VERIFICATION OF MODEL CALCULATIONS FOR THE KAPLAN TURBINE DESIGN

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Abstract
In order to design a water turbine, the Theory of the Physical Similarity of Hydraulic Machines is used in technical practice. This principle has been known and used by manufacturers of turbines and pumps, but is not available to general public. This paper describes author´s calculation program for turbine design that is well accessible to the widest possible range of users of mainly small hydropower sources. Based on the given hydraulic potential (water head and flowrate), the program determines the most suitable turbine type and calculates its main geometric parameters. In addition to numerical results, the program is also endowed with graphic output which renders in true scale hydraulic profiles of rotor blades and guide blades as well as the hydraulic profile of a spiral casing. The process of the Kaplan turbine design is used as an example in this paper. The comparisons of the calculated results with the verified standard 4-K-69 Kaplan turbine confirm the compliance of numerical results with reality.

Key words: blade geometry, calculation program, hydropower, theory of physical similarity, turbine design.

INTRODUCTION

On a global scale, there is a great unused renewable hydropower, especially in small water resources. In this area, besides a wide variety of simple water engines like a bladeless turbine, POLÁK (2013A), split reaction turbine DATE (2010), etc., the conventional blade turbines are used the most, KHAN (2009). The design principles of water turbines have been known and used for a long time by the manufacturers of turbines and pumps, but they use their know-how exclusively for their own needs, keeping it away from the public. The authors present here an original calculation program for turbine design, which has been simplified in order to address as wide an audience as possible. However, the simplification does not mean that the quality of the achieved results would suffer. Moreover, the authors extended the program by adding its own graphic output to it, which immediately renders the calculated key turbine components in true scale. In this way, the end users of small hydropower sources are able to find out detailed information on the device for the most effective use of renewable water energy in their specific local conditions.

For the purpose of the turbine design, the technical practice uses the theory of physical similarity of hydraulic machines. Its principle is based on the geometric similarity of velocity triangles (Fig. 1) of so called model (etalon) and real turbines or in other words on the Euler turbine equation:

$$Y_T = u_1 \cdot c_{u1} - u_2 \cdot c_{u2} \tag{1}$$

Where Y_T [J.kg^{-1}] is specific energy of turbine and u [m.s^{-1}] is circumferential and c_u [m.s^{-1}] absolute velocity according to MUNSON (2006) and ANAGNOSTOPOULOS (2009).

The main turbine design parameters are determined by the recalculations of model (etalon) turbines which are processed and described in detail using nomograms and tabulated values. Other design parameters are determined using fluid mechanics relations.

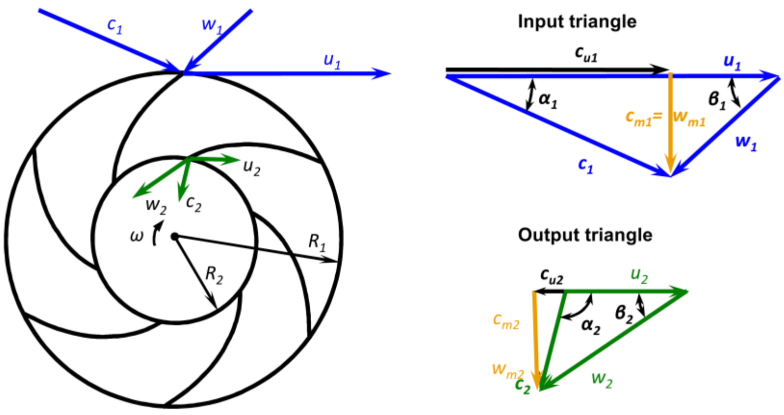


Fig. 1. – Velocity triangles of the radial turbine runner

The hydraulic design of the turbine itself can be converted into a calculation program. However, the tables and nomograms are pitfalls from which it is necessary to "manually" subtract values for other calculations. The authors of the paper converted all necessary tabulated values into mathematical functions. These functions and other fluid mechanics relations were used in order to create a calculation program. This has significantly simplified the whole process of turbine design. In addition to numerical results, the program also offers a graphic extension which displays real scale hydraulic profiles of individual components of the turbine. These are especially blade grid cross sections in stretched stream surfaces which are necessary for the designing and construction of runner blades and guide blades. Moreover, the program allows you to change the angle setting of blades (opening) in the rendered cross-sections and thus display different operating states. Other graphic output draws input and output velocity triangles, another depicts a hydraulic profile of a spiral casing in a cross-section. All the mathematical functions that are the principal of graphic extensions were created by the authors of this paper. The program for the turbine design is possible to edit in any program that can work with mathematical functions (for example MathCAD, Matlab, MS Excel, etc.).

MATERIALS AND METHODS

The calculation program for the turbine design

The following text schematically describes a Kaplan turbine hydraulic design as it is elaborated in the calculation program. Using this program, it is possible to design also Pelton, Francis or Banki turbine. Minding the extent of this paper, these variations are not further discussed and the attention is focused only on the Kaplan turbine. Numerical results of the calculation are subsequently compared with the geometry of the standard 4-K-69 Kaplan turbine (Fig. 6), which is used to verify the model recalculation.

The calculation program has been developed on the basis of the cited literature: MUNSON ET AL. (2006); BRADA ET AL. (1995); SUTIKNO (2011); MELICHAR ET AL. (1998); MELICHAR ET AL. (2002); NECHLEBA (1962); ULRYCH (2007); HUTAREW (1973) and its simplified algorithm is illustrated in Fig. 2.

The values of hydraulic potential, which is water head H [m], flowrate Q [m^3.s^{-1}] and desired turbine shaft speed n [min^{-1}], are entered into program as input variables (green fields in Fig. 5). Subsequently these are used to calculate the specific speed n_q [min^{-1}] by the flow which determines an appropriate turbine type according to DRTINA ET AL. (1999) and TRIVEDI ET AL. (2016):

$$n_q = n \cdot \frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} \qquad (2)$$

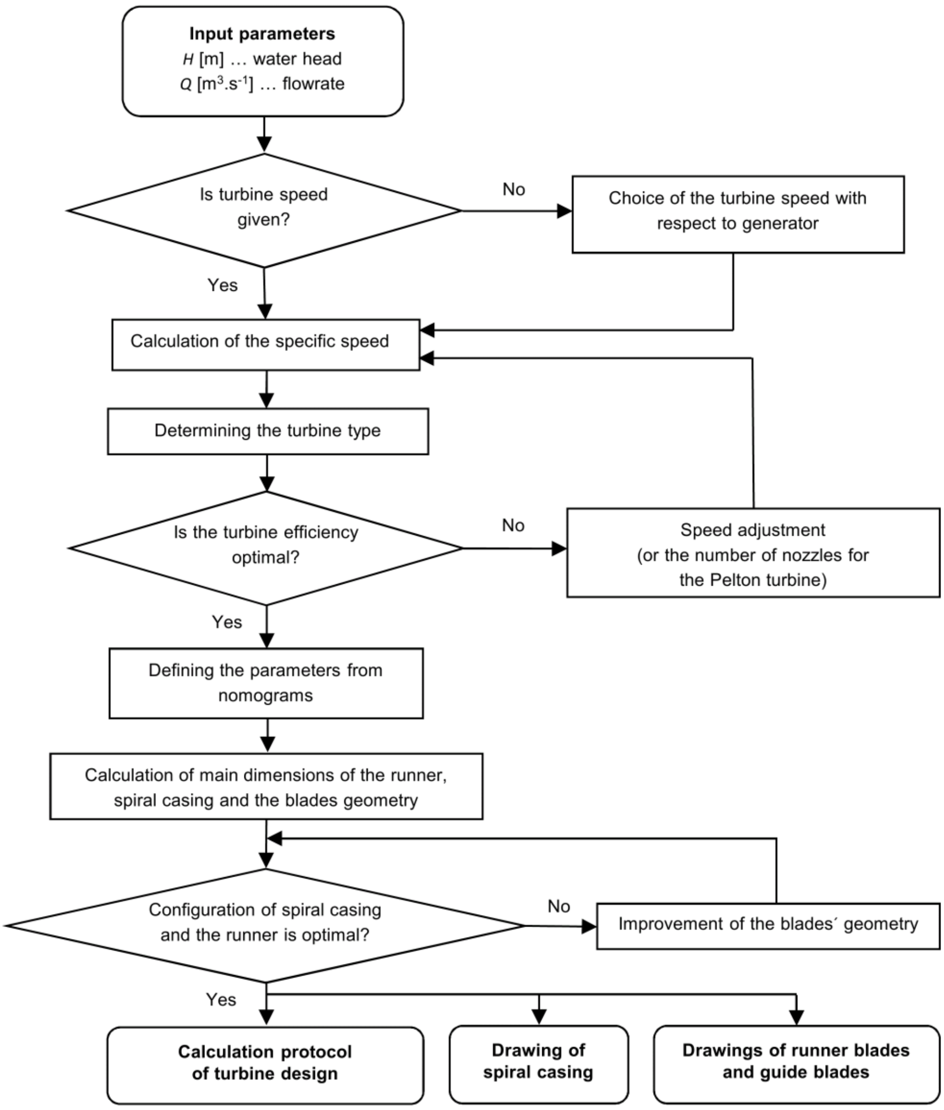


Fig. 2. – Simplified block diagram of the turbine design in the calculation program

Then main dimensions of the turbine runner are recalculated using geometric characteristics of a model turbine. For example the outside runner diameter D_1 [m] is calculated by:

$$D_1 = \left(\frac{Q}{Q'}\right)^{\frac{1}{2}} \cdot \left(\frac{1}{H}\right)^{\frac{1}{4}} \tag{3}$$

The value Q' [m^3.s^{-1}] (flowrate in a model/etalon turbine) is a tabulated value dependent on the specific speed n_q by the flow. Using the correlation analysis, this value and other tabulated values were converted into mathematical equations as polynomial functions of second-, third- or even fourth-degree. The program then calculates the equation which in terms of correlation analysis R^2 proved to be the most suitable. These equations form the basis of the calculation program on which the entire hydraulic design of individual components of the turbine depends. Fig. 3 shows an example of conversion of maximum efficiency values of Kaplan turbine depending on the specific speed n_q. The black line represents a curve of the tabulated values, the red one corresponds to values calculated from determined polynomial function used in the program:

$$\eta_T = -0.0000055 \cdot n_q^2 + 0.0014 \cdot n_q + 0.84 \tag{4}$$

Other design parameters of the turbine are determined using the formulas of fluid mechanics and geometry of the velocity triangles (Fig. 1). The following text, due to the extent of the paper, focuses only on the design of runner blades geometry. When designing the blades, it is possible to use similar principles like those used in the construction of wind power plants propellers. However, it is necessary to take into consideration a number of fundamental differences arising from different operating states, especially the possibility of cavitation BAHAJ ET AL. (2007).

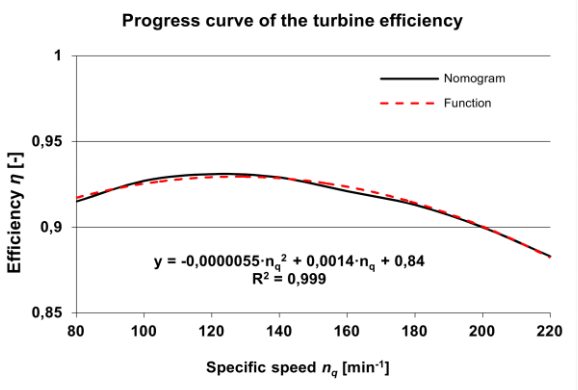


Fig. 3. – Sample of conversion into a mathematical function (PETIT ET AL., 2010)

For the construction of the blade, which is formed by generally curved surface, it is necessary to determine the velocity triangles in several cross-sections along the blade. This is based on the theorem of constant meridian velocity in the flow profile, which in case of the Kaplan turbine has axial direction. Meridian velocity c_m [m.s^{-1}] is determined:

$$c_m = w_m = \frac{Q}{S} \tag{5}$$

Meridian velocity c_m [m.s^{-1}] and w_m [m.s^{-1}] (Fig. 1) and the angular speed ω remain constant throughout the profile. Only the circumferential blade velocity u and the absolute velocity c resulting from it will change. On the basis of the turbine specific energy $Y_T = g{\cdot}H$, the projection of absolute velocity c_1 to direction of the circumferential velocity is determined from the Euler turbine equation (prerequisite being the vortex-free water output $c_{u2} = 0$):

$$c_{u\,i} = \frac{Y_T}{u_1} = \frac{g \cdot H}{u_1} \tag{6}$$

Then the input angle of absolute velocity α_1 [deg] is calculated:

$$tg\ \alpha_1 = \frac{c_m}{c_{u\,1}} \tag{7}$$

This determines the velocity triangle on the outside diameter D_1. From thus defined triangle the geometry of the runner blade is calculated (β_1 correspond to the blade angle on the leading edge and, β_2 the blade angle on the trailing edge), NECHLEBA (1962) and DRTINA ET AL. (1999).

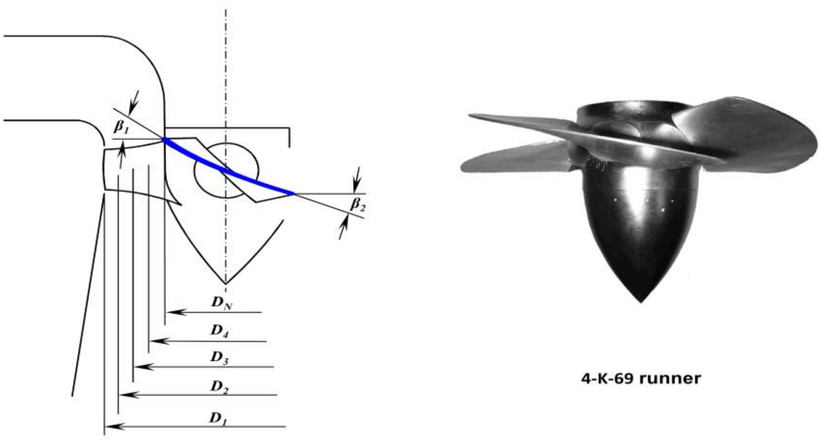
The blade angle β_1 [deg] at the inlet:

$$tg\ \beta_1 = \frac{c_m}{u_1 - c_{u1}} \tag{8}$$

The blade angle β_2 [deg] of the output will be, provided the same circumferential velocity ($u_1 = u_2$) and vortex-free output of water from the turbine:

$$tg\ \beta_2 = \frac{c_m}{u_1} \tag{9}$$

This determines the blade geometry on the outside diameter D_1. By analogy, the input and output blade angles (β_1 and β_2) on the diameters D_1 to D_N are determined – see Fig. 4a).



(a) (b)

Fig. 4. – a) Specification of cylindrical cross-sections for calculation of the blade geometry according POLÁK, (2013B), b) turbine runner 4-K-69

All numerical results of blade runner hydraulic design, including the velocity triangles rendered in real scale, are summarized in a calculation protocol. Fig. 5 shows an example of such a protocol developed for the standard 4-K-69 Kaplan turbine which is used to verify calculated values. Fig. 4b) shows a blade runner and Fig. 6 shows a plan of the entire 4-K-69 turbine.

Kaplan turbine

Input parameters

Water head	$h=$	3,7	m
Flowrate	$Q=$	0,073	$m^3.s^{-1}$
Gear ratio*	$p=$	2,17	-

* with respect to the generator speed N_G = 3 000 min^{-1}

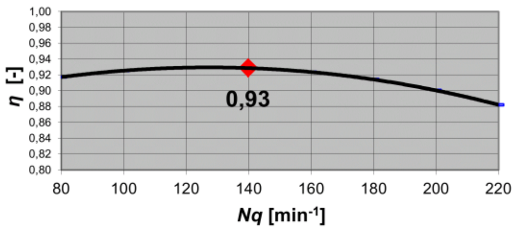
Runner design

Turbine speed	$N=$	1381	min^{-1}
Specific speed	$Nq=$	139,8	min^{-1}
Etalon flowrate	$Q'=$	1,234	$m^3.s^{-1}$
Outside runner diameter	$D_1=$	0,195	m
Channel width ratio	$B_o/D_1=$	0,37	-
Channel width	$B_o=$	0,080	m
Hub diameter ratio	$D_N/D_1=$	0,46	-
Hub diameter	$D_N=$	0,078	m
Angle of suction pipe	$\delta=$	8,7	°
Computed number of blades	$z'=$	3,3	ks
Number of blades	$z=$	4	ks
Blade spacing	$T=$	0,153	m
Relative spacing	$T/L=$	0,87	0,62 - 0,87
Blade lenght	$L=$	0,176	m
Clear opening	$S=$	0,025	m^2
Meridian velocity	$c_m=$	2,9	$m.s^{-1}$
Wheel angular speed	$\omega=$	144,6	s^{-1}
Blade speed on D_1	$u_1=$	14,1	$m.s^{-1}$
Specific head energy	$Y_t=$	36,3	$J.kg^{-1}$
Projection of c_1 to u_1	$C_{u1}=$	2,6	$m.s^{-1}$
Inlet angle of streamline	$\alpha_1=$	48,50	°
Outlet angle of streamline	$\alpha_2=$	90	°
Inlet blade angle	$\beta_1=$	14,18	°
Outlet blade angle	$\beta_2=$	11,67	°
Middle diameter of runner	$D_s=$	0,137	m
Blade-hub clearance	$r_N=$	0	mm
Inside runner diameter	$D_N=$	0,078	m
Turbine efficiency	$\eta=$	0,93	-
Real power output	$P=$	2,46	kW

Cavitation check

Barometric presure	$p_a=$	100	kPa
Pressure of saturated water	$p_d=$	4	kPa
Thoma cavitation coefficient	$\sigma=$	0,51	-
Max. suction height	$h_s=$	7,91	m

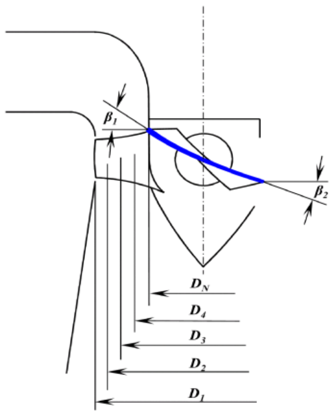
Max. turbine efficiency



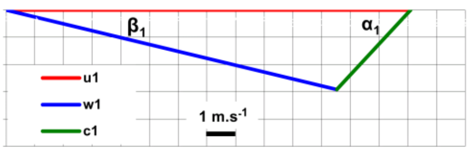
View of computed data - blade geometry

cut	D [m]	u_1 [$m.s^{-1}$]	c_{u1} [$m.s^{-1}$]	α_1°	α_2°	β_1°	β_2°
D_1	0,195	14,1	2,6	48,5	90	14,2	11,7
D_2	0,166	12,0	3,0	43,8	90	18,0	13,7
D_3	0,137	9,9	3,7	38,3	90	25,2	16,4
D_4	0,107	7,8	4,7	31,9	90	43,5	20,6
D_N	0,078	5,6	6,4	24,3	90	74,6	27,3

Sketch of turbine runner



Velocity triangle (D_1) - input



Velocity triangle (D_1) - output

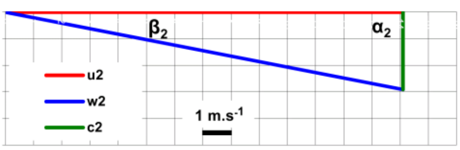


Fig. 5. – Calculation protocol in MS Excel for the standard 4-K-69 Kaplan turbine runner design

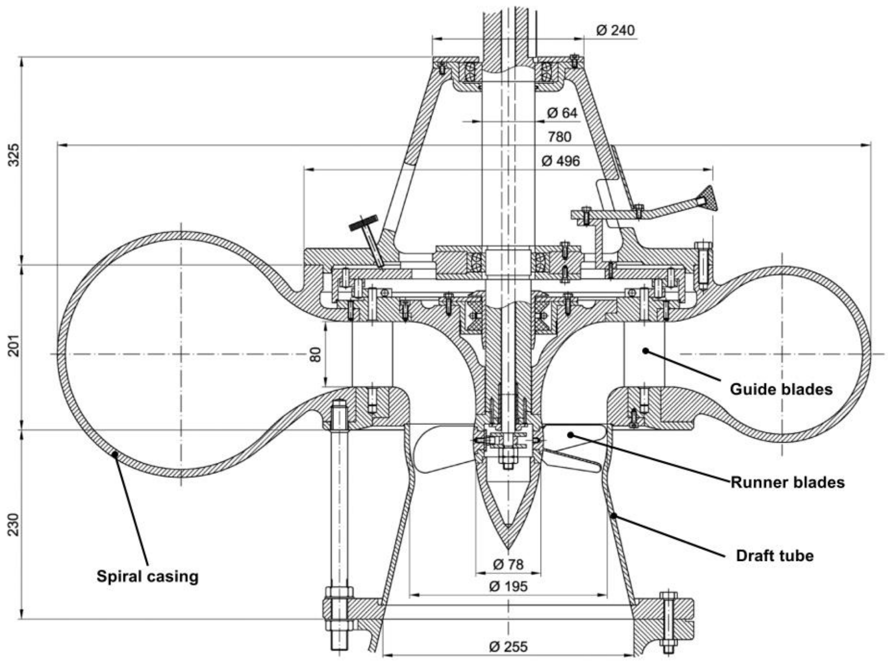


Fig. 6. – The standard 4-K-69 Kaplan turbine

Graphic extension of the program - rotor blades and guide blades

Graphic extension of the program builds on previous calculated angles β_1 and β_2 and renders in real scale longitudinal profiles of runner blades in several cross-sections. The assumption is that in case of structured flow, fluid streamlines in axial runner are distributed along cylindrical stream surfaces. The common axis of these surfaces is the axis of the turbine runner. The stream surfaces intersect runner blades in individual longitudinal profiles according MELICHAR ET AL. (1998). By stretching the particular cylindrical stream surface the flat grid of blade profiles appear, run by the flat flow field as seen in Fig. 7.

The program divides the turbine clear opening by five coaxial cylindrical cross-sections. Their diameters are shown in Fig. 4a. On the stretched cylinder surfaces thus formed are then deposited centerline profiles of the blades. The centerlines are not straight, but curved – due to different angles on the leading and trailing edge β_1 and β_2. The program calculates together with the length of the blade profile the mathematical function of the smooth transition curve between the leading and trailing edge in response to both tangents. The result of this is a centerline which is then „wrapped“ by streamline airfoil that is again converted into a mathematical function. The program works with a NACA 0015 streamline airfoil from BRITO ET AL. (2002) which is, in case of need, possible to change. In the above-mentioned procedure the cross-sections of blade grids are constructed in all five cylindrical stream surfaces.

In addition to displaying the basic position of the blades, the graphic extension allows to „tilt“ them to any angle. It shows the opening/closing of the turbine blades at different operating conditions. The essence of tilt consists again in mathematical description of blades profiles which the program operates with. Fig. 7 shows an example of the standard 4-K-69 turbine graphic output of a hydraulic design of runner blades in two cylindrical cross-sections – on the diameters D_4 and D_3 (see Fig. 4a). Cross-sections are displayed in a grid division of 10 mm. Based on the above-mentioned principles, the blade grid of guide blades are modelled.

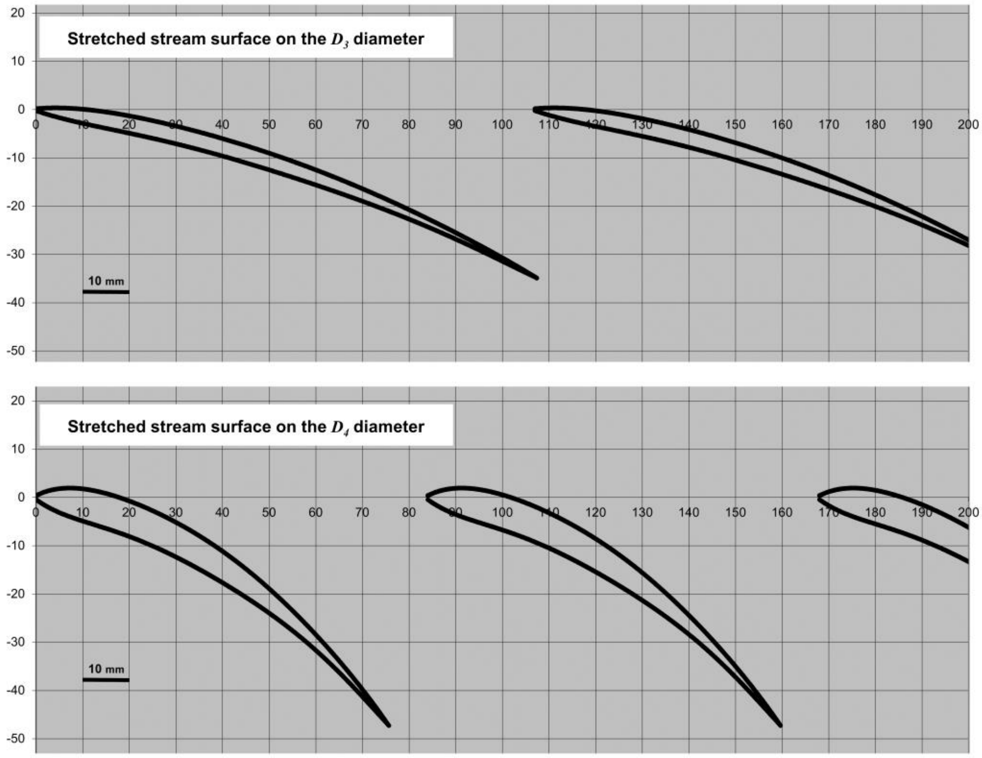


Fig. 7. – Graphic output of the program for the rotor blades design (POLÁK ET AL., 2013)

Graphic extension of the program – spiral casing

Another calculation program extension allows to solve a hydraulic profile of the spiral casing. The calculation program uses one- and two-dimensional stream theory by prof. Kaplan. Nowadays, when the use of CFD models, thanks to its flexibility, has been increasingly popular, such a procedure, based on the classical theory, seems to be a step backwards, PETIT ET AL. 2010). Nevertheless, CFD models require a specialized program and considerable demands on its operation NILSSON ET AL. (2003). These circumstances make the use of CFD model complicated and it hinders its wider usage. Therefore, the calculation program uses more available one- and two-dimensional theory. This is based on the law of constant circulation according to MUNSON ET AL. (2006), NECHLEBA (1962):

$$R_i \cdot c_{u\,i} = K \tag{10}$$

The constant K [m^2.s^{-1}] is determined by the circumferential velocity component c_{u1} on the inlet radius R_1 on the runner (see Fig. 1). On the assumption of a circular profile of a spiral, which is in practice the most common, the inner radius of the circular flow profile is given by:

$$r_i = \frac{\varphi \cdot Q}{720 \cdot \pi \cdot K} + \sqrt{2R_0 \frac{\varphi \cdot Q}{720 \cdot \pi \cdot K}} \tag{11}$$

The calculation program thus calculates total of 72 internal flow profiles over the entire circumference of the spiral casing (where φ [deg] is angular distance from inlet cross section – see Fig. 6a). In addition to numerical results, the graphic output as a radial cross-section (Fig. 8a) and transverse cross sections I - VI (Fig. 8b) is again available. This figure shows a calculated hydraulic profile of a spiral casing for the standard 4-K-69 Kaplan turbine. The indicated network shows the radial dimensions in meters and divides the spiral circumferentially of 5° to the individual calculated sections. This spiral casing is a result of using a one-dimensional stream theory which is more suitable for this case. The graphic output can be compared with the standard 4-K-69 turbine spiral casing in Fig. 8c.

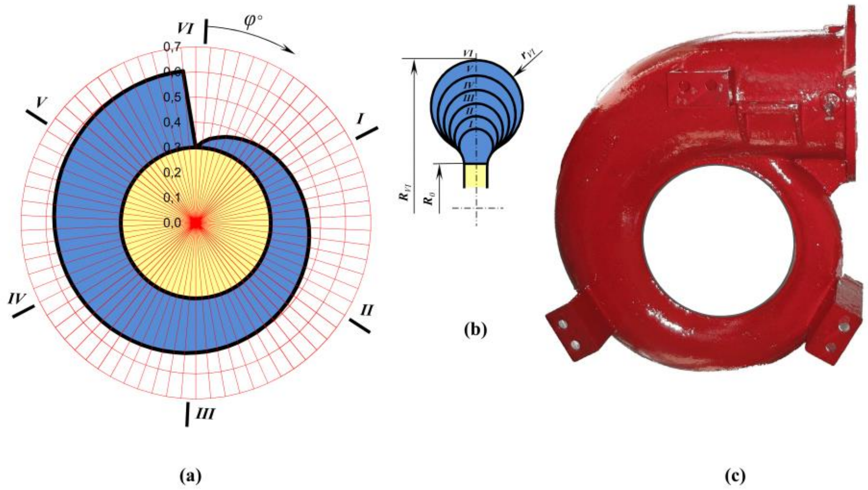


Fig. 8. – Graphic output of calculation program (a, b) and real 4-K-69 turbine spiral casing (c) (POLÁK ET AL., 2013)

RESULTS AND DISCUSSION

The comparison of the runner blade geometry was chosen for a more detailed verification of calculation program results. Numerical results of the calculation program are together with the standard 4-K-69 turbine runner blade (Fig. 4b) summarized in Tab. 1.

In the left part of the table there are the angles for the blade leading and trailing edge of the β_1 and β_2 calculated by the program. In the middle part of the table there are angles of the standard 4-K-69 turbine blade β^*_1 a β^*_2. On the right there are the differences of corresponding values.

Tab. 1. – Comparison of calculated results and geometry of the standard 4-K-69 Kaplan turbine blades

Cross section	Calculated		Standard		Difference of Values	
D [m]	Leading Edge β^*_1	Trailing Edge β^*_2	Leading Edge β_1	Trailing Edge β_2	Leading Edge $\beta^*_1 - \beta_1$	Trailing Edge $\beta^*_2 - \beta_2$
$D_1 = 0.195$	14.2°	11.7°	14°	11°	0.2°	0.7°
$D_2 = 0.166$	18°	13.7°	19°	13°	-1°	0.7°
$D_3 = 0.137$	25.2°	16.4°	28°	17°	-2.8°	-0.6°
$D_4 = 0.107$	43.5°	20.6°	44°	18°	-0.5°	2.6°
$D_N = 0.078$	74.6°	27.3°	72°	32°	2.6°	-4.7°

Fig. 9 shows the dependence of the average angle of the blade setting $(\beta_1 + \beta_2)/2$ on the diameter of the runner, i.e. the position of each cut. The red dashed curve indicates the calculated course of middle-angle setting, the blue line is the course of real values.

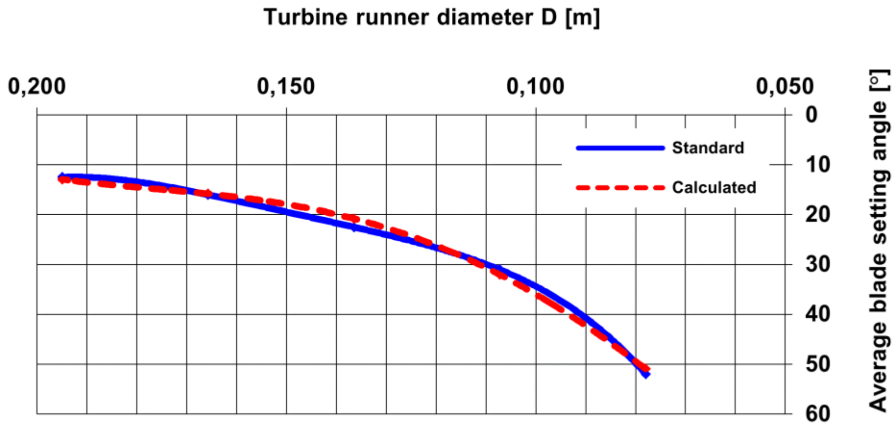


Fig. 9. – The course of the blade average setting angle across cutting surfaces

CONCLUSIONS

Tab. 1 and Fig. 9 show the comparison of calculated values and the reality of runner blade geometry. There is only one case where, in the absolute value, the difference between the calculated and actual angle is greater than 3°, i. e. 10% of the true value (the angle of the trailing edge to the diameter D_N). The difference is probably caused by other operating conditions (cavitation characteristics of runner, etc.) resulting from specific laboratory tests, which calculation program does not include.

In the case of the spiral casing, the difference between the inner diameter at the inlet into a spiral and the calculation (D_0 = 306 mm) is only 2%. As for other dimensions, the differences are not greater than 4%.

In case of other calculated parameters (main dimensions of the runner, guide blades, draft tube, suction height, width of guide blades, etc.), the differences between the calculated values and the real parameters of the standard 4-K-69 Kaplan turbine are always reliably smaller than 10% (compare the calculation protocol Fig. 5 and the turbine drawing Fig. 6).

From the point of view of used methods, the above mentioned results can be considered satisfactory. The essential advantage of the calculation program is a significant simplification of the whole process of turbine design along with graphic outputs of key parts. Thanks to its user accessibility it is a suitable tool for designing the basic parameters of conventional blade water turbine types, especially for the low-potential hydropower resources.

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