Design and Simulation with CFD of 10 kW Kaplan Turbine for Micro-Hydropower Plant (Runner)

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Abstract- In this paper, the design of 10 kW Kaplan turbine runner is design and simulation expressed. The available head and flow rate are 4 m and 0.42 m^3/s respectively. The required speed to produce the expected output power is 980 rpm. There are many essential parts in Kaplan turbine such as guide vane, runner, casing and draft tube. Now, it intends to calculate the design of runner dimensions and blade profile mainly. The calculated runner diameter is 305 mm and the hub diameter is 122 mm. By using the design data of blade profile, 3D modeling blade profiles using AutoCAD software. Moreover the resistance of blade stresses and strains are simulated on the various pressures and forces by CFD software package.

Indexed Terms- power, head, flow rate, speed, turbine, runner, Kaplan, AutoCAD, and CFD.

I. INTRODUCTION

The demand for the use of renewable energy has risen the last few years due to environmental issues. The field of renewable energy includes, for example wind power, solar power and hydropower. Today, hydropower is an important source of producing the electrical energy. Depending on the head and water flow rate of the sites, the hydropower plants have to be equipped with a specific turbine in order to get the higher efficiency. There are several different kinds of water turbines and can be divided into impulse and reaction turbines. The latter are suitable for low head and high flow rate.

The objective of this study is to design the Kaplan turbine that can produce 10 kW output power. Kaplan turbine type is suitable for electricity generation in rural areas far from main power grid because it can be operated under the low head and, it can be constructed with local materials.

II. MAIN COMPONENTS OF KAPLAN TURBINE

Kaplan turbine is in axial flow reaction turbine named in honors of Dr.V. Kaplan and a German engineer, and is suitable for low head. Kaplan turbine were designed to have a minimum number of blades 4 to 6 in number.

The water enters the blades in axial direction from one side and leaves through the other side so that a large quantity of water flows through the runner.

In a Kaplan turbine, the runner blades are adjustable and can be rotated about pivots fixed to the boss of the runner. The blades are adjusted automatically by servomechanism so that at all loads the flow enters them without shock.

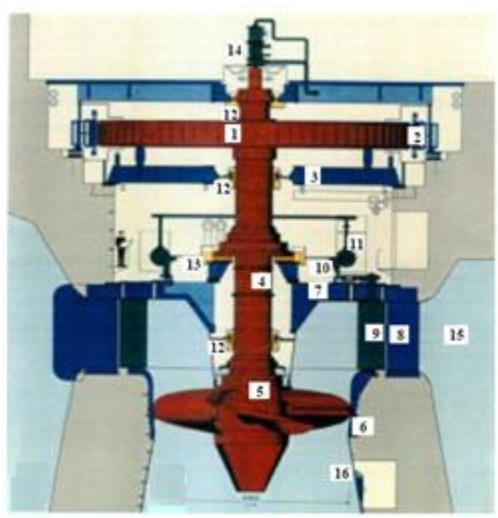

Fig. 1 Schematic diagram of Kaplan turbine [2]

(1) Generator Rotor
(2) Generator Stator
(3) Spider
(4) Turbine Shaft
(5) Runner
(6) Discharge Ring
(7) Turbine Cover
(8) Stay Ring
(9) Guide Vane
(10) Operating Ring
(11) Guide Vane Servomotor
(12) Guide Bearing
(13) Thrust Bearing
(14) Oil Supply Head
(15) Concrete Semi-Spiral Case
(16) Draft Tube Cone

III. DESIGN PROCEDURE OF KAPLAN TURNICE

The required parameters for turbine design are taken from the hydropower plant, Kyauk Ta Gar Dam, Net Mauk Township and design specifications are as follow.

Total expected capacity, P = 10 kW
Flow rate, Q = 0.42 m^3/s
Design head of turbine, H = 4 m
Generator efficiency, η_g = 85%
Mechanical efficiency, η_m = 90%
Number of poles, p = 8 poles

After knowing the design net head and required water flow rate for the expected output power, the specific speed can be specified by the following equation.

$$N_s = \frac{885.5}{H_d^5}$$

(1)

And then, the outside diameter of runner can be determined by the relationship equation of periphery coefficient. The value of periphery coefficient can be calculated by the following equation.

$$\phi = 0.0242 \times N_s^{2/3}$$

(2)

Thus, the runner diameter is

$$D = \frac{84.5 \times \phi \times \sqrt{H_d}}{N}$$

(3)

According to the value of specific speed, the number of blades and the ratio d/D between the diameters of hub and runner can be read.

A. Design of Runner Blade Profile

V.Kaplan, the turbine designer, presented a variation of conformal mapping. The basic construction is the spaces of the runner are divided into several segments. It can be divided into five cylindrical segments have been arbitrarily chosen. For greater accuracy 10 to 15 segments would be preferable. From the number of blades and the angle of overlap (usually 35°-50°) the central angle, Φ.

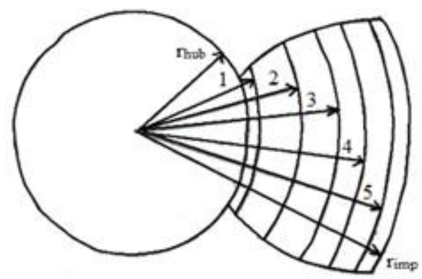

Fig. 2 Five sections of the blade [4]

Section I,

$$r_1 = \frac{d}{2} + (0.015 \text{ to } 0.025)d$$

(4)

Section III,

$$r_3 = \frac{D}{2}\sqrt{\frac{(1+d)^2}{2}}$$

(5)

Section V,

$$r_5 = \frac{D}{2} - (0.015 \text{ to } 0.025)D \quad (6)$$

Section II,

$$r_2 = r_1 + \frac{r_3 - r_1}{2} \quad (7)$$

Section IV,

$$r_4 = r_3 + \frac{r_5 - r_3}{2}$$

(8)

Then, the spacing of the blade can be determined by the following equation.

$$t = \frac{2r\pi}{z}$$

(9)Where, z is the number of the runner blades and r is the radius of runner.

B. Geometrical Characteristics of Airfoils

The most important geometric characteristics of the airfoil are indicated in Fig. 3. It is taken from the profile N.A.C.A (National Advisory Committee for Aeronautics) 4412.

$$\frac{m}{l} = 0.04; \qquad \frac{L}{l} = 0.4;$$

$$\frac{t}{l} = 0.12$$

Where, m is maximum deflection of the central curve of the airfoil, L is distance of maximum deflection of the central curve from the leading edge, l is length of chord and t is maximum thickness of the blade.

C. Aerodynamic Properties of Airfoil

The fundamental aerodynamic properties of airfoils are characterized by the coefficients of resistance c_x, of lift c_z moment which are defined by the following equations

$$P_z = \frac{c_z}{2}\frac{\gamma}{g}V^2 S \qquad (10)$$

$$P_x = \frac{c_x}{2}\frac{\gamma}{g}V^2 S \qquad (11)$$

$$M = \frac{c_m}{2}\frac{\gamma}{g}V^2 S\, l \qquad (12)$$

Where, S is the supporting surface, P_z is the component of the resultant aerodynamic force in the axis of lift, P_x is the component of the resultant aerodynamic force, M is the component of the aerodynamic force, c_z is the coefficient of lift, c_x is coefficient of resistance, c_m is moment of coefficient and V is the flow relative velocity to the airfoil.

The dependence of the coefficient c_z, c_x and c_m on the angle of attack in the majority of cases experimentally determined for certain airfoils. Most important are the following equations.

$$c_z = \frac{\partial c_z}{\partial \alpha}(\alpha - \alpha_0)$$

(13)

$$c_x = c_{xv} + \frac{\partial c_x}{\partial c_z^2}\, c_z^2$$

(14)

$$c_m = c_{m0} + \frac{\partial c_m}{\partial c_z}\, c_z$$

(15)

D. Application of Airfoils to the Design of Runner Blades

The water approaches the blade at the velocity C_1 which incorporate a certain peripheral component C_{u1}. That is in the form of a whirl with its axis in the turbine axis and the circulation:

$\Gamma_1 = 2\pi r C_{u1}$ (16)

The circulation in back of the runner is

$\Gamma_2 = 2\pi r C_{u2}$ (17)

The total circulation in the entire turbine is

$$\Gamma = \frac{\Gamma_1 - \Gamma_2}{z} \qquad (18)$$

The circulation around the blade is

$$\Gamma = \frac{2\pi r}{z}(C_{u1} - C_{u2})$$

(19)

Since t is blade spacing, $t = \frac{2r\pi}{z}$

Thus, $\Gamma = t\ (C_{u1} - C_{u2})$ (20)

In axial flow turbines, according to Fig. 9,

$W_{u2} - W_{u1} = C_{u1} - C_{u2}$ (21)

Euler's equation for an axial flow turbine is

$U(C_{u1} - C_{u2}) = \eta_h g H$ (22)

In horizontal direction,

The relative velocity of the water changes in the horizontal direction = $W_{u1} - W_{u2}$

Mass per second to which flow variation = $W_m\, t\, \frac{\gamma}{g}$

The horizontal component of the force acting on the part of the blade,

$P_h = W_m \; t \; \frac{\gamma}{g} \; (W_{u1} - W_{u2})$

(23)

In vertical direction,

The vertical component of the velocity, W_m, does not change its magnitude.

The vertical component of the force acting on the part of the blade,

$P_v = t\,(p1 - p_2)$ (24)

Where, $(p_1 - p_2)$ is the overpressure of the runner.

According to Bernoulli equation, the resultant force on the blade,

$$P_z^1 = [t^2 (\frac{r}{g})^2 \{W_m (W_{u1} - W_{u2})^2 + (W_{u2} - W_{u1})^2 \frac{(W_{u2} + W_{u1})^2}{2} (1 + \frac{2gh_z}{(W_{u2} - W_{u1})^2})^2 \}]^{1/2}$$

Where, $\Gamma = t\,(W_{u2} - W_{u1})$. Thus,

$P_z^1 = \Gamma \; \frac{\gamma}{g} \; W_\alpha$

(25)

W_α increased W'_α and its inclination changes from β_α to β'_α, because the vertical component does not change, it assumes that the through-flow is constant.

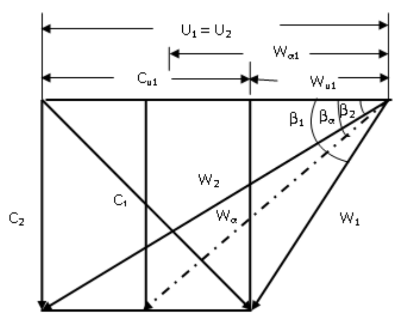

Fig. 3. Velocity triangle of axial flow turbine [4]

With regard to the circumstance that this force is at the same time also the resultant of the lifting component P_z and the resistance component P_x in relation to the velocity W_α, follow from the similarity of the hatched triangle.

$$\frac{P_z^1}{P_z - \frac{P_x}{\tan \beta_\alpha}} = \frac{W'_\alpha}{W_\alpha}$$

(26)

Where, $P_z - \frac{P_x}{\tan \beta_\alpha} = \frac{\gamma}{g} \frac{k_z}{2} L l W_\alpha^2 - \frac{\gamma}{g} \frac{k_z}{2} L l \frac{W_\alpha^2}{\tan \beta_\alpha}$

Thus, the circulation can be expressed by the following equation.

$\Gamma = \frac{1}{2} (k_z - \frac{k_x}{\tan \beta_\alpha}) l W_\alpha$

(27)

The velocity W_α and the angle β_α are needed to know and these are determined according to the construction in Fig. 10. By substituting Eq (20) and Eq (21) in Eq (27) and then introducing the specific velocities, the hydraulic efficiency of the turbine can be expressed by the following equation.

$$\eta_h = u \, w_\alpha \frac{l}{t} \, (k_z - \frac{k_x}{\tan \beta_\alpha})$$

(28)

E. Angle of Attack

The coefficient of the induced drag

$c_{xi} = c_z \sin \alpha_i = c_z \alpha_i$

(29)

Where, α_i is the induced angle of attack. The total drag coefficient of wing of finite span is

$c_x = c_{xo} + c_{xi}$

(30)

Angle of attack is

$\alpha_\alpha = \alpha - \alpha_i$

(31)

Where, α is the angle of the chord with the direction of the relative velocity of the undeflected flow. The induced velocity across the span is constant and defined by the expression.

$$V_i = V \frac{c_z}{\pi \Lambda}$$

(32)

Where, Λ is aspect ratio of the wing.

The induced angle of attack, $\alpha_i = \frac{V_i}{V}$

Therefore, angle of attack is

$$\alpha_\alpha^0 = \alpha^0 - 57.3 \frac{c_z}{\pi \Lambda}$$

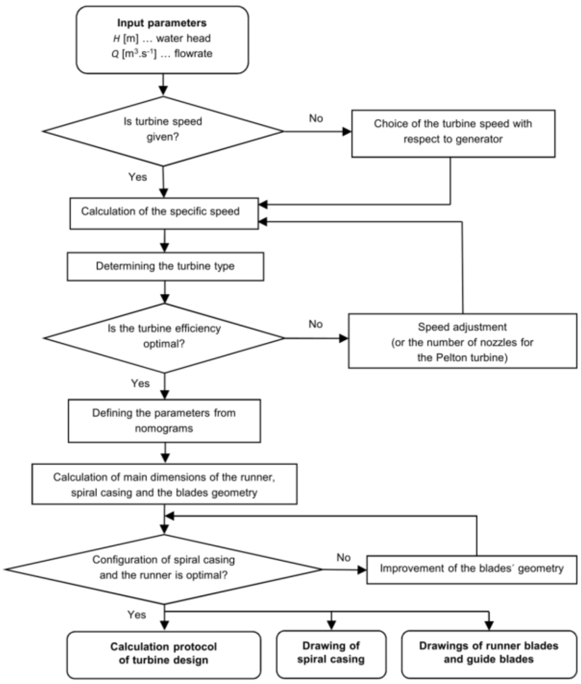

Fig.4. Simplified Block Diagram of The Kaplin Turbine design in the calculation program

IV. RESULTS

The calculated results are expressed in Table I and these are calculated for 10 kW output power of Kaplan turbine based on net head 4 m and water flow rate, 0.42 m3/s. The selected N.A.C.A series is 4412.

Table 1. Result Data of 10 kW Kaplan Turbine for Micro-hydropower Plant (Runner)

No.	Description	Symbol	Value
1	Shaft power	*BP*	13.0719 kW
2	Specific speed	N_s	626.1431
3	Speed of turbine	*N*	980 rpm
4	Runner diameter	*D*	305 mm
5	Hub diameter	*d*	122 mm
6	Number of blade	*z*	4
7	Flow rate of water	*Q*	0.42 m^3/s
8	Net head	*H*	4 m
9	Generator speed	N_g	900 rpm
10	Blade inlet angle	β_1	26.724 deg
11	Blade outlet angle	β_2	23.348 deg

By using calculated results data for blade profile, the dimensional runner blades are drawn by AutoCAD Software tool.

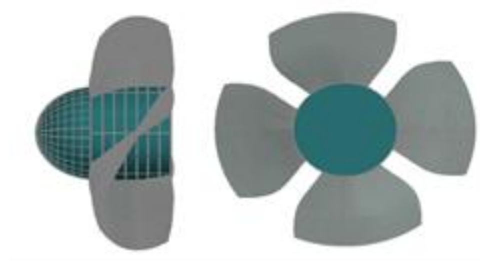

Fig. 5 Three –D Modeling of Runnerfor 10 kW Kaplan Turbine for Micro-hydropower Plant

Table 2. Result data of N.A.C.A- 4412 blade profile dimensions

Parameters	I	II	III	IV	V
R_1 = R_2(m)	0.0656	0.09342	0.121183	0.134661	0.14813
U_1= U_2(m/s)	6.7396	9.5881	12.4364	13.8196	15.2027
V_f (m/s)	6.7654	6.76542	6.765426	6.76542	6.76542
β_1 (degree)	77°	48.69°	35.09°	30.93°	27.66°
β_2 (degree)	45.11°	35.21°	28.55°	26.08°	23.99°
C_{u1} (m/s)	5.1818	3.6424	2.8082	2.5271	2.2972
$W_{\alpha 1}$ (m/s)	4.1486	7.6669	11.0323	12.5561	14.0541
β_α (degree)	58.48°	41.06°	31.52°	28.32°	25.71°
W_α (m/s)	7.9361	10.3006	12.9418	14.2632	15.5983
t (m)	0.09	0.051	0.066	0.074	0.081
Γ (m^2/s)	0.4663	0.46637	0.46637	0.46637	0.46637
l/t	1.1000	1.0125	0.925	0.84	0.74
l= l/t × t (m)	0.099	0.129	0.155	0.16	0.157
β (degree)	41.52°	58.94°	68.48°	71.68°	74.29°
α (degree)	10°	6°	3.5°	2.7°	1.5°

V. CFD SIMULATION

The calculated design of blade is simulated on the various pressures by CFD software. Before simulating, design of the blade profile is drawn with AutoCAD software. And then, mesh generation is made by using SolidWork software. After that, these designed drawings are exported into the Ansys CFD software. After running 2000 numbers of iteratrions, satisfied converged solutions is obtained.

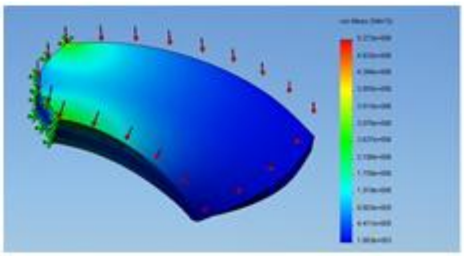

Fig. 6. CFD application of pressure test on stress analysis

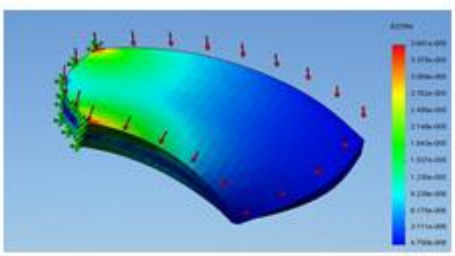

Fig. 7. CFD application of pressure test on strain analysis

VI. CONCLUSION

A Kaplan runner was theoretically designed to reach an efficiency of 94%. . In this paper, the blade runner is divided in five cylinder sections. However, the CFD analyses showed that the same theoretical design only has an efficiency of 50.98%. Here, it can be argued that the theoretical design is low in accuracy particularly due to the numerous simplifying assumptions attached with the calculation process. However, such theoretical calculations should be good to have an approximate design. Therefore, the theoretical design was optimized with AUTO CAD for developing an efficient runner wheel. With the CFD analysis. The efficiency of the theoretically designed runner increased slightly as increasing the number of blades.

However, increasing of number of blades is always not possible and there should be a maximum possible number of blades for a particular size of a runner.

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