Design of a high-specific speed turbine with non-uniform blade cascade

Jakub Stareček¹ *, Pavel Čupr¹, and Miloslav Haluza¹

¹Brno University of Technology, Faculty of Mechanical Engineering, Technická 2896/2, 616 69 Brno, Czech Republic

Abstract. This paper presents the hydraulic design of Kaplan type turbine in fully axial configuration. The turbine model consists of a straight pipe intake, a rib which covers the shaft, five axial guide vanes, a six blade runner and a 10.6°-full angle draft tube. Due to the presence of non-uniform velocity field behind the rib and interaction between stator and rotor, the pressure pulsations occur and cause some excitation forces acting on the runner. To reduce these negative phenomena the six non-uniform blade runner has been designed using computational fluid dynamics modelling (CFD). The runner blades are placed in various axial distances from the guide vanes. This new design should reduce the pressure pulsations caused by interaction between the rib and the runner, which is the main negative effect by the current design. The forces and torque which are acting on the runner blades as well as the pressure inside the turbine are observed in time domain. The amplitudes and frequencies are analysed using Fast Fourier Transform (FFT). The frequencies are compared to the modal analysis results of the runner submerged in water. This hydraulic design is based on previous axial turbine, which was designed for small hydro power plant located at Lužnice River. The design point is defined by head 

H = 1.5 m, volumetric flow rate 

Q = 2.1 m³/s and RPM 

n = 260 min⁻¹. This turbine was designed using six runner blades to reduce the pressure, force and torque pulsations caused mainly by wakes, which were observed behind the rib.

1. Introduction

The small hydropower plant Lužnice, which is located close to the city Tábor, contains six Kaplan turbines without any regulating mechanism. Each machine provides approximately performance of 

\[ P = 16 \text{ kW} \]

which corresponds to the volumetric flow rate of 

\[ Q = 1.5 \text{ m}^3\text{s}^{-1}. \]

It was decided to increase the output of the turbine during the rehabilitation of the first machine, which was reached by increasing of the volumetric flow rate to 

\[ Q = 2.1 \text{ m}^3\text{s}^{-1}. \]

The parameters of the design point for the new turbine are: total head 

H = 1.5 m, volumetric flow rate 

Q = 2.1 m³/s and RPM 

n = 260 min⁻¹.

Each of the original turbines consisted of circular pipe which covered the shaft, nine guide vanes, four runner blades and 10.6° full angle draft tube. The turbine runner had a diameter of 

D = 800 mm

and the hub had a diameter of 

d = 272 mm. As a part of the new design, the pipe which covered the shaft was replaced by a rib. The new design contains five guide vanes and six runner blades with lower wrap angle and higher efficiency. According to the CFD results, the output of the new designed turbine is ca. 

\[ P_{\text{out}} = 25 \text{ kW} \] \[ \text{and the hydraulic efficiency of the whole turbine is more than } \eta = 84 \%. \] The amplitude-frequency characteristics of static pressures, forces and torques were monitored during transient analysis and compared to the results of the modal analysis of the runner mounted on the shaft to avoid the resonance. After finishing the project, we decided to study the influence of blade position in axial direction on the pressure pulsations. This was realized by offsetting of every other blade of the runner in axial direction. The three turbine runners were designed using various offsets (20 mm, 40 mm and their combination). The last design was created by combination of all offsets, i.e. 0 mm, 20 mm and 40 mm. This blade offset had a major impact on the excitation frequencies and their amplitudes. The changes in output parameters of the turbine especially in hydraulic efficiency were not noticed. This article describes the design process of the new turbine using numerical simulations and presents the results of amplitude-frequency characteristics for all cases.

2. Hydraulic Design of the Turbine

The specific speed of the new turbine calculated by (1) has a value of 

\[ n_q = 278 \text{ min}^{-1}. \] The specific speed can be calculated by following equation:

\[ n_q = 333 \cdot n \cdot \frac{\sqrt{Q_v}}{\sqrt{H^3}} \] \[ \text{(1)} \]

where 

\[ Q_v \text{ [m}^3\text{s}^{-1}] \] is the volumetric flow rate, 

\[ Y [\text{J} \cdot \text{Kg}] \] is the specific energy and 

\[ n \text{ [min}^{-1}] \] is RPM. The unit RPM 

\[ n_{11} \text{ [min}^{-1}] \] and unit flow rate 

\[ Q_{11} \text{ [m}^3\text{s}^{-1}] \] can be calculated using equations (2) and (3) which are based on the theory of hydraulic similarity. In this equations 

\[ H [\text{m}] \] is a head of water, 

\[ D [\text{m}] \] is a runner diameter.

\[ n_{11} = \frac{n \cdot D}{\sqrt{H}} \] \[ \text{(2)} \]

\[ Q_{11} = \frac{Q_v}{D^2 \cdot \sqrt{H}} \] \[ \text{(3)} \]

The preliminary design considered the use of the swirl turbine without guide vanes. The velocity triangles were used for the beta angle estimation. The assumption of straight intake on the blade without circumferential component of absolute velocity 

\[ c_{u1} \text{ [m}^3\text{s}^{-1}] \] was used at the leading edge of the blade. Meridional velocity component was calculated from continuity equation, circumferential component of absolute velocity 

\[ c_{u2} \text{ [m}^3\text{s}^{-1}] \] was obtained from Euler’s turbine equation (4) with hydraulic efficiency estimation and circumferential velocity 

\[ u_{12} \text{ [m}^3\text{s}^{-1}] \] was based on RPM.
represents the volumetric flow rate which is monitored at the turbine inlet and the draft tube outlet. The hydraulic efficiency \( \eta_h \) was evaluated using equation (7) \([3][4]\)

\[
\eta_h = \frac{P_{\text{out}}}{P_{\text{in}}} \quad \text{(7)}
\]

Various quantities were monitored during the transient analysis to capture all excitation forces acting on the mechanical parts of the turbine, i.e. static pressure, forces, and torques. The corresponding amplitude-frequency characteristics were obtained by Fast Fourier Transformation (FFT). The overview of all points, which were used to monitor the static pressure, is shown in the following picture (Fig. 2). The forces and torque were evaluated on the surface of the runner blades and guide vanes. The duration of the transient simulation was set to 4096 iterations, which corresponded to more than 22 revolutions of the runner.

2.2. Design of runner with offset blades

As a next step, the influence of blade position in axial direction was studied. Three configurations of runner blades were designed. The offset was realized by displacement of meridian blade cut in axial direction. The first and second configuration were created by displacement of every other blade. All the blades with modified position were moved by 20 mm in the first case and 40 mm in the second case (Fig. 4).

The last configuration was designed as a combination of various values of blade displacement. The first blade had a position of 0 mm, the second one was moved by 20 mm and third was moved by 40 mm. (Fig. 3)[1][2]
3. Numerical Simulation

3.1. Transient CFD analysis

The runner and guide vanes were modelled separately using ANSYS BladeGen. The computational mesh was created using ANSYS Turbogrid and ICEM CFD. The volumetric losses were taken into account and therefore the blade gap between the runner and chamber was included in the model. The computational model of the whole turbine consisted of 6 040 000 hexahedral cells (6 274 000 nodes). The mesh was created as structured with boundary layer refinements in the boundary layer.

All simulations were calculated as transient analysis using ANSYS CFX. The time-step was defined to turn 2° per time step, i.e. for RPM $n = 260 \text{ min}^{-1}$ was used time step $\Delta t = 0.001282 \text{ s}$ with six internal iterations. The analysis was carried out using k-ε realizable turbulence model. All parts were set as stationary with the exception of the runner. The transient Rotor-Runner interface (marked by green colour in the figure (Fig. 6)) was applied between rotating and stationary domain. The general grid interface – GGI was applied between the inlet domain and domain with guide vanes (marked by yellow colour) as well as between non-periodic blade-channels in the runner (purple marked). These interfaces enabled to connect the computational meshes with different mesh distributions at the common face. The no-slip wall boundary condition was applied on all faces where fluid can’t flow through. The counter rotating wall was applied on the chamber in runner rotating domain. The total pressure boundary condition ($P_{\text{tot}} = 15 000 \text{ Pa}$) was applied at the inlet (marked by blue colour) to simulate a head of water over the turbine. The zero static pressure ($P_s = 0 \text{ Pa}$) was applied at the outlet of the domain (marked by red colour). The reference pressure was set to $P_{\text{ref}} = 1 \text{ atm}$.

The transient analyses were carried out for various RPM, i.e. $n_1 = 160 \text{ min}^{-1}$, $n_2 = 210 \text{ min}^{-1}$, $n_3 = 260 \text{ min}^{-1}$ and $n_4 = 310 \text{ min}^{-1}$, to observe the influence on output characteristics.

3.2. Acoustic modal analysis

The acoustic modal analysis was carried out to ensure that there is no coincidence between the excitation frequencies coming from the flow and natural frequencies of the runner mounted on the shaft. When the runner is submerged in water its modal properties are influenced by added mass effect. The computational model consisted of the runner and shaft fixed in bearings and an acoustic domain representing the water around the runner. The material properties of linear elastic material model which was used for structural parts are in Table 1. The runner was made of chrome-nickel steel and the shaft was made of construct steel (class 12).

Table 1 Structural and fluid material properties

| Material                  | Chrome - nickel steel | Construct steel |
|---------------------------|-----------------------|-----------------|
| Density [Kg/m$^3$]        | 7700                  | 7800            |
| Young modulus [MPa]       | 206000                | 210000          |
| Poisson ratio [-]         | 0.288                 | 0.33            |
| Density [kg/m$^3$]        |                       |                 |
| Speed of sound [m/s]      |                       |                 |
| Water                     | 998.2                 | 1482.1          |

The water was represented by acoustic elements. The computational mesh was created in ANSYS Meshing using 590 000 linear hexahedral and tetrahedral elements (354 000 nodes). For material properties of water see Table 1. The computational mesh is shown in the figure below (Fig. 7).
The shaft, fully connected to the runner, was fixed in the bearings by displacement boundary conditions. All degrees of freedom were fixed except the rotation around shaft axis. The FSI interface was applied on the surface where the hub and runner are in contact with water. The zero acoustic pressure was applied at the inlet and outlet of the domain.

The first six natural frequencies of the runner submerged in still water are shown in the table below.

| Mode | \( f_{\text{WATER}} \) [Hz] |
|------|---------------------|
| 1    | 0                   |
| 2    | 57.833              |
| 3    | 134.36              |
| 4    | 149.61              |
| 5    | 187.62              |
| 6    | 347.53              |

4. Results

The unit RPM and unit flow rate were calculated using equations (2) and (3) to compare the \( n_{\text{11}} - Q_{\text{11}} \) characteristics for all three configurations to the original one (Fig. 8).

The \( n_{\text{11}} - Q_{\text{11}} \) characteristics show, that a small blade displacement (compared to the runner dimensions) has no major impact on the flow characteristics. A light drop of hydraulic efficiency is significant, but it does not exceed more than one percent. The performance characteristics of all configurations correspond to the original one with maximum difference of 0.5 % (Fig. 9).

4.1. Observed excitation frequencies

All presented results correspond to the RPM value of \( n_3 = 260 \text{ min}^{-1} \).

4.1.1. Pulsations in the impeller domain

The pressure pulsations were monitored in various points (Fig. 2). The most significant impact of the blade offset on the static pressure pulsations was observed at the outlet of the impeller domain (points P4, P5 and P6 in Fig. 2). The maximum detected amplitudes and the corresponding frequencies are summarized in Table 4. Each column in the table shows the maximum detected amplitudes plotted in one of the figures below (P4 - Fig. 10, P5 - Fig. 11 and P6 - Fig. 12).

The highest amplitude decrease (28 % - 45 %) was obtained for the configuration with blade offset 20 mm. While the distance from the hub in radial direction is increasing, the amplitude drop is decreasing. The highest amplitude drop was observed in point P6, which is located close to the hub. The amplitude drop was also observed for the configurations with various blade offsets (16 % - 20 %) and 40 mm offset (10 % - 11 %).
4.1.2. Pulsations behind the rib

The offset of runner blades has a positive impact on the amplitudes of static pressure pulsations evaluated near the trailing edge of the rib (points P1 – P3 in Fig. 2). The maximum detected amplitudes and the corresponding frequencies are summarized in Table 5.

The highest amplitude drop was reached by configuration with various blade offsets (24% - 32%). Compared to the original design without blade offset, the excitation frequency 8.8 Hz occurred. The significant amplitude decrease (15% - 16%) was also observed by configuration with 20 mm blade offset. The amplitude-frequency characteristics evaluated in point P2, that is located in the centre of flow channel behind the rib, are shown in the figure below (Fig. 13).

Table 4 The overview of maximum static pressure amplitudes behind the runner

| Offset [mm] | Monitor point | P4 | P5 |
|-------------|---------------|----------------|----------------|
| 0           | 2250 [26Hz]   | 0%             | 1750 [26Hz]    | 0%             |
| 20          | 1620 [26Hz]   | -28%           | 1180 [26Hz]    | -33%           |
| 40          | 2030 [26Hz]   | -10%           | 1560 [26Hz]    | -11%           |
| Comb.       | 1880 [26Hz]   | -16%           | 1440 [26Hz]    | -18%           |
| Offset [mm] | Monitor point | P6 |
|-------------|---------------|----------------|
| 0           | 1240 [26Hz]   | 0%             |
| 20          | 680 [26Hz]    | -45%           |
| 40          | 1100 [26Hz]   | -11%           |
| Comb.       | 980 [26Hz]    | -20%           |

Fig. 10 Amplitude-frequency characteristics calculated from point 4 for static pressure (at 90% height of turbine chamber)

Fig. 11 Amplitude-frequency characteristics calculated from point 5 for static pressure (at 50% height of turbine chamber)

Fig. 12 Amplitude-frequency characteristics calculated from point 6 for static pressure (at 10% height of turbine chamber)

Fig. 13 Amplitude-frequency characteristics calculated from point 2 for static pressure (behind the rib at centre streamline)
### Table 5 The overview of maximum static pressure amplitudes behind the rib

| Offset [mm] | Monitor point | P1      | P2      |
|------------|---------------|---------|---------|
| 0          | 186 [26Hz]    | 0 %     | 188 [26Hz] | 0 %   |
| 20         | 156 [26Hz]    | -16%    | 159 [13Hz] | -15%  |
| 40         | 211 [13Hz]    | +13%    | 181 [13Hz] | -4%   |
| Comb.      | 141 [8.8Hz]   | -24%    | 128 [8.8Hz] | -32%  |

### 4.1.3. Pulsations in the draft tube

The draft tube is the only part of the turbine where the impact of blade offset on the static pressure pulsations was slightly negative (see Table 6). The amplitudes of pressure pulsations in point P10 located near the wall rapidly increased up to +471 %. The excitation frequencies observed in this point are multiples of blade frequencies. All maximum amplitudes in the draft tube are in the orders of Pascal and their impact on the dynamic behaviour of the turbine is negligible.

### Table 6 The overview of maximum static pressure amplitudes in the draft tube

| Offset [mm] | Monitor point | P7      | P8      |
|------------|---------------|---------|---------|
| 0          | 33 [26Hz]     | 0 %     | 43 [26Hz] | 0 %   |
| 20         | 19 [62Hz]     | -42%    | 41 [26Hz] | -5%   |
| 40         | 32 [62Hz]     | -3%     | 42 [26Hz] | -2%   |
| Comb.      | 42 [17.3Hz]   | +27%    | 38 [26Hz] | -11%  |

### 4.1.4. Forces acting on the stator blade

The offset of runner blades has major impact on the forces acting on the guide vane surface. The increase of force amplitudes between 10 % - 80 % is observed by the offsets of 20 mm and 40 mm (Table 7). The slight decrease of force amplitude (-9 %) was observed by the configuration with various blade offsets (Fig. 14). All detected frequencies are multiples of the blade frequency (26 Hz).

### Table 7 The overview of maximum force amplitudes acting on the guide vane

| Offset [mm] | Monitor point | Guide vane |
|------------|---------------|------------|
| 0          | 25.57 [26Hz]  | 0 %        |
| 20         | 28.2 [13Hz]   | +10%       |
| 40         | 46 [13Hz]     | +80%       |
| Comb.      | 23.38 [8.8Hz] | -9%        |

No significant impact of the blade offset on the amplitudes and frequencies calculated for the forces acting on the runner blades were detected.

The all excitation frequencies detected from all ten points P1 – P10 were compared to the results of acoustic modal analysis. No coincidence between the excitation frequencies and natural frequencies of runner submerged in water was found.

### 5. Conclusion

The influence of runner blade offset in axial direction on the static pressure pulsations was investigated. Three configurations of blade offset were designed. The offset was realized by displacement of meridian blade cut in axial direction. The first and second configuration were created by displacement of every other blade. All the blades with modified position were moved by 20 mm in the first case and by 40 mm in the second case. The last configuration was designed as a combination of various values of blade displacement.

The decrease of pressure amplitudes behind the rib and behind the runner was achieved by all three configurations. The performance characteristics of all configurations correspond to the original one with maximum difference of 0.5 %. The only increase of pressure amplitudes was observed near the wall of the draft tube. These pulsations have no significant effect on the machine dynamics, as the amplitudes are in orders of...
The similar trend of pressure pulsations was also observed for different values of RPM, i.e. $n_1 = 160 \, \text{min}^{-1}$, $n_2 = 210 \, \text{min}^{-1}$ and $n_3 = 310 \, \text{min}^{-1}$.

The highest static pressure amplitude drop was reached by 20 mm blade offset in all parts of the turbine (15% - 16% behind the rib, 28% - 45% at the outlet of impeller domain and 15% - 42% in the draft tube). The only negative effect of the blade offset for all configurations was observed near wall of the draft tube.

The increase of pressure amplitudes was by all configurations in the range of 328% - 471%. The absolute value of these amplitudes are in orders of Pascal and have insignificant impact on the dynamic behaviour of the turbine.

The significant amplitude decrease was also observed by the configuration with various axial blade offset (24% - 32% behind the rib, 16% - 20% at the outlet of impeller domain).

The pressure amplitudes decreased by all offset configurations. Another important fact is, that new excitation frequency occurred. By the offset of 20 mm and 40 mm, the new excitation frequency 13 Hz occurred. This frequency has a half value of the dominant frequency detected by the original runner without offset. By configuration with various blade offsets the new excitation frequencies 8.8 Hz and 17.3 Hz were detected.

The acoustic modal analysis was also carried out to ensure that there is no coincidence between the detected excitation forces and natural frequencies of the runner. This analysis enables to calculate natural frequencies and corresponding mode shapes for runner submerged in the water, which is more accurate. The first six natural frequencies were computed. These frequencies were compared to all excitation frequencies detected in the whole turbine and no coincidence was found.

The future steps will continue with the analysis of the runner with all six blades moved about 20 mm.

This work has been supported by specific research project no. FSI-S-17-4615 “Multiphysics Problems in Fluid Mechanics”.

References

1. J. Veselý, F. Pochylý, J. Obrovský, J. Mikulášek. A New Concept of Hydraulic Design of Water Turbine Runners. International Journal of Fluid Machinery and systems. Turbomachinery Society of Japan, Korean Fluid Machinery Association, Chinese Society of Engineering Thermophysics, IAHR, 383-391 (2009)

2. F. Pochylý, M. Haluza, J. Veselý, The Francis Pump Turbine With Stochastic Blades. Procedia Engineering. Elsevier, 68-75 (2012)

3. J. Bednář, Turbíny: (malé vodní elektrowně). Češkovice: Marcela Bednářová, (357), (2013) ISBN-978-80-905437-0-6

4. M. Nechleba, Vodní turbíny, jejich konstrukce a příslušenství. Praha: Státní nakladatelství, (539) (1962), ISBN-04-241-61