Design of Reversible Pump Turbine for its prospective application in Nepal

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Abstract- Most hydropower plants in Nepal are run-off type, which cannot supply the designed amount of energy during dry season and peak demand period resulting in energy crisis. Therefore, pumped storage plants can be an ideal solution to meet the current energy needs of the country. Most of these storage plants make use of a single unit acting both as turbine as well as pump; hence aptly called Reversible Pump Turbine (RPT). This paper explains the new application concept for use of such RPTs as auxiliary unit to supplement the main power unit in hydropower plants of Nepal. It also summarizes the design process of such runner with reference to the parameters available at Chilime region in central part of Nepal. The design is performed in two steps. The first step is an analytical design, which gives an initial geometry of the RPT runner. The process resulted in a runner of larger size than a normal Francis runner for same parameters since it has to work as pump as well. The next step is an optimization procedure involving CFD analysis under which the simulation of the RPT in turbine mode yielded an efficiency of 88.71%. The detail laboratory experiments on a model RPT will be performed later to validate the results from CFD and determine the characteristics of the designed runner at different mode and operating conditions.

Index Terms- CFD, Chilime, Design, Pumped Storage plant, Reversible Pump-Turbine (RPT)

I. Introduction

Located at the laps of young Himalayas, Nepal is a small country with abundant rivers and other water resources. The glaciers originating from the Himalayas feed the river systems in Nepal, which accompanied by small rivers, flow through the slope terrains of the country and provide an excellent opportunity for hydropower generation [1]. Study reveals that a total of 43000 MW hydro-electricity can be generated from the water resources in Nepal [2] out of which only 762 MW has been generated [3], which is less than 2% of the total generation capacity. Most hydropower plants in Nepal are run-off type [4] and hence, has to operate at off design condition during dry season resulting in low power generation. Moreover, the fluctuation in energy production from hydropower plants and their inability to meet the demand in peak hours has beckoned a dire need for construction of pumped storage plants [5]. A Pumped-storage plant stores energy by pumping water from a lower reservoir at off peak hours of electric demand by means of surplus power into a high level reservoir, in order to utilize the stored energy at periods when it is most needed [6]. It is probably the best way to compensate for the gap between produced and consumed power [7]. These pumped storage system use a single pump/turbine unit i.e. Reversible Pump Turbine to efficiently and economically store electrical energy during periods of low demand to meet peak load demands [8]. This paper proposes a new design model for application of pumped storage concept in Nepalese hydropower and elaborates the design process of such reversible pump turbine.

II. Application concept for Nepal

The pumped storage system take benefit of the variation in electricity price during peak and off demand period. The water is pumped when the electricity price is low and the same water is run down through the turbine during peak demand period when electricity price is high [9]. Since Nepal does not have a concrete tariff plans for electricity consumption and the local electricity demand is always high [10], the idea however is different. Hydropower plants in Nepal during monsoon have an excess amount of water supply while in dry season it is difficult to meet the minimum supply of water required to run all units [11]. But, there are several sites in Nepal with two rivers close enough with different head. RPT units can act as a link between such rivers. As shown in Fig. 1, water can be pumped from level H1 to the upper reservoir during dry season to meet the need to run all the turbine of the main unit. Due to the head difference between H1 and H3, relatively less amount of energy is used to pump a fixed amount of water and a higher amount of energy is generated from the same amount of water using the main turbine unit. While in monsoon when the water supply is more, the excess amount of water is sent through the RPT generating an excess amount of electricity. The technology is thus feasible for country like Nepal where there are a huge number of perennial rivers that flow very close to each other and have different heads [5].

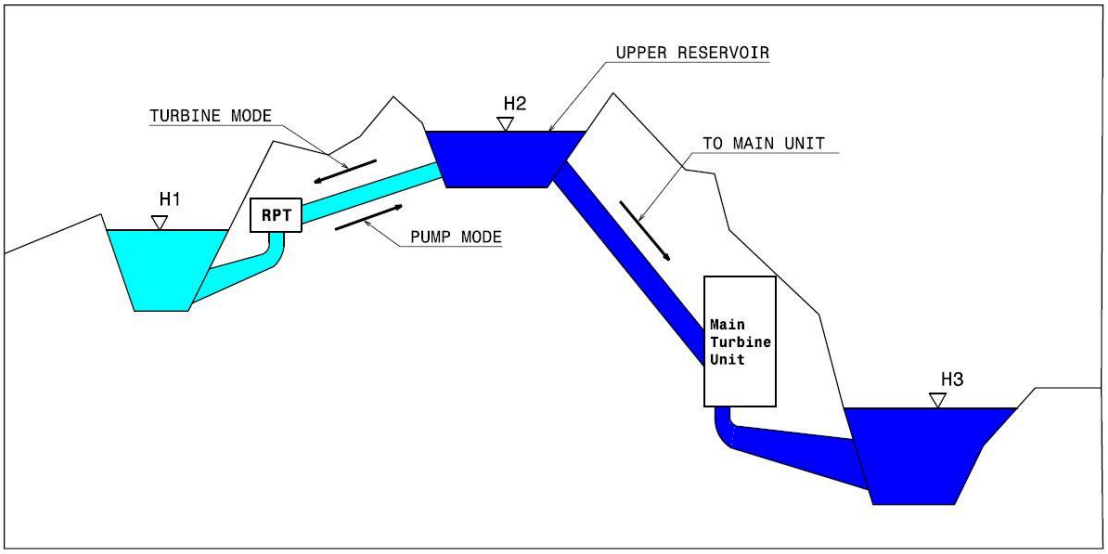


Figure 1: Schematic diagram of pumped storage plant

III. Hydraulic Design Issues

An RPT acts as a water passageway with a shiftable body component selectively displaceable to achieve alternatively, either an energy generation or an energy accumulation mode [12]. The efficiency of this system is typically between 70% and 85%, making it one of the most efficient methods for storing energy [13]. The design of a pump turbine is similar to the design of a high head Francis turbine. However, there are certain factors that need to be considered while designing an RPT [14].

A. Pump head should be higher than in turbine mode due to loss in the waterways.

According to the Euler's equation for turbine and pump we have,

$$\eta_t = \frac{U_1 C_{u1} - U_2 C_{u2}}{H_t \times g} \quad (1) \qquad \eta_p = \frac{H_p g}{U_1 C_{u1} - U_2 C_{u2}} \quad (2)$$

Now, we assume frictionless flow i.e. $H_p = H_t$, and no swirl at the outlet during turbine mode i.e. $c_{u2} = 0$. The reduced speed for turbine and pump can then, be calculated as

$$U_{1t}^* = \frac{U_1}{\sqrt{2gH_{nt}}} \rightarrow U_1^* = \frac{\eta_t . \sqrt{0.5gH_{nt}}}{C_{u1t}} \quad (5) \qquad U_{1p}^* = \frac{U_1}{\sqrt{2gH_{np}}} \rightarrow u_1^* = \frac{\sqrt{0.5gH_{np}}}{\eta_p C_{u1p}} \quad (6)$$

Assuming speed is the same in both pump and turbine i.e. $U_{1t}^* = U_{1p}^*$

$$\frac{\eta_t . \sqrt{0.5gH_{nt}}}{C_{u1t}} = \frac{\sqrt{0.5gH_{np}}}{\eta_p C_{u1p}} \quad (7) \qquad \therefore C_{u1t} = \eta_t . \eta_p . C_{u1p} \quad (8)$$

This means, $C_{u1t} < C_{u1p}$

Therefore, equation 8 shows that the pump turbine has to be designed for a higher head than the theoretical head such that $H_{nt} < H_{np}$. This ensures that the pump head should be greater than the turbine head. But, the function of the RPT defines it using as both pump and turbine which means the design head should be compatible with both modes ensuring highest efficiency delivered. The solution can be met by having fewer and longer blades than the traditional Francis runner.

B. The pump should be stable pump and not oscillating.

For stability of the pump, the inlet angle β_1 should be less than 90˚ such that for higher value for flow (Q) the power stabilizes. With β larger than 90˚, the slope will be positive. When the value of U_1 is increased the inlet angle β_1 gets smaller which is preferable considering pump characteristics.

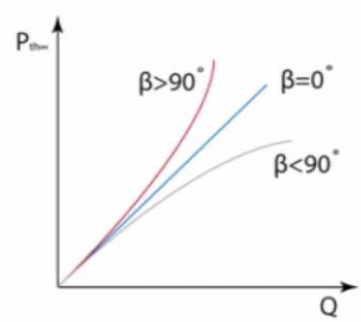


Figure 2: Effect of β angle on stability

C. The system should deliver high efficiency in both modes.

The higher efficiency in both modes can be achieved by designing the system for best efficiency head rather than the required head at the site. Adjustment of the best efficiency head for pump is possible by varying its rotational speed as head produced is dependent on

the rpm of the impeller in the pump mode. High specific speed pumps have relatively steep head-discharge curves so that they are able to operate at wide variation of head. However, the allowed variation of operating head range is narrow for RPT than in case of real turbine such that the head range is only allowed between 65% to 125% of the design head.

D. The relative velocity should remain almost constant.

Acceleration through the runner is undesired since it turns into deceleration when shifting operation mode. A decelerated flow is more vulnerable to secondary flow effects and separation. Thus, a small difference between the magnitudes of relative velocity at the inlet and the outlet should be the goal of the design, which can be achieved by increasing U_1. To increase U_1, Brekke [15] suggests that while designing, the value of U_1 reduced should be chosen near to 1. This value gives steep pump characteristics and just a small increase of W_1 through the runner channel.

IV. Design Process

The first choice in the design process is to find a suitable existing plant site that can be used as a starting point. The power plant should have a turbine with a speed number between 0.27 and 0.35. After analyzing the available data, Chilime Hydropower plant in Central region of Nepal was taken as the reference site for design. The available head and flow at the site is 270 m and 4 m^3/s respectively.

The design for RPT is done at best efficiency point. The design procedure starts with calculating the outlet diameter, D_2, number of poles in the generator and synchronous speed. With these values known, the dimensions at the inlet are calculated. These comprises of diameter, D_1, inlet angle, β_1, and height, B_1 [14-16].

A. Main Dimensions

The dimensioning of the outlet starts with assuming no rotational speed at best efficiency point (BEP) i.e $C_{u2} = 0$. In addition, the values for outlet angle, β_2, and peripheral speed, U_2, are chosen from empirical data:

$13° < \beta_2 < 22°$ Lowest value for highest head

$35 \text{ m/s} < U_2 < 42 \text{ m/s}$ Highest value for highest head

The outlet diameter and the speed are found by reorganizing the expressions for flow rate and rotational speed, respectively. C_{m2} is obtained from the known geometry in the velocity triangles. The number of poles, Z, in the generator depends on the rotational speed and net frequency. With a net frequency of 50 Hz, the number of poles is calculated using equation 12.

$$D_2 = \sqrt{\frac{4Q}{\pi C_{m2}}} \text{ [m]} \quad (9) \qquad n = \frac{U_2 \cdot 60}{\pi D_2} \text{ [rpm]} \quad (10) \qquad c_{m2} = U_2.\tan\beta_2 \text{ [m/s]} \quad (11) \qquad Z_{poles} = \frac{50 .60}{n} \quad (12)$$

Since the number of poles is an integer, the value obtained from equation 11 must be round up. With the correct number of poles, equation 9 is used to find the synchronous speed, $n_{corrected}$, which in turn, is again used to calculate the corrected diameter at the outlet, $D_{2corrected}$ (see equation 13) .

$$\tan\beta_2 = \frac{C_{m2}}{U_2} = \frac{C_{mcorrected}}{U_{2corrected}}$$

$$n_{corrected} D_{2corrected}^3 = n D_2^3$$

$$\frac{4Q/\pi D_2^2}{\pi n D_2/60} = \frac{4Q/\pi D_{2corrected}^2}{\pi n_{corrected} D_{2corrected}/60}$$

$$D_{2corrected} = \sqrt[3]{\frac{n D_2^3}{n_{corrected}}} \text{ [m]} \quad (13)$$

To avoid cavitation at the runner outlet, high head turbines usually need to be submerged. The level of submergence is calculated using equation 14. The $NPSH_{required}$ is calculated from equation 15 [15].

$$h_s = h_b - h_{va} - NPSH \quad (14) \qquad NPSH_{required} = a.\frac{C_{m2}^2}{2g} - b.\frac{U_2^2}{2g} \quad (15)$$

The next step in the design is to calculate the inlet parameters, diameter, D_1, height of the inlet, B_1, and inlet angle, β_1. In order to find these values, the Euler Equation i.e. equation 1 is used. By introducing reduced dimensionless values and assuming no rotation at the outlet, the equation 1 can be rewritten as

$$\eta_h = 2\, \underline{u}_1 \cdot \underline{c}_{u1} \quad (16) \qquad U_1 = \underline{U}_1 \cdot \sqrt{2gH} \text{ [m/s]} \quad (17)$$

The efficiency, η_h, is set to 0.96. This value accounts for the friction in the runner and draft tube. For the high head Francis Turbine, $\underline{U}_1$ is chosen in the interval of 0.7 to 0.75. However, in case of RPT, the $\underline{U}_1$ is taken to be nearly equal to 1. U_1 is obtained from equation 16.

The inlet diameter can now be found by using equation 18. From the velocity triangle, the expression for the inlet angle can be derived (equation 19). The value for C_{m1} in equation 18 is calculated by using the continuity equation i.e. equation 20. Assuming minimal acceleration of 10% in the runner, the inlet height is calculated by using equation 21.

$$D_1 = \frac{U_1.60}{n\pi} \text{ [m]} \quad (18) \qquad \tan\beta_1 = \frac{c_{m1}}{U_1 - c_{u1}} \quad (19) \qquad C_{m1} . A_1 = C_{m2} . A_2 \quad (20) \qquad B_1 = \frac{1.1\, D_2^2}{4D_1} \text{ [m]} \quad (21)$$

After the main dimensions of the runner are known, the shape of runner blade can be designed. The procedure starts by determining the shape of the blade in axial view, then the radial view is established, and finally, the runner blade can be plotted in three dimensions [15].

B. Guide vanes

Since the flow from the guide vanes outlet to the runner inlet is not affected by the runner blades, free vortex theory is used to define the velocities in the guide vane path. In general, longer the guide vane, better the directed path of water but it also results in more friction loses. Hence, suitable length with overlapping of about 12% need to be selected. Afterwards, the number of guide vanes is selected such that no water enters the runner in its full closed condition. For the RPT design, appropriate number of guide vanes was chosen to be 17 and NACA 2412 profile was selected for guide vane shape. The guide vane shaft was fixed at 2/3 of the guide vane length from guide vane outlet.

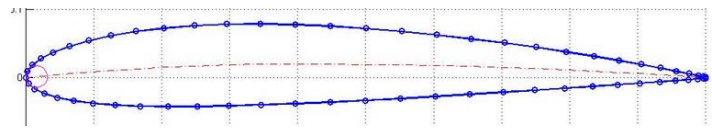


Figure 3: NACA 2412 profile

C. Design of Spiral Casing

For the spiral casing design, the C_m component of velocity remains same throughout the sections of the spiral casing. The flow of the turbine is 4 m^3/s and C_m velocity at inlet is about 10.84 m/s. The section of the spiral casing is selected such that it is slightly more than the number of stay vanes, which results in even flow inside the casing. For our design case, we have selected 20 sections for spiral casing where there are 17 stay vanes. Hence, each section is at an angular interval of 18°. Now the diameter of each section of spiral casing is calculated using equation 22. The flow in the next section is then calculated using equation 23. Equation 22 and 23 are used to calculate diameter and flow for respective sections.

$$d = \sqrt{\frac{4Q'}{\pi . C_{m1}}} \quad (22) \qquad Q' = \frac{360 - \theta}{360} . Q \quad (23)$$

D. Draft tube

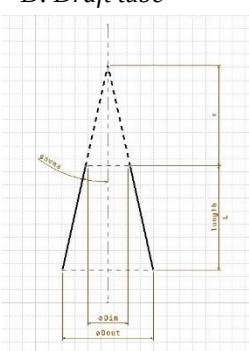


Figure 4: Draft tube cone

The draft tube recovers kinetic energy at the outlet of the turbine to pressure energy. It allows the turbine outlet pressure to be lower than the atmospheric pressure by gradually increasing the cross section area of the tube. A cone type draft tube was used for the design of RPT. To avoid unfavorable flow patterns as backflow, the angle ϒ between centerline and the wall was chosen to be 3°. Inlet diameter of the draft tube cone $D_{cone\ i}$ is equal to the outlet turbine diameter D_2. Equation …. Gives the outlet diameter of the draft tube cone which depends on the length, which is selected to be little larger than suction head of the turbine to minimize cavitation effect.

$$D_{cone\ o} = D_{cone\ I} + 2 . L_{cone} . \tan \Upsilon \quad [m] \quad (24)$$

V. Design Parameters and Outputs

The design of RPT was performed with the procedure described in chapter 4. In order to simply the design process, a Graphic User Interface (GUI) program to create and modify design of Francis runner has been developed in MATLAB [17]. The program is capable of creating 3-D runner profile based on given basic design data. Since the design of RPT is almost similar to design of Francis runner, the same program was used for designing the RPT with variation in certain input parameters. The parameters used in the design and its outputs are listed in table 1 and table 2. The velocity triangle obtained from the MATLAB program is shown in fig. 5. Fig. 6(a) and Fig 6(b) show the axial and radial view of the runner blade developed. The 3-D profile obtained from the program was modelled and modified in 3D CAD software and the final design was prepared as shown in Fig. 6(c).

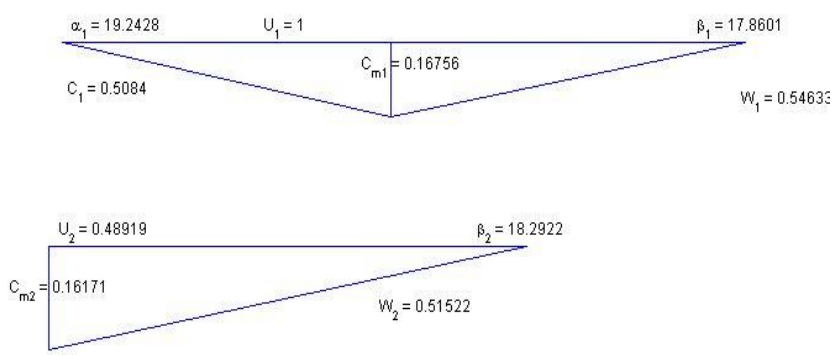


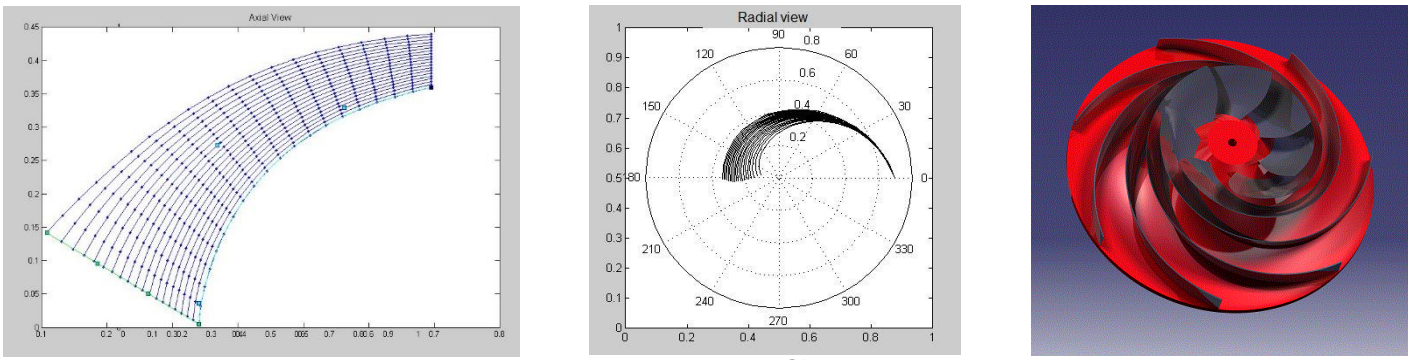
Figure 5: Velocity triangle obtained from the program

Table I: Design parameters

Symbol	Parameter	Unit	Value
H	Head	m	270
Q	Discharge	m^3/s	4
Z_{poles}	Number of poles in generator	-	3
H	Effficiency	-	0.96
$\underline{U_1}$	Reduced velocity at inlet	-	1.00

Table II: Design outputs

Symbol	Parameter	Unit	Value
D_2	Outlet Diameter	M	0.68
Ω	Speed Number	-	0.337
NPSH	Net Positive Suction Head	m	11.46
D_1	Diameter at inlet	m	1.39
B_1	Breadth at inlet	m	0.09
N	Number of blades	-	7
Ω	Angular Velocity	rad/s	104.72
R.R.	Reaction Ratio	-	0.73
P	Power output	MW	10.17



(a) (b) (c)

Figure 6: (a) Axial view of blade (b) Radial view of blade (c) CAD model of runner

VI. CFD Analysis

The CFD analysis of the RPT was carried out inside the premises of ANSYS Workbench 14.0. The code uses finite volume approach to solve the governing equations of fluid motion numerically on a user-defined computational grid. In this simulation, there are in total four different domains which were meshed separately in ANSYS meshing. In this study, all the simulations have been performed in turbine mode and the setups were done for the same condition. The details of the numerical models for each of the domains are described below.

A. Spiral Casing

Spiral Casing was discretized with tetrahedral mesh with total node count of 241929. The discretization was made finer towards the interface between the casing and the guide vane, with mapped mesh property. The inlet of the spiral casing is the inlet of the whole domain, which have mass flow rate of 4000 kg/m. Towards the outlet of this casing, an interface was defined with the guide vane inlet. Other regions were defined to be no slip wall. The roughness of the wall was not included in this analysis.

B. Guide Vane

Guide vane was meshed with a total node count of 1038232. In this case, the size of the mesh is uniform throughout the domain, whereas inlet and outlet of the domain was considered to be mapped in order to have proper mapping with neighboring domains. The inlet of the domain was interfaced with spiral casing outlet and the outlet was interfaced with the runner inlet. The mixing model between the stationary and rotating domain was done through Stage averaging between the blade passages. Other boundaries in this domain were considered to be no slip walls, including the blades.

C. Runner

As runner is the most important region of the whole domain, it was meshed with finest discretization. The total node count of 5635095 was used to mesh the runner with finer mesh towards the leading and trailing edges. All boundaries except the interfaces with stage mixing model were considered to be no slip walls.

D. Draft tube

Draft tube consists of hexahedral mesh with total node count of 832477. The outlet of the domain was taken as outlet with average static pressure of 1atm. The inlet of the domain was interfaced with the runner having stage mixing model. Other boundaries were considered to be no slip wall without roughness. The type of mesh selected for all the cases is defined in Fig 7. A total of 7747733 mesh nodes were used for the analysis.

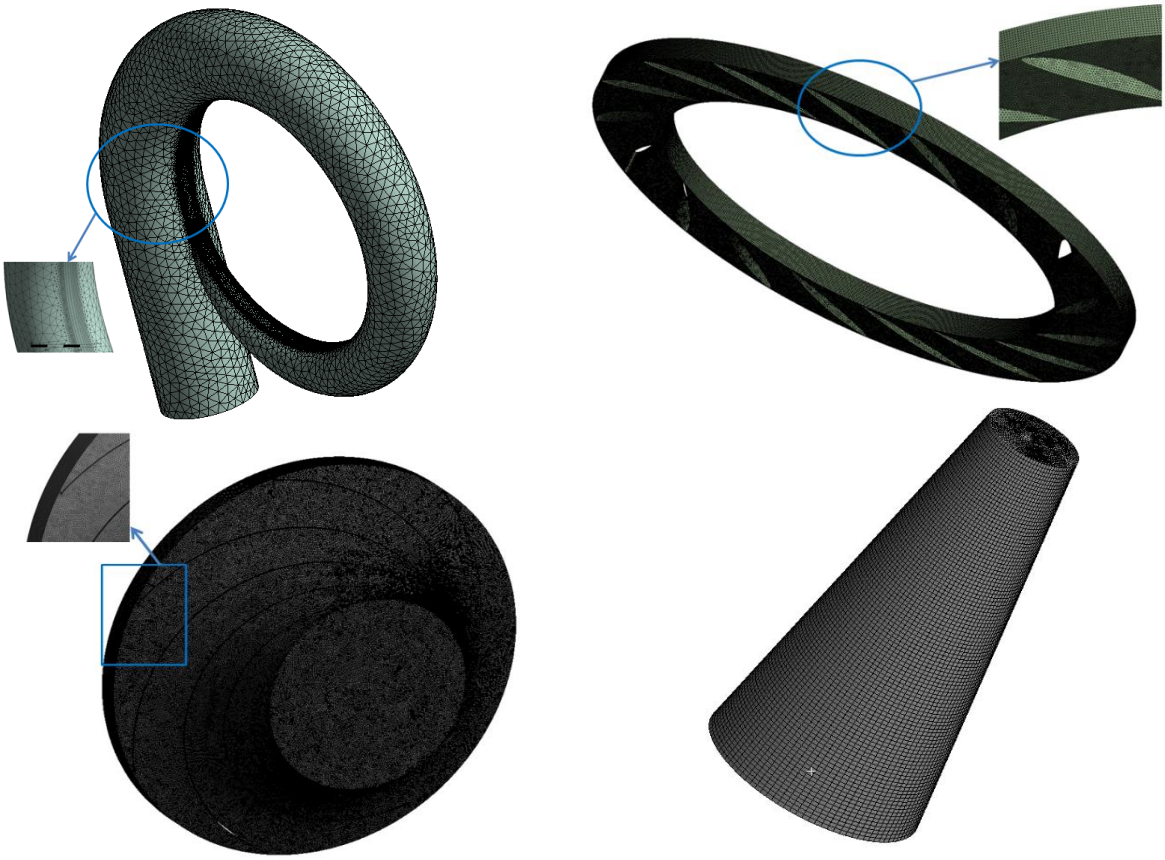


Figure 7: Mesh selected for all domains

E. Results of CFD analysis

The velocity streamlines with the above setup is shown in Fig 8. The efficiency of this turbine was measured based on the output power obtained from the torque produced on the runner and the input power based on the available head. The efficiency of the turbine was found to be 88.71%. Fig 8 also shows the streamlines in the draft tube region. From the runner, the water flows towards the draft tube with high velocity. This velocity head is converted into the pressure head generating more power and efficiency. The diverging passage of the draft tube allows for the decrease in the velocity at the outlet from the equation of continuity. As it can be seen, the maximum velocity of the flow in the draft tube is around 17 m/s towards inlet, which is reduced to less than 1 m/s towards the outlet boundary.

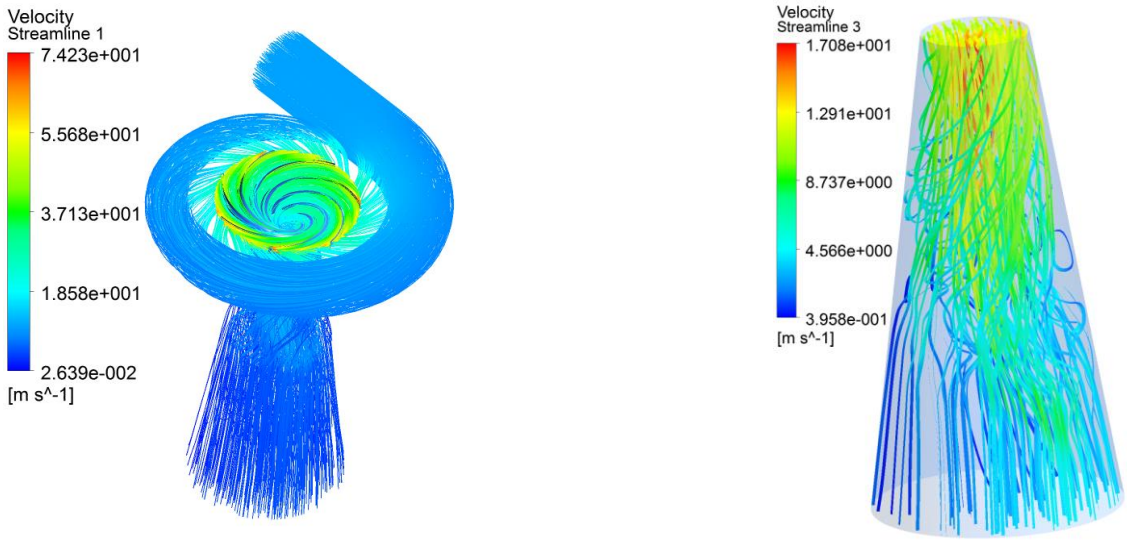


Figure 8: Velocity Streamline in the turbine mode

VII. Analysis and Interpretation

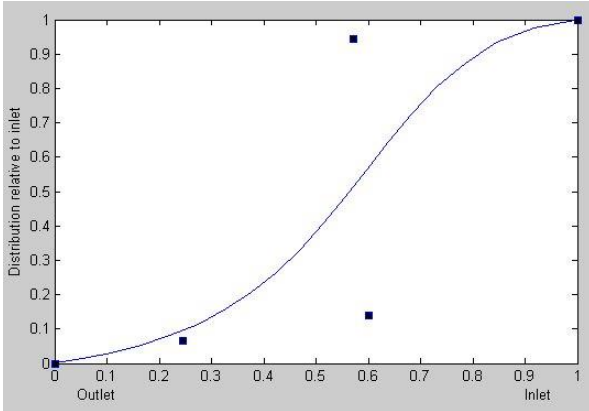


Figure 9: UC_u distribution

When designing a high head RPT, there are certain variations than when designing a high head Francis. The main changes are that the water flows in both directions and the dimensioning pressure height gets larger because of friction losses in the conduct system. In order to avoid the losses in the system, the UC_u distribution is flattened at both ends, which can be achieved by using longer blades. This also results in good cavitation performance and flow characteristics at off design operating points [16]. Therefore, it is common to choose the reduced velocity U_1 to be approximately equal 1. However, the longer blades will lead to more stability problems due to decelerated flow and backflow as indicated by the drop in relative velocity, W at the middle of the blade. This risk increases when low relative velocities occurs through the runner. To avoid this, one have to "stretch" the blades, and the UC_u distribution as well [15].

Basic calculations for a high head Francis runner was also done for the same site using same parameters and conditions using the MATLAB program. The output was a runner of 0.99 m inlet diameter, which is almost 71% smaller than its RPT counterpart. Thus, the RPT is quite larger with longer and less number of blades than a Francis runner of same capacity. This is justifiable since it has to operate as pump as well for which larger blade size would result in more output.

VIII. Conclusion

The RPT can be used as a secondary unit to supplement the existing main plant so that it can operate at best efficiency point for most part of the year. A RPT for Chilime Hydropower plant in central region of Nepal has been designed and the CFD analysis of the RPT in turbine mode of operation has been completed. The RPT is a hybrid of centrifugal pump and Francis runner with speed number 0.337. It has an inlet diameter of 1.39 m and outlet diameter of 0.68 m with inlet height 0.09 m. Model test of the RPT will be performed at Turbine Testing Lab in Kathmandu University to learn more about the performance of the unit. Despite the simple design techniques involved in the design, the runner performance is quite satisfactory with a numerical computed efficiency above 88 percent at BEP.

IX. Further Works

The design of runner has already been completed. Now, the next task is to analyze the competence of the designed runner by combining CFD simulations and laboratory measurements. Proper model design and test rig installation with up-to-date measuring instruments will be required to get the accurate data on its performance.

Appendix

Nomenclature

B – Runner height [m]
C – Absolute velocity [m/s]
D – Runner diameter [m]
U – Peripheral velocity [m/s]
W – Relative velocity [m/s]
N – Synchronous speed [m/s]
Z_p – Number of poles in generator [-]
C_m – Meridian component of C [m/s]
C_u – Tangential component of C [m/s]
h_{va} – Vapour Pressure [m]
h_b – Barometric pressure [m]
h_s – Submergence [m]
β – Angle between the relative and peripheral velocity [degree]
$_1$ – Inlet
$_2$ – Outlet
$_t$ – Turbine mode
$_p$ - Pump mode
η – Efficiency

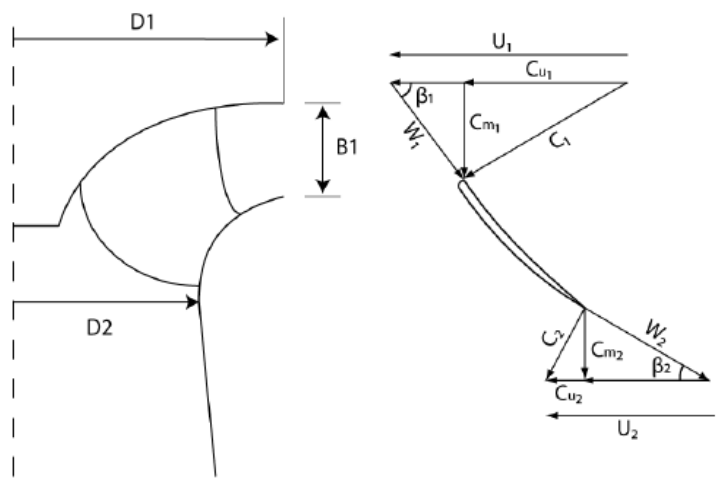


Figure 10: Main dimensions and velocity triangle in turbine mode for RPT

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