Study on the effect of the runner design parameters on 50 MW Francis hydro turbine model performance

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Abstract

Francis hydro turbine is the dominant turbine in the hydropower generation. Francis turbine has been installed at most 60% of the hydropower in the world at present. Although the basic design for the Francis turbine has various method regarding the specific speed. The runner meridional shape varies with different specific speed. Despite having, the basic design but there is still some room for the optimization. In this study 50 MW, Francis hydro turbine with specific speed 323 m-kW was designed and considered for the optimization. The various parameter as runner meridional shape (curve profile of hub, shroud, leading edge and trailing edge), blade angle and its distribution, blade thickness, runner inlet width that has been considered for the optimization of the runner for enhancement of the performance.

Keywords: Francis hydro Turbine, Design, Meridional Shape, Blade Angle.

1. Introduction

Francis hydro turbine has a wide range of operating condition from low head to high head. There have been lots of study on the high head Francis runner but unfortunately, there is not sufficient study on low head Francis runner. Due to lack of study on low head Francis turbine the exploitation of low head potential energy is limited and mainly depend on the axial turbine. The low head Francis runner has different characteristics than the high head one. The difference between low head and high head Francis runner is the dependency of the flow discharge with rotational speed, which is inverse in high head Francis Runner, and direct with low head Francis runner [1]. Francis runner design is influenced by numerous parameter due to the shape complexity and nature of the three-dimensional flow. The major parameter for the design of Francis runner is rotational speed of the turbine, guide vane height, meridional shape of the runner, runner inlet and outlet diameter, blade angle, blade angle distribution from leading edge to trailing edge, thickness distribution profile [2]. Every power plant has its own specification for turbine design to operate at maximum efficiency. In this article, the effect of runner design parameters on the performance of the turbine has been discussed. The specification of the 50 MW Francis turbine is shown in the table 1.

Table 1: Parameter of 50 MW Francis Turbine.

| Parameter       | Value       |
|-----------------|-------------|
| Head            | 38.7 m      |
| Discharge       | 132 m³/s    |
| Power           | 46.3 MW     |
| Rotational Speed| 150 rpm     |
| Specific Speed  | 323 m-kW    |
2. **Turbine Blade Design Parameter**

Many runner design parameters affect the turbine performance. Runner meridional shape, blade angle distribution, and thickness distribution in developing the blade shape has a major contribution to increase or decrease the performance of the turbine. Based on non-dimensional specific speed, Francis turbine design parameter is determined. P Henry and M Hussain Khan [3] [4] have used the equation 1 for determination of non-dimensional specific speed. The parameters for the meridional shape are the function of the non-dimensional specific speed.

\[ \nu = \frac{\omega \sqrt{Q/E}}{(2.4)^{1/2}} \]  \hspace{1cm} (1)

2.1. **Meridional Shape**

The meridional shape is the main concern with blade shape and design. The meridional shape has an influence on the flow characteristics of runner. The meridional shape of the runner is varied according to the specific speed of the runner. The meridional shape is generated from the equation developed by BOVET [3]. The meridional shape can adjust according to the need of the turbine shape. The curve for the meridional shape generally belongs to above curve family represented by equation 2.

\[ \frac{y}{y_{\text{max}}} = \frac{16}{3\sqrt{3}} \sqrt{\frac{x}{l} - \frac{x^3}{l^3}} \]  \hspace{1cm} (2)

The characteristics curve shown in figure 1 represent the curve profile for hub and shroud with the variable with x represents the length of the curve. The maximum vertical height of the curve at the 25% of the total length of the curve. Similarly, with help of the characteristic curve, the meridional shape for the 50 MW Francis runner has been generated in Blade Gen ANSYS 18.1 [5]. The l and r are the length and radius, subscript i and e represent hub and shroud profile respectively, and subscript 1 and 2 represent leading edge and trailing edge respectively. However, in the meridional shape, the shroud profile length is 25% of the total length of the curve, which is the location of maximum camber.

![Figure 1: Characteristic curve for Flow Channel](image1.png)

![Figure 2: Nomenclature of the Meridional Shape](image2.png)

The empirical dimension of meridional shape as shown in figure 2 can be calculated by using equations 3 to 8 [3]. After calculating the empirical values, it is should be converted to the real physical value by using some constant and represented by the upper case in figure 2. The constant
value will vary according to the required runner dimension. It can determine with help of energy coefficient and discharge coefficient of the runner.

\[ b = 0.8(2 - \nu)\nu \]  

(3)

\[ l_i = 3.2 + 3.2(2 - \nu)\nu \]  

(4)

\[ x_r = 0.52 - 0.2\nu \]  

(5)

\[ l_v = 2.4 - 1.9(2 - \nu)\nu \]  

(6)

\[ r_{ie} = 0.4 + \frac{0.18}{\nu} \]  

(7)

\[ r_{ve} = \frac{1}{0.96 + 0.2\nu} \]  

(8)

3. Numerical Method

The commercial CFD software ANSYS CFX 18.1 [5] has been used to perform all the analysis of the turbine. The analysis of both one pitch and the full domain has been conducted. The accuracy of the simulation is dependent on the parameters such as numerical scheme, computational grid quality, boundary condition and proper turbulence model. The casing and draft is unstructured grid consists of a tetrahedral pyramid and prism elements. Stay vane and guide vane consists of unstructured hexahedral elements from ANSYS ICEM 18.1 [5]. Hexahedral elements for the runner is generated from ANSYS TURBOGRID 18.1 [5].

Table 2: Boundary Conditions for Analysis

| Analysis Type   | Steady State       |
|-----------------|--------------------|
| Turbulence Model| SST                |
| Inlet           | Total Pressure     |
| Outlet          | Static Pressure    |
| Walls           | No slip            |
| Working Fluid   | Water at 25°C      |

The result of the mesh independency test has been shown in figure 4. The efficiency and power output for the different mesh number has been normalized by the efficiency and power output of fine mesh. The deviation of normalized efficiency and power with respect to the variation of a mesh is less than 10%. From the mesh independency test, the minimum node number for the simulation is 450000 to reduce the result deviation less than 0.5%. Similarly, the result of turbulence model test is presented in figure 5. In turbulence model test the efficiency and power output have been normalized by efficiency...
and power output of SST turbulence model. The difference in the normalized efficiency and power output is less than 5%, which is compared with six different turbulence model. From the mesh independency and turbulence model test it can be verified the result of the simulation.

4. Result and Discussion

4.1 Blade Angle distribution
The inlet and outlet blade angle is vital for the proper runner design. The conventional way of the designing considered the inlet blade angle is fixed from the hub to shroud and outlet angle vary with the diameter. However, in the low head Francis runner, both inlet and outlet blade angle varies from hub to shroud. The inlet and outlet blade angles are calculated as per the equation 9 and 10 respectively [6]. The inlet and outlet blade angle are dependent upon the diameter of the runner. At the shroud, the inlet and outlet blade angle is nearly equal due to the minimum difference in the inlet and outlet diameter whereas in the hub due to the high difference inlet and outlet angle deviation is also high.

Beside inlet and outlet angle, the shape of the runner is dependent on blade angle distribution from leading and trailing edge. The beta angle distribution for the low head Francis runner is not determined exactly. As shown in figure 7 there are six types of beta angle distribution from the leading edge to trailing edge of the runner. Profile 1 is the rear blade loading profile, which means the decrease in the blade angle initially and gradually increases towards the outlet. Profile 2 is the front blade loading
which angle distribution increases at the inlet and gradually decreases at the outlet. Profile 4 is the linear distribution from an inlet to outlet. In figure 8, the blade cross section of 2 and 4 is shown which indicate that the variation of the blade shape with change in the blade angle distribution from an inlet to outlet.

\[
\cot \beta_1 = \frac{\pi D_b B}{Q} \left( \frac{\pi D_1^2 N}{60} - \frac{60 g H_n}{\pi D_1^2 N} \right)
\]

(9)

\[
\tan \beta_2 = \frac{Q/A}{\pi D_2^2 N/60}
\]

(10)

Figure 8: Blade cross section for profile 2 and profile 4 at mid-span

4.2 Thickness Distribution
The thickness distribution of the runner plays the vital role in the strength and performance of the runner. The smooth thickness distribution reduces the friction losses and flow separation in the runner [7]. There is three thickness profile variation, which has been studied and shown in figure 9. The thickness distribution is with positive, negative and neutral variation from leading edge and trailing edge.

Figure 9: Thickness distribution from leading edge to trailing edge

4.3 Performance
The performance of the runner is affected by the blade angle distribution. The front loading blade profile shows the better performance compare to the other cases as shown in figure 10. The blade angle distribution for strictly increasing and decreasing from the inlet to outlet has the adverse effect on the efficiency due to the swirl flow at the outlet of the runner. The blade angle must follow the front blade loading for the better efficiency and minimization of the swirl flow in the outlet. However, the
profile 3 has highly poor performance in comparison to the other blade loading profiles as shown in figure 10. The sudden decrease in the performance of the profile 3 is due to the swirl flow in the draft tube. The flow angle decreases to a minimum value in the mid-span and suddenly start to rise, which is the main cause of the swirl flow at outlet and sudden pressure drop in the mid-span. In profile 6, blade angle after the mid-span is slightly lesser than the outlet angle and it slight effect on the pressure drop at that specific location. Therefore, the blade angle of the runner throughout span should be greater than the outlet blade angle to minimize the swirl flow and have the smooth pressure drop in the blade.

The negative thickness profile has better performance but the low power generation than thicker blade profile as shown in figure 11. Rather than the thicker blade, a thinner blade has the better streamline flow by reduction of secondary flow at the suction side to enhance the performance of the runner by 0.1%. The pressure drop in the suction side of the thicker blade is less than the thinner one. Overall, the uniform thickness blade performance is better than thicker and thinner profile. The effect of the thickness is not significant to improve the performance of the drastically but slight improvement has been possible from the thickness of the blade.

![Figure 10: Efficiency versus blade angle distribution profile](image1)

![Figure 11: Efficiency and power versus thickness distribution profile](image2)

The swirl flow in the draft tube is shown in figure 12, which implies the reason for profile 2 of having better performance. The comparison of the outward flow in the draft tube in figure 12 clarifies that with blade angle distribution such as profile 3 has the most swirl flow because the blade angle at the mid-span is lower than the outlet blade angles. Figure 13 shows the flow angle at the draft tube inlet from the hub to the shroud of the front loading, rear loading, and no loading. There is not any difference in the flow angle at the outlet of the runner in three different loading. However, the
circumferential velocity at the outlet of the runner indicates the difference between the three types of blade loading. In an ideal case, the circumferential velocity should be zero at the outlet for the Francis runner for maintaining the uniform flow. Figure 14 indicates that the circumferential velocity at the outlet is approximately zero with the slight deviation in the case of front loading.

Figure 12: Streamline in the draft tube

Figure 13: Flow angle versus normalized span
5. Conclusion

After the studying various blade loading profile, it is concluded that blade loading had the direct impact on the performance. The better performance runner can be designed by following the front blade loading profile. The swirl flow in the draft is suppressed when the blade angle throughout the span is greater than outlet blade angle. When the blade angle at any span is lesser than outlet blade angle causes a change in the curvature of the runner blade causing drastic pressure loss in the runner. Additionally, the better performance and uniform flow in the runner is achieved by front blade loading with blade angle throughout span is greater than outlet blade angle.

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Nomenclature

- $v$: Non-dimensional Specific Speed
- $\omega$: Angular Speed of Runner
- $Q$: Volumetric Flow Rate
- $E$: Specific Hydraulic Energy
- $N$: Rotational Speed of Runner
- $\beta_1$: Inlet Blade Angle
- $\beta_2$: Outlet Blade Angle
- $D_1$: Inlet Diameter of Runner
- $B_1$: Inlet Width of Runner
- $D_2$: Outlet Diameter of Runner
- $D_1^*$: Variable Inlet Diameter with Meridional Plane
- $D_2^*$: Variable Outlet Diameter with Meridional Plane

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