Influence of guide vane clearance on internal flow of medium-specific speed Francis turbine

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Abstract. In Francis hydro turbines, a small clearance between the guide vane blade and the facing plate is crucial in pivoting the guide vane blade and controlling the flow rate of the turbine. This clearance-to-blade height ratio is inversely proportional to the scale of the hydro turbine. Smaller hydro turbines have higher clearance-to-blade height ratio than larger ones. Most of the time, this clearance is not included in the simulation model for it can cause inconsistencies between the computation and model turbine. This paper is focused on the various guide vane clearance and its influences on the flow. Firstly, steady numerical calculation of the turbine model (casing, stay vanes, guide vanes and draft) with and without guide vane reference clearance was carried out using the commercial code ANSYS CFX. Experiments were conducted on model turbine of specific speed 180 [m-kW] with guide vane reference clearance. The inter-blade pressure near the leading edge (LE), mid-chord and the trailing edge (TE) of the guide vane were obtained using the pressure sensor located along the circumference of the turbine casing. The result shows a good agreement between numerical computation and experiment. Furthermore, guide vane models with different clearance heights were simulated and the impact on runner inlet energy loss was investigated. In conclusion, it can be clarified that the roll-up flow from the guide vane clearance interacted with the main flow upstream of the runner which then caused loss to the turbine performance.

1. Introduction

Francis turbine is an inward-flow reaction turbine that combines radial and axial concepts. It converts both the pressure energy and kinetic energy in a fluid to the mechanical energy at the runner. For achieving high efficiency at the runner, the fluid has to enter into the runner with appropriate designed angle. These optimized entering angles of the fluid are guided by the guide vanes (GV). These guide vanes have pivoted support with the external control mechanism to direct them to the runner vanes. Although, guide vanes are stationary, they perform their periodic movement based on the flow or the load variation of the turbine. These movements are either automatically controlled through servo mechanism or manually controlled by an operator.

In order to aid the movement mechanism of guide vane, a small clearance gap is unavoidable between the facing plates and the guide vane. In Francis turbine, the clearance-to-blade height ratios are inversely proportional to the scale of the hydro turbine. So, these clearances gaps have higher influence on the small-scale hydro turbines and their efficiency.

Limited researches have been conducted for clearance gaps. Zhao W et al [1], studied leakage flow through the clearance gap between two stationary walls using a simplified model developed at Water
Power Laboratory. Eide S et al [2], studied the effect of clearance gap induced by head cover deflection and its effects on the flow around guide vanes. Watanabe T et al [3], computed the flow through both a stationary turbine cascade and a rotating turbine using compressible Naviers-Stokes solver. Many times, these guide vane clearance numerical investigations are carried out using simplified models or they are not included in the simulation model causing inconsistencies between the computation and model turbine. Therefore, this study focuses on the guide vane clearance and investigation of its influence on the runner upstream using numerical computation, and the accuracy of the numerical computation were verified with the real model turbine operating under same condition.

This paper unifies major prospects related to the clearance gap effects with reference to that research. The focus of this work is to investigate the accuracy of the numerical computation of the experimental model turbine followed by the various guide vane clearance-to-blade height ratio interactions with the runner-upstream main flow and its effects on energy loss.

2. Guide vane clearance-to-blade height ratio

Figure 1 shows the cross-sectional view of the horizontal type model of Francis turbine with the location of its guide vane clearance gap. Guide vanes are responsible for the flow control in the turbine. The gaps on the passage and the presence of the bluffed body and shaft promote leakage and secondary arouse flow respectively.

Brekke H [4], in 1988 illustrated loss at different region from inlet to outlet of high head Francis turbine. The total loss in the high head Francis turbine is around 5% to 6%, during the operation in Best Efficiency point (BEP). With minimum clearance gap, around 1.5% loss occurs through leakages. Hence, the loss due to leakage is one of the major portions of the inside-turbine loss.

Theoretically, these gap ratios must be directly proportional to the scale of the hydro turbine (the size of the runner inlet). This means either the runner inlet diameter $D_1$ or guide vane height $B_g$ increases or decreases the clearance ratio to its proportionality. Even so, in real practice, the guide vane clearance is not designed based on the proportionality ratio of the scale of the hydro turbine. Yet, it is designed based on the assembly perspective in order to make a less-trouble assembly of the guide vane with the facing plates.

Figure 2 on the graphical representation shows the comparison between the experimental clearance values to its power station clearance values of a low specific speed Francis turbine based on $D_1$ and $B_g$. It is very clear that the experimental assumption of the guide vane clearance-to-blade height ratio does not match with the power station clearance-to-blade height ratio. To be more precise, the experimental clearance-to-blade height ratio is directly proportional to the scale of the hydro turbine. On the other hand, in power station application clearance-to-blade height ratio is inversely proportional to the scale of the hydro turbine.
Based on the above data, the study of the research has been designed to investigate the guide vane clearance flow mechanism when there is an inverse shift of clearance value referring to its experimental assumption using the numerical computational method. Furthermore, a comparison has been made between the computational results and the experimental results for the accuracy study of numerical computation method.

![Figure 2](image-url). Experimental and power station guide vane clearance value based on D1 and Bg.

3. **Computational/ Experimental conditions and models**

A specific speed of Francis turbine model with various guide vane clearance was investigated using the numerical computational method. A guide vane clearance model of 0.18mm was analysed numerically and experimentally to clarify the inconsistencies of the computational model and the experimental model.

3.1. **Computational and experimental conditions**

A specific speed of 180 [m-kW] model turbine was selected for this research. Table 1 presents the relevant design values for the model turbine at BEP.

| S.N | Parameter                        | Symbol | Unit  | Value  |
|-----|---------------------------------|--------|-------|--------|
| 1   | Head                            | H      | m     | 13.0   |
| 2   | Flow rate                       | Q      | m³/s  | 0.4    |
| 3   | Rotational speed                | N      | min⁻¹ | 700    |
| 4   | Specific speed                  | Nₛ     | m-kW  | 180    |
| 5   | Runner inlet diameter           | D₁     | m     | 0.3892 |
| 6   | Number of runner blades         | Zₘ     | -     | 15     |
| 7   | Guide vane passage width        | Bᵥ     | m     | 0.0861 |
| 8   | Chord length of GV              | lᵥ     | m     | 0.0887 |
| 9   | Number of guide vanes           | Zᵥ     | -     | 18     |
| 10  | Number of stay vanes            | Zₛ     | -     | 18     |
| 11  | Guide vane clearanceᵃ           | -      | mm    | 0.00, 0.08, 0.13, 0.18, 0.23, 0.28 |
| 12  | Guide vane clearanceᵇ           | -      | mm    | 0.18   |

ᵃ Numerical computational model clearance values.
ᵇ Experimental model clearance value.
The above-mentioned parameters are used to build up both computational model and experimental model. From table 1, it is very clear that the conditions of both computational and experimental method are similar. First, the numerical computational analysis was carried out with non-clearance model followed by various clearance-to-blade height ratio models. The reason for the choice of this study condition was to investigate the influence of the guide vane clearance when it alters from its designed clearance-to-blade height ratio value based on the assembly perspective and to probe its influence towards the runner upstream flow.

On the other hand, an experiment was conducted in a well-equipped hydraulic laboratory creating the same conditions as that of the numerical computational model along with the designed guide vane clearance of 0.18mm on both hub and shroud sides for a comparison study with the numerical computational model including the designed clearance value.

3.2. Numerical computation model
A commercial code ANSYS CFX was used for the steady numerical calculation of the turbine casing, stay vanes, guide vanes and draft. The meshes were generated considering 5% convergence criteria in Grid Independent Analysis. Separate computation was performed for each of the domain with its operating boundary conditions.

Figure 3 shows the mesh models. ICEM (powerful meshing software) and Turbo Grid meshing features of ANSYS (Analysis System Software) 18.0 were used to generate structural mesh for the computational models. The flow cascade consists of 15 runner vanes, 18 stay vanes and 18 guide vanes. The stacking near the hub, casing and tip was close enough to approximate y+ values in single digits. Table 2 presents the mesh details of the components.

| S.N | Domain             | Total nodes   |
|-----|--------------------|---------------|
| 1   | Spiral casing      | 1361987       |
| 2   | Stay vane (SV)     | 3305502       |
| 3   | Guide vane (GV)    | 6877368       |
| 4   | Runner blade (RB)  | 7851480       |
| 5   | Draft tube         | 1303231       |
| 6   | Clearance mesh layer | 10           |

In contradiction with the practice, this computational approach implemented the practice of using the guide vane clearance along with the computational model. Non-clearance domain and five different guide vane domains with clearance gaps of 0.08, 0.13, 0.18, 0.23, 0.28mm were used. These selections were made based on the site observation with its assembly perspective.

3.3. Experimental model

3.3.1 Test set-up
Development of the experimental test set-up was targeted to produce a real plant site with an advantageous, adjustable head and flow rates to test various model turbines on specific speed (N_s).

Figure 4 shows the experimental set-up of the model turbine. The number mentioned in the diagram represents various parts of the experimental set-up. It is an open looped system, in which an intake pump (6 units) sucks the water from the base tank and the output flow from the pump passes through the upper tank and reaches the inlet sluice valve. Later when the inlet valve opens, the water from the inlet passes through the hydro turbine and flows back to the base tank taking the route of the
flow path. The air pressure inside the tank is maintained as the atmospheric pressure using the adjustable valves. An overflow valve has been placed on both the tanks which help to test the models with various heads.

Figure 4. Experimental test set-up.

3.3.2 Guide vane static inter-pressure measurement design
A special type of design has been employed in order to measure the inter-blade pressure of guide vane. Figure 5 (a) and (b) represent the sectional view of the model turbine and the location where the static inter-blade pressure recording position is designed.

Figure 5. (a) Sectional view of model turbine (b) static inter-blade pressure recording locations.

Pressure sensors are placed on the guide vane wall at two locations along the circumference of the turbine casing. The inter-pressures are recorded at 11 circumferential points with their respective
measuring angle (θ) along the chord length of the guide vane from leading edge, mid-chord to the trailing edge. The equipment used to measure the static inter-blade pressure is differential-pressure sensor (DSA3207). All the measurement of this study is conducted with flow rate of 0.4 m³/s. This is the design flow rate at BEP used in the numerical computation. The static pressure measured at the casing inlet and draft outlet is used to determine the hydraulic head (H) of the turbine. All the experimental measurements are recorded with Japanese Industrial Standards (JIS) permissible limits.

4. Results and discussions

This paper presents the results of two cases; first, a comparison between the computational and the experimental results measuring the static inter-blade pressure of the guide vane along the leading edge, mid-chord and the trailing edge. Next, the result of guide vane models with various clearance-to-blade height ratio and the impacts on the energy reduction towards the runner inlet.

4.1 Computation and experiment static inter-blade pressure comparison (BEP guide vane opening)

This section compares the overall performance of the computational model and the experimental model in more quantifiable terms. The static inter-blade pressure of location A and B is measured respectively on the measuring point angle (θ) as referred in figure 5(b). The static inter-blade pressure is presented as a pressure coefficient as defined bellow:

\[ C_p = \frac{P_{\text{inlet}} - P_{\text{local}}}{\rho g H} \]  

(1)

Here, \( P_{\text{inlet}} \) value for both computational and experimental is measured from the mid span at an angle 122.5 [location-A], 282.5 [location-B].

Figure 6 shows the measured static pressure coefficient at location A and B for designed guide vane opening of 80% with inlet diameter (d_{in}) 0.509m and outlet diameter (d_{out}) of 0.422m. All the data is measured at the BEP relative to its guide vane openings. The measuring points are divided into leading edge (4-points), mid-chord (3-points) and trailing edge (4-points) along with the circumferential location of the turbine casing.

The comparison results in figure 6 shows a good agreement between the experimental and numerical computational model. The static pressure coefficient values for both model matches with each other at their respective points.

Figure 6. Computation and experimental model static inter-blade pressure comparison graphs [BEP].

The static pressure drops are higher at the points located near the blades mid-span at leading edge angle (117.5, 122.5, 277.5, 282.5) and trailing edge angle (137.5, 142.5, 297.5, 302.5) in location A and location B when compared to the points located near the guide vane blade surfaces (suction side and pressure side of the adjacent blades) of the leading edge and trailing edge angle. On the other side, the mid-chord pressure drop increases along the tangential flow direction. This concludes that the computational result matches with the experimental result at BEP guide vane opening. This result paved the root to continue the study for the various guide vane openings. From figure 6, it is clear that
the result does not change with respect to the locations. So, for the further comparison study, location-A is selected.

4.1.1 Computation and experiment static inter-blade pressure comparison (various guide vane openings)
In this session, the inter-blade pressure for guide vane openings (110%, 90%, 70%, 50% and 30%) are presented for both the computational and experimental model at location-A. Figure 7 shows the results for various guide vane openings. It is notable that the normalized pressure of the trailing edge of the computational model starts to deviate from the experimental model as the guide vane opening changes from its designed value. However, the leading edge and the mid-chord normalized pressure values still show an acceptable match.

Figure 7. Computation and experiment model static inter-blade pressure comparison graphs [various guide vane openings].
The analysis of the static pressure measured values for both models shows negligible pressure drop as the flow passes from the leading edge to trailing edge as expected. Based on the results, it is clear that the pressure-drop near the surface is less when compared to the blades mid-span locations. This is due to the flow near the blade surface that moves with less turbulence and the turbulence increases towards the mid-span due to the reaction flow among blades. It can be estimated that the velocity and the pressure circulation along the circumferential direction of the trailing edge for larger guide vane openings become non-uniform. Hence, this non-uniform pressure circulation might also be one of the reasons for the mismatch of the pressure drop between the computational and the experimental model.

Since the computational model shows a good agreement with the experimental model with their respective guide vane openings, therefore, it fixes the erratic problem of the computational model. Refering to the results, further study has been continued to analyse the effects of the guide vane clearance when it deviates from its designed clearance-to-blade height ratio.

4.2 Effects of various guide-vane clearance towards the runner-inlet energy loss

A study has been conducted to understand the clearance flow mechanism and its effects on the energy loss towards the runner inlet when the designed clearance ratio value changes. Keeping 0.18mm clearance as the designed clearance value, a study was conducted at BEP guide vane opening by varying the clearance ratio by 0.08mm, 0.13mm, 0.23mm and 0.28mm.

Several cases of clearance flow at the guide vane have also been reported by past authors [5, 6, 7]. In this study, figure 8 shows the total head loss value from guide vane leading edge to trailing edge particularly to its guide vane clearance value. From figure 8 (a), it is clear that the clearance head loss also increases as the clearance ratio increases as expected. The maximum head loss of 0.34% was measured at 0.28mm. From figure 8 (a), it is evident that the head loss increasing ratio decreases as the clearance value increases.

Figure 8. Head loss: (a) total loss from LE to TE (b) divided loss from LE to TE.

Figure 8 (b) shows the divided head loss percentage from LE to TE. It explains that the average head loss percentage from the guide vane LE to mid-chord is around 27.5% and the head loss from the mid-chord to the TE is around 72.5% in all the cases respectively.

Figure 9. Flow field pressure loss vector contour: (a) 0.00mm clearance (b) 0.18mm clearance.
Figure 9 (a) and (b) shows the flow of the field pressure loss vector contour in the computational non-clearance model and clearance model. The vector planes for both cases are placed at 0.17% from the shroud side. The result explains that the pressure-loss vector close to the leading edge starts to develop cross-flow but still its direction and magnitude are similar to that of the main stream flow. Later, from mid-chord to TE, the cross-flow becomes stronger and the magnitude and direction become perpendicular to the main stream. Subsequently, the pressure loss is much higher at mid-chord to TE when compared to LE to mid-chord. The cross-flow indicates that there might be a possibility of change in the velocity component value from the designed BEP guide vane velocity component. Figure 10 explains about the convergence height along $B_g$. The convergence height is kept as 0mm to 4mm from the shroud side at which the turbo components ($V_m$ and $\alpha$) starts to converge to the designed values after recovering from the cross-flow effects. In this study the shroud side alone is analyzed due to the reason of matching the experiment model.

Figure 11 shows the effects of clearance flow on the velocity meridional at the guide vane outlet. Based on the cross-flow result, equivalent velocity triangles are drawn from shroud to the converging height. The results of figure 11 (a) explains that the velocity meridional ($V_m$) at shroud starts with a lower velocity value form the designed meridional velocity and it gradually increases until 0.02% of the guide vane passage width. Then, it decelerates and reaches the convergence at 0.23% of the guide vane passage width. In the case of the clearance models, the convergence height has been doubled up to 0.46% of the guide vane passage width when compared to the non-clearance model.

Figure 10. Height of convergence along guide vane passage width ($B_g$).

Figure 11. Velocity component at guide vane TE: (a) from shroud side to convergence height (b) at convergence height.

Figure 11 (b) shows the results of the velocity meridional at the convergence height. The meridional velocity value of the clearance model has decreased by 2.75% from the non-clearance...
model. It can also be described that the velocity of meridional decreases as the clearance increases until 0.18mm of the clearance value, and from 0.18mm to 0.28mm of the velocity meridional remains unchanged even the clearance value increases. Thus, it explains that the velocity meridional decreasing ratio starts to converge from 0.18mm clearance value.

Figure 12 shows the results of the guide vane outlet flow angle at 0.00mm to 0.28mm clearance values. Near the shroud, the flow angle is higher and it gradually decreases as it reaches the convergence height. The convergence height for both non-clearance and clearance models is around 0.46% of the guide vane passage width.

![Figure 12. Outlet flow angle at guide vane TE: (a) shroud to convergence (b) at convergence.](image)

The clearance-model flow angle is higher when compared to the non-clearance model until 0.23% of the Bg. On the contrary, the flow angle of the clearance model starts to decrease from the non-clearance model when it reaches its convergence height.

Figure 12 (b) explains the flow-angle results of both models at convergence height (0.46% of the Bg). It concludes that the velocity of flow angle decreases as the clearance value increases. Maximum of 2.51% of flow-angle decrease is recorded at higher clearance ratio. The decreasing flow angle ratio convergence is similar to that of the velocity meridional convergence. Such changes on the velocity meridional (V_m) and the outlet velocity flow angle (α) at guide vane outlet have considerable consequences on the turbine performances by loss of energy towards runner inlet.

### 5. Conclusion

A model of Francis turbine with specific speed of 180 [m-kW] has been developed to investigate the inconsistence between the numerical computational model and the experimental model by comparing their guide vane static inter-blade pressures. An experimental model was designed in such a way to record the static pressure of the guide vane at LE, mid-chord and TE along the circumference of the turbine casing. Presented measurement has been conducted at 0.18mm clearance value by varying the guide vane openings. Moreover, a study was conducted using numerical computational model to investigate the flow mechanism by varying its guide vane clearance-to-blade height ratio from its designed ratio. Head loss and the velocity component analysis are scrutinized to evaluate the energy loss at the guide vane outlet.

It is concluded from this study that the static inter-blade pressure comparison result between the computational model and the experimental model has a good agreement for its designed and larger guide vane openings. For smaller guide vane opening (GV-30%), the computational result at TE shows a difference of 8.69% from the experimental result.

The investigation of the various guide vane clearance-to-blade height ratio flow mechanism result concludes that the head loss increases as the clearance-to-blade height ratio increases. The maximum head loss of 0.34% was recorded at the higher guide vane clearance of 0.28mm.

A strong cross-flow occurs from the guide vane clearance due to the high pressure difference between the guide vane surfaces. Meridional velocity (V_m) decreased by 2.75% and the flow angle decreased by 2.51% when compared to non-clearance model at higher clearance value of 0.28mm.
Such changes on the meridional velocity \( (V_m) \) and the flow angle \( (\alpha) \) at guide vane outlet may cause energy loss towards the runner inlet which affects the turbine performances.

An estimate conclusion can be derived from the above result that the leakage cross-flow from the guide vane clearance might be the trigger factor for the development of vortex induced flow towards the runner. Thus, further investigation of the characteristics of the cross flow and its effects on the runner inlet flow is necessary, to identify its effects on turbine performances. An experimental method would be developed to investigate more about the consistencies between the computational model and the experimental model based on the turbo velocity component.

**Nomenclature**

- \( P \) Static pressure
- \( C_p \) Static pressure coefficient, \( C_p = \frac{P_{\text{inlet}} - P_{\text{local}}}{\rho g H} \)
- \( y^+ \) Non-dimensional turbulence wall function
- \( \text{LE} \) Guide vane leading edge
- \( \text{MID} \) Guide vane mid-chord
- \( \text{TE} \) Guide vane trailing edge
- \( \text{BEP} \) Best efficiency point
- \( V_m \) Velocity meridional
- \( \alpha \) Guide vane outlet flow angle
- \( B_g \) Guide vane passage width

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