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Impact of Blade Quality on Maximum Efficiency of Low Head Hydraulic Turbine

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Paper presents considerations based on 1D Euler equation for hydraulic turbine stage efficiency with the help of Energy–Losses diagram. It has been shown that the character of rotor blading strongly depends on the ratio of guide blade losses to rotor blade losses. This is the crucial point of presented analysis. The presented method gives the information what has to be done in order to achieve maximum stage efficiency for designed profiles. The problem how profiles should be shaped is left to either experience or to higher order computation for the details of profile characteristics. It has to be emphasized that flow deflection angle and losses are coupled and they have to be matched according to presented results.

Presented approach based on 1D model has to be correlated with results obtained from higher order models like 2D or 3D. These higher order models, giving more details of flow through the turbine stage, are not suitable for the prediction of turbine kinematics to assure maximum efficiency for given by investors input data of designed turbine stage.

Keywords: Hydraulic turbines, 1D theory, Euler turbine basic equation, turbine stage design

1. Introduction

The recent policy of renewable energy increases the interest to hydraulic turbine of very low heads. From one side the progress in technology makes small hydraulic turbine installation reasonable from economical standpoint [3]. From another side the developments in fluid mechanics of turbine blading improve the efficiency of energy extraction from the small heads, which stay at disposal of potentials investors. This second reason pronounces by the fact of better blade quality of blading which can be used for low head turbines. By the better quality of blading we mean here as low as possible dissipation (losses) for a given deflection angle of the stream flowing through the guide vanes and rotor blade cascades. It is not sufficient to have a well elaborated blading for guide vanes and rotor cascades. They have to be matched properly in order to achieve the maximum efficiency of the stage. Low level of loss coefficients of blading should correspond to high efficiency of turbine stage, which is not only the function of blading losses. The aim of present paper is to elucidate the relation between blade losses and deflection angles of the flow in order to maximize the efficiency of a turbine stage. The problem can be denominated as proper matching of bladings in turbine stage. The recent analysis is based on traditional 1D model, well known in classical literature on turbomachinery [4, 6]. But the way of analysis of stage efficiency is new and may support the higher order computations based on 2D or 3D models. Performed in the paper considerations have fundamental character for validation of experiments and higher order numerical computation (2D, 3D).

2. Representation of energy conversion for water turbine

Energy conversion process in thermal turbo machinery is very commonly represented by means of entropy–enthalpy diagrams [1, 2]. By analogy, a similar representation can be applied for water turbine. Although this is not very popular technique, as it follows from recent literature on hydraulic turbine [5], there is as a reason for such an approach due to the fact of very transparent way of presenting the processes as it was shown in [6].

The fluid element in classical mechanical approach is a carrier of the sum of four forms of energy:

- kinetic energy connected with absolute velocity $\vec{c}$ and defined as $c^2/2$;
- potential energy in gravity or centripetal force field denoted as Π;
- pressure energy defined for density ρ as p/ρ;
- internal energy connected with specific heat c_v and temperature T of fluid element denoted as c_vT.

The sum of energies of fluid element

$$E = \frac{c^2}{2} + \Pi + \frac{p}{\rho} + c_vT \tag{1}$$

remains either constant (guide vanes) or diminishes in a rotor domain of turbine. This fundamental principle of energy conversion is subjected to certain restriction. For incompressible fluid, neglecting a heat transport, no matter how the sum E behaves along a trajectory of fluid element, we have always

$$T(t_2) > T(t_1) \tag{2}$$

where time instants fulfil $t_2 > t_1$. This is due to dissipation effects existing always during energy conversion process. It can be expressed by the entropy increase what is a common technique in thermal turbo machinery but not in hydraulic machinery. Hydraulic literature prefers to present dissipation by means of losses defined commonly by the means of loss coefficient ζ.

$$\Delta E_{1,2} \overset{def}{=} \zeta \frac{c^2}{2} = c_v(T_2 - T_1) \tag{3}$$

Let us consider two positions of fluid element on its trajectory in gravity field, as it is shown in Fig. 1. If there is no energy extraction between position 1 and position 2, the process can be presented in (ΔE, E) diagram as it is shown in Fig. 2.

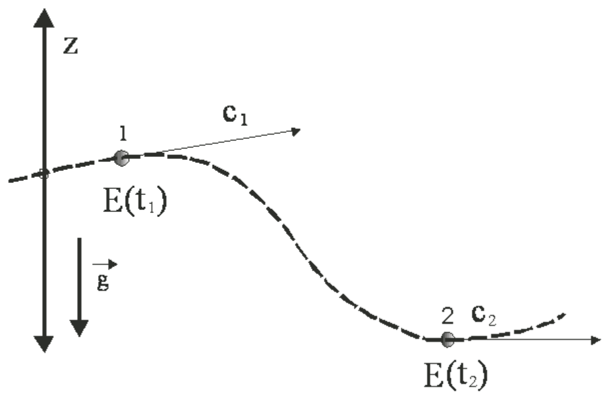

Figure 1 Two positions of fluid element on its trajectory line

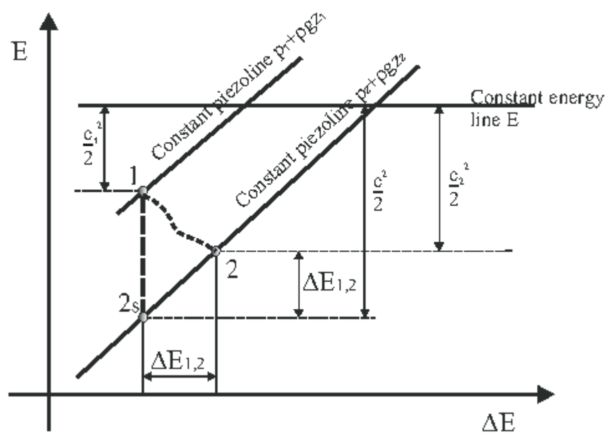

Figure 2 Representation of energy conversion process without energy extraction

The line 1-2s represents a process without dissipation, a curve 1-2 is the process with losses: $\Delta E_{1,2}$. The lines of constant $p + \rho gz$, the so–called piezolines, are inclined as it is shown in Fig. 2 proportionally to ΔE. If there are no losses the kinetic energy obtainable from the change of piezometric pressure is .

For the position 1 in Fig. 2 we can write

$$\frac{c^2}{2} = \frac{c_1^2}{2} + \left(\frac{p_1 + \rho\, g z_1}{\rho} - \frac{p_2 + \rho\, g z_2}{\rho}\right) \tag{4}$$

For the position 2 we have

$$\frac{c^2}{2} = \frac{c_2^2}{2} + \Delta E_{1,2} \tag{5}$$

Comparing right sides of above relations we get

$$\frac{c_1^2}{2} + g z_1 + \frac{p_1}{\rho} + c_v T_1 = \frac{c_2^2}{2} + g z_2 + \frac{p_2}{\rho} + c_v T_2 \tag{6}$$

This means energy conservation in process 1–2.

3. Energy extraction in water turbine stage

Applying the above technique we can present the process in hydraulic turbine stage with energy extraction. First of all one can introduce two reference coordinates. The absolute coordinate is tied with guide blading and energy conservation is expressed by relation (6). The second coordinate system is moving with rotor blading. Energy conservation is expressed by relative velocity and additional potential energy due to centripetal acceleration is determined by rotor velocity . In this coordinate system energy conservation has the form

$$\frac{w_1^2}{2} + g z_1 + \frac{p_1}{\rho} - \frac{u_1^2}{2} + c_v T_1 = \frac{w_2^2}{2} + g z_2 + \frac{p_2}{\rho} - \frac{u_2^2}{2} + c_v T_2 \tag{7}$$

Changing the nomenclature of particle position as follows:

- 0 – for the position in front guide blade;
- 1 – for the position between guide vanes and rotor blading;
- 2 – for the position behind rotor;

the process of energy extraction in (ΔE, E) diagram can be presented as in Fig. 3.

The energy extracted from the flow, denoted as a in Fig. 3, given by an Euler formula, is

$$a = \frac{c_1^2}{2} - \frac{w_1^2}{2} + \frac{u_1^2}{2} + \frac{w_2^2}{2} - \frac{u_2^2}{2} - \frac{c_2^2}{2} \tag{8}$$

Let us define the parameters being in common use to characterize a turbine stage performance. The set of parameters defined below appears in the turbo machinery literature in many variants. Therefore the definitions are necessary at every case,

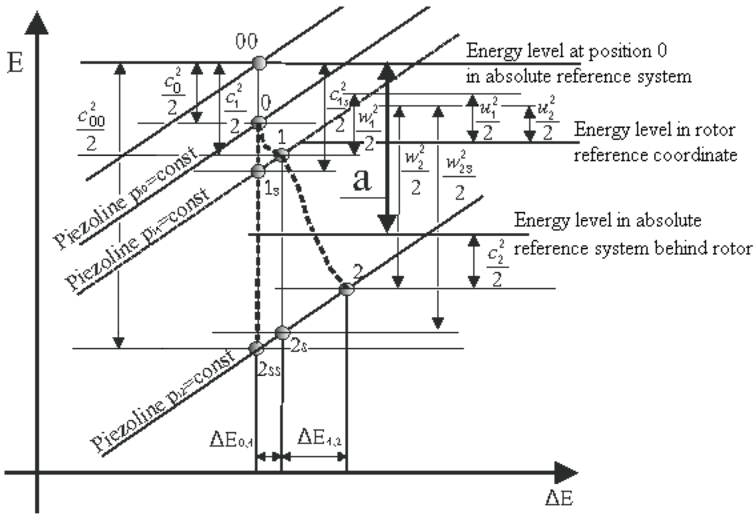

Figure 3 Representation of energy extraction process in hydraulic turbine stage

to avoid confusion. The notations used below are shown in Fig. 3. The velocity c_{oo} is related to the head H, so that

$$c_{oo} \stackrel{def}{=} \sqrt{c_o^2 + 2gH} \tag{9}$$

The hydraulic efficiency (neglecting leakage losses) of the stage is

$$\eta \stackrel{def}{=} \frac{a}{c_{oo}^2/2} \tag{10}$$

The reaction of the stage, which determines the division of the head between guide vanes and rotor bladings, is

$$\gamma \stackrel{def}{=} \frac{(p_{i1} - p_{i2})/\rho}{c_{oo}^2/2} = \frac{(c_{oo}^2 - c_{1S}^2)/2}{c_{oo}^2/2} \tag{11}$$

Mass flow rate coefficient is

$$\mu \stackrel{def}{=} \frac{m}{\rho S c_{oo}} \tag{12}$$

where: m[kg/s] is mass flow rate, S[m^2] is axial cross-section in front of a guide vanes cascade.

For the sake of simplicity let us assume $u_1 = u_2 = u$, what is reasonable for low head turbine design, where cross–sections do not change essentially between positions 0–1–2. The velocity coefficient can be defined as

$$\nu \stackrel{def}{=} \frac{u}{c_{oo}} \tag{13}$$

per analogy to values μ and ν used in turbomachinery [4].

The quality of blading, with respect to dissipation processes, is characterized by loss coefficient. For a guide vanes cascade the loss coefficient is

$$\zeta_1 \stackrel{def}{=} \frac{(c_{1S}^2 - c_1^2)/2}{c_{1S}^2/2} = 1 = \frac{c_1^2}{c_{1S}^2} \tag{14}$$

For a rotor cascade the definition is

$$\zeta_2 \stackrel{def}{=} \frac{(w_{2S}^2 - w_2^2)/2}{w_{2S}^2/2} = 1 = \frac{w_2^2}{w_{2S}^2} \tag{15}$$

The above defined values can be tied up by kinematic of the flow through blade cascades in turbine stage.

4. Kinematic of turbine flow

Within the frame of one–dimensional model (1D), the sketch of blading shown in Fig. 4, will be essential for developing the formulas according to above definitions.

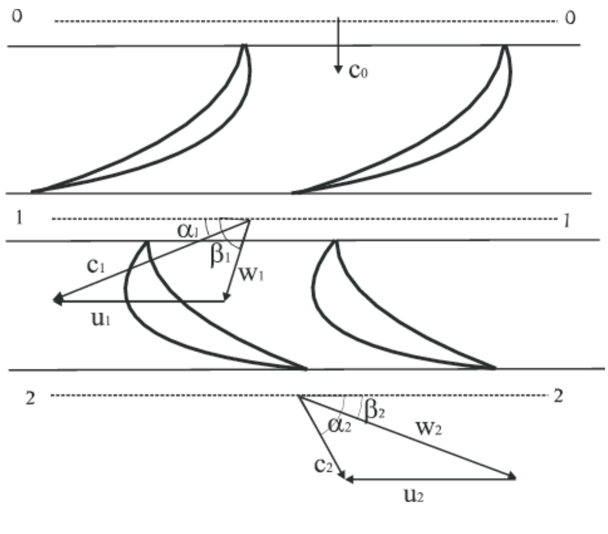

Figure 4 The sketch of blading with description of velocity triangles

Two types of rotor blades are possible. The first type shown in Fig. 5 is for high deflected rotor blades (unreal deflection) where the flow turns in rotor blade by the angle $\Theta = \beta_1 - \beta_2$ (the notation of β_1 has been changed comparing to the sketch in Fig. 4)

The second type of blading with triangles shown in Fig. 6 is characterized by a low deflection (common deflection in Kaplan turbines). For the description of β angles as in Fig. 6 the flow deflection given by $\Theta = \beta_1 - \beta_2$ is low.

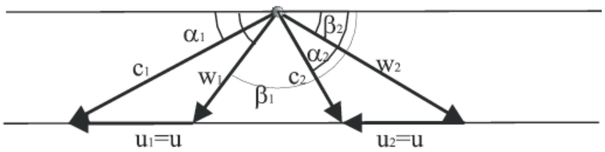

Figure 5 Velocity triangles for high deflected rotor blades

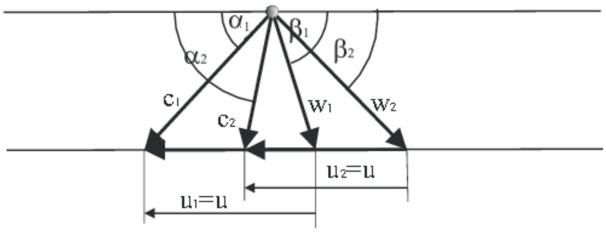

Figure 6 Velocity triangles for low deflected rotor blades

The question in which conditions the high or low deflected rotor blades are suitable can be answered by the criteria of maximum available efficiency. It is worth noting that according to definition (10) maximum efficiency assures maximum power of the stage for a given head H and kinetic energy at inlet to the guide vanes because power is given by

$$N_{max} = \eta_{max}\, m \frac{c_{oo}^2}{2} \tag{16}$$

In order to introduce kinematic values into efficiency defined by (10), let us rewrite η with the help of Euler formula

$$\eta = \left(\frac{c_1}{c_{oo}}\right)^2 - \left(\frac{w_1}{c_{oo}}\right)^2 + \left(\frac{w_2}{c_{oo}}\right)^2 - \left(\frac{c_2}{c_{oo}}\right)^2 \tag{17}$$

The group of brackets can be expressed by means of definitions and triangle formulas. So we have

$$\left(\frac{c_1}{c_{oo}}\right)^2 = (1-r)(1-\zeta_1) \tag{18}$$

$$\left(\frac{w_1}{c_{oo}}\right)^2 = (1-r)(1-\zeta_1) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\rho_1)} + \nu^2 \tag{19}$$

$$\left(\frac{w_2}{c_{oo}}\right)^2 = (1-\zeta_2)\left[r + (1-r)(1-\zeta_1) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + \nu^2\right] \tag{20}$$

$$\left(\frac{c_2}{c_{oo}}\right)^2 = (1-\zeta_2)\left[r + \left(\frac{w_1}{c_{oo}}\right)^2\right] - 2\nu\cos\beta_2\sqrt{(1-\zeta_2)}\sqrt{r + \left(\frac{w_1}{c_{oo}}\right)^2} + \nu^2 \tag{21}$$

Summing up the above relations we have the shape of the function

$$\eta = \eta\left(\overbrace{\zeta_1, \alpha_1}^{vanes.char}, \overbrace{\zeta_2, \beta_2}^{rot.char}, \overbrace{r, \nu}^{stage.char}\right) \tag{22}$$

with three groups of arguments. The first group characterizes a guide vanes cascade, the second group characterizes a rotor cascade and the third group characterizes the stage. The fact that the shape of this function is rather complicated and not transparent at all, was the reason for not paying too much attention in the literature to analyse the behaviour of this function. The function has the shape as follows

$$\eta = 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + 2\nu\cos\beta_2\sqrt{(1-\zeta_2)}\sqrt{1-\zeta_1(1-r) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + \nu^2} - 2\nu^2 \tag{23}$$

Two parameters can be eliminated from the list of arguments. Let us eliminate reaction r and outlet angle from the rotor β_2. For the cross–section 1 behind the guide vanes cascade mass flow rate is

$$m = \rho S c_1 \sin\alpha_1 \tag{24}$$

Neglecting leakages the same mass flow rate is for the cross-section 2 where

$$m = \rho S w_2 \sin\alpha_1 \tag{25}$$

With the help of mass flow rate coefficient the above relations can be rewritten as

$$\mu = \sqrt{(1-r)(1-\zeta_1)}\sin\alpha_1 \tag{26}$$

$$\mu = \sqrt{(1-\zeta_2)}\sqrt{1-\zeta_1(1-r) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + \nu^2}\sin\beta_2 \tag{27}$$

which allows eliminating r and β_2 in function (23) and rearranging it to the form

$$\eta = \eta\left(\overbrace{\zeta_1,\zeta_2,\alpha_1}^{blade\ char.},\ \overbrace{\mu,\nu}^{stage\ char.}\right) \tag{28}$$

Now it is possible to define the problem of efficiency maximization. The mass flow rate coefficient μ is given when head H, mass flow rate m and cross–section S are the input values at the starting point of every project. If the quality of bladings (ζ_1, ζ_2) can be approximately estimated on the base of experience, then only two parameters are left for maximizing the function (28). The shape of the function (28) is algebraically complicated, as one can see below

$$\eta = 2\nu\left[\mu\cot\alpha_1 - \nu + B\sqrt{1-\zeta_2}\sqrt{1+\nu^2-2\mu\nu\cot\alpha_1 - \frac{\zeta_1}{1-\zeta_1}\left(\frac{\mu}{\sin\alpha_1}\right)^2}\right] \tag{29}$$

where

$$B = \sqrt{1 - \frac{2(1-\zeta_1)(\mu\sin\alpha_1)^2}{(1-\zeta_2)C}} \tag{30}$$

$$C = (1-\zeta_1)(1+\nu^2) - 2\zeta_1\mu^2 - (1-\zeta_1)[(1+\nu^2)\cos 2\alpha_1 + 2\mu\nu\sin 2\alpha_1] \tag{31}$$

nevertheless it embraces three sources of efficiency losses. The efficiency falls down due to three components as below

$$\eta = 1 - \Delta\eta_1 - \Delta\eta_2 - \Delta\eta_{exit} \tag{32}$$

where efficiency loss in guide vanes cascade is

$$\Delta\eta_1 = \frac{E_1 - E_{1S}}{c_{oo}^2/2} = \zeta_1(1-r) \tag{33}$$

and efficiency loss in rotor cascade is

$$\Delta\eta_2 = \frac{E_2 - E_{2S}}{c_{oo}^2/2} = \zeta_2[1-\zeta_1(1-r) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + \nu^2] \tag{34}$$

and efficiency loss due to exit kinetic energy is

$$\Delta\eta_{exit} = \frac{c_2/2}{c_{oo}^2/2} = \left(\frac{\mu}{\sin\alpha_1}\right)^2 \tag{35}$$

5. Maximizing stage efficiency

Due to complexity of function (29) it is not possible to find an analytical solution for the maximum of such function (only numerical approximation can be obtained). It is worth noting that in a hypothetical case when there is no dissipation in guide vanes and rotor domain $\Delta\eta_1 + \Delta\eta_2 = 0$ we have always for $\Delta\eta_{exit} > 0$, the efficiency of the stage $\eta < 1$. Then maximum η_{max} exists for axial outlet velocity i.e. $\alpha_2 = 90^\circ$ as it can be seen from (35).

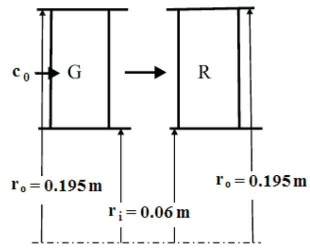

Figure 7 Meridional cross–section of stage

Because of character of function (29) it seems reasonable to examine the behaviour of this function supported by the example. Among five arguments, let us leave free two of them, namely: (α_1, ν), assuming that three arguments (ζ_1, ζ_2, μ) are given. For the sketch of model turbine stage in a meridional cross section as in Fig. 7 let us consider the following data for the example:

mass flow rate: $m = 235$ [kg/s];
head: $H = 2$ [m];
water density: $\rho = 999.1$ [kg/m^3];
gravity: $g = 9.81$ [m/s^2].

The set of such values is usually known to an investor as the input data. From these figures and dimensions as in Fig. 7, simple computation gives

$$S = \pi(r_0^2 - r_i^2) = 0.108149\,[\mathrm{m}^2] \tag{36}$$

$$c_o = \frac{m}{\rho S} = 2.1749\,[\mathrm{m/s}] \tag{37}$$

$$c_{oo} = \sqrt{c_o^2 + 2gH} = 6.631\,[\mathrm{m/s}] \tag{38}$$

$$\mu = \frac{m}{\rho S c_{oo}} = 0.327987\,[–] \tag{39}$$

Let us consider quality of cascades with the loss coefficients like $\zeta_1 = 0.10$, $\zeta_2 = 0.15$. This is rather a pessimistic estimation. Now one can perform the computation of function (29). The shape of the function above plane (α_1, ν) is shown in Fig. 8. The maximum point is marked on the surface by a dot. It has the following coordinates ($\alpha_1 = 25.1742$ [$^\circ$], $v = 0.555542$ [-], $\eta_{max} = 0.75598$ [-]).

To get the idea how sharp is the extreme illustrates the contours plot shown in Fig. 9.

Having the parameters for η_{max} it is easy to get the rest of essential values for the stage kinematics like: reaction of the stage given by the expression

$$r = 1 - \frac{\mu^2}{(1 - \zeta_1)(\sin \alpha_1)^2} \tag{40}$$

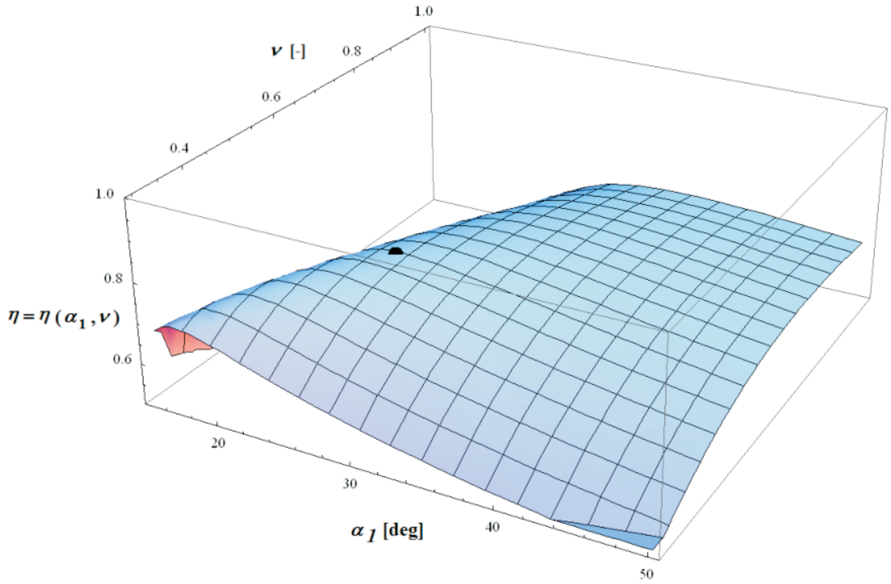

Figure 8 The shape of function $\eta = \eta(\alpha_1, V)$ for the sample calculation

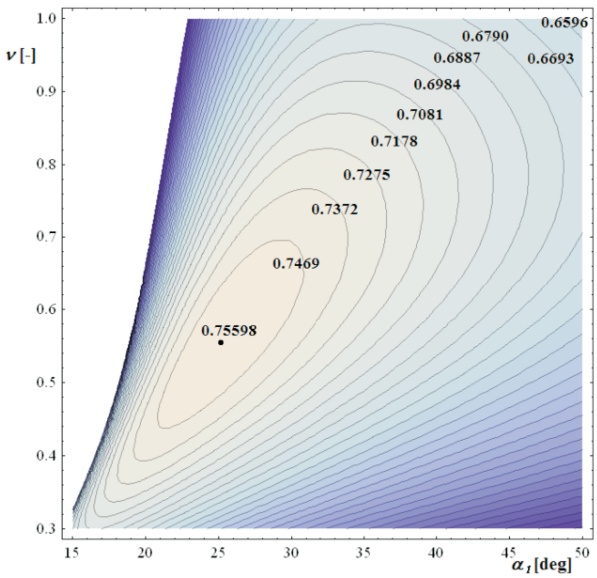

Figure 9 The contours plot of function $\eta = \eta(\alpha_1, V)$ (position of max is shown by a dot).

rotational speed (rpm) of rotor for nominal radius $r_n = 0.5(r_1 + r_2)$ is given as follows

$$n = 1 - \frac{30 v c_{oo}}{\pi \, r_n} \tag{41}$$

triangle parameters one can compute from

$$c_1 = c_{oo}\sqrt{(1-r)(1-\zeta_1} \tag{42}$$

$$w_1 = c_{oo}\sqrt{(1-r)(1-\zeta_1) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + \nu^2} \tag{43}$$

$$\beta_1 = a\sin\left(\frac{c}{w_1}\sin\alpha_1\right) \tag{44}$$

$$w_2 = c_{oo}\sqrt{(1-\zeta_2)}\sqrt{r + (1-r)(1-\zeta_1) - 2\nu\cos\alpha_1\sqrt{(1-r)(1-\zeta_1)} + \nu^2} \tag{45}$$

$$\beta_2 = a\cos\left(\frac{\sqrt{1-(1-\zeta_1)}\mu^2}{(1-\zeta_2)Z}\right) \tag{46}$$

$$Z = 1 + \nu^2 - 2\nu\mu\cot\alpha_1 - \zeta_1(1 + \nu^2 + \frac{\mu}{\sin 2\alpha_1}(\mu - \nu\sin 2\alpha_1)) \tag{47}$$

and further

$$c_2 = \sqrt{w_2^2 + (\nu c_{oo})^2 - 2\nu c_{oo} w_2 \cos\beta_2} \tag{48}$$

$$\alpha_2 = a\sin\left(\frac{w_2}{c_2}\sin\beta_2\right) \tag{49}$$

Finally one can calculate the maximum power

$$N = \eta_{max} m c_{oo}^2/2 \tag{50}$$

For the numerical example considered here, adequate figures are tabulated below

Table 1 Hydraulic turbine stage characteristics

$\boldsymbol{m}$ [-]	ν [-]	η_{max} [-]	α_1 [$^\circ$]	$\boldsymbol{c}_1$ [m/s]	$\boldsymbol{w}_1$ [m/s]	β_1 [$^\circ$]
0.328	0.5555	0.75598	25.17	5.113	2.371	113.45
$\boldsymbol{w}_2$ [m/s]	β_2 [$^\circ$]	$\boldsymbol{c}_2$ [m/s]	α_2 [$^\circ$]	$\boldsymbol{r}$ [-]	$\boldsymbol{n}$ [rpm]	$\boldsymbol{N}$ [kW]
4.179	31.36	2.178	86.96	0.3394	275.9	3.906

A few comments follow from the above table. First of all we can see that according to the notation in Fig. 5, the rotor has a highly deflected cascade $\beta_1 > 90^\circ$. Secondly the angle $\alpha_2 \neq 90^\circ$ what means that maximum efficiency appears at not axial exit velocity c_2. These two remarks are not obvious and cannot be deduced without performing the above computation.

6. Impact of blade quality

The essential presupposition for the present analysis is the quality of blading expressed by coefficients ζ_1, ζ_2. In order to establish the influence on the stage maximum efficiency, the calculation has been executed for the following range of these coefficients:

for: $\zeta_1 = 0.05$, $\zeta_2 = 0.005 \div 0.25$;
for: $\zeta_1 = 0.075$, $\zeta_2 = 0.005 \div 0.25$;
for: $\zeta_1 = 0.10$, $\zeta_2 = 0.01 \div 0.25$;

The value of coefficient ζ_1 is chosen inside the interval of coefficient ζ_2. The results of computation are presented in figures as below. The limiting case is when $\zeta_1 = \zeta_2$. The border line is marked in Fig. 10. Low (commonly known) deflected profiles appear when $\zeta_1 > \zeta_2$ then $\beta_1 < 90^\circ$. This situation repeats for other values of coefficients ζ_1 as it can be seen in Fig. 11 and Fig. 12.

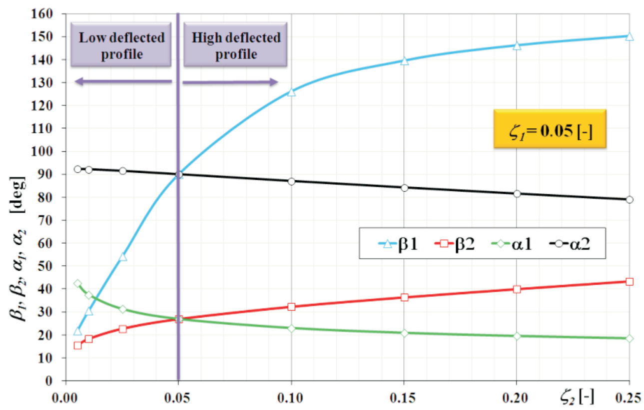

Figure 10 The influence of the loss coefficient 2 on the deflection in cascade for $\zeta_1 = 0.05$

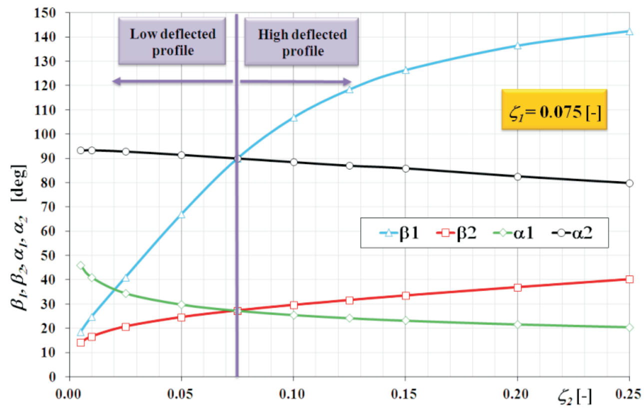

Figure 11 The influence of the loss coefficient 2 on the deflection in cascade for $\zeta_1 = 0.075$

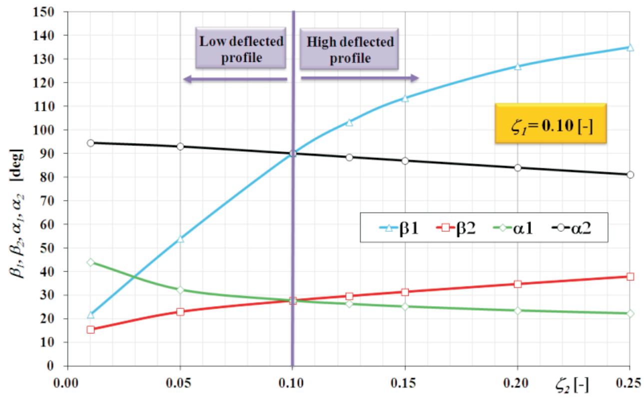

Figure 12 The influence of the loss coefficient 2 on the deflection in cascade for $\zeta_1 = 0.10$

In Fig. 13 the influence of ζ_2 coefficient on maximum efficiency η_{max} of the stage is shown for $\zeta_1 = 0.05$. The similar tendency one can find also for other values of ζ_1. It has to be pointed out that a fall of efficiency for higher values of ζ_2 is by intuition justified. The increase of losses in rotor cascade for higher flow deflection also seems to be clear. The analysis presented here does not discover any relation between increasing of deflection angle and level of losses in rotor cascade. The analysis shows only how the kinematics of the stage flow should correspond to the level of losses for being designed profiles.

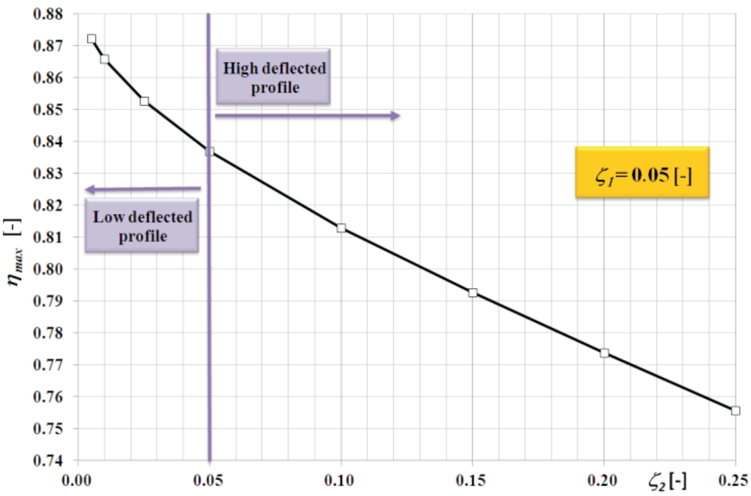

Figure 13 The influence of the loss coefficient ζ_2 on the maximum efficiency η_{max} in the cascade for $\zeta_1 = 0.05$

As the examples two more tendencies are illustrated in Fig. 14 for degree of reaction, and in Fig. 15 for optimal rpm of the stage for $\zeta_1 = 0.075$ and $\zeta_1 = 0.10$ respectively.

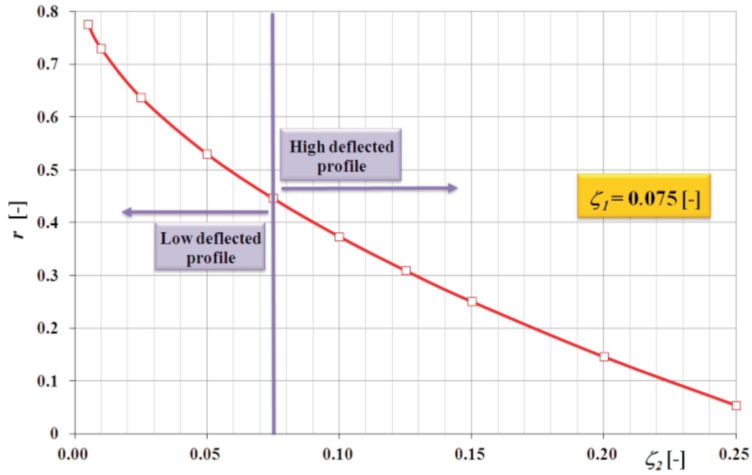

Figure 14 The influence of the loss coefficient ζ_2 on the reaction r in cascade for $\zeta_1 = 0.075$

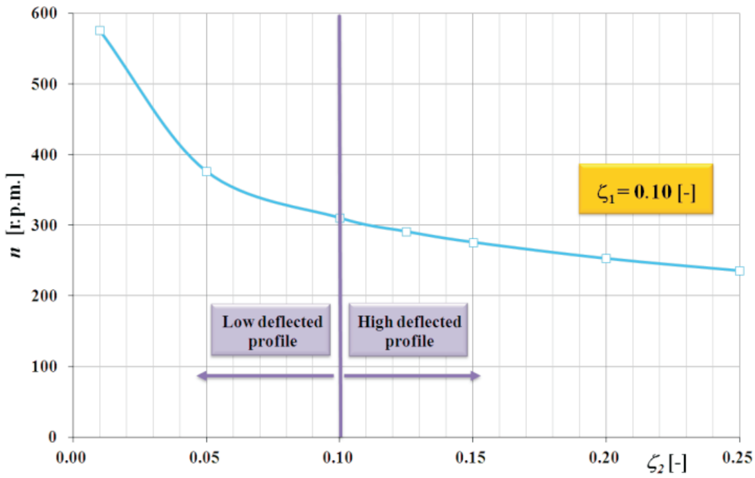

Figure 15 The influence of the loss coefficient ζ_2 on the rotational speed n in cascade for $\zeta_1 = 0.10$

7. Conclusions

The performed analysis is a very important procedure at the early stage of turbine design for a given essential investment parameters.

The character of rotor blading strongly depends on the ratio of guide blade losses to rotor blade losses. This is the crucial point of presented analysis.

The presented method gives the information what has to be done in order to achieve maximum stage efficiency for designed profiles. The problem how profiles should be shaped is left to either experience or to higher order computation for the details of profile characteristics. It has to be emphasized that flow deflection angle and losses are coupled and they have to be matched according to presented results.

The performed approach based on 1D model has to be correlated with results obtained from higher order models like 2D or 3D. These higher order models, giving more details of flow through the turbine stage, are not suitable for the prediction of turbine kinematics to assure maximum efficiency for given by investors input data of designed turbine stage.

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