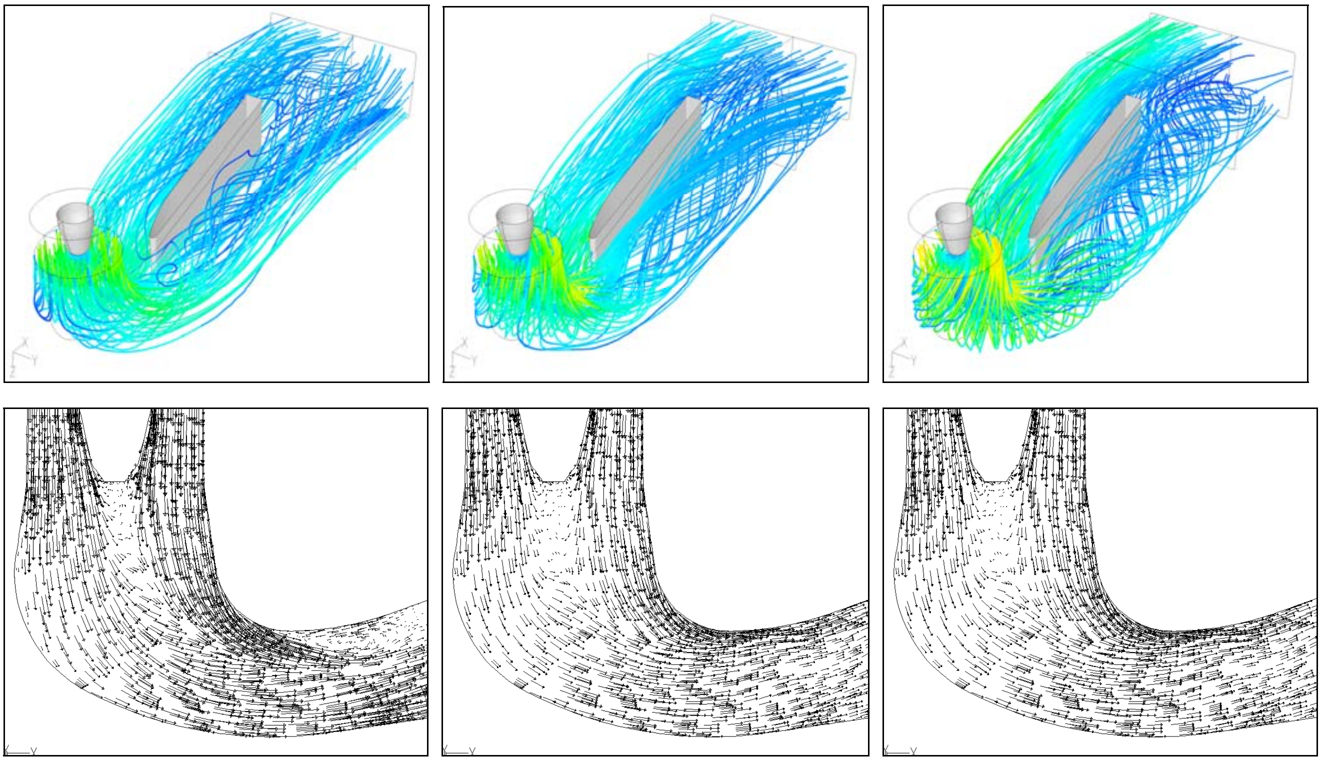


Fig. 6 Direct coupling of turbine-generator

4. Optimization of the draft tube geometry

In case of low head hydro power plant the draft tube is one of the most important parts of a turbine. The performance of the draft tube is influenced by draft tube geometry and by the inlet velocity distribution. Although the runner was not yet designed at this stage of the hydraulic design it was necessary to optimize draft tube geometry and inlet conditions as well. The commercial CFD software Fluent was used for purposes of the flow in the draft tube analysis. The manual optimization process started with usual velocity distribution known from Kaplan turbine runners design [Lit.3, 4]. After receiving of the final draft tube geometry the different velocity distribution at the draft tube inlet was tested and evaluated.



A) S_r=0.00 , k_a=1.0 B) S_r=0.15 , k_a=1.2 C) S_r=0.30 , k_a=1.4

Fig. 7 Flow in the draft tube for different inlet conditions

The inlet boundary condition was prescribed by axial and tangential velocity components, the linear velocity profiles from hub to outer ring are considered including boundary layer profiles. The axial velocity distribution is expressed by coefficient km defined as ratio of axial velocities near outer ring and near hub (Eq.1). Tangential velocity distribution is well described by known expression for swirl intensity coefficient (Eq.2). The draft tube characteristic can be evaluated and it is sketched in Fig.8. The flow for selected inlet conditions is in Fig. 7. At low swirl condition (A) the separation downstream of the elbow causes additional losses. On the other hand the highly swirling flow (C) evokes internal frictional losses. The flow at point (B) is very close to the optimal one.

k_m axial velocity ratio coefficient $$k_a = \frac{v_{a1}}{v_{a0}} \quad (1)$$

S_r swirl intensity coefficient $$S_r = \frac{1}{R_1 . Q} \iint_S \frac{r . v_t}{v_a} dQ \quad (2)$$

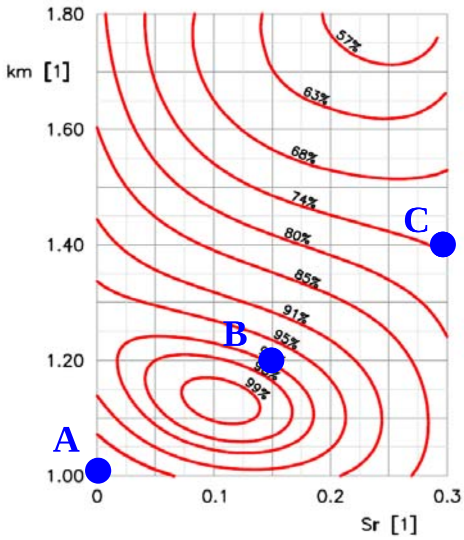


Fig. 8 Draft tube characteristic

5. Optimization of the runner blades

After successful design of the draft tube geometry the automatic optimization process of the shape of the runner blades was carried out. The in house software implementing direct search Nelder-Mead simplex method was used [Lit.1]. The blade geometry is described by 20 independent parameters. The mathematical algorithm searches the local minima of the objective function concerning the quantities evaluated from CFD results. The previous experience from the Kaplan blade shape optimization [Lit.2] shows the importance of the output runner velocity tuning for the certain draft tube. In the above mentioned literature the simulation of the flow in the separated runner was applied for the predetermined velocity distribution achievement. Such approach required additional term in the objective function and this made the solution of the optimization process more complicated. This term can be excluded by coupling of the runner with the draft tube in one computation. The achievement of the velocity distribution is not the portion of the optimization process. The objective function is then composed from efficiency term, head term and cavitation term only (Eq.3).

$$f_O = w_E(1-\eta) + w_H\left(\frac{|H - H_R|}{H_R}\right) + w_K\left(\sum_{p_S < p_V}(p_V - p_S)\right) \Rightarrow \min imum \quad (3)$$

η actual efficiency
H actual head
H_R required head
p actual static pressure on blade
p_V vapor pressure
w individual weighting factors

The automatic optimization process optimizes blade geometry in order to reach the maximal efficiency at required operational point. In the CFD simulation the flow is prescribed by inlet velocity value while the actual head is evaluated from total pressure difference. The history of the objective function value is in Fig.9 and corresponding history of the actual head end actual efficiency is in Fig,10. Consequently the simulation of the runner coupled with the draft tube needs larger computational grid and more computational time then in case of individual runner. Economic structural grid of the entire computational domain consists from about 300,000 cells. The CPU time for converged solution of the one geometry in Fluent 6.3 was approximately 25 minutes. For final optimized geometry for about 400 steps were necessary.

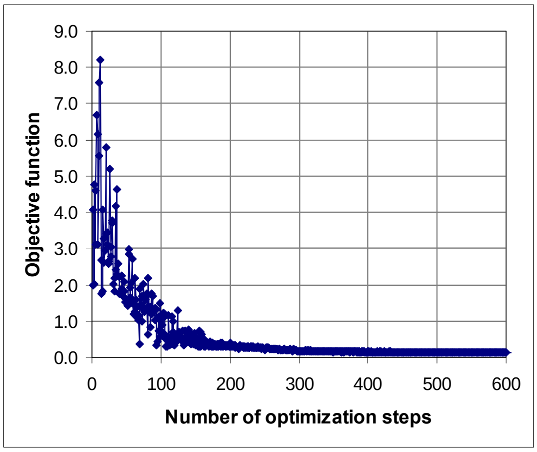


Fig. 9 History of the objective function

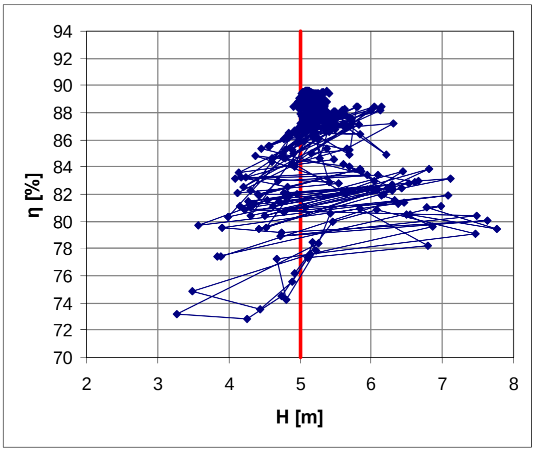


Fig. 10 History of the efficiency and actual head

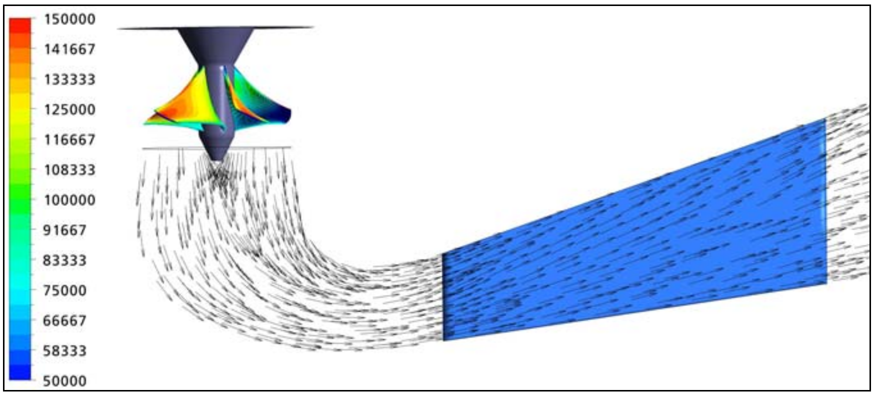


Fig. 11 Flow at the best efficiency point after optimization

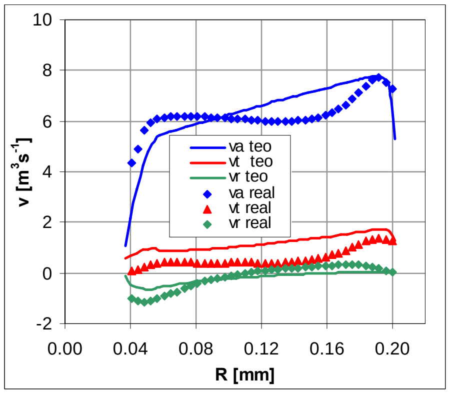


Fig. 12 Velocity components

The result of the draft tube optimization process was not only the draft tube geometry but also the optimal distribution of the inlet velocities. The resulted velocity distribution was not used in the optimization process, however similar velocities were expected downstream of the optimized runner. The comparison of theoretical and real velocities depending on radius is in Fig.12. The velocities from individual draft tube corresponding to point B from characteristic (theoretical) are marked with lines, velocities downstream of the optimized runner coupled with the draft tube (real) are marked with points. Lower swirl ratio and lower axial velocity ratio of the real velocities agree well with draft tube optimum at draft tube characteristic (Fig.7). Of course the real velocities profiles are not linear as in case of theoretical profile. At the turbine best efficiency point the flow in the draft tube is without any separation.

The optimized runner called “Mixer” (Fig.13) was finally analyzed as coupled with draft and spiral case including guide vanes. The total entire turbine performance for 5 guide vane openings was modeled and analyzed in CFX Ansys 11. The SST turbulence model was used. The CFD based turbine hill chart was evaluated (Fig.14). According to the best efficiency point the turbine specific speed n_q=250 (n_S=900) was evaluated.

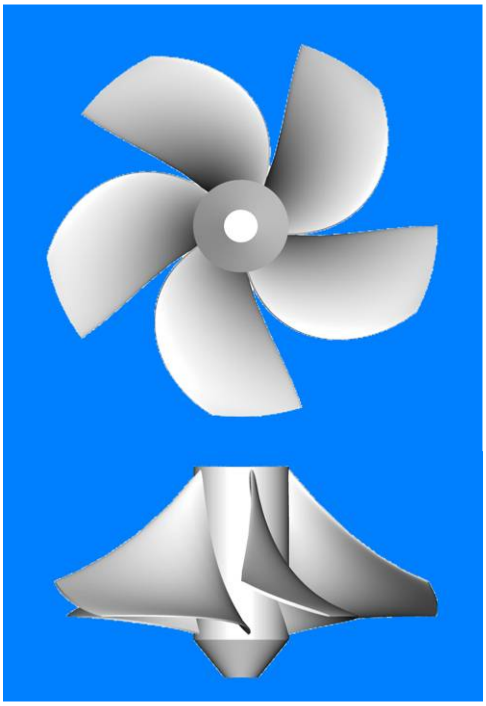


Fig. 13 “Mixer” runner geometry

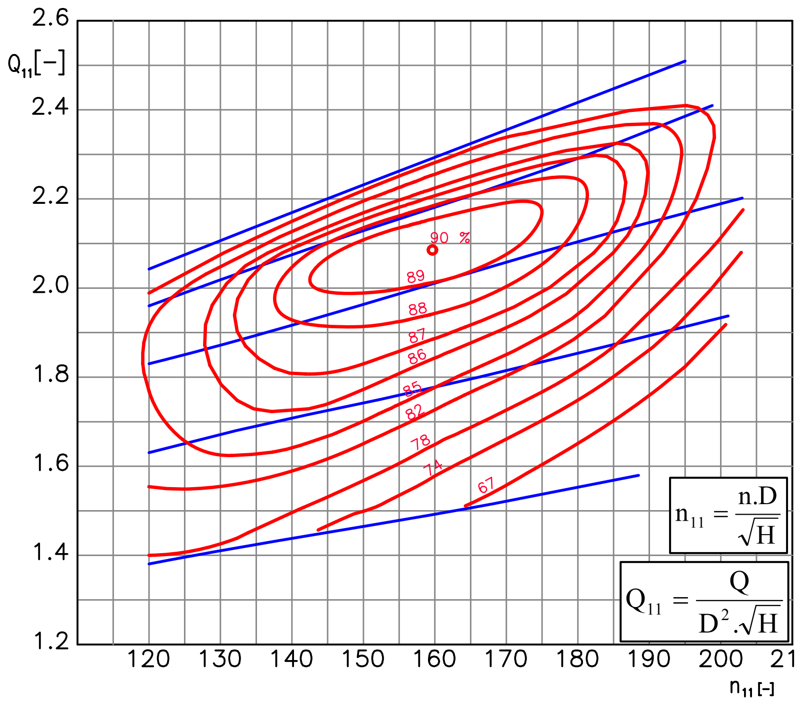


Fig. 14 Turbine hill chart from CFD simulation

6. Modal analysis of the runner

In the prototype real size runner the dangerous cracks may occur when the exciting frequency is in resonance with one of the natural frequencies. Therefore the runner with diameter 2250mm and with rotational speed 120 rpm was checked for possible resonance problems. As a most dangerous exciting frequency on the runner the blade passing frequency was evaluated (Eq.4).

$$f_E = z_S \cdot f_n = 24 \cdot 2 = 48Hz \qquad (4)$$

z_S number of guide vanes
f_n runner rotational frequency

The natural frequencies of the real prototype runner was computed without (at air) and with (at water) influence of surrounding environment. The Ansys software was used. The computation at air is fast and simple, but the results can not be often simply corrected for water. For the new type of the runner the full computation with water surrounding environment was carried out. The ratio between natural frequencies at air and at water is expressed by coefficient k_f. This value depends on mode number and varies in a range 0,55 ÷ 0,80 (Fig.15). The nearest natural frequency to the exciting one is that of the 8th mode (Fig.16), however even it is sufficiently far from possible dangerous resonance. The displacements and modal stress of the 8th mode are in Fig.17.

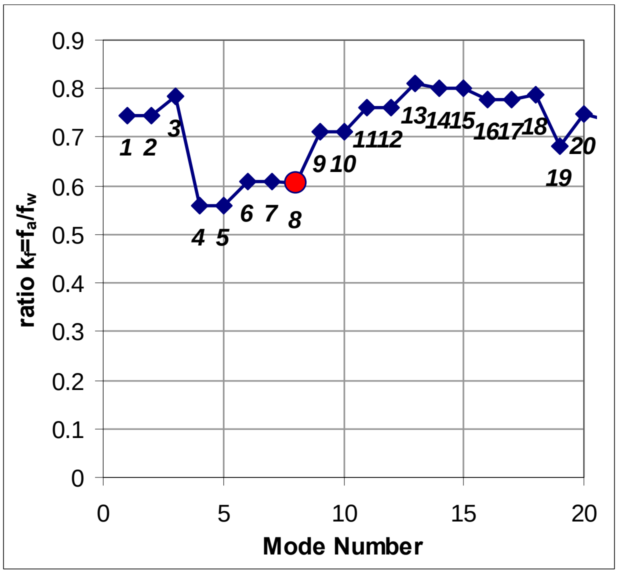


Fig. 15 Ratio of air/water frequency

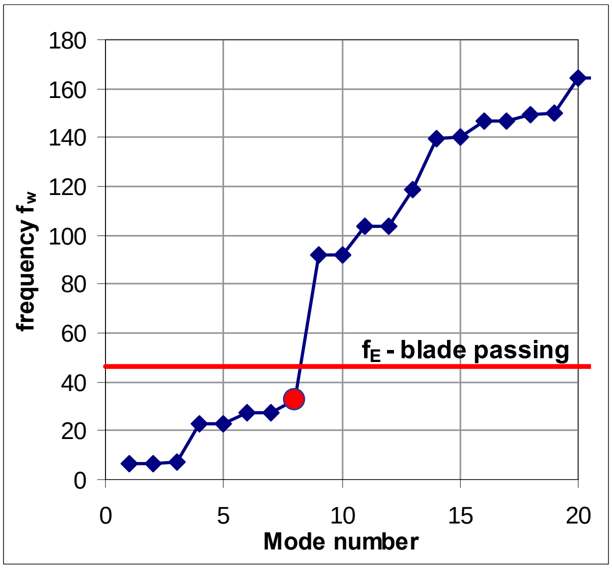


Fig. 16 Natural and exciting frequency

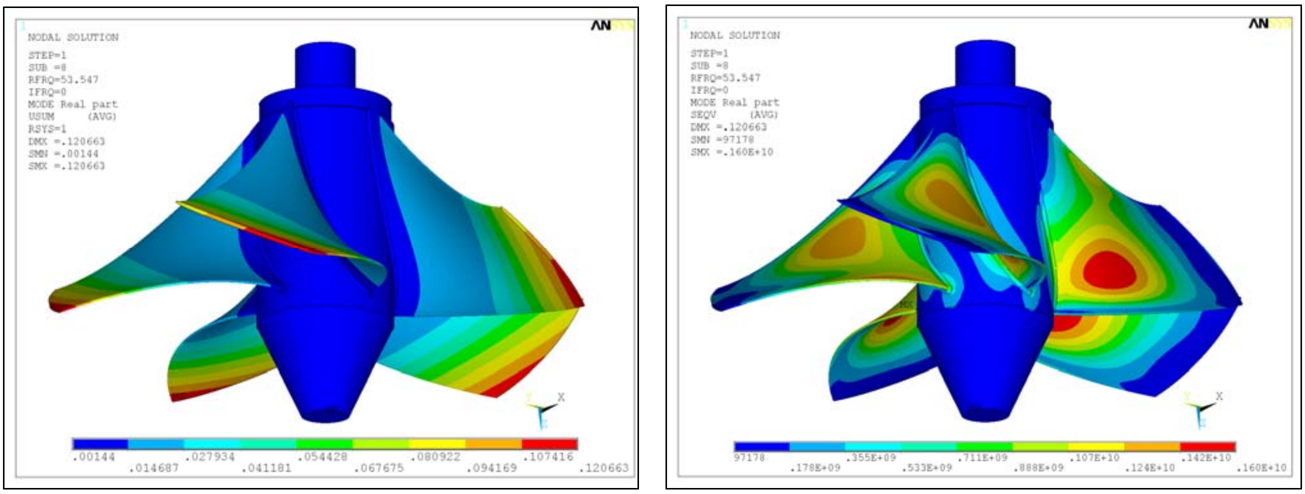


Fig. 17 Modal displacement and modal stress at mode No.8

7. Stress analysis of the runner

During the normal operation the runner is loaded by the hydraulic load and by the centrifugal forces. The hydraulic load effect prevails at normal operation, while the centrifugal forces prevail at the runaway speed. There are some simplifications known from previous methodology of Kaplan turbine runners FEM computation. One of the simplifying assumptions coming from history is to substitute the real pressure load by constant static pressure load. The value of the static pressure is estimated according to the assumption of equivalent torque on the shaft. However, nowadays methods allow transformation of the real pressure load known from the CFD computation to the FEM solver. This is so called one directional fluid structure interaction (FSI). The comparison between those two approaches was carried out and the results are in Fig.18. The analysis shows, that results of both computations are very similar. Both, the deformation and stress are higher in case of real pressure distribution application. The maximal value of von-Misses stress for real pressure load application is 28 MPa, while in the case of constant pressure it is 24 MPa.

The next analysis has been done for runaway speed operation. At this operation the torque on the shaft is zero due to the almost balanced static pressure distribution on the runner blade. The static pressure load was neglected in this computation. Therefore the centrifugal forces only can cause the displacements. The direction of displacement is opposite to the displacement from normal load (Fig.19). The highest analyzed stress is about 60 MPa, which is still safety enough even in case of carbon steel runner blade material.

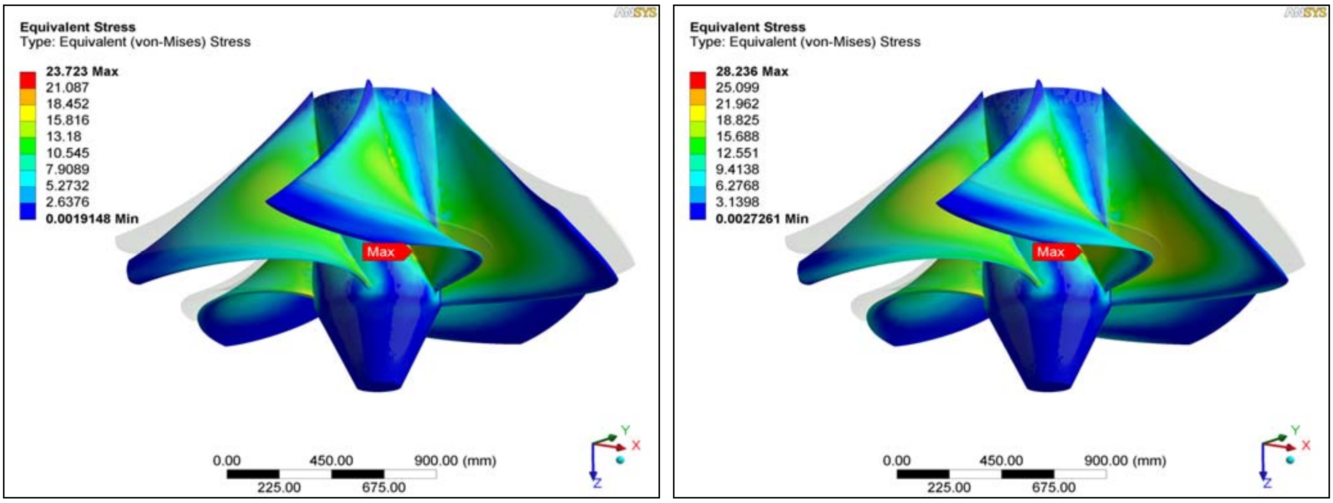


Fig. 18 Comparison of constant pressure load (left) and real pressure load (right)

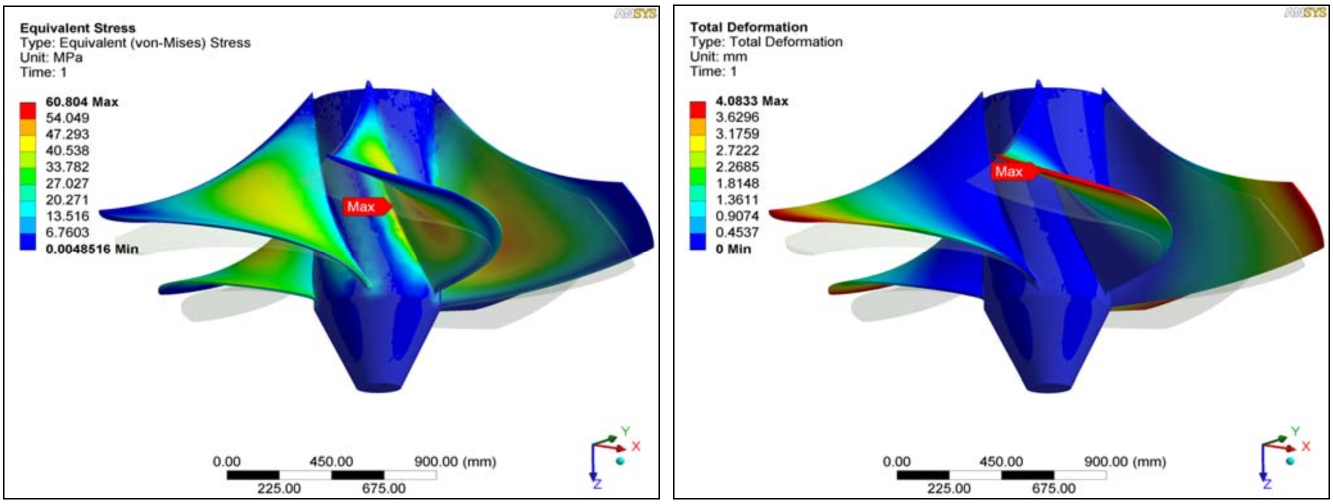


Fig. 19 Von-Misses stress (left) and total deformation (right) at runaway speed operation

8. Model tests

The model runner with diameter 350 mm has been manufactured for model testing in hydraulic laboratory. The manufacturing of the model scale runner was based on machining from one piece of bronze material (Fig.20), which differs from the prototype production welding technology. In the finalizing of this paper the model has been installed in the hydraulic laboratory and has been prepared for testing. The results of tests confirmed predicted hydraulic parameters.

Fig. 20 CNC machining of the model runner

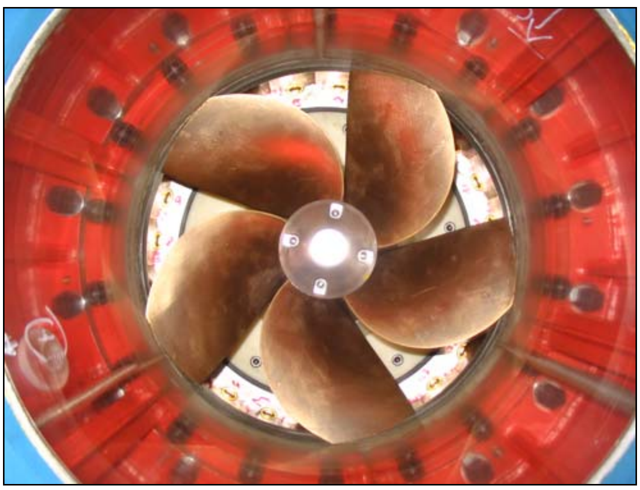


Fig. 21 Installed model in laboratory

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Conclusion

The new fixed blade runner suitable for the old low head hydropower plants uprating was successfully developed. The modification of the original draft tube was designed in order to improve entire turbine performance. The high specific speed "Mixer" runner allows direct coupling of runner with the generator. The mechanical design of the turbine is fulfilled as simple as possible and minimizes turbine maintenance. The propeller high specific speed turbine installation to the old low head hydropower plants allows shorter payback period in case of rehabilitation projects..

References

[1] Nelder, J.A., Mead, R.: A simplex method for function minimization, Computer Journal 7, pp. 308-313, 1965

[2] Skotak, A., Obrovsky, J.: Shape Optimization of a Kaplan Turbine Blade, paper 233, 23rd IAHR Symposium on Hydraulic Machinery and Systems, Yokohama, Japan, 2006

[3] Skotak, A., Obrovsky, J.: Low swirl flow separation in a Kaplan turbine draft tube, 2nd IAHR International Meeting of the WorkGroup on Cavitation and Dynamic Problems, Timisora, Romania, 2007

[4] Skotak, A., Obrovsky, J.: Analysis of the flow in the water turbine draft tube in Fluent and CFX, 25th CADFEM Users' Meeting ,Dresden , Germany, 2007

[5] Skotak, A., Obrovsky, J., STEGNER, P., OBROVSKY, J.: The New Approach to the Old Francis Units Uprating, HydroVision 2008, Sacramento, California, USA, 2008