A Case Study of Upgrading of 62.5MW Pelton Turbine

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ABSTRACT

Upgrading and refurbishing of existing Pelton turbines becomes more beneficial. ČKD Blansko Engineering, a.s. applies both new methods of numerical flow modelling and model testing at the test stand as a part of upgrading and refurbishment process of Pelton turbines. Application of this process results in improvements of basic hydraulic characteristics and extension of live time of refurbished turbines. This paper presents a study of upgrading and refurbishing procedure of four nozzles 62.5 MW vertical Pelton turbine for water power plant Tillari – India- see Fig. 1. Mentioned power plant was commissioned in the year 1986. The upgrading procedure is prepared in the present time. This Pelton turbine unit operates at a speed 500 rpm and net head of 623.7÷625.36 m. Some aspects of that rehabilitation process are presented.

With refurbished runner and nozzles the rated capacity will be increased up to 68.2 MW at net head of 624.8 m. Main goals of the Tillari HPP refurbishment are power and efficiency improvement. The power of the new runner increases by 9 % and efficiency increases by 1.4%. The power and efficiency improvement of the mentioned turbine were reached with application of the new runner, new design of straight flow nozzle tips, straight nozzles strike enlargement and modification of turbine housing.

The refurbishment process started with numerical flow simulations. These simulations through main parts of refurbished turbine were provided and some results of these simulations are presented in this paper. Flow simulations over Pelton turbine bucket were performed with software specially developed in ČKD Blansko Engineering, a.s. The commercial CFD software Fluent was used for the flow simulations through the other parts of rehabilitated turbine. Finite element stresses analysis of the runner and some components of straight flow nozzle were used as well. Results of this analysis as a basic data for the life analysis and mechanical behaviour were given.

INTRODUCTION

The Tillari HPP has been in operating without serious problems for 15 years. On the other hand the live time of the Pelton runner used at the Tillari HPP is over and it is necessary to change the Pelton runner and some other parts at the HPP. Very important point of the rehabilitation process was the costs minimization. The sphere of supply and hydraulic profile changes were focused on a new runner and modification of the turbine housing and nozzle tips. Hydraulic design all of the mentioned parts was tested at hydraulic laboratory. Laboratory tests were conducted during year of 2000. The Tillari Pelton turbine is four nozzles vertical machine, however two nozzles will be changed at the turbine only. The other two straight flow nozzles will be repaired from the two old straight flow nozzles. This fact evoked basic request of the customer for the nozzle refurbishment. The request was that basic parts of new and old nozzles should be interchangeable.

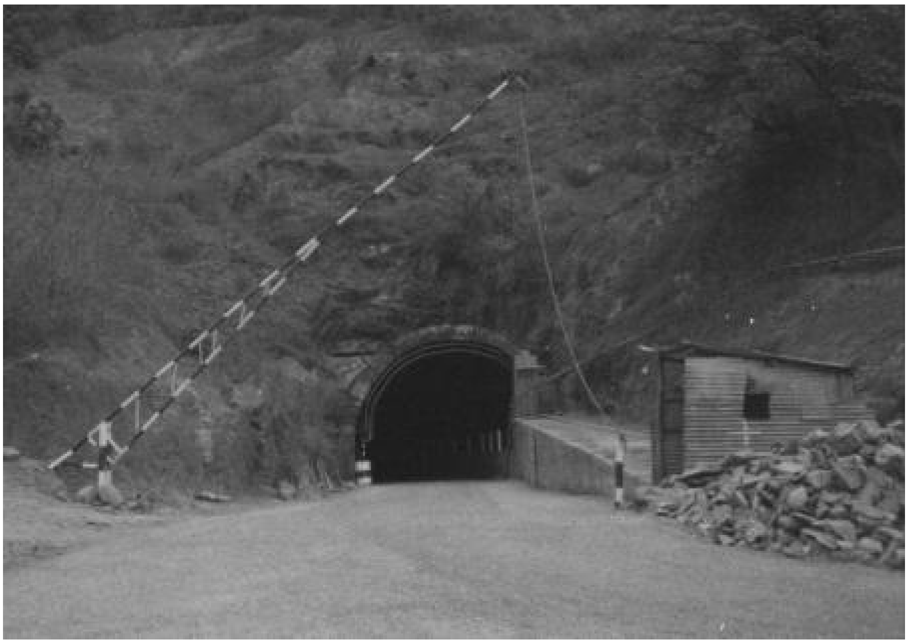

Fig. 1. Tillari HPP – entry tunnel 1986

HYDRAULIC DESIGN

The Tillari HPP operates as a peak-load power station therefore the major preference for upgrading process was an afford to achieve of maximum electricity power. Figure 2. shows efficiency and power for Tillari HPP. The efficiency and power were measured on a model at the test stand.

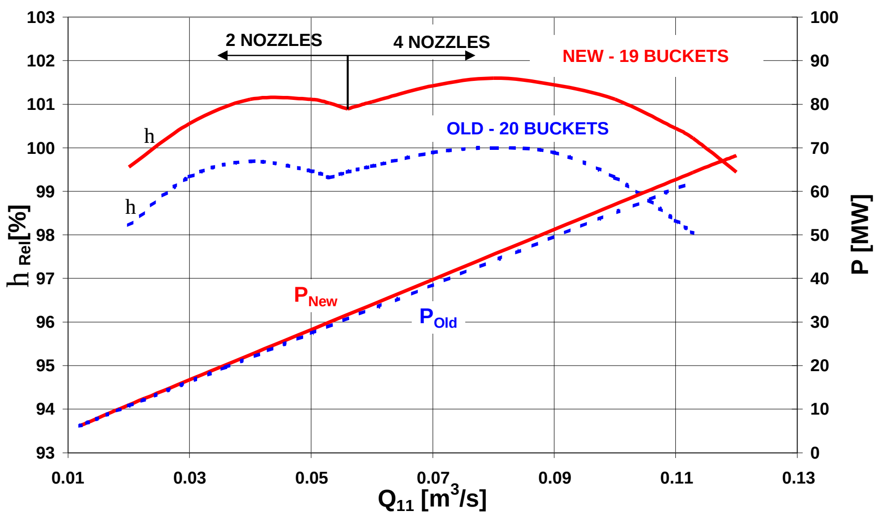

Fig. 2. Expected relative efficiency and power for Tillari HPP

The specific speed of the turbine had to be changed to achieve the power increase. The specific speed of the existing turbine runner is 22.8 [min^{-1}] and specific speed for the new turbine runner is 23.9 [min^{-1}]. Dimensions of Pelton turbine buckets of the runner with higher specific speed are different from those of the existing runner buckets. The pitch diameter of the new and the old turbine runner remained the same. Hydraulic design of the existing runner was provided more than 20 years ago. Since this time the hydraulic design of Pelton runners have been improved. Up to date experiences were applied at the hydraulic design of the new turbine runner. Hydraulic design of the new bucket shape allowed make the new runner for mentioned HPP with 19 bucket – see Fig. 3. The existing runner was designed with 20 bucket. The change of bucket number was advantage for runner production technology.

Turbine Housing

The specific speed and dimensions of the runner were modified during upgrading process. A comparison between the geometry of the Tillari runner and turbine housing for old and new design is in the Fig.4. A part of the turbine housing around turbine shaft interfere with the new runner bucket position. New runner was tested at the test stand in the existing turbine housing. Some small part of existing turbine housing near the turbine runner had to be cut out for this model test. Water at the runner outlet flows to sealing gap causing an efficiency decrease measured by the model test. These problems were eliminated by turbine housing optimization. Several turbine housings modifications were tested at the test stand in the hydraulic laboratory. According these model tests the optimal turbine housing shape was determined. Minimization of turbine housing re-shape works, turbine efficiency and power maximization and technology of the turbine housing modifications were factors choosing the optimized turbine housing design. Result of the optimization procedure is shown in the Fig. 4.

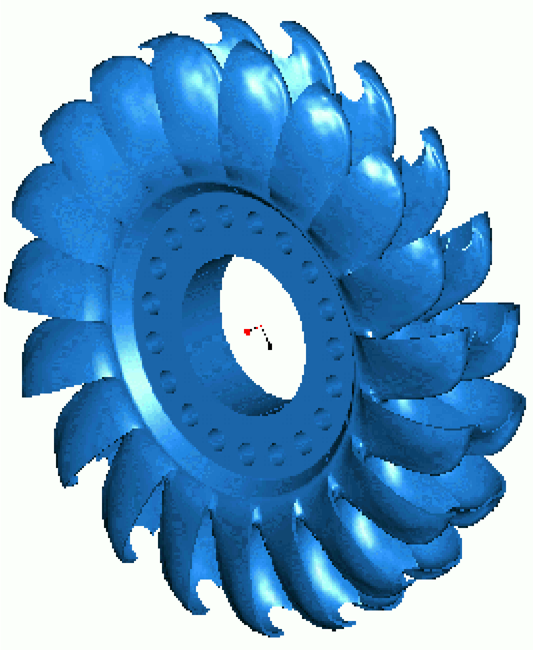

Fig. 3. Runner CAD model – new design

The last step of the turbine housing investigation was optimization of the sealing gap. The sealing gap is important for natural air admission of the turbine housing and for prevention of water flow to the turbine bearing. The position and size of sealing gap was optimized according turbine efficiency. It was found that there exists an optimal sealing gap for whole range of Tillari HPP. This optimized sealing gap was tested at the model and it will be used at the prototype.

Meridian section of turbine housing and Pelton turbine runners were modified during rehabilitation process. Some parts of existing turbine housing have to be removed for hydraulic design improvement.

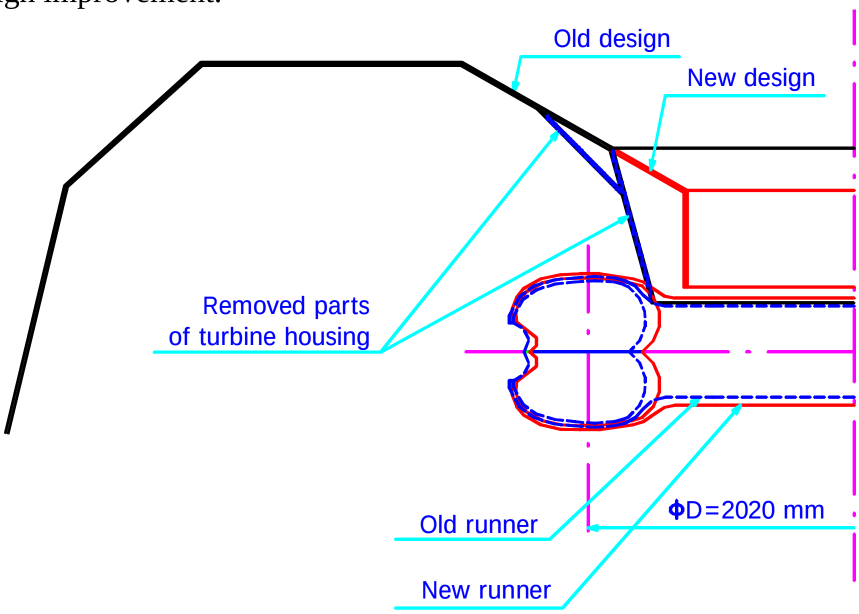

Fig. 4. Turbine housing modifications and Pelton runner dimensions

Turbine Runner

Based on flow simulation over the bucket the hydraulic design of the turbine runner for the Tillari HPP was carried out.

For mathematic modelling of the flow over the bucket following simplifications were applied:

- Friction forces between water and the bucket were neglected
- Attack between water and the bucket was neglected
- Gravity acceleration was neglected
- Normal acceleration is constant at the line perpendicular to inner surface

The numerical analysis of the flow over a Pelton turbine bucket was realized as a task of moving and geometry changing elements. The simulation of this problem was based on the equation of water motion on the bucket surface. Similar system of the flow simulation over Pelton turbine bucket published Kubota – see [1].

Results of the numerical flow simulation over Pelton turbine are velocity and pressure distribution over the bucket. The velocity distribution at the runner inlet and at the runner outlet was used for the runner efficiency evaluation – see Veselý – [2] and [5].

Hydraulic efficiency of turbine runner is defined by Euler turbine equation (1).

$$h_h = (u_1 \cdot c_{u1} - u_2 \cdot c_{u2})/(gh) \qquad (1)$$

Where: u circumferential velocity,

c_u component of absolute velocity to the circumferential direction.

This equation allows evaluating the hydraulic efficiency of the flow over the bucket based on knowledge of inlet and outlet velocities. Index number 1 means velocities at the inlet and index number 2 means velocities at the bucket outlet.

Numerical Flow Simulation

Mentioned software equipment was used for evaluation of the transfer of energy from water jet to the Pelton turbine bucket. Some differences of the existing and new turbine runner could be seen at efficiency curves – see Fig. 5. Water supply is different due to different number of buckets at solved runners. The number of frame describes time of water flow to the solved bucket.

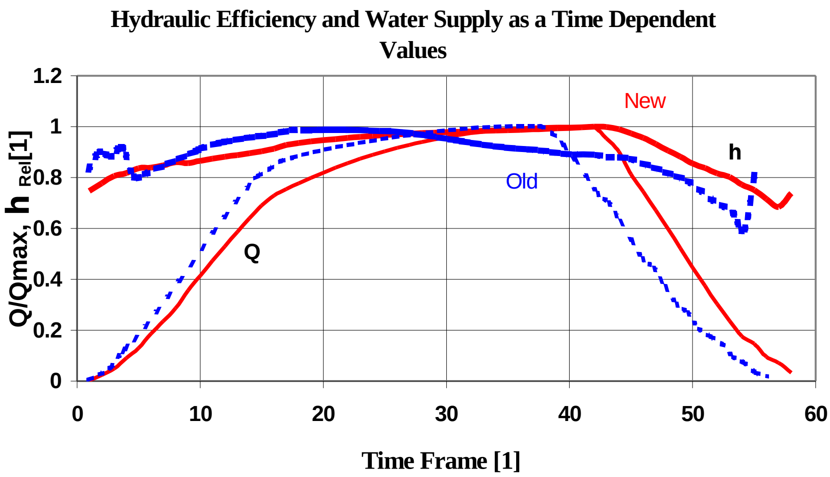

Fig. 5. Hydraulic efficiency for existing and new design

Maximum of hydraulic efficiency for existing design is situated to the time frame 20 and maximum of the discharge is situated to the time frame 37. The disharmony of the maximums of the discharge and efficiency is a source of the existing runner efficiency level – see Fig. 2.

The new bucket was designed in such a way that the maximum of discharge and maximum of efficiency is situated to the same frame number. This fact is very important for maximisation of integral value of hydraulic efficiency. Integral values of the numerically simulated hydraulic efficiency for existing and new design were evaluated. The hydraulic efficiency for the new design was evaluated by 2.0 percents higher than that for the old design. The relative comparison between efficiency of the Pelton turbine – see Fig. 5 and result of numerical simulation are in accordance.

Straight Flow Nozzle

The commercial CFD software Fluent was used for the flow simulations through straight flow nozzle of rehabilitated turbine. CFD flow simulation through straight flow nozzle was realised as a two phases turbulent axisymmetric flow – see Fig.6. The VOF method for the free surface modelling and RNG k-ε turbulence model was used – see lit. [3]. The same system for the free surface and turbulence flow modelling used Muggli in lit. [6].

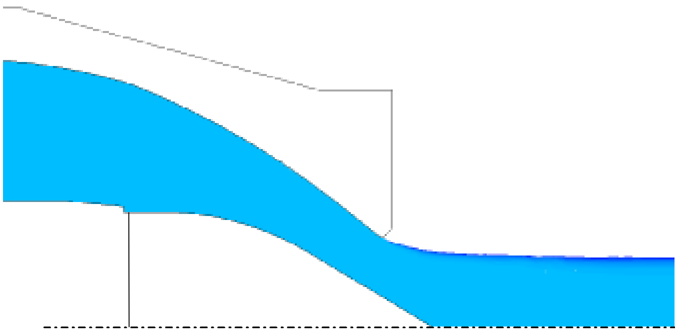

Fig.6. Free surface shape for maximum power

The ratio of the energy at the nozzle inlet to energy at the outlet describes this technical problem from the energy aspect - see equation (2). Efficiency of the nozzle depends on the nozzle and needle shapes and the needle opening. Numerical flow simulation was applied for the straight flow nozzle hydraulic efficiency evaluation.

Efficiency evaluation: $$\eta = 1 - \frac{p_{t1} - p_{t2}}{p_{t1}} \qquad (2)$$

In this equation p_{t1} is the total pressure at the inlet and p_{t2} is the total pressure at the outlet. Results of numerically simulated efficiency are presented in Fig. 7. The figure shows numerically simulated efficiency of Pelton nozzle depending on the unit discharge of the nozzle $\mathbf{Q_{11}}$ - see equation (3).

Nozzle unit discharge: $$Q_{11} = \frac{Q}{D^2 \sqrt{H}} \qquad (3)$$

In this equation Q is discharge of one nozzle, D is nozzle tip diameter and H is net head. Unit nozzle opening is defined as a ratio of the nozzle opening and nozzle diameter.

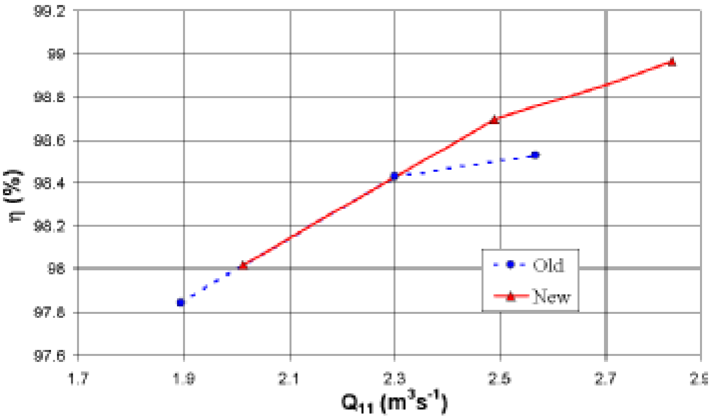

Fig. 7. Nozzle hydraulic efficiency

The new straight flow nozzle design achieves the same hydraulic efficiency as the existing one for the maximum turbine power. It means a maximum unit nozzle discharge Q_{11}=2.2[m^3s^{-1}] (unit nozzle opening z_1=0.6) – see Fig. 7.

Dependence of straight flow nozzle discharge for new and existing design is shown in Fig. 8. There are some differences in the CFD and model tests discharge evaluation. CFD results show higher discharge than that measured at the test stand. On the other hand the relative comparison of

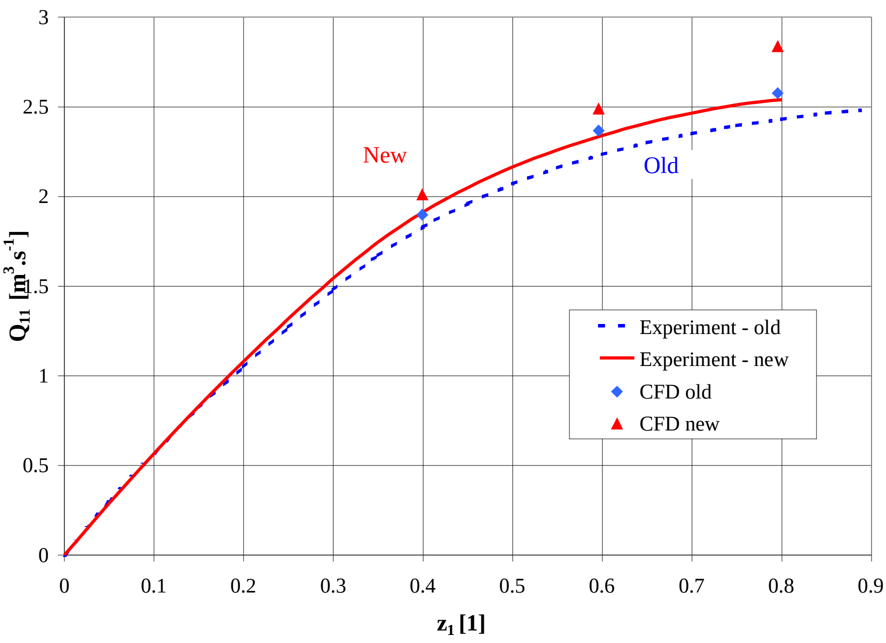

Fig.8. Unit discharge existing and new nozzle

the straight flow nozzle discharge for CFD flow simulation and experiment are in accordance. The turbine efficiency for existing and new straight flow nozzle design was measured at the test stand. Values of the turbine efficiency were in the same levels for both mentioned nozzles tested during the model tests with the new model turbine runner.

A shape of separation of the flow at the nozzle tip was compared with flow separation published in [6]. The flow separation obtained from CFD simulation by Fluent could be characterised as a correct flow separation.

FATIGUE STRENGHT EVALUATION OF THE RUNNER

Pelton runners are structures of a very complex shape and beside other loads (centrifugal forces, thermal loading) they are loaded by pulsating loading due to jet force action. Typical number of load cycles is 10^{10} and more. Therefore the most dangerous mode of Pelton runners failure is material fatigue. Fatigue life assessment of the structure must be based on the knowledge of both material characteristics and dynamic stress. It is well known, that from the fatigue life point of view, the bucket roots are the critical places of the Pelton runner. To achieve acceptable runner fatigue lifetime the shape of those places has to be carefully optimised using CAE tools. Geometry model of the Tillari runner (see Fig.3) was prepared in Pro/Engineer CAD package. Stress levels for different load cases were determined using Cosmos/M finite element program. Finite element mesh with considerable element refinement in the bucket root area is shown in Fig. 9. The magnitude of stress due to

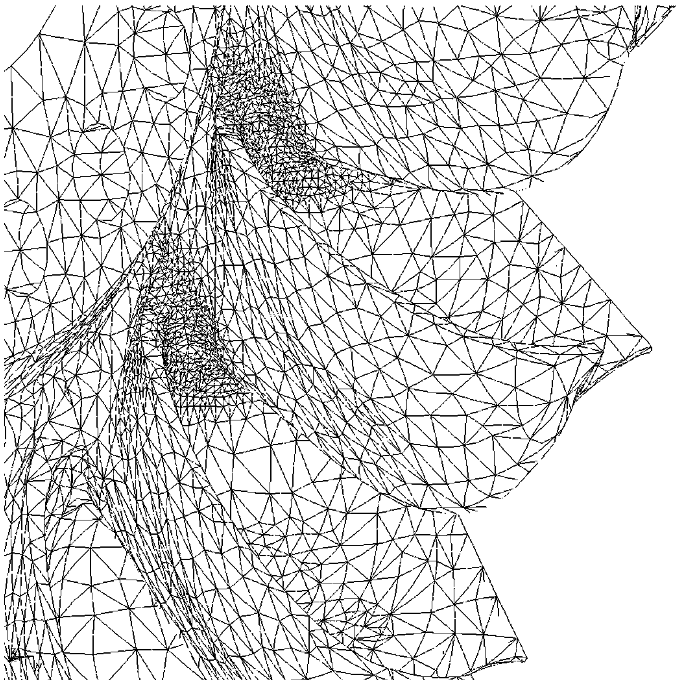

Fig. 9. Finite element mesh of the runner

possible bucket vibration was estimated using procedure described in [7]. Computed stress amplitudes and the Haigh diagram of the runner (see Fig. 10) were utilised for corrosion fatigue life evaluation of the runner. Twelve hours a day full power operation for the whole 20 years lifetime was considered.

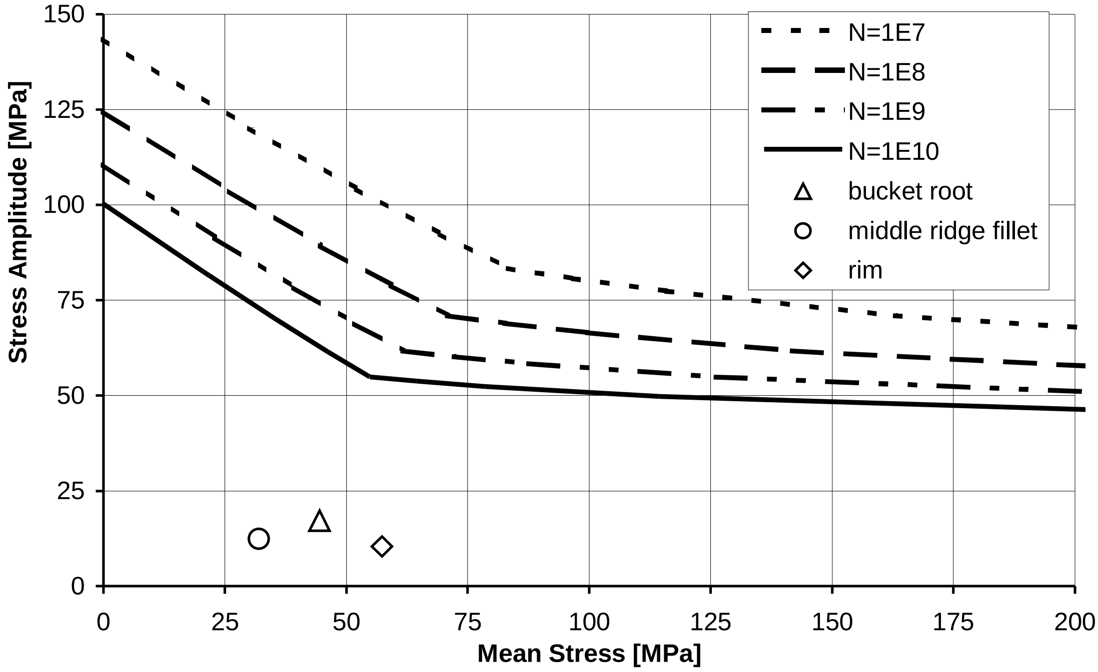

Fig. 10. Haigh diagram

The curves of the Haigh diagram for 10^8 up to 10^{10} load cycles were obtained by extrapolation based on experimental data [8, 9] and corrosion fatigue crack initiation findings [10, 11]. The values of the computed stress and fatigue stress safety factor are shown in table 1.

	Bucket root	Middle ridge fillet	Rim
Mean stress [MPa]	44.5	32.0	57.3
Stress amplitude [MPa]	17.2	12.4	10.4
Fatigue stress safety factor *k* [-]	2.9	3.9	4.3

Table 1 Calculated stresses and Fatigue stress safety factor

The fatigue stress safety factor values $k \geq 2.9$ show high level of the runner resistance against corrosion fatigue crack initiation.

MANUFACTURING PROCESS

A high quality of the manufacturing process is necessary to ensure long lifetime and hydraulic quality of the new runner and straight nozzles.

The Pelton runner for the Tillari HPP is cast of stainless steel CrNi 13.6. The casting process was realised and tested according international standards ASTM A802 and CCH 70-3. To ensure the quality of the runner a lot of tests were performed such visual examination, chemical composition test, Charpy V, dimensional check, ultrasonic examination, and magnetic particle examination. Casting defects found by means of non-destructive tests were grinded, repair welded and checked again. 3D CAD model – see Fig. 3 was a base for casting and manufacturing process.

Fig. 11. Tillari - casted runner during inspection

The straight flow nozzle manufacturing process includes producing and testing all of straight flow nozzle parts. Nevertheless assembly of nozzles, assembly group check, pressure test, leakage test and function test was carried out. The manufacturing process of straight flow nozzles has been finished and nozzles are prepared for shipping.

Fig. 12. Straight flow nozzle during pressure test

CONCLUSION:

Up to date methods of numerical simulation of the fluid flow as well as the structural analysis were applied to refurbish of 62.5 MW Pelton turbine. The application of numerical flow simulation of Pelton turbines is a new step in research and development of Pelton turbines. Using new developed procedures the optimisation of hydraulic design of the Pelton turbine to be refurbished was carried out. The paper presents some comparison of application of numerical flow simulations for existing design, new design and model test of new designed turbine.

- Results of numerical flow simulation of the runner hydraulic efficiency and efficiency of whole model turbine evaluated at the test stand are in accordance.
- Flow simulation through straight flow nozzles is a power full tool for nozzle optimisation
- The increase of turbine power was demonstrated at the ČKD Blansko Engineering Laboratory – vertical Pelton test stand.
- Runner corrosion fatigue lifetime conforms to nowadays requirements.

The numerical simulations of the flow through main parts of Pelton turbine become to be a powerful tool making the Pelton turbines refurbishment process more effectively.

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