Comparison of the effects of leakage flow from guide vanes of different hydrofoils using alternative clearance gap approach

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Abstract. Sediments flowing along with water through guide vanes induces abrasive and erosive wear in the GV surface. The effect of these wears differs with the shape of GV profiles. This study presents an alternative clearance gap method to compare the effects of clearance gap (CG) in the GV of Francis turbine with different hydrofoils. Numerical prediction of the performance of Francis turbine with symmetrical and asymmetrical hydrofoils is studied with the CG of 0 mm, 2 mm, and 4 mm thickness. Increase in the size of the gap deteriorates the performance of the runner. In case of asymmetrical GV profile, the effects in the runner is found to be less than the symmetrical profile. It is due to lesser pressure difference between the adjacent sides of the GV profile. With the alternative clearance gap approach, it distinguishes the effect of CG and the pressure pulsation due to rotor-stator interaction for a single numerical model.

1. Introduction
Sediment erosion in hydraulic machinery is one of the most affecting factors in terms of frequent maintenance of turbine components as well as the degraded performance of turbine at desired operating conditions [1]. Despite of having higher opportunity for harnessing the available hydro-energy sources in Asian rivers, same extent of challenges arises due to the large sediment concentration in the water. Asian rivers alone contribute to the 6.3E12 kg of sediment flux annually [2]. This has induced great challenges in operation and maintenance of the turbines during the turbine operation. Erosion in turbine component is related with the action of sediment flowing through the water that has an impact against a solid surface. Despite its own characteristics of having smaller size and higher roughness value, erosion in turbine components inside turbine is governed by flow of fluid [3].

Reaction type of machine, Francis turbines are widely used as well as highly affected hydro-turbines due to sediment erosion in most of the hydroelectric projects in South Asia [1]. These machines are capable of utilizing pressure and kinetic energy for imparting rotational energy in runner. Hydraulic losses occur when fluid flows through these turbines. The total loss in Francis turbine was found to be 5-6% at best efficiency point, out of which losses through the GV alone contributes to 1.5% of the loss or even higher for the losses originating from the clearance gap of GV [4]. Guide vanes are one of the main components of Francis turbines that are capable of controlling flow inside the turbine. For making such flexibility allowances are made near the top cover and bottom cover such that GV can freely rotate from its axis. This allowance is termed as ‘Clearance Gap’. During no flow condition size of clearance gap varies from 0.1mm to 0.3mm [5]. Size of the gap gradually increases due to continuous effects of abrasive and erosive wear inside clearance gap due to sediment erosion [6-7].
Guide vanes of Francis turbines are designed to contribute to required velocity of fluid flow entering turbine runner. The available pressure energy at spiral casing is converted into 50% of kinetic energy before it enters inlet of runner [8]. This causes highly accelerated fluid flow from the outlet of GV. The highest acceleration of fluid flow in GV causes unstable fluid flow which further drives to wear in GV surface and local cavitation. Shape of GV profile has significant effects on performance of Francis turbine for sediment laden projects. Past studies suggest the use of asymmetrical GV profile in these regions. Asymmetrical hydrofoil having 4% camber at 40% of the chord i.e. NACA 4412 type hydrofoil was found to be suitable instead of symmetric NACA0012 hydrofoil [9-10] for sediment affected projects.

The presented work studies fluid flow from clearance gap of asymmetrical and symmetrical hydrofoil contributing pressure oscillations in the runner. Alternative clearance gap method is used to study the overall flow inside clearance gap and compute pressure oscillations [11-12]. First part of the study includes numerical and experimental validation of the reference case. The validated numerical model is used in latter section to examine and compare leakage flow from clearance gap of both GVs.

2. Methodology

2.1. Numerical Model

The numerical model of the current case was designed for prototype turbine of Jhimruk Hydropower Plant, Nepal. Geometry of runner, guide vane and clearance gap were modelled in ANSYS ICEM CFD 17.2. Performance of designed geometry resembles real turbine of the power plant with specified head of 201.5m and flow of 2.35 m$^3$/s. Thus, acceptable runner geometry was used for further numerical analysis with several operating conditions. Operating conditions in this study refers to the condition of runner with and without leakage flow. Geometry thus created was used to generate structure hexahedral grid with overall quality of 0.3 and minimum angle of 18 degrees. In order to resolve high gradients near walls mesh were refined near walls as shown in figure 1. Separate blocks were employed to generate the hexahedral grid to each domain. Thus, there was three separate domain domains one rotating runner and others stationary GV and CG. Current case study uses full passage modelling to avoid shifting vortex filament in interface due to non-uniform pitch angles between stationary domain and rotating domain. The fluid flow conditions, and solver criteria is listed in Table 1.

![Figure 1. Structure hexahedral grid of the reference geometry](image.png)

Due to its robustness in predicting near and wall boundary flows for hydraulic machines, SST turbulence model was used. For a similar flow conditions on similar numerical study, SST model was found to give close prediction to the experimental results [13]. Commercial CFD solver ANSYS-CFX-17.2 was used for the steady state and transient state numerical simulations. The simulations use high resolution discretization in advection scheme and first order upwind scheme in turbulence equation.
Table 1. Boundary conditions and solver set up

| Boundary condition     | Mass flow rate: 2350 kg/s with cylindrical flow component (0, -0.34, -0.94) |
|------------------------|-------------------------------------------------------------------------|
| Inlet boundary condition|                                                                         |
| Outlet boundary condition| 0 Pascal, Average static pressure                                          |
| Turbulence intensity  | 5% at the inlet                                                          |
| Walls                  | No slip wall                                                             |
| Turbulence model       | Shear Stress Transport (SST)                                              |
| RV-GV interface        | Steady analysis: Frozen Rotor; Transient analysis: Transient blade row    |
| Solver precision       | Double                                                                   |
| RMS criteria           | 1E-4                                                                     |

2.2. Numerical validation

The numerical estimation of uncertainties was done using GCI technique [14], which is found to be effective in predicting the numerical uncertainties in flow field phenomena in a reaction type hydraulic machine, i.e. Francis turbines [15-16]. In this case, three different mesh of the computational domain were created. The first mesh being coarse one, other two mesh were successively increased such that the grid refinement factor with between two mesh remains at least 1.3. For this, elements consisting of each block were updated by 1.5x in i, j and k directions maintaining the overall quality of the mesh. Thus, three different mesh with number of elements 0.48M, 2.01M and 4.98M were generated. The uncertainty analysis was computed for no leakage condition. With each mesh scheme the flow conditions employed was as shown in Table 1. By computing the results with each mesh scheme, the final value of GCI was computed as,

\[
GCI_{fine}^{21} = \frac{1.25e_a^{21}}{r_p^{21} - 1}
\]  

Since, the purpose of the study was to observe the effects flow flowing from the GV on the runner, the pressure field built up inside the runner was assumed to be the critical parameter for measuring uncertainties. Thus, four different points were located inside the rotating domain as shown in figure 2 to monitor the pressure value.

![Figure 2. Location of points inside runner](image)

Table 2 presents the uncertainties in the numerical uncertainties in the pressure measurements with different mesh scheme. The numerical uncertainty was found to be 0.198% at location Point 3. Since, the computational domain consists only of runner and GV, fine mesh with 4.98M elements was selected for further numerical investigation.
Table 2. Uncertainties in the numerical solutions

|        | Pressure (kPa) at different locations |        |
|--------|--------------------------------------|--------|
|        | Point1  | Point2  | Point3  | Point4  |
| Coarse | 1450.70 | 1212.90 | 1080.68 | 1424.97 |
| Medium | 1273.85 | 1042.48 | 902.90  | 1264.64 |
| Fine   | 1258.14 | 1038.23 | 884.63  | 1260.53 |
| $GCI_{fine}$ | 0.1004 | 0.0075  | 0.198   | 0.006   |

Figure 3 and figure 4 shows velocity distribution and normalized total pressure distribution in between GV and RV, 50% location above the GV shroud. The location corresponds trailing of edge of 4 GV’s, making 60 degrees of circular arc. Normalized total pressure distribution in figure 4 refers to the ratio of difference of instantaneous total pressure and mean total pressure to the total available hydraulic energy as suggested by [17].

$$p^*[-] = \frac{P - \overline{P}}{(\rho E)_{BEP}}[-]$$

(2)

Since, the location of these measurements contributes to flow leaving 4 GV trailing edge, 4 peaks with higher uncertainty could be observed in any measurements. The plotted velocity and normalized pressure values were averaged by considering one complete runner rotation. Maximum discretization uncertainties were found to be near location corresponding the GV trailing edge. Highest uncertainty at these locations can be predicted because of wakes.

Figure 3. Velocity at GV outlet with extrapolated values and discretization error bars
Fig. 4. Normalized total pressure at GV outlet with extrapolated values and discretization error bars

Fig. 5. Pressure loading on GV

Numerically validated model was further investigated with experiment on 3-GV cascade rig developed by [18]. For this pressure measurement at different points in the mid-span of GV was carried out. Fluid entering GV hits leading edge resulting high pressure. The divided flow from leading edge travels from pressure side and suction side. The fluid travelling the pressure side experiences high pressure and thus lower velocity and opposite to the case of fluid travelling the suction side. Pressure difference across pressure side and suction side was observed when fluid flow from the GV. It was found that using current numerical model maximum error in pressure measurement was 8.8% towards the leading edge in pressure side of GV as shown in figure 6. This may be the influence of inappropriate stagnation as compared to numerical solution. At all other locations of pressure measurement error in
pressure measurement was less than 5%. Thus, validated model was used for further numerical investigations.

3. Results and discussion

3.1. Effect of leakage flow with clearance gap

Flow leaving the GV hits inlet of runner at certain design angle. Angle of fluid hitting runner is directly related to the performance of turbine in terms of efficiency as suggested by Euler turbine equation. Thus, at first section investigation was carried out to check if using new GV profile effects on stagnation angle. The stagnation angle of NACA 0012 GV profile was used for reference case. For new NACA 4412 GV profile stagnation angle was checked at inlet of runner from hub to shroud. Maximum change of 0.9 degrees at hub region and shroud region were observed at No Leakage condition. Higher change in stagnation angle near hub and shroud was due to effect of wall near those regions. However, at all other points change was less than 0.01 degree. Thus, investigation of NACA 4412 GV profile was done to maintain such impact angle in runner inlet.

![Figure 6. Position from Hub (0) to Shroud (1) (Left: 2mm leakage, Right: 4mm leakage)](image)

Further investigations were carried out to investigate the effects of leakage flow from GV. In this case, leakage of 2mm and 4mm were introduced at each NACA 0012 and NACA 4412 GV. As stagnation angle was not changing at no leakage condition, change could be seen introducing leakage flow. Figure 6 shows the effect of introducing leakage flow in stagnation angle. Increasing size of clearance gap was found to have inverse relationship to stagnation angle in case of NACA 0012 GV profile. The maximum reduction of 35% of stagnation angle was found with 4mm leakage in NACA 0012 GV. Oppositely, increase in size of clearance gap did not show significant effect on stagnation angle. Section 3.3 examines possible causes of change in stagnation angle with these two GV profiles.

3.2. Flow inside clearance gap

Fluid flow inside clearance gap is highly unpredictable. Fluid flow uncertainty arises due to effect of walls of GV surface and GV covers. The domain of clearance gap usually contains 0.1 - 0.3 mm of height termed as ‘Dry Clearance’ [5]. With high pressure fluid entraining sediment particles into GV domain, small size and roughness of these particles gradually shear off the walls due to abrasive and erosive wear. Size of clearance gap thus increases after certain operating hours. In such condition fluid can easily pass through leakage domain. Inside narrow clearance gap, high pressure fluid experiences the effects of pressure difference. Pressure difference arises from nearest GV domain having pressure side and suction side within itself. Pressure difference of fluid beneath the clearance gap influence similar pressure field inside the clearance gap. An attempt to measure velocity field inside the clearance gap was done by [5]. The detailed examination of velocity field by changing shape of GV profile was done by [19]. Figure 7 shows the PIV measurement of flow inside clearance gap of NACA 0012 and NACA 4412 GV profile.
In case of NACA 4412 GV profile flow was found to be less diverted from the chord of GV. This is due to lesser pressure difference between pressure side and suction side of adjacent GV surface. CFD calculation of velocity field inside the NACA 4412 GV profile also shows similar flow pattern. Oppositely, in case of NACA 0012 GV profile velocity field inside clearance gap was opposite from experiment as shown in figure 8. In case of experimental result for NACA 0012 GV profile, high intensity vortex resulted in localized cavitation flows that disturbed the PIV measurements. However, no such local cavitation occurs during CFD measurements. Thus, very high velocity fluid leaves the GV trailing edge instead forming the vortex.

3.3. Leakage flow
High velocity fluid leaves clearance gap of GV due to pressure difference on adjacent side. Figure 9 shows velocity streamline of fluid leaving clearance gap towards inlet of runner with both GV profile. As discussed in section 3.1, fluid flow from leakage is along main flow flowing through GV in case of NACA 4412 GV. In case of NACA0012 GV profile, fluid forms vortices beneath clearance gap. The possible cause of change in stagnation angle with increased size of clearance gap is due to behaviour of fluid flow travelling from clearance gap. Since in case of NACA 4412, flow from clearance gap were carried down by main flow, the effect in stagnation angle with increased size of clearance gap was negligible. However, in case of NACA 0012 GV profile, fluid flow from the clearance gap forms vortices with very high velocity that has tendency to deviate main flow from design angle.
3.4. Pressure pulsation due to leakage flow

Numerical model for current case consists of 24 GVs and 17 RV. Flow field leaving GV has a complex pressure and velocity field due to RSI [12][17]. In the first section of unsteady pressure measurement numerical simulations were carried out for both GV profiles with no leakage condition. A point near clearance gap inside runner domain was introduced to monitor pressure field from GV. This point would monitor pressure at each degree of runner rotation. Fourier analysis of pressure-time data was performed to determine prevailing frequency in turbine at the located position. Near clearance gap region, it showed frequency of 400 Hz corresponding to blade passing frequency (BPF). Both GVs were investigated at same transient operating condition to observe the fluctuations in pressure oscillation. Figure 10 shows FFT of the pressure-time signal at the monitor point. It was seen that NACA 4412 has least effect of pressure oscillation than that of NACA 0012 GV.

More detail investigations were carried to investigate effect of leakage flow alone in the performance of runner. Investigation of leakage flow on pressure oscillation were done by [10]. However, distinguished effect of leakage flow alone was not seen. The effects of leakage added up in first harmonics corresponding to BPF. Current case uses 12 clearance gaps alternatively to 24 GVs. Thus, there were 12 GVs consisting of clearance gaps and 12 GVs without clearance gap. Both GVs were further investigated using this technique and compared with the pressure oscillation. With this technique, first harmonics occurred at 200 Hz instead of 400 Hz. Thus, result of 200 Hz corresponds to effect of leakage flow.
Figure 11 show Fourier analysis of pressure-time data with 2mm leakage and 4mm leakage. On increasing size of clearance gap effect of leakage flow was also higher in case of NACA 0012 GV profile. It was seen that NACA 0012 GV profile has higher effect of pressure oscillation than that of NACA 4412 GV profile. The amplitude of pressure oscillation due to leakage flow in case of NACA 0012 GV profile was close to that of BPF for 4mm clearance gap. However, in case of 2mm GV profile it was lower. In case of NACA 0012 GV profile, vortex leaving the CG is carried down by main flow thus shifting its position. Thus, pressure amplitude due to leakage flow is least affected near clearance gap. In case of NACA 4412 GV profile, since flow leaving clearance gap mixes up with main flow, the vortex shifting does not occur. Thus, flow leaving clearance gap of NACA 4412 GV profile has higher pressure amplitude at 200 Hz than that for BPF. Though, effect in NACA 4412 GV profile was higher than RSI, the corresponding amplitude of NACA 0012 GV profile was higher in any case.

4. Conclusion

Numerical investigation of effects of leakage flow from clearance gap of GV was carried out with symmetrical NACA 0012 and asymmetrical NACA 4412 GV profile. Numerical simulations were carried out at best operating point and validated with reference experimental data. Leakage flow from clearance gap is one of the most effecting parameters for operation of hydro-turbine exposed to sediment erosion because of higher amplitude of pressure oscillation adding up to RSI. This causes unwanted vibrations in turbine. The detailed analysis of flow field inside the clearance gap of two hydrofoil showed formation of vortices due to pressure difference in both GV profile. The unsteady vortex flow passes into runner domain. The effect of vortex flow travelling from the asymmetrical GV was found to be lesser than that for symmetrical GV. These vortex flow flowing from clearance gap has tendency to deviate main flow flowing into runner. It was seen that increase in size of clearance gap has least effect in runner stagnation angle for asymmetrical GV whereas it was highest for symmetrical one. This would cause reduced performance of turbine in terms of hydraulic efficiency.

Fourier analysis of the pressure time signal at transient operating condition of runner indicated least effect of pressure pulsation for asymmetrical GV profile than that of symmetrical one. Alternative clearance gap approach was capable to distinguish effect of pressure pulsation due to leakage flow separately at 200 Hz. Thus, leakage flow was found to have significant contribution to pressure oscillation in runner.

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