Classic design of a kaplan turbine runner wheel

Dimensionamento clássico de um rotor hidráulico do tipo kaplan

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ABSTRACT
The Brazilian electric matrix is based on renewable sources of energy, and most of it comes from hydroelectric power plants, its structure is composed of a dam, a capture system, a spillway, and a powerhouse, in this installation are the water turbines The main types are Francis, Bulbo, Helix, Kaplan, and Pelton each one has particularity for dimensioning. In this context, the objective of the article is to carry out the classic design of a Kaplan turbine runner wheel and the computational design. For this, first, the runner wheel case study was defined considering those installed in Brazil, using as criteria the power energy efficiency. Later, based on the input data: flow, fall height, and runner rotation, the calculation memorial was developed, preliminary calculations to define the specific rotation, complementary calculations to determine the minimum runner flow, number of poles, specific design rotation, an outer diameter of the runner blade, hub diameter, the height of the distributor, solid shaft diameter, fluid flow velocity, profile lengths, and profile thickness. The Hydroelectric Power Plant case study was Luis Eduardo Magalhães, installed on the Tocantins River in the state of Tocantins, has five generator units with Kaplan runners of five blades and power of 180.5 MW each, the flow rate of 700 m³/s, drop height of 29 m and 100 revolutions per minute. The values obtained were organized in electronic spreadsheets and later the computational design of the runner was performed using CAD software. The dimensioned runner wheel has dimensional and power values similar to the case study. Thus, it is concluded that the application of the

classic dimensioning allowed the design of the runner wheel containing five blades and power of 183.4 MW, and the computational design allowed us to visualize the similarities with the case study.

Keywords: hydroelectric plants, flow machines, mechanical design.

RESUMO
A matriz elétrica brasileira é baseada em fontes renováveis de energia, sendo a maior parte proveniente de usinas hidrelétricas na qual a sua estrutura é composta basicamente por barragem, sistema de captação da água, vertedouro e a casa de força onde estão as turbinas hidráulicas. Os principais tipos de turbinas hidráulicas são Francis, Bulbo, Hélice, Pelton e Kaplan. Cada uma apresenta particularidade para o dimensionamento. Neste contexto, o objetivo do artigo é realizar o dimensionamento clássico de um rotor hidráulico do tipo Kaplan e o desenho computacional do rotor. Para isso, primeiramente foi definido o rotor estudo de caso considerando as instaladas no Brasil, utilizando como critério o rendimento energético de potência. Posteriormente, baseado nos dados de entrada: vazão, altura de queda e rotação do rotor, foi desenvolvido o memorial de cálculo, cálculos preliminares para definição da rotação específica, cálculos complementares para determinação da vazão mínima do rotor, número de polos, rotação específica de projeto, diâmetro externo da pá do rotor, diâmetro do cubo, altura do distribuidor, diâmetro do eixo maciço, velocidade de escoamento do fluído, comprimentos do perfil e espessura do perfil. A Usina Hidrelétrica estudo de caso foi a Luis Eduardo Magalhães, instalada no rio Tocantis no estado de Tocantis no Brasil, possui cinco unidades geradoras com rotores do tipo Kaplan de cinco pás e potência de 180,5 MW cada, vazão de 700 m³/s, altura de queda de 29 m e 100 rotações por minuto. Os valores obtidos foram organizados em planilhas eletrônicas e posteriormente foi realizado o desenho computacional do rotor utilizando software CAD. O rotor dimensionado apresenta valores dimensionais e de potência similares ao estudo de caso. Desta forma, conclui-se que a aplicação do dimensionamento clássico permitiu a elaboração do projeto do rotor contendo cinco pás e uma potência de 183,4 MW e o desenho computacional permitiu visualizar a similaridade com o rotor estudo de caso.

Palavras-chave: usinas hidrelétricas, máquinas de fluxo, projeto mecânico.

1 INTRODUCTION

In 2000 in Brazil electricity consumption was 27,2% in the residential sector, 42,7% in the industrial sector, 15,5% in commerce, and 14,6% in others. In 2016 consumption per class was 28,9% residential, 35,6% industrial, 19,2% commercial and 16,3% other. This represents an increase of 6,25% in the residential class, 23,87% in the commercial class, and 11.64% in others, while the industrial class presented a decrease of 16,6% in the participation relative to the period from 2000 to 2016. By 2026, the consumption of the commercial and residential sectors will continue to gain share in total consumption, while the industrial sector will lose share, with the result that the estimated structure of electricity consumption by class in 2026 will be 29,7% for residential, 19,9% commercial, 33,5% industry and 17,0% other (EPE, 2017).

According to the National Electrical Energy Agency (ANNEL, 2019), in Brazil non-renewable energy sources are responsible for 12,9%, while renewable sources generate 87,1% of the country's energy. Of this, 80,4% come from the hydraulic energy of the operation 217 Hydroelectric Power Plants (HPP's), 426 Small Hydroelectric Plants (SHP's), and 698 Hydroelectric Generating Plants (CGH's).

In Brazil, there are several Hydroelectric Plants that use turbines with Kaplan runner wheel, among them, the Passo Real Hydroelectric Plant, located on the Jacuí River, in the municipality of Salto do Jacuí (Rio Grande do Sul), containing two turbines with 79 MW each. The Ferreira Gomes Hydroelectric Power Plant, located on the Araguarí River in the municipality of Ferreira Gomes (Amapá), with three 84 MW turbines. The Estreito Hydroelectric Power Plant, located on the Tocantins River in the municipality of Estreito (Maranhão), has eight Kaplan type 135 MW turbines. The Peixe Angical Hydroelectric Plant, located on the Tocantins River, between the municipalities of Peixe, São Salvador do Tocantins and Paranã (Tocantins), with three 166MW turbines. The Aimorés Hydroelectric Power Plant, located on the Rio Doce in the municipality of Aimorés (Minas Gerais) on the border with Espírito Santo, contains three turbines with 110 MW of power each unit. The Porto Estrela Hydroelectric Power Plant, located on the Santo Antônio River in the municipality of Joanésia (Minas Gerais), which has two 56 MW turbines. And, the Lajeado Hydroelectric Power Plant, located on the Tocantins River, between the municipalities of Lajeado and Miracema do Tocantins (Tocantins), with five 180,5 MW turbines per unit, is the one with the highest nominal power per turbine among those mentioned (ANA, 2014).

According to Viana (2010), the five main parts of a hydraulic turbine are (i) spiral casing; (ii) pre-distributor; (iii) distributor; (iv) runner wheel and shaft, being this part the focus of the study of this article; and (v) suction pipe, which is an outlet pipe that slows down the flow after passing through the turbine, returning it to the river.

According to Souza (2011a) and Henn (2019), the flow machine itself can be defined as an energy transformer, necessarily using mechanical work as one of the forms of energy, in which the fluid passes through the machine and interacts with a rotating element, in this case, the runner, consequently performing mechanical work. So, water turbines are flow machines designed to work with water as a working fluid, when it passes through the machine it promotes rotation on the axis from the kinetic energy and potential present in the flow (HENN, 2019).

The turbine, more specifically the runner, is designed to meet the design specifications of each hydroelectric plant. Thus, each hydroelectric power plant is unique and requires a properly designed runner that provides the best performance (KAVURMACI et al., 2017).

Given this, it is a challenge for the design engineer to carry out the design of the hydraulic runners. For this, there are improved methods of calculations that make it possible to obtain results for the dimensioning, as proposed by Macintyre (1983), Souza (2011b and 2011c), and Henn (2019). These procedures present a high degree of complexity and require decision making in sizing by the design engineer, and may go as far as to result in more than one possible solution for the design of flow machines (HENN, 2019). In this case the Kaplan type hydraulic runners.

In this context, there are several studies in the literature on the dimensioning and analysis of hydraulic runners, mainly Francis, Pelton, and Kaplan type. With emphasis, Almeida et al., (2008), Santos (2012), Bozz (2012), Ost and Kraulich (2013), Soares Júnior (2013), Macedo (2015), Araujo Filho et al. (2016), Magri Júnior (2016), Hantsch and Abeykoon (2017), Lima (2018), Bottlender (2019) and Conte (2020). In these works, the complexity of the dimensioning and analysis of the hydraulic runners is highlighted. Due to the complexity of the hydraulic runner's projects, computational techniques have been used for fluid dynamics, the application of these simulation tools has assisted in the optimization of the runner's projects as highlighted in the works of Nennemann and Vu (2007), Janjua, Khalil and Saeed (2013), Motycak, Skotak, and Obrovsky (2010) and Kumar and Bhingobe (2015).

For Souza (2011b) the steps for the design of axial hydraulic turbine, in this group are the Kaplan turbine, are divided into five: (i) preliminary or initial sizing: aims to establish the physical limits of the flow, also known as the hydraulic flow path; (ii) basic dimensioning: elaborates the sketches of the hydraulic runner wheel, defines the desired characteristics and the geometry adopted with the equations resulting from the principles of mechanics, through the application of equations; (iii) manufacture of the prototype; (iv) tests and (v) manufacturing. In this article, the focus is on the application of steps (i) and (ii).

In this sense, the article aims to perform the sizing of a Kaplan-type hydraulic turbine and the computational drawings, using as a case study a Hydroelectric Power Plant installed in Brazil.

2 MATERIAL AND METHODS

The Hydroelectric Power Plant selected as a case study is Luis Eduardo Magalhães, located in the state of Tocantins, this enterprise has the highest nominal power among the Hydroelectric Power Plants with Kaplan turbine in Brazil. There are five generating units, each with an output of 180,5 MW. The initial data used for sizing are, the flow rate of 700 m³/s, drop height of 29 m, and 100 revolutions per minute.

For the dimensioning, steps (i) and (ii) were used for Souza's (2011b) axial hydraulic turbine project.

(i) Preliminary sizing: First the type of runner wheel to be sized was determined, for this, equations (1) and (2) were used.

$$n_{qa} = 10^3 n \frac{Q^{1/2}}{(Y)^{3/4}} \quad (1)$$

At where:
n_{qa} = specific rotation speed (dimensionless)
n = runner rotation (rps)
Q = project flow (m³/s)
Y = specific energy jump (J/kg)

$$Y = g\,h \quad (2)$$

At where:
g = gravity (9,81m/s^2)
h = design height (m), design height 29 m.

(ii) Basic sizing: In this step was the sizing of the blade runner wheel, for this, the following data were considered: Torsion $\tau_{t_{ad}} = 200\ (kgf/cm^2)$, volumetric yield $\eta_v = 0{,}99\ (dimensionless)$, internal yield $\eta_i = 0{,}94\ (dimensionless)$, nominal yield $\eta_m = 0{,}99\ (dimensionless)$, as recommended by Souza (2011b).

$$Q_i = 0{,}25\ \mathrm{Q} \tag{3}$$

At where:
Q_i = minimum runner flow (m³/s)

$$Q_{r_{\frac{1}{1}}} = \eta_v\,\mathrm{Q} \tag{4}$$

At where:
$Q_{r_{1/1}}$= flow in the runner to the distributor all open (m³/s)
η_v = volumetric performance (0,99 dimensionless)

$$z_p = \frac{3600}{n} \tag{5}$$

At where:
z_p= number of pole pairs of the electric generator (dimensionless)

$$Q_r = 0{,}62\ Q_{r_{1/1}} \tag{6}$$

At where:
Q_r = design flow for Kaplan axial turbine with mobile distributor (m³/s).

$$\mathrm{n}_{qA_r} = 3\ \mathrm{n_r}\ \frac{{Q_r}^{0{,}5}}{\mathrm{H}^{0{,}75}} \tag{7}$$

At where:
n_{qA_r}= project specific rotation (dimensionless)

$$\mathrm{D} = 9{,}64\ \frac{H^{0{,}112}{Q_r}^{0{,}269}}{{n_r}^{0{,}483}} \tag{8}$$

At where:
D = External diameter of blade (m)

$$D_c = \left(0{,}297 + \frac{68{,}19}{\mathrm{n}_{qA_r}}\right) D \tag{9}$$

At where:
D_c = Hub Diameter (m)

$$b_o = \left(0{,}449 - \frac{34{,}026}{n_{qA_r}}\right) D \tag{10}$$

At where:
b_o = distributor height (m)

$$P_e = 9{,}81\, Q_{r_{\frac{1}{1}}}\, H\, \eta_i\, \eta_m \tag{11}$$

At where:
P_e = power on TH axis (kW)
η_i = internal performance (dimensionless)
η_m = nominal yield (dimensionless)

$$d = 0{,}7914 \sqrt[3]{\frac{P_e}{\tau_{t_{ad}}\, n_r}} \tag{12}$$

At where:
d = solid shaft diameter (m)
$\tau_{t_{ad}}$ = Torsion (kgf/cm^2)

$$z_r = 19{,}993 - 4{,}932.10^{-2}\, \mathrm{n}_{qA_r} + 4{,}65\,.\,10^{-5}\, \mathrm{n}_{qA_r}{}^2 - 1{,}412.10^{-8}\, \mathrm{n}_{qA_r}{}^3 \tag{13}$$

At where:
z_r = number of runner blades (dimensionless)

$$c_m = \frac{4\, \mathrm{Q}_r}{\pi\left(D^2 - D_c{}^2\right)} \tag{14}$$

At where:
c_m = theoretical southern speed (m/s)

$$u_c = \frac{\pi\, D_c\, \mathrm{n}_r}{60} \tag{15}$$

At where:
u_c = tangential velocity not cube (m/s)

$$U_c = \frac{\mathrm{g}\, H\, \eta_i}{2\, u_c} \tag{16}$$

At where:
U_c = half of the difference of the absolute velocity projections in the tangential direction of the cube (m/s)

$$\beta_{\infty_c} = \text{arctg}\,\frac{c_m}{u_c - U_c} \tag{17}$$

At where:
β_{∞_c} = angle that the direction of the undisturbed relative velocity of the flow W_∞ makes with the direction in the hub (degrees)

$$W_{\infty_c} = \frac{c_m}{sen\,\beta_{\infty_c}} \tag{18}$$

At where:
W_{∞_c} = undisturbed relative velocity of flow in the runner near the hub (m/s)

$$L_c = 3{,}173\left(1{,}689 + 2{,}368.10^{-3}\,\mathrm{n}_{qA_r} - 7{,}235.10^{-6}\,\mathrm{n}_{qA_r}{}^2 + 4{,}302.10^{-9}\,\mathrm{n}_{qA_r}{}^3\right)\frac{D_c}{Z_r} \tag{19}$$

At where:
L_c = length of the profile rope next to the cube (m)

$$C_{sc} = \frac{1177{,}2\ \mathrm{H}\,\eta_i}{z_r\,n_r\,W_{\infty_c}\,L_c} \tag{20}$$

At where:
C_{sc} = Coefficient of support of the profile next to the cube, without considering the drag (dimensionless).

From equation 21 to 52, the other characteristics of the runner wheel were calculated in three different profiles. The first with D_j = D_C = 3,37 m, that is, the diameter of the runner wheel hub, the second with D_j = D_m = 5,56 m, this being the diameter of the average profile, and finally, the third with D_j = D_e = 7,76 m, resulting in the profile of the outer diameter.

$$D_j = D_{j-1} + (j-1)\frac{D - D_C}{j-1} \tag{21}$$

At where:
D_j = Diameters of the cylindrical sections (m);

$$u_j = \frac{\pi\,n_r\,D_j}{60} \tag{22}$$

At where:
u_j = Tangential speeds (m/s);

$$U_j = \frac{4{,}905\ H}{u_j\ \eta_i} \tag{23}$$

At where:
U_j = Components resulting from c_j in direction u (m/s);

$$\beta_{\infty_j} = \text{arct}\ \frac{c_m}{(u-U)_i} \tag{24}$$

At where:
β_{∞_j} = angle between speeds W_{∞_c} e u_j (degrees);

$$W_{\infty_j} = \frac{c_m}{sen\ \beta_{\infty_j}} \tag{25}$$

At where:
W_{∞_j} = Relative speeds of flow in the runner (m/s);

$$t_j = \frac{\pi\ D_j}{Z_p} \tag{26}$$

At where:
t_j = Steps (m);

$$L_j = \frac{\pi\ D_j\ cos\beta_{\infty_j}}{D_C\ cos\beta_{\infty_j}} \tag{27}$$

At where:
L_j = Length of the ropes of the profiles (m);

$$e_{cg_c} = 0{,}22\ (\mathrm{D} - D_C)\ \sqrt{\frac{\mathrm{H}}{\sigma_{fad}}\left(1 + \frac{L}{L_c}\right)} \tag{28}$$

At where:
e_{cg_c} = Thickness of the profile next to the cube in its center of gravity (m);

And the thickness of the profile at the end:

$$e_{cg} = e_{cg_c}\ 0{,}33 \tag{29}$$

$$e_{cg_j} = \left(\mathrm{D} - 0{,}33\ D_C - 0{,}67\ D_J\right) \frac{e_{cg_c}}{\mathrm{D} - D_C} \tag{30}$$

At where:
e_{cg_J} = Thickness of other profiles (m);

For GÖ 480 profile:

$$f_{e_c} = \frac{(e_{cg}/L)_c}{0{,}112} \tag{31}$$

For GÖ 428 profile:

$$f_{e_e} = \frac{(e_{cg}/L)_e}{0{,}08} \tag{32}$$

Being the index and corresponding to the diameter D;

At where:

f_{e_e} = Coefficient of thickening or thinning of the profile next to the hub (dimensionless);

For GÖ 480 profile:

$$f_{e_{480}} = \frac{(e_{cg}/L)_{480}}{0{,}112} \tag{33}$$

For GÖ 428 profile:

$$f_{e_{428}} = \frac{(e_{cg}/L)_{428}}{0{,}08} \tag{34}$$

At where:
$f_{e_{480}}$ e $f_{e_{428}}$= Coefficient of thickening or thinning of the other profiles (dimensionless);

For GÖ 480 profile:

$$Y_{cg_c} = 0{,}5\ e_{cg_c} + 0{,}0088.\ f_{e_c}\ L_c \tag{35}$$

For GÖ 428 profile:

$$Y_{cg_e} = 0{,}5\ e_{cg_e} + 0{,}0038\ \ f_{e_e}\ L_e \tag{36}$$

Being the index and corresponding to the diameter D.

At where:
Y_{cg_c} e Y_{cg_e}= The ordering of the profiles next to the hub and at the end in their respective centers of gravity (m);

$$Y_{cg_j} = \frac{Y_{cg_c}\ D - Y_{cg_e}\ D_c - \left(Y_{cg_c} - Y_{cg_e}\right) D_c}{D - D_c} \tag{37}$$

At where:
Y_{cg_j}= Sorting of other profiles in their centers of gravity (m);

$$L_{cg_j} = 0{,}3875\ L_j \tag{38}$$

At where:
L_{cg_j}= Distance from the attack edge of each profile to its center of gravity (m);

$$L_{máx_j} = 0{,}3\ L_j \tag{39}$$

At where:
$L_{máx_j}$= Distance from the leading edge of each profile to its maximum order (m);

For GÖ 480 profile:

$$Y_{máx_c} = 1{,}0463\ e_{cg_c} + 0{,}011\ f_{e_c}\ L_c \tag{40}$$

For GÖ 428 profile:

$$Y_{máx_e} = 1{,}0313\ e_{cg_e} + 0{,}003\ f_{e_e}\ L_e \tag{41}$$

Being the index and corresponding to the diameter D.

At where:
$Y_{máx_c}$ e $Y_{máx_e}$= Maximum ordering of the profiles next to the hub (m);

$$Y_{máx_j} = \frac{Y_{máx_c}\, D - Y_{máx_e}\, D_c - \left(Y_{máx_c} - Y_{máx_e}\right) D_c}{D - D_c} \tag{42}$$

At where:
Y_{cg_j}= Maximum order of the other profiles (m);

$$\left(\frac{Y_{máx}}{L}\right)_j \tag{43}$$

At where:

$\left(\frac{Y_{máx}}{L}\right)_j$ = Relationship between the maximum ordinates and the respective string lengths of the profiles (dimensionless);

$$C_{s_j} = \frac{120\ g\ \mathrm{H}\ \eta_i}{z_r\ n_r\ W_{\infty_j}\ L_j} \tag{44}$$

At where:

C_{s_j}= Coefficient of support not considering drag (dimensionless);

$$\left(\frac{\mathrm{t}}{L}\right)_j \tag{45}$$

At where:

$\left(\frac{\mathrm{t}}{L}\right)_j$ = Relationship between the steps and the lengths of the profile strings (dimensionless);

$$\left(\frac{C_s^*}{C_s}\right)_j = 1 + \left(1{,}34\frac{\mathrm{t}}{L}\right)_j \left[tg(55{,}131 - 0{,}29\,\beta_\infty - 0{,}0881\,\beta_\infty^2 + 0{,}00122\,\beta_\infty^3)\right]_j \tag{46}$$

At where:
$\left(\frac{C_s^*}{C_s}\right)_j$ = Relations between the support coefficients of the profiles considering the drag and the influence of one shovel over the other and considering only the drag (dimensionless);

$$C^*_{sj} = \left[\left(\frac{C^*_s}{C_s}\right) C_s\right]_j \tag{47}$$

Note the following limitations, to:
For GÖ 480 profile – 0,60 < C^*_s <1,20;
For GÖ 428 profile – 0,10 < C^*_s <0,80;
At where:
C^*_{sj} = Coefficients of support considering the two effects (dimensionless);

$$\delta_j = 10{,}8696 \left[C^*_s - 4{,}8 \left(\frac{Y_{máx}}{L}\right)_j\right] \tag{48}$$

At where:
δ_j = Angle of attack (graus);

$$\beta_j = (\beta_\infty - \delta)_j \tag{49}$$

At where:
β_j = Angle that the ropes of the profiles form with the horizontal (degrees);

$$\alpha_{\infty_j} = arctg\left(\frac{C_m}{U_j}\right) \tag{50}$$

At where:
α_{∞_j} = Undisturbed flow angles at the runner inlet which will be used to fix the outlet angle of the distributor blades at the design point, and in the space between the outlet of the distributor blades and the inlet of the runner blades the potential vortex should be observed (degrees);

$$\varphi_m = \frac{2\, C_m}{u_c + u} \tag{51}$$

$$\varphi_m = 0{,}383$$

At where:
φ_m = Average speed coefficient (dimensionless);

$$\eta_r = \frac{1 - \dfrac{0{,}01\ a\ 0{,}04}{\varphi_m}}{1 + (0{,}01\ a\ 0{,}04)\ \varphi_m} \tag{52}$$

$$\eta_r = 0{,}8821$$

At where:
η_r = Average runner yield not considering the losses close to the inner diameter 0,01 < ε_m<0,04 ((dimensionless);

The values resulting from the equations were organized in tables and later the computational drawings were performed.

3 RESULTS AND DISCUSSIONS

The values resulting from step (i) Preliminary sizing are presented in Figure 1 by equations 1 and 2, while the values for equations 3 to 19 result from step (ii) Basic sizing.

Figure 1 – Resulting values for step (i) and (ii).

Eq.1	Eq.2	Eq.3	Eq.4	Eq.5	Eq.6	Eq.7	Eq.8	Eq.9	Eq.10
--	J/kg	m³/s	m³/s	--	m³/s	--	m	m	m
637,85	284,49	175,00	693,00	36,00	429,66	497,60	7,76	3,37	2,95
Eq.11	Eq.12	Eq.13	Eq.14	Eq.15	Eq.16	Eq.17	Eq.18	Eq.19	
MW	m	---	m/s	m/s	m/s	graus	m/s	m	
183,46	1,65	5,00	11,17	17,67	7,57	47,97	15,05	4,29	

As three current lines have been defined, in the inner crown (Dc), in the outer crown (De) and an average (Dm), Figure 2 shows the profile values for the respective sections.

Figure 2 – Results of profile calculations for sections Dc, Dm and De used in dimensioning.

Section	Eq.21	Eq.22	Eq.23	Eq.24	Eq.25	Eq.26	Eq.27
	m	m/s	m/s	graus	m/s	m	m
D_c	3,37	17,64	7,57	47,99	15,04	2,64	4,29
D_m	5,56	29,14	4,58	24,47	26,98	4,37	7,09
D_e	7,76	40,65	3,28	16,65	39,01	6,09	9,88
Section	Eq.30	Eq.31	Eq.38	Eq.39	Eq.42	Eq.43	Eq.44
	m	-	m	m	m	-	-
D_c	0,24	0,50	1,66	1,28	0,28	0,065	1,24
D_m	0,16	0,10	2,74	2,12	0,18	0,025	0,42
D_e	0,08	0,28	3,82	2,96	0,08	0,008	0,21
Section	Eq.45	Eq.46	Eq.47	Eq.48	Eq.49	Eq.50	Eq.51
	-	-	-	graus	graus	graus	-
D_c	0,62	0,62	0,79	5,15	42,83	55,86	0,383
D_m	0,62	1,17	0,49	3,97	20,49	67,68	0,383
D_e	0,62	1,44	0,30	2,80	13,84	73,60	0,383

For the design of the Kaplan turbine used the Göttingen profile - 428 for the external and medium chain line and the Göttingen profile - 480 for the internal chain line were adopted. In Figures 3, 4 and 5 the values of thickness and length are shown

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considering the rope of the profile Lj = 9,88 m for the external current line, Lj = 7,09 m for the medium current line and Lj = 4,29 m for the internal current line.

Figure 3 – Thickness and length of the external crown chain line.

De

L= 9,88

PERFIL GO 428	0	1,25	2,5	5	7,5	10	15	20	30	40	50	60	70	80	90	100
x	0	0,0125	0,025	0,05	0,075	0,1	0,15	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1
Ys	1,25	2,75	3,50	4,89	6,05	6,50	7,55	8,20	8,55	8,35	7,80	6,80	5,50	4,20	2,15	0,00
Yp	1,25	0,30	0,20	0,10	0,00	0,00	0,05	0,15	0,30	0,40	0,40	0,35	0,25	0,15	0,05	0,00
Ys - Yp	0,00	2,45	3,30	4,79	6,05	6,50	7,50	8,05	8,25	7,95	7,40	6,45	5,25	4,05	2,10	0,00
x final	0,00	0,12	0,25	0,49	0,74	0,99	1,48	1,98	2,96	3,95	4,94	5,93	6,92	7,90	8,89	9,88
e (m)	0,00	0,03	0,03	0,05	0,06	0,07	0,08	0,08	0,0864	0,08	0,08	0,07	0,05	0,04	0,02	0,00
e(mm)	0,00	25,66	34,56	50,16	63,36	68,07	78,55	84,31	86,40	83,26	77,50	67,55	54,98	42,41	21,99	0,00

Figure 4 – Thickness and length of the medium current line.

Dm

L= 7,091

PERFIL GO 428	0	1,25	2,5	5	7,5	10	15	20	30	40	50	60	70	80	90	100
x	0	0,0125	0,025	0,05	0,075	0,1	0,15	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1
Ys	1,25	2,75	3,50	4,89	6,05	6,50	7,55	8,20	8,55	8,35	7,80	6,80	5,50	4,20	2,15	0,00
Yp	1,25	0,30	0,20	0,10	0,00	0,00	0,05	0,15	0,30	0,40	0,40	0,35	0,25	0,15	0,05	0,00
Ys - Yp	0,00	2,45	3,30	4,79	6,05	6,50	7,50	8,05	8,25	7,95	7,40	6,45	5,25	4,05	2,10	0,00
x final	0,00	0,09	0,18	0,35	0,53	0,71	1,06	1,42	2,13	2,84	3,55	4,25	4,96	5,67	6,38	7,09
e (m)	0,0000	0,0543	0,0732	0,1063	0,1342	0,1442	0,1664	0,1786	0,1830	0,1763	0,1641	0,1431	0,1165	0,0898	0,0466	0,0000
e(mm)	0,00	54,35	73,20	106,25	134,20	144,18	166,36	178,56	183,00	176,35	164,15	143,07	116,45	89,84	46,58	0,00

Figure 5 – Thickness and length of the internal chain line

Di

L= 4,293

PERFIL GO 480	0	1,25	2,5	5	7,5	10	15	20	30	40	50	60	70	80	90	100
x	0	0,0125	0,025	0,05	0,075	0,1	0,15	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1
Ys	2,55	5,10	6,15	7,65	8,85	9,80	11,25	12,10	12,85	12,60	11,60	10,00	7,85	5,45	2,85	0,00
Yp	2,55	0,80	0,30	0,05	0,00	0,00	0,45	0,70	1,10	1,45	1,55	1,50	1,25	0,85	0,40	0,00
Ys - Yp	0,00	4,30	5,85	7,60	8,85	9,80	10,80	11,40	11,75	11,15	10,05	8,50	6,60	4,60	2,45	0,00
x final	0,00	0,05	0,11	0,21	0,32	0,43	0,64	0,86	1,29	1,72	2,15	2,58	3,01	3,43	3,86	4,29
e (m)	0,0000	0,1025	0,1394	0,1811	0,2109	0,2335	0,2574	0,2717	0,28000	0,2657	0,2395	0,2026	0,1573	0,1096	0,0584	0,0000
e (mm)	0,00	102,47	139,40	181,11	210,89	233,53	257,36	271,66	280,00	265,70	239,49	202,55	157,28	109,62	58,38	0,00

From the data in Figures 1, 2, 3, 4, and 5, the Kaplan type hydraulic runner wheel blade was computed. First, the inner (Dc), middle (Dm), and outer (De) crown diameters were drawn, as illustrated in Figure 6a. In Figures 6b, 6c, and 6d, on the drawing of the diameters Dc, Dm, and De, the drawing of the current lines were carried out considering the angles of Equation 49, present in Frame 2.

Then in Figure 6e the profile Göttingen - 428 for the external current line (De – 7765 mm) and medium (Dm – 5567 mm) and Göttingen - 480 for the hub current line (Dc – 3370 mm) was drawn. The shovel design was completed by joining the profiles of the current lines, obtaining a solid element as shown in Figure 1f. Finally, Figure 2 shows the runner wheel blade with the inlet and outlet edge of the fluid.

Figure 6 – Roadmap for the computational design of the runner wheel blade.

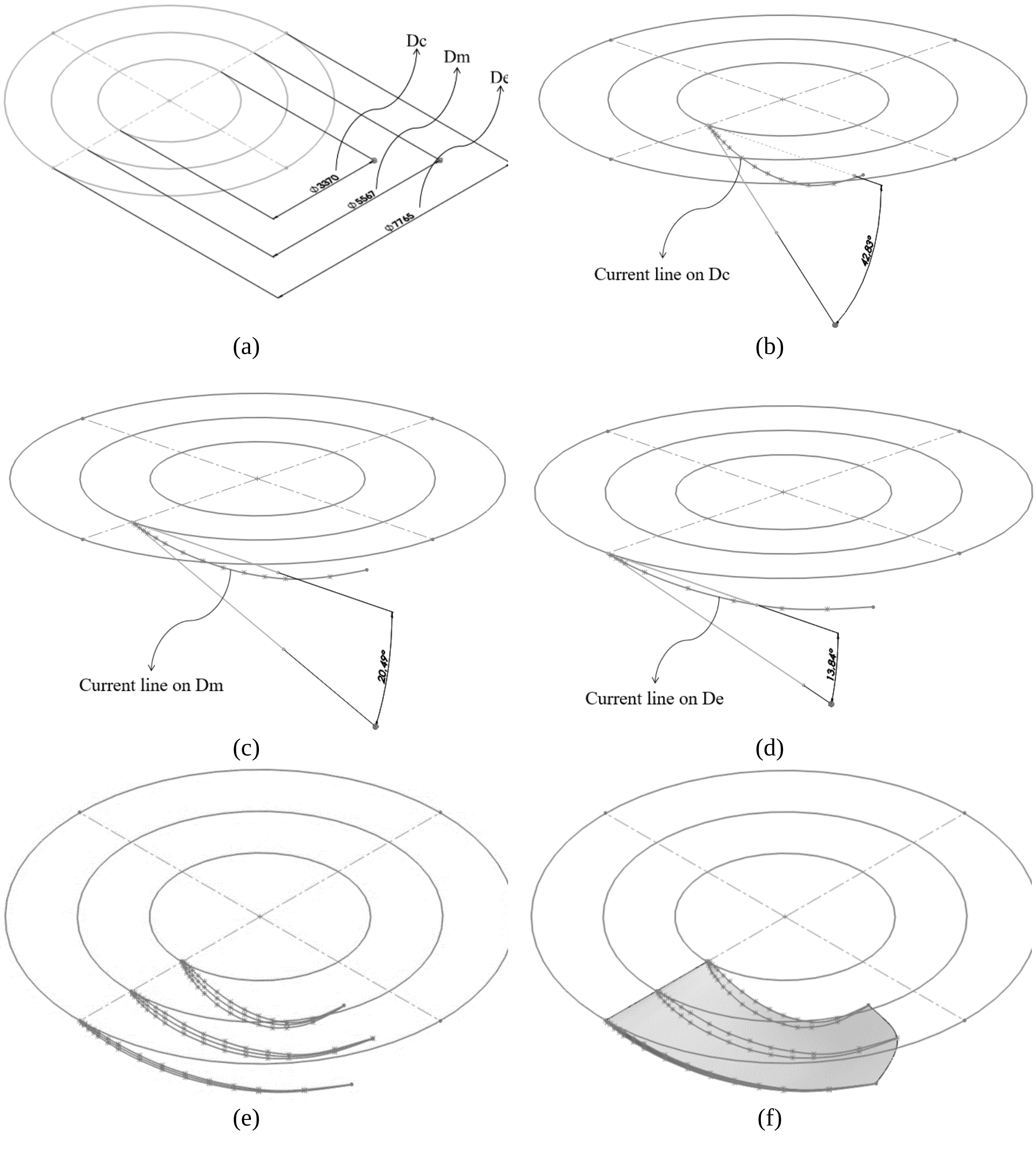
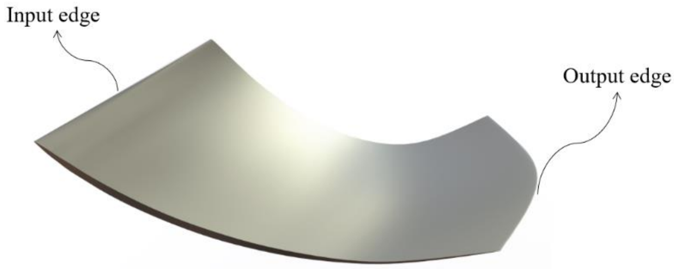
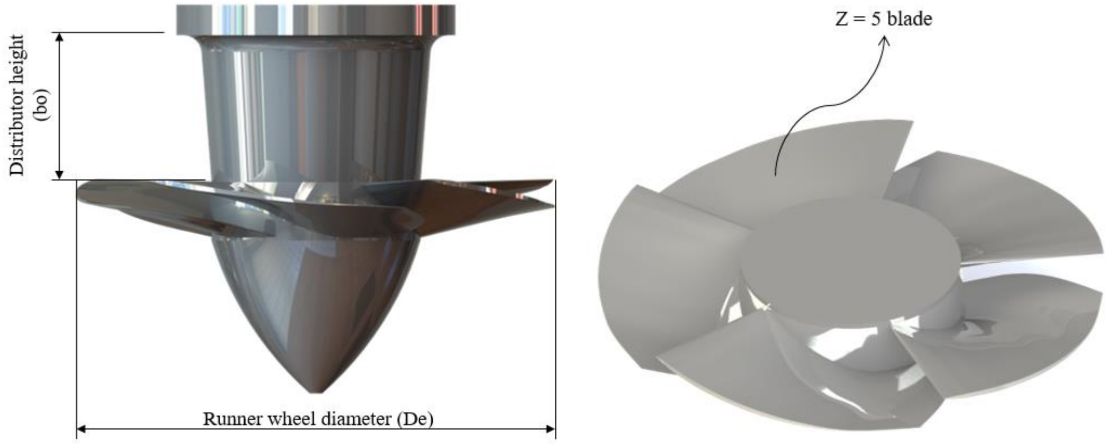


Figure 7 – Computational design of the Kaplan runner blade.



With the modeling of the blade (Figure 7) and the dimension of the distributor's height (bo) the runner whell drawing was elaborated, Figure 8a. As calculated by equation 13, the number of blades is five, this way in the drawing this number of blades was applied (Figure 8b).

Figure 8 – Drawing of the Kaplan runner with the dimensioned blade.



(a) (b)

Although each hydraulic runner wheel design is unique as a function of input data, flow rate, rotation, and height of the waterfall, the unique characteristics of each hydroelectric plant. It is possible to assure that the calculation memorial used for the sizing presented in this article allowed the elaboration of the runner wheel blade layout as well as the use of the aerodynamic profiles that shaped the blade, as well as in the work of Almeida et al., (2008) and Hantsch and Abeykoon (2017).

4 CONCLUSIONS

Despite the complexity for the sizing of hydraulic runner wheel, the method used for the sizing of the Kaplan turbine allowed satisfactory results to be obtained, since the type, shape and profile of the blade resemble the runner used as a case study. In the same way that the number of blades and the power resulting from the dimensioning are equivalent to the runner case study of the Luis Eduardo Magalhães Plant, respectively five blades and a power close to 180 MW. Finally, it should be noted that computational modeling allows the visualization of dimensioned elements.

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