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Original Paper

Selection of Optimal Number of Francis Runner Blades for a Sediment Laden Micro Hydropower Plant in Nepal

Binaya Baidar[1], Sailesh Chitrakar[2], Ravi Koirala[3] and Hari Prasad Neopane[4]

[1, 2, 3] Turbine Testing Lab, Department of Mechanical Engineering, Kathmandu University
Dhulikhel, Kavre, P.O. Box – 6250, Nepal
[1] binayabaidar@ku.edu.np, [2] sailesh@ku.edu.np, [3] ravikoirala@ku.edu.np
[4] Department of Mechanical Engineering, Kathmandu University
Dhulikhel, Kavre, P.O. Box – 6250, Nepal, hari@ku.edu.np

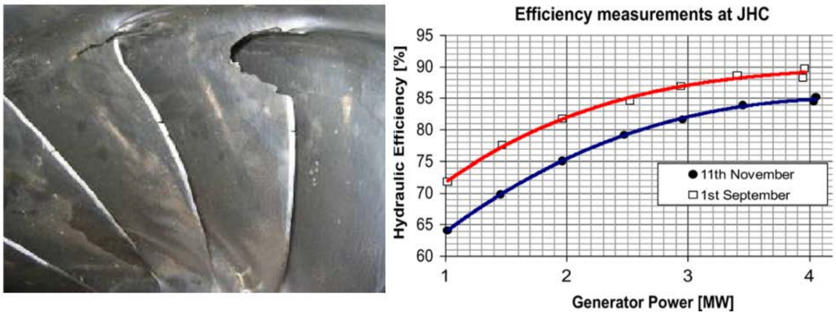
Abstract

The present study is conducted to identify a better design and optimal number of Francis runner blades for sediment laden high head micro hydropower site, *Tara Khola* in the Baglung district of Nepal. The runner is designed with in-house code and Computational Fluid Dynamics (CFD) analysis is performed to evaluate the performance with three configurations; 11, 13 and 17 numbers of runner blades. The three sets of runners were also investigated for the sediment erosion tendency. The runner with 13 blades shows better performance at design as well as in variable discharge conditions. 96.2% efficiency is obtained from the runner with 13 blades at the design point, and the runners with 17 and 11 blades have 88.25% and 76.63% efficiencies respectively. Further, the runner with 13 blades has better manufacturability than the runner with 17 blades as it has long and highly curved blade with small gaps between the blades, but it comes with 65% more erosion tendency than in the runner with 17 blades.

Keywords: CFD, Francis turbine, Sediment erosion, Efficiency, Performance, Optimum number of blades

1. Introduction

Micro hydropower plants (MHPs) play an important role in the rural electrification of Nepal. Majority of the MHPs use Pelton and Cross Flow turbine while the use of Francis turbine is limited. Francis turbine design and manufacture has much more complications compared to other types of turbines, at least in the context of Nepal. It is a site specific and therefore needs specific design to meet the site's specifications. Moreover, Nepal lying in the Himalayas with rivers heavily loaded with sediments creates severe erosion problem in the hydraulic turbine components. It is mostly due to the Quartz as it has enough hardness to erode the turbine material. Figure 1(a) shows the eroded Francis turbine runner blade during the first 4000 hours of operation at Jhimruk power plant in Nepal and Figure 1(b) shows its efficiency drop in one monsoon season. The drop at Best Efficiency Point (BEP) is 4% and estimated that 50% loss was due to leakage of labyrinth seals whereas 25% loss in efficiency was due to erosion [1]. Power plants have erosion problems despite of having well designed flushing and sediment handling system. Many researches have been conducted in order to make sediment friendly designs [2-4]. Most of the previous works are concentrated on material selection, mechanics of material, flow phenomena inside the turbine and to analyse their effects on design improvements.



(a) (b)

Fig.1 (a) Sediment damage in runner at Jhimruk power plant after one year of operation and (b) efficiency drop [1]

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Corresponding author: Binaya Baidar, Research Fellow, binayabaidar@ku.edu.np

Francis runner is exposed to hydraulic forces, both static and dynamic. Static forces are created due to pressure difference between pressure side and suction side; whereas dynamic forces are associated with the pressure pulsations induced by the secondary flows due to blade passing, leakage flows, draft tube surge and cavitation. Two considerations are generally taken for the selection of blade numbers in a runner; the first is the blades should withstand hydraulic forces and the other is that flow in a runner should not have much loss that deteriorates the hydraulic efficiencies at the design point as well as part flow conditions. However, it is also of great interest to see how the runners with different numbers of blades affect the sediment erosion problem, which is the point of interest in the current study.

Current technology using Computational Fluid Dynamics (CFD) is capable of design optimization of the hydraulic turbines both statically and dynamically [5, 6]. Fluid Structural Interaction (FSI) analysis is performed for the structural analysis of the runner after it is hydraulically designed [7, 8].Wang et al. [9] demonstrated that reducing comprehensively the number of runner blades can increase the turbine capacity and found 13 blades is the optimal choice for the case of a small hydropower plant. In spite of considering these aspects, sometimes, runner with splitter blades are necessary to improve the performance of a turbine. It is because the fewer blades on the runner outlet side helps in lengthening the blades towards the downstream side improving variable discharge characteristics and decreasing vibration in the draft tube, and increased blades on the runner inlet side eased the blade loading on the runner outlet side reducing load per blade and decreasing flow separation at inlet and cavitation [10]. In addition, other parameters like surface roughness also have the impact on Francis turbine performances which can also be estimated using CFD [11].

For high head Francis turbines, most of the torque is carried by the hub and hence it is desirable to have higher loading at the beginning of the blade. The number of blades *(Z)* in a runner is determined by the momentum at the runner blade *(M)* given by relation 1.

$$M_{runner} = Z \cdot a \cdot b \cdot R_M \cdot \Delta p \tag{1}$$

Where, a is the length of the blade, b is the height of the blade, R_M is the mean radius and Δp is the differential pressure.

The present paper deals with Francis runner design for the *Tara Khola* micro hydropower site and attempts to study the effect of blade number in a high head Francis runner exposed to sediment erosion using ANSYS CFX. The investigation is done in terms of minimizing the swirl flow, losses in the runner and improving efficiencies in the best efficiency point as well as at the variable discharge conditions considering sediment erosion.

2. Methodology

2.1 Turbine Design Methodology

Francis turbine design is a site specific and its blade shape is complex. Specific speed is a key to the turbine type selection and performance prediction. Lower specific speed corresponds to high head turbine with smaller inlet dimensions, while high specific speed has bigger inlet dimensions and corresponds to low head turbine. The specific speed (N_s) of the turbine depends on rotational speed of turbine *(N)*, shaft power *(P)* and net head *(H)* and given by equation 2.

$$N_s = \frac{N.\sqrt{P}}{H^{\frac{5}{4}}} \tag{2}$$

Shaft power is calculated from equation 3.

$$P = \eta_0 . w . H . Q \tag{3}$$

Where η_0 is the overall efficiency, w is the weight density of water and Q is the discharge through turbine. From the relations, it is clear that both head and flow are important parameters to define the specific speeds and hence the head.

Tara Khola MHP is supposed to have two units in this study, each having 114.6 m net head and 0.265 m^3/s flow rate. Considering 9.81 kN/m^3 as weight density of water, rotational speed of 1500 rpm and overall efficiency of 85%, the specific speed of the turbine is 63.66. In this range, either Pelton or Francis turbine can be a choice but Francis turbine is more compact and economical. The calculated specific speed i.e. 63.66 is quite low, which corresponds to the high head Francis turbine. Thus Francis turbine is chosen for the considered micro hydropower site.

The turbine is designed with the aid of the sophisticated in-house design code ‘Khoj’. The software is programmed in Matlab and capable of creating 3D runner profile and also handles sediment erosions, as discussed in [1, 12, 13]. The software is also capable of design modifications involving blade shape, load distribution and blade leaning. After a satisfactory design is made, the software generates curve files compatible to ANSYS and Pro/Engineer or PTC Creo Parametric for CAD, hydraulic and mechanical analysis. The process of designing a Francis runner starts with calculating the outlet diameter, number of poles in the generator and synchronous speed. From these known values, the inlet dimensions like the inlet diameter, the inlet angle and the blade inlet height are calculated. The main parameters determining these values are head and discharge of the available hydropower site. It begins with dimensioning at runner outlet considering no rotational speed at BEP. Detail documentation of the software can be found in [1].

Table 1 presents the turbine design data. These parameters were kept constant for all the cases considered here. The velocity triangles for the design turbine are shown in Figure 2. The 3D views of the runners with different blade numbers are shown in

Figure 3. It can be seen from the figures that keeping the main dimensions of the turbine constant, the shape of the blades are adjusted accordingly.

Table 1 Turbine design data

Turbine Parameters	Value
Rotation speed (rpm)	1500
Angular velocity (rad/s)	157.08
Outlet diameter (m)	0.261
Inlet diameter (m)	0.434
Blade height at inlet (m)	0.041
Peripheral velocity at inlet, U_1 (m/s)	34.14
Absolute velocity at inlet, C_1 (m/s)	31.95
Tangential comp. of C at inlet, C_{u1} (m/s)	31.61
Meridian comp. of C at inlet, C_{m1} (m/s)	4.66
Relative velocity at inlet, W_1 (m/s)	5.30
Peripheral velocity at outlet, U_2 (m/s)	20
Synchronous velocity, U_2 synchronous (m/s)	20.57
Meridian comp. of C at outlet, C_{m2} (m/s)	5.13
Relative velocity at outlet, W_2 (m/s)	21.20
Angle between rel. and peripheral velocity, β_2	14

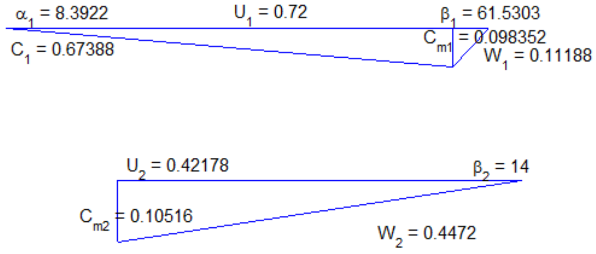
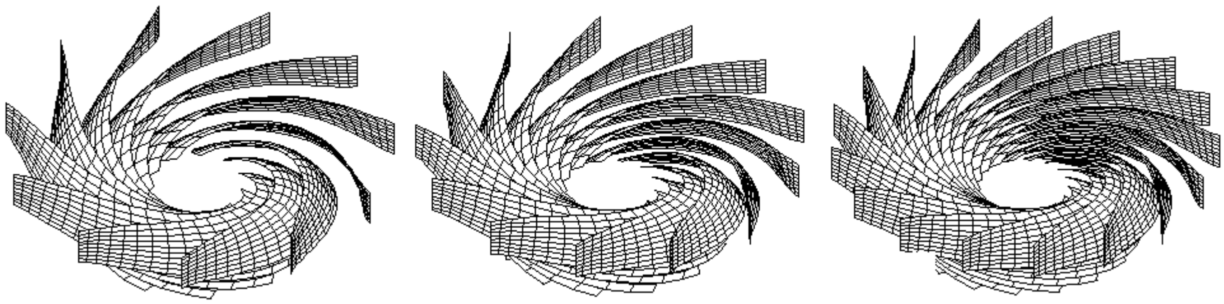


Fig.2 Velocity triangles



(a) (b) (c)

Fig.3 3D view of the runners with (a) 11 blades, (b) 13 blades and (c) 17 blades

2.2 CFD Methodology

The turbine design data and boundary parameters with the opening angle at the best efficiency point from 'Khoj' are imported to ANSYS CFX. The inlet velocities for all the cases are kept same for the consistency in the simulation.

The governing equations of viscous flow are based on conservation of mass, momentum and energy. The commercial code utilizes the finite-volume, multi-block approach to solve the set of unsteady Navier-Stokes equations in their conservation form. The temperature effect is negligible in the analysis. The instantaneous equation for mass, momentum and energy are summarized in the equations 4-8. [14]

The continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U) = 0 \tag{4}$$

The momentum equations

$$\frac{\partial (\rho U)}{\partial t} + \nabla \cdot (\rho U \otimes U) = -\nabla \rho + \nabla \cdot \tau + S_M \tag{5}$$

Where the stress tensor, τ, is related to the strain rate by

$$\tau = \mu(\nabla U + (\nabla U)^T - \frac{2}{3} \delta \nabla \cdot U) \tag{6}$$

The energy equation

$$\frac{\partial (\rho h_{tot})}{\partial t} - \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U h_{tot}) = -\nabla \cdot (\lambda \nabla T) + \nabla \cdot (U \cdot \tau) + U \cdot S_M + S_E \tag{7}$$

Where h_{tot} is the total enthalpy, related to the static enthalpy *h (T,p)* by

$$h_{tot} = h + \frac{1}{2} U^2 \tag{8}$$

Mesh Study

Meshing was performed in ANSYS TurboGrid. Automatic Topology and Meshing (ATM optimized) is set (Figure 4), which is easier to use than the traditional topologies. Traditional topologies like H/J/C/L, J, H allows a separate choice of topology type however there is need of control point adjustments and hence time consuming. On the other hand, ATM optimized allows to control over the global mesh size and create high-quality meshes also in the boundary layer with minimal effort, thus there is no need for the control point adjustment. It automatically selects the appropriate topology on the basis of the blade style and blade angles.

A factor ratio of 1.5 with RMS value of 1E-4 as residual criteria for convergence is chosen in the present study. The mesh independency test in Figure 4 shows that the convergence of under 1% occurred after 0.28 million nodes, hence to reduce computational cost, mesh count of 0.28 million is used. The y+ method is used as near wall element size specification and for the considered mesh the average value of y+ around the blade was 84.56. This value is satisfactory since the recommended value is between 30 and 100 [9].

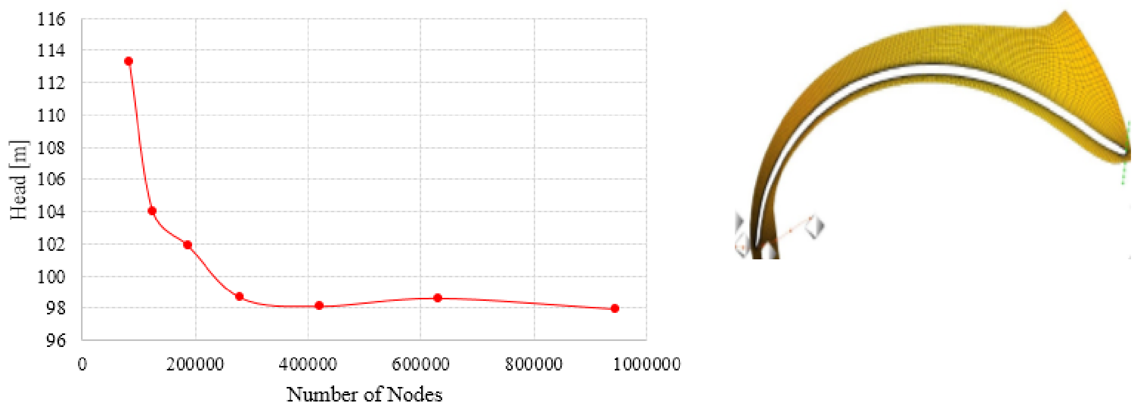


Fig. 4 Mesh independency test for the factor ratio of 1.5, RMS of 1E-5 with head (m) as monitoring point and ATM mesh

Erosion model

The model of Tabakoff and Grant [14] is used for the erosion modeling on the blade with Quartz-Aluminum erosion parameters. The erosion rate E is determined by the relation 9.

$$E = k_1 . f(\gamma). V_P^2 . \cos^2(\gamma)[1 - R_T^2] + f(V_{PN}) \tag{9}$$

Where, V_P is the particle impact velocity, $f(\gamma)$ is a dimensionless function of the impact angle in radian, parameters k_1, R_T and $f(V_{PN})$ are constants that depends on the particle/wall material combination and the impact velocity. This erosion model is relatively more reliable than Finnie erosion model as it is dependent on more parameters. The parameters for sediment data are included in Table 2.

Table 2 Sediment parameter data

Data	Value
Material	Quartz
Density (g/cm^3)	2.65
Diameter (mm)	0.1
Molar mass (kg/kmol)	1
Shape factor	1
Flow rate (kg/s)	0.07

CFD parameters

The mass flow at inlet and static pressure at outlet has been defined as the boundary condition. Water is taken as the working fluid with the mass flow rate of 265 kg/s at the inlet with an angle suggested by 'Khoj' for the design condition. It can be seen from Figure 5 that only a single runner blade passage has been modeled for the analysis. Although the velocity profile at the inlet of the runner lacks the deficits due to wakes from the guide vane, it is assumed that the presence of these wakes is negligible for steady analysis. Furthermore, a significant amount of computational time has been reduced.

Turbulence of medium intensity has been defined at the inlet. At the outlet static pressure of 1 atm was defined. The Shear Stress Transport (SST) type turbulence model was used. SST model is the hybrid type of model which shows good performance in the separated flows and adverse pressure gradients [14].

Table 3 includes the general CFX parameters which are also used in previous related researches [9, 15].

Table 3 General CFX parameters

Parameter	Type	Value
Total Node Count	Fine	282534
Factor Ratio		1.5
Reynolds no.		500000
Turbulence	SST	
Mass and Momentum	No Slip wall	
Analysis type	Steady	
Numerical scheme	Advection with high resolution, first order turbulence numerics	
Erosion model	Tabakoff	
Morphology	Particle transport fluid	
Rotational speed		1500 rpm

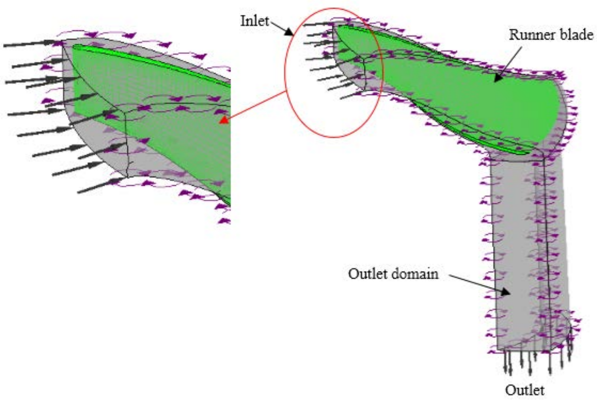


Fig. 5 Computational domain with inlet and outlet

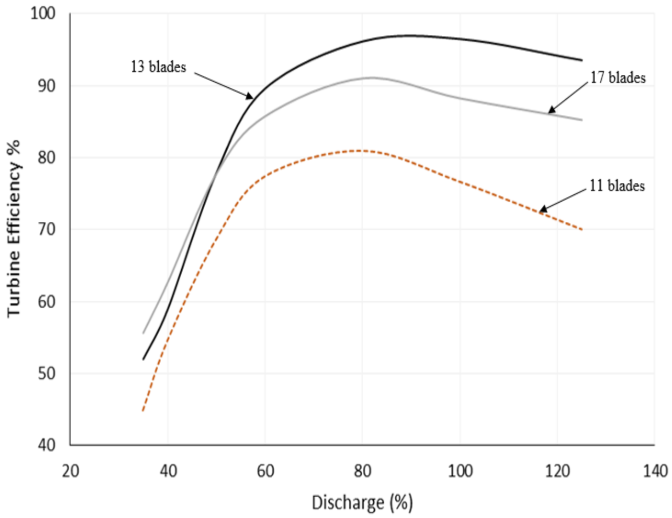
3. Results and Discussions

Prime number of runner blades is usually chosen turbine design. It is done principally to avoid harmonic interference with guide vane numbers and minimize the effects of rotor stator interactions. Hence numerically evaluated results are analyzed and compared for the runners with 11, 13 and 17 number of blades, which is under standard practice of choosing Francis runner blade number. The results are presented in two headings, first with the performance of the runner with different sets of blades and secondly with the sediment erosion effects in these sets.

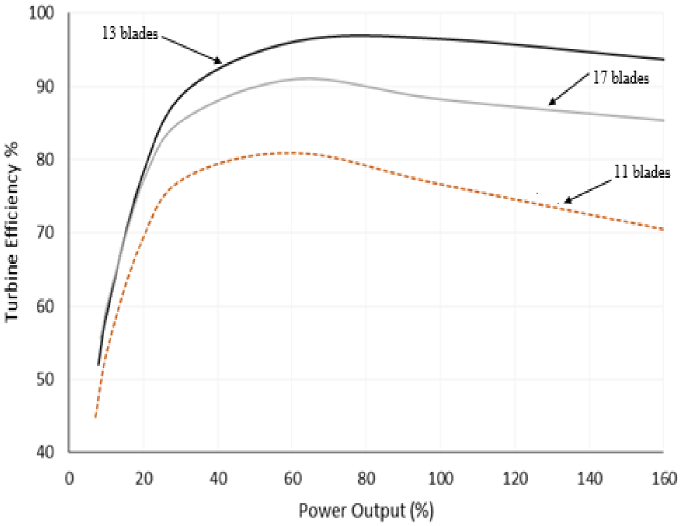
3.1 Performance of 3 sets of runners

The CFD predicted results are shown in Figure 6-8. Figure 6 (a) shows the turbine efficiency with respect to the relative discharge. The efficiency for the runner with 13 blades in the design condition is higher than the other runners. The runner with 13 blades shows 96.5% efficiency at design point whereas runner with 11 and 17 number of blades yields only 76.6% and 88.25% efficiencies respectively. These are hydraulic efficiencies which do not incorporate losses and leakages, and expect to drop. 75% - 85% efficiency is commonly achieved in the hydropower plant with Francis turbine [1] or even higher. Moreover at off-design conditions, the runner with 13 blades still performs better than others. It can also be noted that the curve for 13 blades has smooth performance at variable discharge conditions. However, below 50% part flow conditions, the runner with 17 blades is showing a better result than the other runners. At 35% part flow condition, the runner with 17 blades shows slightly better result, about 3% more efficiency than the runner with 13 blades. The blades for the runner are longer than the normal Francis blade for this specific site, and with higher number of blades the blockage effect in the runner could be the cause of lower efficiency at design discharge or higher and slightly better performance in lower discharge conditions than the runner with 13 blades. Referring Figure 6 (b), at different power output conditions, efficiency for the runner with 13 blades shows better performances than the other runners.

The velocity streamlines for all the runners at design condition is shown in Figure 7. There is maximum swirl flow in the runner with 11 blades. It is because the runner with 11 blades for the present flow condition has more gaps between the blades, resembling the partial flow condition, as a result the flow separates. The flow is separated from the inlet and vortices are generated. This causes pressure fluctuations and can cause severe vibration in the turbine. It can even cause vibration in draft tube when the swirl flow is decelerated resulting vortex breakdown causing pressure fluctuations [16]. Towards the outlet of the blades, losses occur due to cross flows and thus decreasing turbine efficiency. The flow is streamlined for both 13 and 17 number of blades. Hence it is concluded that the losses is not that significant in both of these cases, but the overall performance is better with 13 blades.



(a)



(b)

Fig. 6 CFD results showing (a) Efficiency vs. relative discharge, (b) Efficiency vs. relative power output

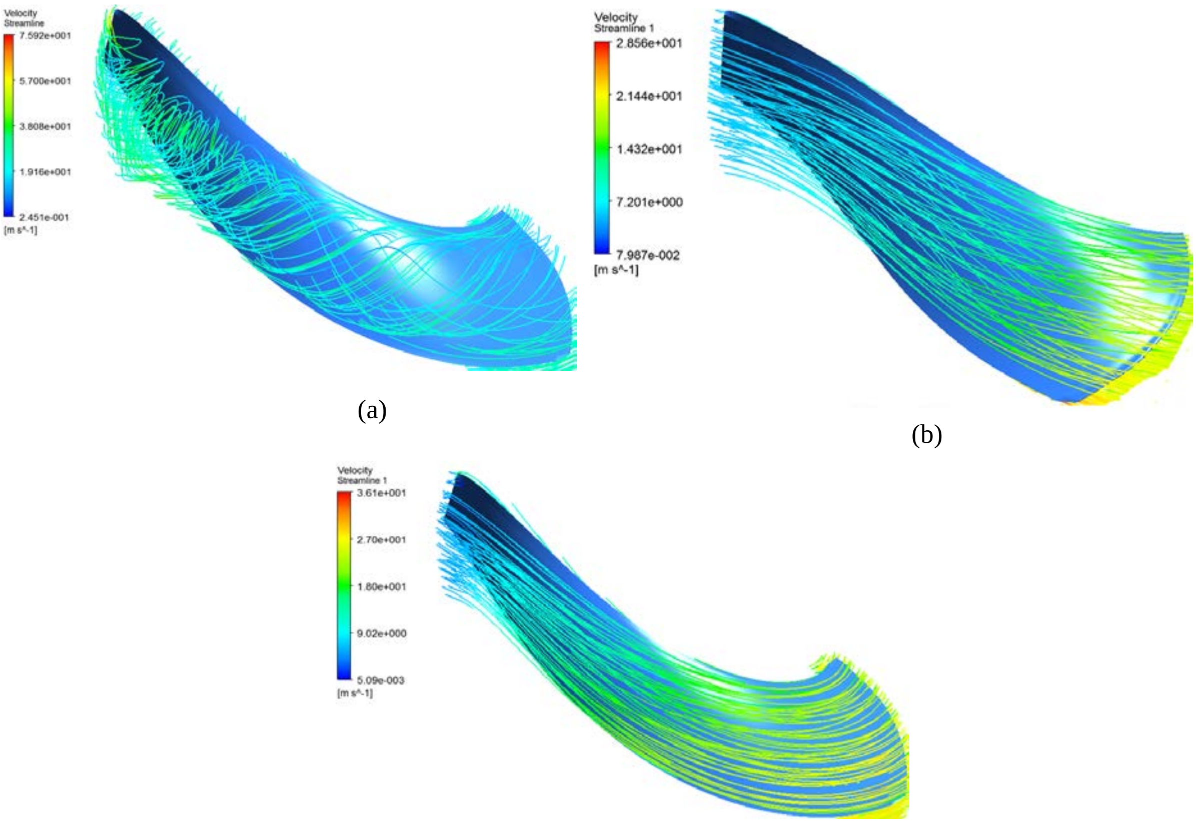
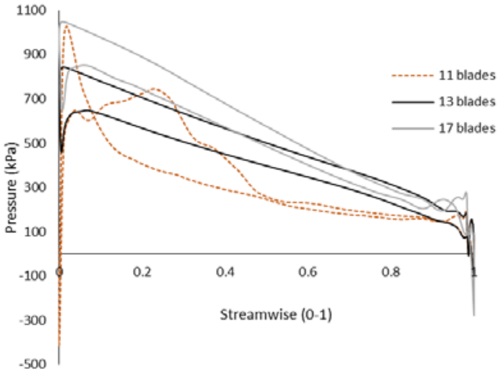
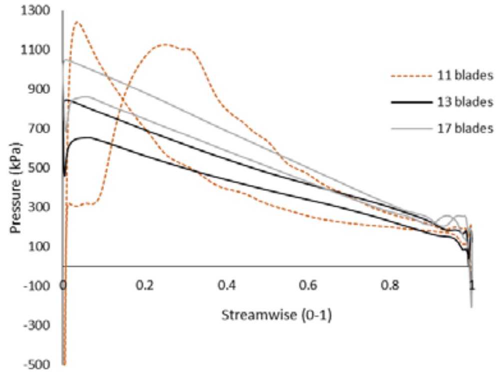


Fig. 7 Velocity streamlines at full load in (a) 11 blades, (b) 13 blades and (c) 17 blades

The blade loading curves in Figure 8 show the pressure distribution in the blade's streamwise locations at different spans. Higher number of blades reduces loading per blade, also at the outlet. It is desirable to utilize most of energy at the beginning of blade in the high head turbines. High tensions created due to this larger pressure difference need thicker profile than towards outlet. This analysis showed the blade loading is eased at the outlet with 17 blades than for 13 blades. For 11 blades, there is not smooth transition of pressure at all span positions, which also revealed that there are high pressure fluctuations due to unsteadiness. Special attention must also be given regarding the cavitation potential towards the outlet region of the blades. Low pressure towards the outlet of the blade signifies ample probability of reaching the vapor pressure of water at the given operating condition.



(a)



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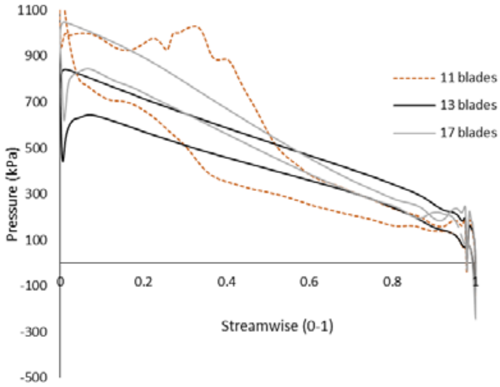

(c)

Fig. 8 Blade loading curves at full load in (a) 20% span, (b) 50% span and (c) 80% span

3.2 Sediment erosion on 3 sets of runners

Apart from the hydraulic performance of the runner, it is also important to consider erosion tendency in the region like Nepal where the rivers are heavily loaded with sediments causing severe destruction of the turbine parts and efficiencies as shown in Figure 1. The CFD analysis has been conducted to evaluate the effects of the change in blade numbers on sediment erosion in the runner blade surfaces. The erosion can be studied by observing the erosion pattern on the blade and quantitatively by calculating the erosion rate density, which is defined as the mass of the eroded particle per unit surface area of the blade per unit time. The unit is kgm-2s-1. Figure 9 shows the sediment erosion rate density on the runner blade surface with the particle tracking and Figure 10 shows the average erosion rate density on the blade surface with their corresponding efficiencies. The erosion rate density is highest in the runner with 11 blades, as expected. From the particle tracking in Figure 9, it can be seen that the runner with 11 blades has the maximum swirl from the starting of the inlet which has higher impact on the erosion. The figures signify that the erosion begins with the unsteadiness of the flow and the eroded region and the pattern in the runner configuration with 13 and 17 blades is similar to that shown in Figure 1.

Blade loading plays an important role in erosion, and decrease in the blade loading reduces the erosion tendency as studied in [15]. The graphs in Figure 8 confirms that the blade loading is eased at the outlet in the runner with 17 blades than 13 blades, thus the average erosion rate density is lower in the runner with 17 blades as shown in Figure 9-10. Runner with 13 blades has 65% more erosion tendency than with 17 blades. However, this value only represents a single blade and the calculation for the full runner reduces the overall erosion in the 13 blades runner, since the number of blade is lower. The runner configuration with 17 blades comes with a bit lower efficiency, about 8% at the design point and also in other part load conditions.

A final consideration has to be made regarding the manufacturing and repairing complexities of the turbines. Nepalese manufacturers do not have sufficient competences to produce Francis turbines due to complicated blade profiles. In this condition, the runner having long and curved blade profiles may add more difficulties to cast, weld and grind. The blade models shown in Figure 3 and Figure 7 infer that the runner with 13 blades would have shorter and less curved blades, which could be an additional benefit considering the practical issues.

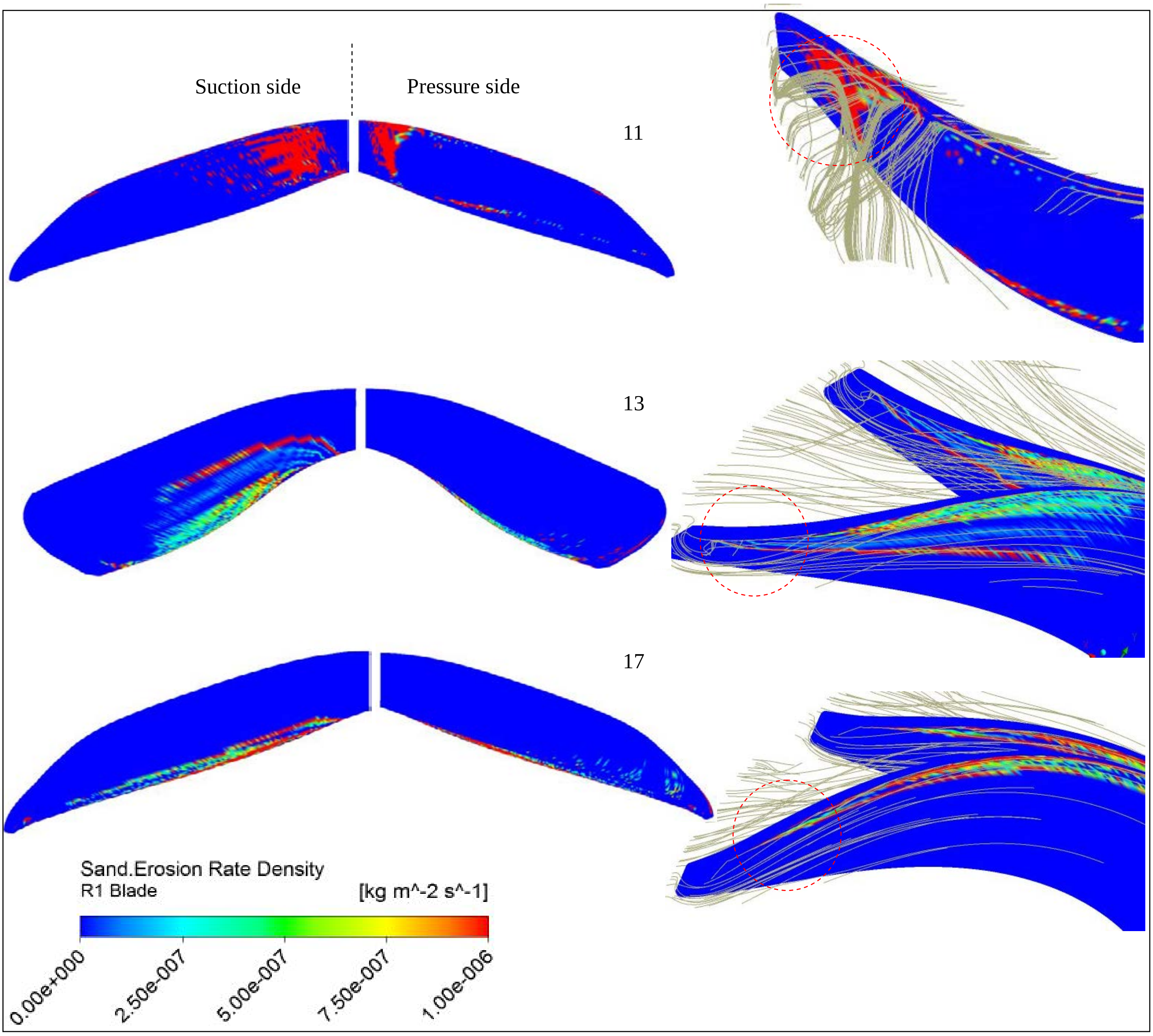

Fig. 9 Sediment erosion rate density and particle tracking

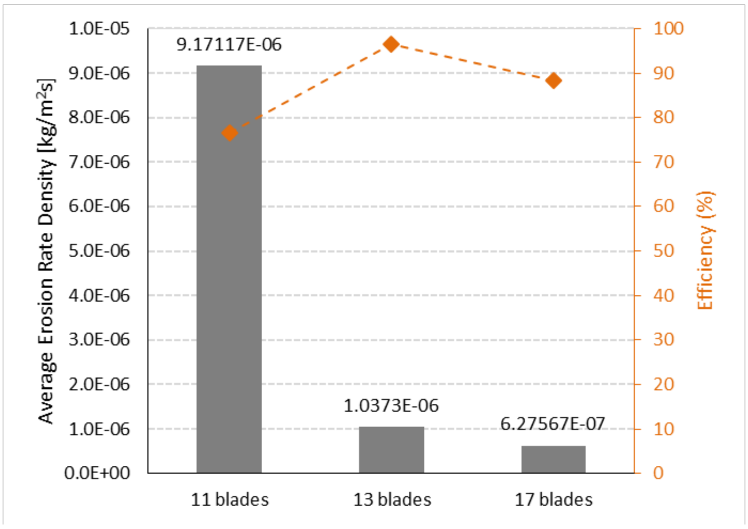

Fig. 10 Average erosion rate density for runner sets and corresponding efficiencies

4. Conclusions

CFD analysis of the high head Francis turbine for the sediment laden *Tara khola* micro hydro project in the Baglung district of Nepal was performed to investigate the optimum number of blades. The forces created by the hydraulic forces (static and dynamic) are important criteria for choosing the right number of blades in a Francis runner. The flow streamlines and blade loading in the runner is also highly dependent on the blade numbers chosen to ensure the minimum losses and higher efficiency. Further, in Nepal and the region where the rivers are heavily loaded with sediments, it is necessary to consider the sediment erosion tendency. Flow simulation using CFD has clearly shown that the runner with 11 blades has poor performance in its full load as well as part load conditions. It is also highly susceptible to sediment erosion. The runner with 13 blades has optimum efficiency and good variable discharge characteristics. The runner with 17 blades has 8% efficiency lesser than the runner with 13 blades, but slightly better performance at part load (3% higher at 35% part load condition). From the study it was found that the optimum blade number selection is 13 that yields higher efficiency, better variable discharge performances and manufacturability; however it comes with the cost of higher blade loading and thus higher sediment erosion tendency than the runner with 17 blades.

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Nomenclature

Δp	Differential pressure	S_M	Momentum source
h, h_{tot}	Specific static, total enthalpy	S_E	Energy source
M	Momentum at runner	λ	Thermal conductivity
Z	Number of runner blades	μ	Dynamic viscosity
τ	Stress tensor		

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