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Analytical and Computational Analysis of Blade Parameter on the performance of L.P axial flow turbines

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Abstract*— Since axial flow turbine are widely been used in order to achieve adequate compressibility of fluid in concern to develop power. However axial flow turbines are generally used in gas turbine engine or power plant in form of steam turbine which are more efficient than radial inflow turbine in most operation range.In this paper low head axial flow turbine theoretical as well as computational model is been examined to internal performance of runner with it dependable functionality and compare the relationship blade height and blade number.A computational model of runner is developed with the help of ANSYS 14.5 Blade gen module and compared with the experiment result of 3 runners. The obtained results are well validated with shows good result agreement with experimental result. In results the effect of blade parameter such as Blade height and Blade number is been analyzed at different operating variable such as vortex angle, Runner loss coefficient factor, absolute flow angle, relative flow angle..

***Keywords*— Axial Flow turbine, Blade angle, Efficiency, Power, CFD, velocity.**

I. INTRODUCTION

The modern axial-flow turbine developed from a long line of inventions stretching back in time to the aeolipile of Heron (aka Hero) of Alexandria around 120 B.C. Although we would regard it as a toy it did demonstrate the important principle that rotary motion could be obtained by the expansion of steam through nozzles. Over the centuries many developments of rotary devices took place with wind and water driven mills, water driven turbines, and the early steam turbine of the Swedish engineer Carl de Laval in 1883.

Enormous research and development is going on in the field of Axial flow turbine in various countries from 18th century. Many patents and innovative designs are there in the name of NACA scientific division formally NASA. The focus of the major scientific community is on large axial flow turbines, where the design and operating requirements are completely different. In the special context of micro-hydro, the optimization study becomes extremely important because of challenging boundary conditions, like large variation of flow and yet optimum power generation, notwithstanding the economic constraints that emphasize on simplicity of design and manufacture.

II. LITERATURE REVIEW

In 1969 Yahay and Doyle presented the aerodynamic losses in a flat-bladed rotor operating under partial admission conditions. The derive loss formulae by verified three turbine rotors of different blade pitch.

CRANE 1973 presented the methods for predicting deposition for drops on low pressure steam turbine. He also discusses the context of wet stream flow in low pressure turbine.

Silva and Negus 1976explain some optimization techniques for the design of turbine blade shape and develop an automated structural synthesis capability for turbine blade profile. They developed an automated structural synthesis capability for turbine blade profiles by a high degree of non-linearity problem formulation.

Rowland 1977 determined the overall dimensions of a radial gas turbine rotor for a given certain operating conditions. They conclude that the loss prediction due to passage losses, disk friction and clearances alters the

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performance and should be consider in order to develop an optimal design.

Chattopadhyay and Chang 1993 developed efficient turbine airfoil design by optimization procedure and compare the tow leading section i.e. elliptical and circular and revealed that the elliptical section is leaner as compared to circular section

Habali and Saleh 2000 reported the design of blade and root of the airfoil selection and aerodynamic design of the blade for a small wind turbine. They used a composite material Glass Fibre Reinforced plastic for designing the rotor blade. They proof the blade could with stand loads ten times the normal working thrust in 15KW grid-connected-pitch-controlled machine.

Wenbin and Andy 2002 use FEA to design a turbine blade fir-tree and implement ICAD and they used a product model from Rolls-Royce in the optimization.

Jureczko and pawlak 2005 developed a computer program language to optimisation of wind turbine blades. They use ANSYS to create various blade models. They develop a computer program that would enable optimisation of wind turbine blades with required criteria.

Jun and Ren 2006 investigated the results of a series of turbine blade containment tests. This process involves impacts, plastic deformation and penetration. They use numerical simulations and non-linear finite element method to study the impact process.

Yang and Hua 2007 conducts and experiment for a L.P axial fan with integration of skewd rotor blade circumferentially.The results showed that compared to the radial rotor and the back ward skewed rotor could result a higher total pressure loss near the hub and shroud region and lower loss in the mid-span region.

Vitale and Rossi 2008 presented wind turbine blades by computational methods. They verified by comparing rotor power and rotor efficiency for different designs. The comparisons were similar to those provided by commercial wind generator manufactures.

Hsing-nan and Ming-huei 2012 investigated the performance of horizontal-axis water turbines and conforming to blade element momentum theory.the obtained result shows good agreement with experimental data. The effect of blade parameter i.e. blade radius, blade number, velocity and revealed that the increase in blade number decease the power production.

Puertas and Luri 2013 present an innovative design of Francis turbine blade by integrating channel angular extrusion and isothermal forging. And compare the mech. Properties of AA1050 with conventional manufacturing processes. They also conclude that the sub micrometric microstructure can be achieved by improving mechanical properties

Hermod Brekke 2013 reported the turbine design is to increase the efficiency and avoid cavitation and fractures during operation. He also discusses stress analyses and fatigue problems. He also includes an ancient runner design with plate steel blades moulded in cast steel in crown and band.

Kim and Lee 2013 developed software for designing optimum shape and analysing aerodynamic performance of multi-MW wind turbine blades. They verified its performance analysis module and the aerodynamic shape design module. The result of optimization blade design efficiency increased by 1% while thrust coefficient decreased by 7.5%.

III. MATHEMATICAL MODELLING

(The performance of the turbine mainly depends up on the main variable and functionalities

$$Q, P_{shaft}, \eta = f(N, gH)$$

(1)

The propeller internal variables and functionalities are assigned with the help of velocity triangle along with the cascade approach are then utilized to develop a deeper understanding of the physical significance of the internal variables in the context of propeller geometry.

Stipulating the internal variable in equation (1) in order to realize the significance of the internal variable of the propeller geometry

Now specifying the internal variables of (1) the Discharge (Q) depend up on the axial velocity and flow are which is given by this equation (2)

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$$Q = c_x \cdot (\pi \cdot D^2 / 4) \cdot (1 - (d_h / D)^2)$$
(2)

Here, $c_x = f(Q, d_h / D)$

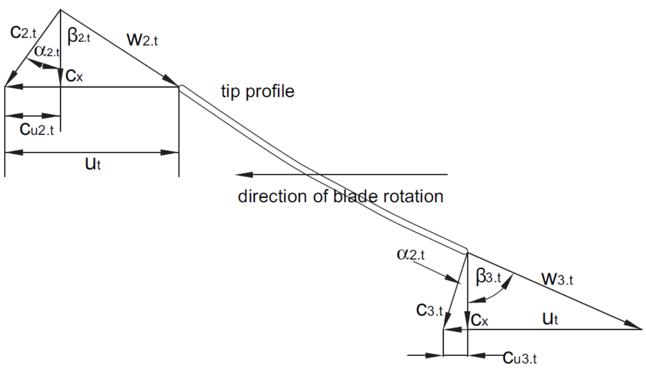

Figure.1 The velocity triangle at tip and hub of the L.P. axial flow turbine

However, Dixon[10] and Pfleiderer[2] discussed the Euler turbine fundamentals in hich momentum and leakage loss involve, which is stated in equation, (3)

$$P_{shaft} = m \cdot (\Delta c_u \cdot u) - \text{Leakage loss}$$
(3)

If this leakage loss are not considered the (3) becomes

$$P_{shaft} = \left(\rho(\pi^2 \cdot D^2 / 4) \cdot (d_h / D)^2 \cdot c_x\right) \cdot (\Delta c_u \cdot u)$$
(4)

Now the net tangential flow velocity Δc_u is a crucial parameter which signifies rotational momentum.

Since from equation, (4)

$$\Delta c_u = f(P_{shaft}, d_h / D)$$

From velocity triangle figure 1 Δc_u can be written in terms of c_x ,α_2, β_3 in equation (5) keeping blade velocity constant i.e. u

$$\begin{aligned} \Delta c_u &= (c_x \cdot \tan\alpha_2) - (u - c_x \cdot \tan\beta_3) \\ &= c_x \cdot (\tan\alpha_2 + \tan\beta_3) - u \end{aligned} \quad (5)$$

Again, $\Delta c_u = f(c_x, \beta_3, \alpha_2)$

Now moving toward the Head while it is an independent variable however it is converse by euler in terms of shat work and the new terms is introduce over here i.e. total system loss coefficient in quation (6)

$$gH = \Delta c_u \cdot u + h_{losses} = \Delta c_u \cdot u + \zeta_{system} \cdot \frac{c_x^2}{2}$$
(6)

From above equation on expanding three variables i.e. discharge, shaft power and head other internal variables such as loss coefficient, flow angle, vortex angle, axial and net tangential flow velocity can be computed.

On combining equation (5) and (6) a second-degree polynomial for c_x can be developed as shown in equation (7) where $\Delta c_u = f(\zeta_{system}, \beta_3, \alpha_2)$

$$\frac{\zeta_{system}}{2} \cdot (c_x^2) + u \cdot (\tan\alpha_2 + \tan\beta_3) \cdot (c_x) - (gH + u^2) = 0$$
(7)

Another substitution can be done by substituting equation (5) in equation (7) Functionality for the net tangential flow velocity and shown in equation (8)

$$\begin{aligned} &\Delta c_u^2 \cdot \frac{\zeta_{system}}{2} + \Delta c_u \cdot u \cdot (\zeta_{system} + (\tan\alpha_2 + \tan\beta_3)^2) \\ &+ \frac{\zeta_{system}}{2} \cdot u^2 \cdot (\tan\alpha_2 + \tan\beta_3)^2 = 0 \end{aligned}$$
(8)

$\Delta c_u = f(\zeta_{system}, \beta_3, \alpha_2)$ For Constant gH, u

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Some other Control variable such as Blade height and Blade number the behaviour of the frictional losses can be analyzed. Apart from this runner loss coefficient which also involve over all other losses in component are taken in consideration like guide vane ring spiral volute, draft tube.

For Inlet vortex angle (α_2) it can be determined theoretically by using continuity equation between guided vans and runner. Shown in equation (9)

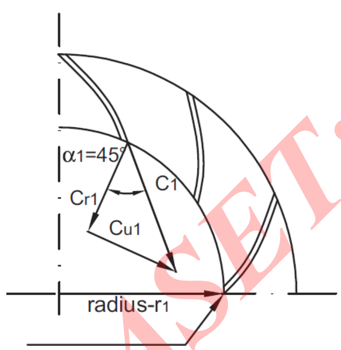
$$A_{guidevane} \cdot c_{r1} = A_{runner} \cdot c_{x2} = \left(\pi \cdot D^2 / 4\right) \cdot \left(1 - (d_h / D)^2\right) \cdot c_{x2}$$

$$\alpha_{2t} = \tan^{-1}\left(\frac{c_{u2}}{c_{x2}}\right) = \tan^{-1}\left(\frac{c_{u1} \cdot (D_1 / D)}{A_{guidevane} \cdot c_{r1} / \left(\pi \cdot D^2 / 4\right) \cdot \left(1 - (d_h / D)^2\right)}\right)$$

$$= \tan^{-1}\left(\frac{\tan\alpha_1 \cdot (D_1 / D)}{A_{guidevane} / \left(\pi \cdot D^2 / 4\right) \cdot \left(1 - (d_h / D)^2\right)}\right)$$

(9)

By knowing the angle of incidence (i) and the relative flow angle at the exit (β_3) the inlet and exit velocity triangle are constructed and are shown in figure 2



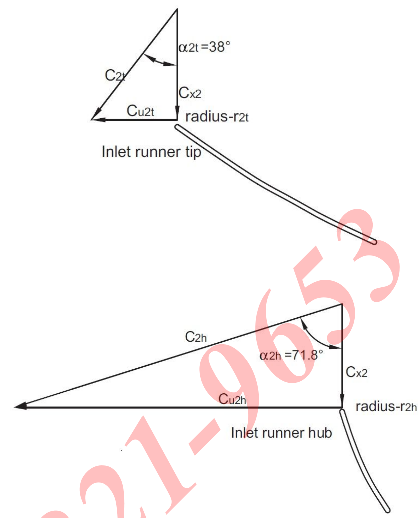


Figure 3 Velocity triangles at the guide vane exit, inlet runner hub and inlet runner tip

IV. METHODOLOGY

(An experiment is conducted by Singh and Nestmann [] comprises of the open loop hydraulic test bed

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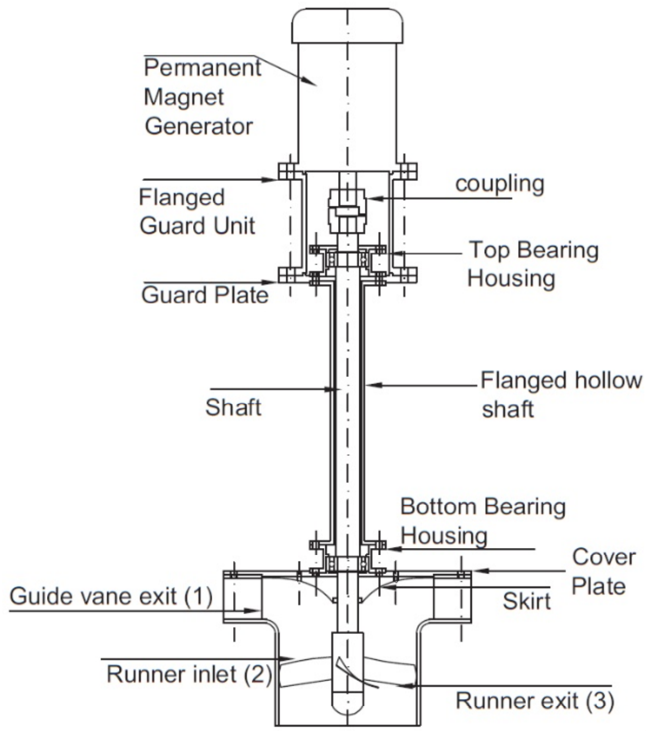

Figure 4 Experimental step of Singh and Nestmann [16]

In their experiment they implement three different types of runner and compare the turbine performance and other parameter such as shaft power efficiency. Now in present work a computational analysis is been carried out, and runner is developed with the help of ANSYS BLADE gen module and the obtained result is been compared with Singh and Nestmann [16] result and an optimized runner is been proposed with better performance and efficiency.

Table 1 and figure 5 the shows the runner configuration and the mesh model of runner D is shown in figure 6

TABLE I

RUNNER CONFIGURATION

Parameter	Runner A	Runner B	Runner C	Runner D
d_h	0.25	0.3	0.42	0.45
α_{2h}	71.8	68.4	61	61.7
α_{2t}	38	37.2	34.5	33
β_{2t}	65	66	63	61.8
β_{3t}	74	70	71	71
z	5	5	6	6
s/l	1.17	1.17	1.15	1.15

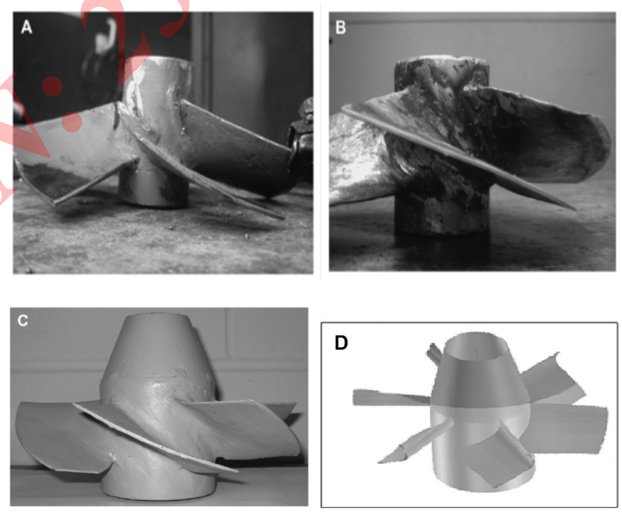

Figure 5 Picture of runner (A). Picture of runner (B). Picture of runner (C). Picture of runner (D)

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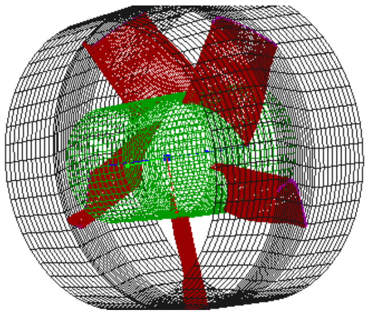
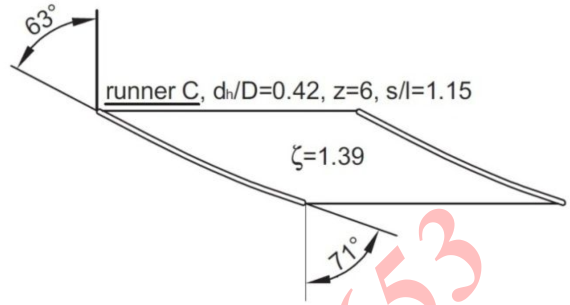
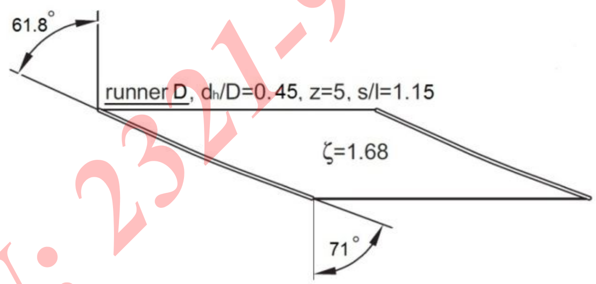


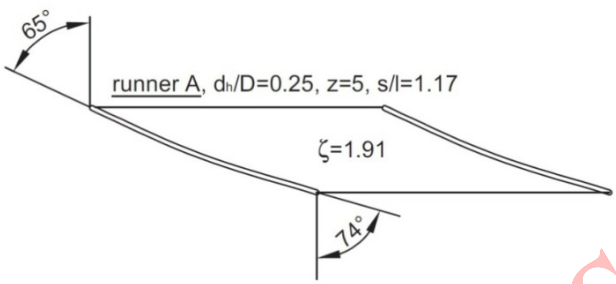
Figure 6 Turbo grid Mesh model of Runner D



Runner C

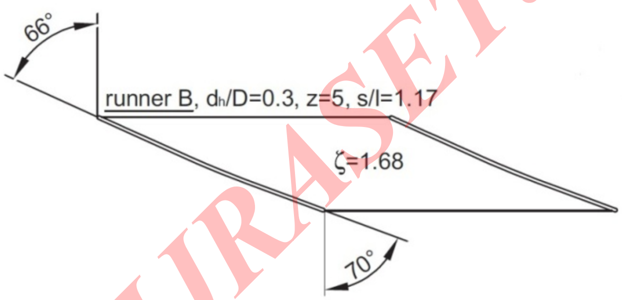


Runner D



Runner A

Figure 7 Cascades of the runner A, B, C and runner D.



Runner B

TABLE II

VALIDATION OF RUNNER DISCHARGE AT DIFFERENT SPEED

Speed, rpm	Runner A Expt. Ref [19]	Runner A Present	Percent Variation	Runner B Expt. Ref [17]	Runner B Present	Percent Variation
900	63.115	63.25	±0.135	68.82	69.01	±0.19
1200	68.211	68.32	±0.109	78.36	78.85	±0.49
1400	75.12	75.19	±0.070	85.01	85.21	±0.2

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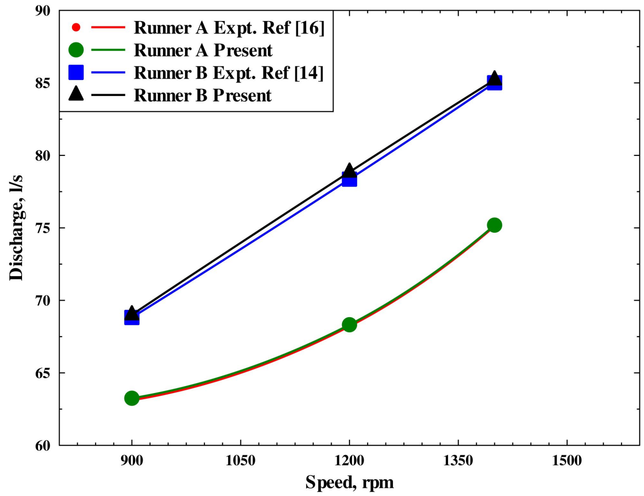

Figure 8 Validation of Runner Discharge at different Speed

Table 2 and Figure 8 show the Validation of Runner Discharge at different Speed. From this it can be conclude that the Discharge goes on increasing as the runner speed is increased and the obtain result shows the good agreement with the experiment result and the percent variation is also mention here is due to the variation in mesh configuration.

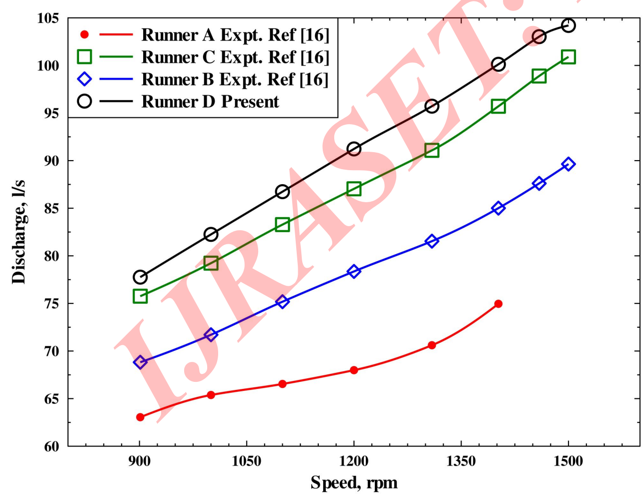

Figure 9 Variation of Turbine Discharge at Different Speed

From figure 9 shows the Variation of Turbine Efficiency at Different Speed for different runner. From this it seems that the turbine Discharge goes on increasing as the speed increases. It can also be conclude that the Runner D has more efficiency as compared to other runner. This is because of runner design variation i.e. number of blade and blade angle affects the turbine performance and ultimately optimize the efficiency, here Exit blade angle is changed due to which the discharge of the turbine affects at different speed.

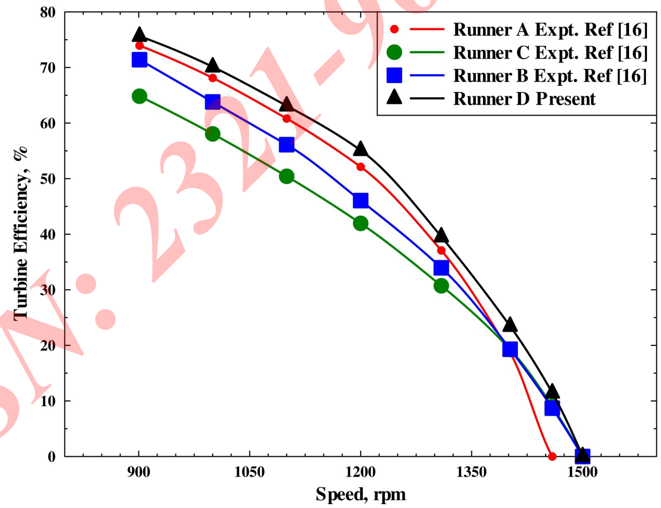

Figure 9 Variation of Turbine Efficiency at Different Speed

From figure 9 shows the Variation of Turbine Efficiency at Different Speed for different runner. From this it seems that the turbine efficiency goes on decreasing as the speed increases. It can also be conclude that the Runner D has more efficiency as compared to other runner. This is because of runner design variation i.e. number of blade and blade angle affects the turbine performance and ultimately optimize the efficiency.

V. CONCLUSIONS

It can be concluded that the computational model developed for turbine runner is well validated with theoretical model and it is found that the results show good agreement with the experimental result. In whole analysis process basic principle of classical turbomachinery are used and the

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developed computational model evolved a functionality of the internal variables. The influences of blade height and blade number both are important design parameter are related to other control variable such as flow angle, relative velocity. It can also be revealed that on increasing blade number the flow guidance improved which ultimately affects the performance of turbine.

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Nomenclature

c	absolute velocity, m/s
d	local diameter, m
D	tip diameter, m
g	acceleration due to gravity, m/s2
H	head parameter, m
I	angle of incidence, deg
l	blade chord length
m	mass flow rate, kg/s
N	speed, rpm
P	power, kW
Q	discharge, l/s or m3/s
s	blade pitch
T	torque, Nm
u	tangential blade velocity, m/s
w	relative velocity, m/s
z	blade number
u	tangential blade velocity, m/s
Greek symbols	
η	efficiency, %
ρ	density, kg/m3
Α	absolute flow angle, degrees
β	relative flow angle, degrees
ζ	pressure loss coefficient
	Greek symbols

Subscripts	
1	guide vane exit
2	runner inlet
3	runner exit
g	generator
h	hub region
m	mechanical
t	tip region
u	tangential component
x	axial direction

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