Chapter 2

A Review on Sediment Erosion Challenges in Hydraulic Turbines

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Additional information is available at the end of the chapter

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Abstract

Sediment constitutes several mineral compositions depending upon the geological formation and geography. In many of the rivers in Himalayas and Andes, Quartz is found as a main constituent (more than 50%), along with feldspar and other hard minerals. These particles have hardness more than 5 Moh's scale, which is capable to erode turbine components. In hydraulic turbines, flow is highly turbulent and unsteady, which can aggravate the erosion problems. Depending upon the nature of the flow, different components of turbines are eroded with different mechanisms. This chapter will provide a review on how various flow phenomena is responsible for particular types of erosion in turbines and their potential consequences. Some examples of the effect in existing power plants will be shown. This chapter will also discuss about some preventive measures that have been proposed and implemented to reduce the impact of the sediment particles in hydraulic machineries.

Keywords: sediment, erosion, Francis, Pelton, flow, turbine

1. Introduction

Hydropower is one of the cleanest forms of energy and has been used in many countries as a principle source of electricity. However, it has been reported that two-third of the world's feasible hydropower resources are still undeveloped. Out of these potential resources, more than 55% lies in Asia alone [1]. Despite the future prospects in hydropower development in this region, the geological problem seems to be a major obstacle. It has been studied that out of 20 billion tons of global sediment flux from rivers to the oceans per year, around 6 billion tons is contributed by Asian rivers, particularly from Indian subcontinent [2]. The problem of sediment handling, maintenance and operation of the power plants has become a serious issue.

The study of wear and erosion in general applications is found to be abundantly studied in literature. However, there are only few literatures related to erosion problems in hydraulic machineries. Brekke [3] reported that the hydraulic machineries working in the range of high Reynolds number, between 10^6 and 10^8 will normally be exposed to micro erosion, secondary flow vortex erosion and acceleration erosion. There have also been several endeavors made to predict and quantify erosion rates. The most general expression for erosion as studied by Truscott [4] was erosion $\propto$ (velocity)^n. This relation is an indication that higher velocity will increase the erosion because of higher turbulence. Apart from velocity, erosion is also a function of the shape, size and material of the particles hitting the turbine surfaces. Some more factors affecting the turbine's wear are explained in Section 2.

This chapter focuses specifically on the erosion of Pelton and Francis turbines. A detailed investigation of the eroded components of these turbines has shown that turbines erode depending upon the particular types of flow phenomena in particular regions. It has been explained that the secondary flow and sediment erosion in turbines are simultaneous in nature [5], which means that one phenomenon aggravates another. In most of the cases, the condition gets worse because of the cavitation induced after the surfaces are eroded. The combined effect of erosion, cavitation and secondary flow affects the turbine's performance drastically, causing the efficiency loss, damage and fatigue problems. Different mechanisms of flow in Francis and Pelton turbines inducing different types of erosion are explained in Sections 3 and 4.

Section 5 of this chapter explains some recent studies that have been done in the field of erosion in guide vanes of Francis turbines. Sediment particles wear the surface out of the clearance gap, which results in the leakage flow through the gap. This leakage flow mixes with the main flow, causing disturbances in the form of a vortex to downstream turbine components. This concept is one of the examples of the two-way effect between erosion and secondary flow. This section also explains some possibilities of minimizing the leakage flow by changing the guide vane profiles.

2. Wear and erosion

Erosion can be classified under one of the various forms of wear. According to the standard of ASTM 640-88, wear is defined as "damage to a solid surface, generally involving progressive loss of material, due to relative motion between that surface and a contacting substance or substances." Hence, the major cause of wear and energy dissipation is friction and it is estimated that one-third of the world's energy resources in present use is needed to overcome friction in one form or another [6]. The classification of the wear, in recent years have taken a very broad concept, compared to 1950s, when only adhesive, abrasive, surface fatigue and corrosion were considered as the types of wear [7]. The addition of erosion by solid particle and droplet impact were done by Bhusan [8]. The wear rate, which is the rate of material removal, depends upon the geometry of the interacting surfaces, types of interaction, material properties, load and surface pressure, surrounding temperature, humidity, atmosphere, surface properties and relative velocities between interacting surfaces [9].

Erosive wear or erosion is one form of wear caused by the impacts of solid or liquid particles on a solid surface. The flow medium contains particles that possess enough kinetic energy to damage metallic surface. The mechanism of the erosive wear is quite similar to the abrasive wear, but in the case of the abrasive wear, the eroding agent is much bigger in size and the angle of impingement is lower. The erosive wear on the other hand, is accompanied with relatively small particles with several number of wear mechanisms. These mechanisms are differentiated based on the impingement angle, size, shape and speed of the particles and mechanical properties of the base material. Stachowiak and Batchelor [6] have discussed seven different possible mechanisms for solid particle erosion, including abrasive erosion, surface fatigue, brittle fracture, ductile deformation, surface melting, macroscopic erosion and atomic erosion. In the case of hydraulic machinery, the first four mechanisms out of the seven are applicable, as shown in **Figure 1**. A low angle of impingement is favorable for the abrasive wear, as the particles are drawn across the surface after the impact. Similarly, if the speed is low, the stresses at impact are insufficient for plastic deformation or brittle fracture, which induces surface fatigue depending upon the endurance limit of the base material. If the shape of the eroding particle is blunt or spherical, the plastic deformation is more likely to occur, whereas, if the particles are sharp, the cutting or abrasive wear is more common. It is seen that for the ductile mode, the maximum erosive wear is found close to an angle of 30° whereas, for the brittle mode, the maximum erosive wear is found at around 90° impingement angle [6]. **Figure 2** shows the rotation of a particle due to the velocity profile near a boundary layer. When the particle is bigger than the boundary layer, it experiences different velocities along its height, causing it to rotate. Because of the irregular shape of the particle, the rotating movement enhances the abrasive wear.

A detailed survey on abrasive wear in hydraulic machinery was done by Truscott [4]. This study consisted of some of the critical findings related to the factors affecting wear that were

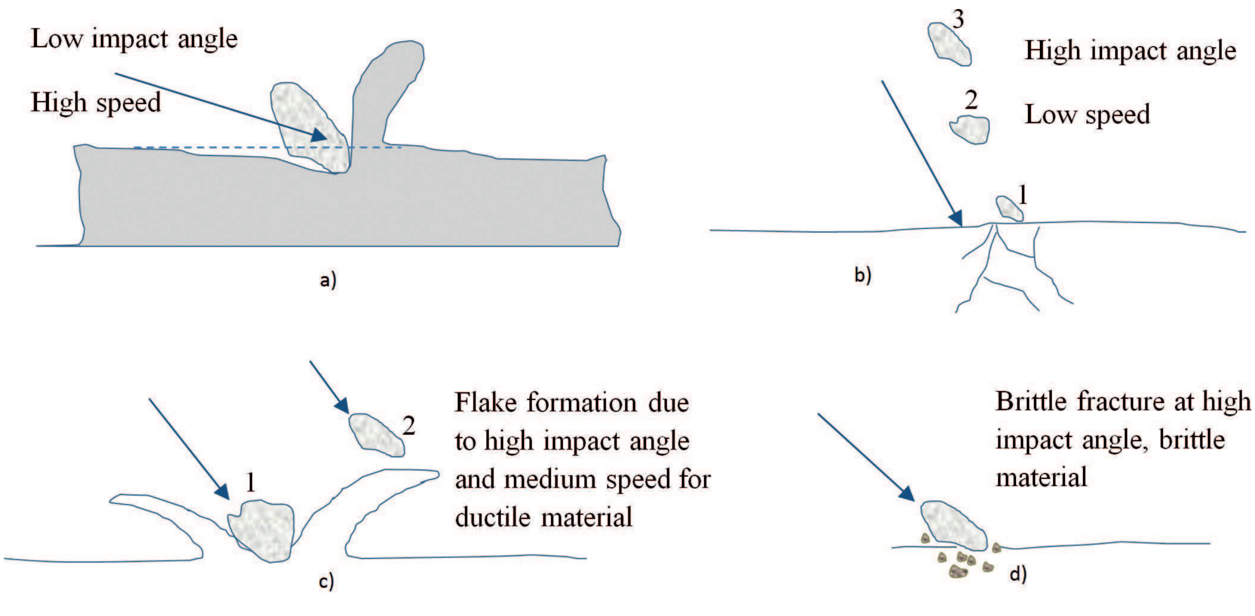

Figure 1. Various forms of erosive wear mechanisms: (a) abrasive/cutting erosion; (b) fatigue erosion; (c) plastic deformation; and (d) brittle fracture [6].

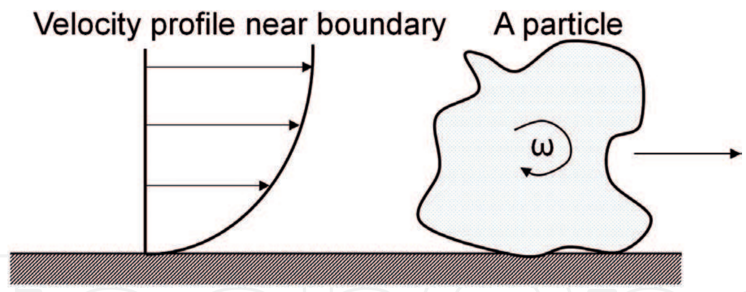

Figure 2. Rotation of a particle in a boundary layer.

studied in several works since 1950s. Brekke [3] characterized sand erosion in turbo-machines in three categories: (i) micro erosion, where fine particles with grain size less than 60 μm strikes the turbine surfaces with high velocity, (ii) secondary flow vortex erosion, caused by obstacles in the flow field or secondary flow in the corners of conduits and (iii) acceleration erosion, where acceleration of particles normal to the flow direction separates the particles from the flow direction and collide with the surface.

The basic factors affecting wear of hydraulic machines are (i) the properties of the solid particles (sand) like hardness, size, shape, relative density and concentration, (ii) properties of the eroded material like composition, structure and hardness and (iii) the operating condition like flow-speed, temperature and impact angle. Laboratory tests conducted by Wellinger and Stauffer, according to Truscott [4] show that for metals in general, wear increases rapidly once the particle hardness exceeds that of the metal for both scouring and impact abrasion. While most literature [10] state that in general, the absolute wear rate increases with grain size and sharpness, Wellinger and Worster, according to Truscott [4] state that wear is directly proportional to size for sliding abrasion, but is independent of size for direct impact. Wellinger also shows the effect of particle shape, with angular grains causing about twice the wear as compared to the round ones. Although the particle shapes are described qualitatively such as round, angular and semi-round, the actual shapes of particles are complex and cannot be quantified in simple mathematical terms. In the case of river sedimentation, concentration of particles is expressed in terms of parts per million (ppm), which is equivalent to milligram/liter or kilogram of particles in 1000 m^3 of water. It is mostly accepted that the wear increases with the concentration of particles [11]. Similarly, the increase in the temperature of the operating condition softens the material and hence the erosion rate increases. The conveying medium such as air, water or oil also plays a significant role on erosion rate depending upon their density, viscosity, nature of the flow (laminar or turbulent) and microscopic properties. Stachowiak and Batchelor [6] show that small amount of lubricant in the liquid medium reduces erosion rate due to restriction in the change in material properties during particle impingement.

Velocity of the fluid carrying particle and impingement angle are characteristics which affect wear significantly. The expression for the relation between erosion and velocity is often quoted as erosion $\propto$ (velocity)^n, where n depends on the material and operating conditions. This value is mostly taken to be 3, but Truscott [4] reported different value of this exponent, for instance 1.4 for steel St37 to 4.6 for rubber. The dependency of erosion rate on the impingement angles show contrasting results in many studies. While Bhusan [8] showed no erosion up to certain

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low impingement angles, Stachowiak and Batchelor [6] have shown erosion rates at even zero degree impingement. According to IEC 62364, the hydro-abrasive erosion depth in a Francis turbine can be estimated by using following equation [12]:

$$S = \frac{W^{3.4} \times PL \times K_m \times K_f}{RS^p} \quad (1)$$

where PL (particle load) is the integral of the modified particle concentration over time:

$$PL = \int_0^T C(t) \times K_{size}(t) \times K_{shape}(t) \times K_{hardness}(t)\mathrm{d}t \quad (2)$$

W is the characteristic velocity. In the case of guide vanes, it is the flow through unit divided by the minimum flow area at the guide vane apparatus at best efficiency point.

$W_{gv} = \frac{Q}{\alpha \times Z_0 \times B_0}$, Q is the discharge, α is the average shortest distance between adjacent guide vanes. Z_0 is the total number of guide vanes in a turbine and B_0 is the height of the distributor in a turbine. In the case of runner, it is the relative velocity between the water and the runner at best efficiency point.

$W_{run} = \sqrt{u_2^{\ 2} + c_2^2}$, $u_2 = n \times \pi \times D$, $c_2 = \frac{Q \times 4}{\pi \times D^2}$.

K_m is the material factor, which characterizes how the hydro-abrasive erosion relates to the material properties of the base material.

K_f is the flow coefficient $\left[\frac{\mathrm{mm} \times \mathrm{s}^{3.4}}{\mathrm{kg} \times \mathrm{h} \times \mathrm{m}^{\alpha}}\right]$, which characterizes how the hydro-abrasive erosion relates to the water flow around each component.

RS is the turbine reference size (m), which is the reference diameter (blade low pressure section diameter at the band) for the case of Francis turbines.

p is the value of the exponent, which is 0.25 for guide vanes, facing plates and runner inlet, whereas 0.75 for labyrinth seals and runner outlet.

C is the concentration of particles (kg/m^3).

K_{size} is the size factor (median particle size d50 (mm)), K_{shape} is the shape factor (round = 1, sub-angular = 1.5 and angular = 2) and $K_{hardness}$ is the hardness factor (fraction of par0ticles harder than Mohs 4.5 for stainless steel).

The erosion resistance of a material is seen to depend upon the material hardness and the impingement angle. The difference in the erosion resistance for two impingement angles can be seen in **Figure 3** [6]. Some materials such as cobalt have a very good erosion resistance at low impingement angle but the resistance decreases severely once the impingement angle is high. The formation of the martensites results in the improved hardenability and erosion resistance except at low impingement angles and in the case of low allow steels, the ferritic phase with sufficient spheroidal carbide to induce strengthening is very effective against erosion wear. The erosion resistance of austenitic steels (21Cr4Ni) strengthened with nitrogen was seen to be higher than a martensitic steels (13Cr4Ni) due to the distribution of hard carbides in the matrix of stabilized austenite [13].

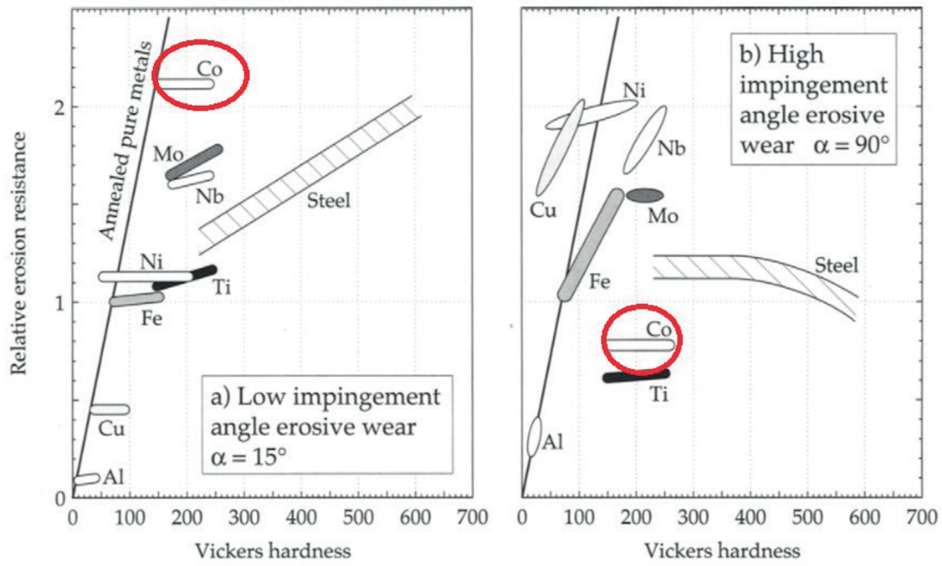

Figure 3. Erosive wear for materials with different impingement angles [6].

Some materials such as ceramics are recommended for applications where the working temperature is high. However, these materials are brittle and might result in the brittle fracture. In the case of hydro turbines, the prevention of the erosion is also done by applying coating on the surfaces of the vulnerable regions. The most common type of the coating is the tungsten-carbide (WC-Co), that uses 86–88% WC and 6–13% Co [14]. These coatings have excellent hardness, with better adhesion and large toughness.

3. Erosion in Pelton turbines

In the case of Pelton turbine, all the hydraulic energy of water is converted into kinetic energy before runner inlet. Because of the high head application of Pelton turbines, the velocity of water in the jets is usually higher than 100 m/s. Since velocity is the dominant factor for erosion (erosion $\propto$ velocity^3), the rise in velocity make the flow more turbulent, increasing the erosion rate. Brekke [3] has classified the erosion of Pelton turbine into four parts: (i) inlet and valve, (ii) nozzles, (iii) runner and (iv) wheel pit. Out of these components, nozzle system and runner are mostly vulnerable to erosion. In the nozzle, a strong turbulent effect occurs on the needle surface due to very high velocity at the nozzle outlet and the needle surface. In Chilime Hydropower Plant (2 × 11 MW, gross head: 350 m), average erosion in the needle tip during the maintenance was found to be around 0.3 mm [15]. In this region, the average quartz particle in the sand is more than 75%.

Some eroded components of Pelton turbines are shown in **Figure 4**. **Figure 4a** is the runner bucket of Khimti HPP. It shows that most of the erosion in this bucket is inside the bucket surface. However, in **Figure 4b**, a more severe case has been shown, where erosion is predominant in surface as well as in splitter. In **Figure 4c**, formation of ripple and grooves ahead of the webs are seen, which are explained later. In **Figure 4d**, the effect of both erosion and cavitation is seen.

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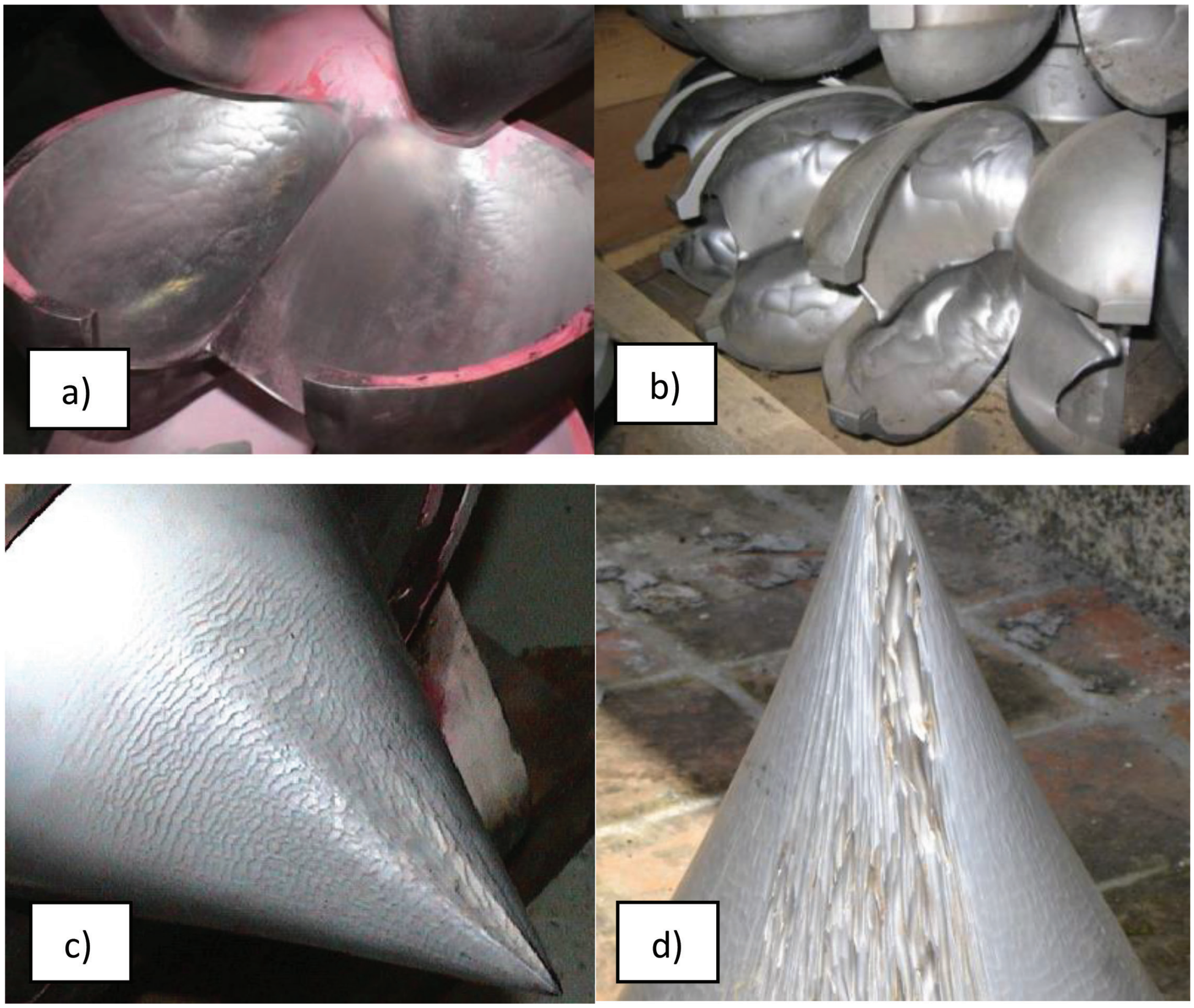

Figure 4. Erosion in Pelton turbines: (a) Khimti HPP [16], (b) Rangjung HPP [16], (c) Khimti HPP [9], (d) Chilime HPP [15].

3.1. Erosion of the nozzle system

The erosion pattern and intensity on the nozzle system depends on the operating condition. In the case of full opening of the needle, the mean velocity increases along the surface of needle due to gradual contraction of the fluid passage. This decreases the pressure on force towards the needle tip. Although the velocity is maximum towards the tip, the force at this point is minimum, which reduces the intensity of the erosion in this region. The erosion pattern is formed as ripples with circular grooves when viewed in axial direction (shown in **Figure 4c**). In the case of partial opening, the contraction of the fluid passage increases further, which reduces the pressure to such an extent that it can give rise to cavitation (shown in **Figure 4d**). In this case, a combined effect of erosion and cavitation can be seen on the needle surface. In the case of Khimti Hydropower Plant (5 × 12 MW, gross head: 684 m), two distinct grooves were seen ahead of the two webs to support needle guide to the vortices from the trailing edges of the webs [9].

The mechanism for the formation of the ripple pattern during erosion has not been specifically explained for hydraulic turbines. One explanation is the micro-roughness in the surface of the needle forming a turbulent boundary layer with eddies that erode the surface with both fine sediment and cavitation. Another explanation is the formation of Kelvin-Helmholtz vortex due to the velocity gradient between the interface of water and air. **Figure 5** explains the erosion types in the case of full and partial opening of needles as explained earlier. Apart from the needles, the outer ring of the nozzle also faces similar erosion problems. In summary, the nozzle system in Pelton turbines is one of the most eroded components due to a highly turbulent flow. The erosion intensity aggravates during the part load operations due to the combined effect of erosion and cavitation. It has also been explained that the erosion challenges in the nozzle system cannot be solved by a hydraulic design, but can be minimized by using hard material for turbine nozzle as well as its coating.

3.2. Erosion of the runner

It was analytically explained that the acceleration normal to the bucket of the Pelton turbine can reach up to 100,000 m/s^2 [3] At such a high acceleration, the sand particles separate from the flow and collide on the bucket surface. It has been shown that the erosion in the bucket is sensitive to its curvature (R). The location and types of erosion have been classified according to the size of the sand particles [9]. The coarse particles most likely hit the bucket inlet, eroding the surfaces around splitter. Fine particles glide along with water and strike on the outlet surface. It was also seen that the damages in the splitter and entrance lip were severe due to direct hitting of the particles. The erosion on the surface of the bucket on the other hand, was seen like a hammering effect. It has also been explained that the efficiency loss in Pelton turbines is primarily due to the erosion of the entrance lip. **Figure 6** explains the particle distribution inside the bucket and the erosion patterns due to different sizes of materials hitting the bucket.

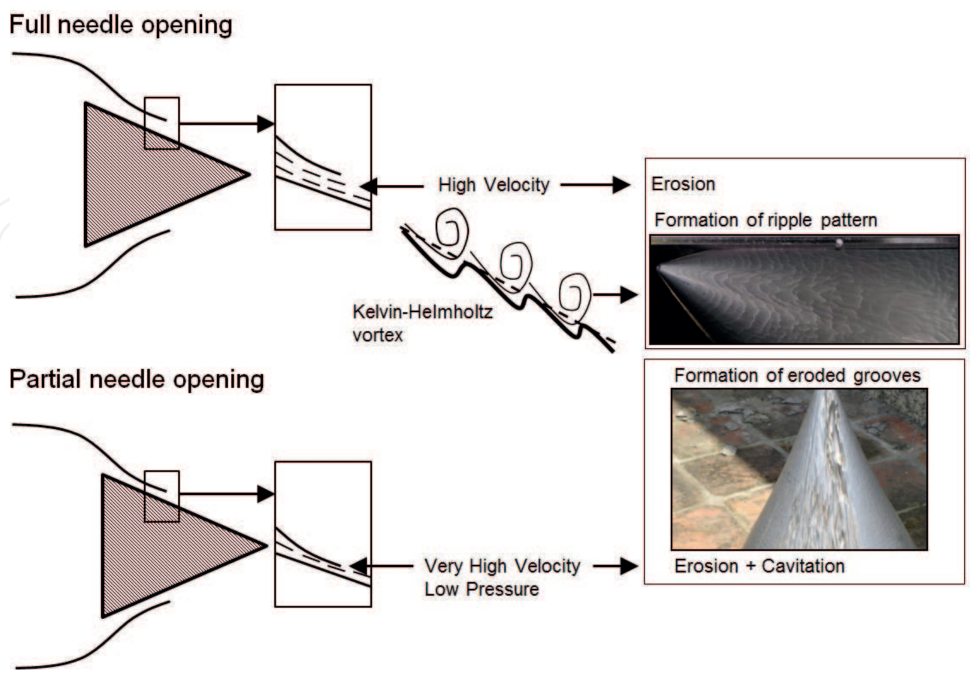

Figure 5. Erosion mechanisms in full and partial needle openings.

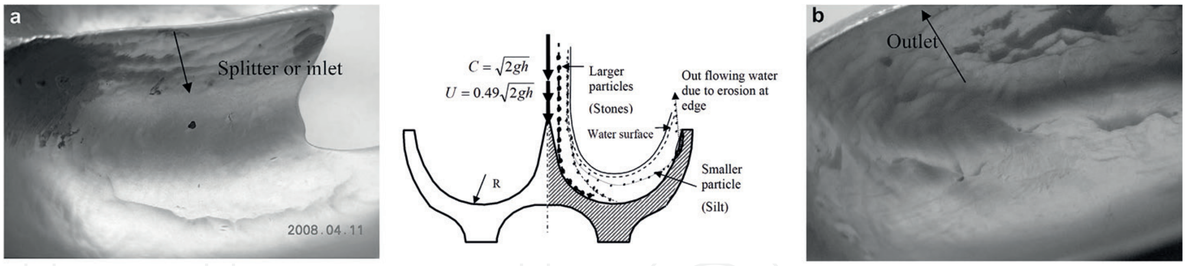

Figure 6. Erosion distribution at inlet and outlet of the bucket [9, 17].

Another experimental study also showed that the coarse particles with high velocity have tendency to move with jet and hit the splitter blade causing erosion due to shearing action of the sand on the surface [17]. The smaller particles on the other hand are flown out of the jet causing erosion on the depth and outlet of the bucket. The splitter tip was found to be eroded by plastic deformation and indentation whereas the depth of the bucket by plastic deformation and plowing.

Figure 7 shows pictures of eroded buckets from Andhi Khola Power Plant (5.1 MW) in Nepal. Erosion of such intensity causes the drop in efficiency of the turbine significantly. In a largely sediment affected power plants like these, the turbines are repaired on a regular basis. The efficiency loss due to erosion in Pelton turbines can be estimated from the graph shown in the **Figure 7**. Compared with the repaired runner, the efficiency drop in the non-repaired one is more than 10% for part load operations.

Based on these illustrations, it can be concluded that the fine sand particles mostly erode the nozzle system and coarse particles mostly affect the buckets. The erosion in the bucket due to fine particles is towards the outlet end, which is due to the separation of the particles at high acceleration. The coarse particles are not easily detached from the water stream, which on the direct impact with the bucket, can damage the splitter and region around it. Although the minimization of the erosion in Pelton turbines depend mostly on the material and types of coating used, some hydraulic design preferences can be used. They are: (i) using large radius of curvature in the location where the flow direction changes, (ii) using low number of jets and (iii) using large radius of the bucket and the nozzle. It has also been found [18] that an increase of D/B, that is, the ratio of pitch circle diameter and bucket width and/or decrease of specific speed enhances erosion.

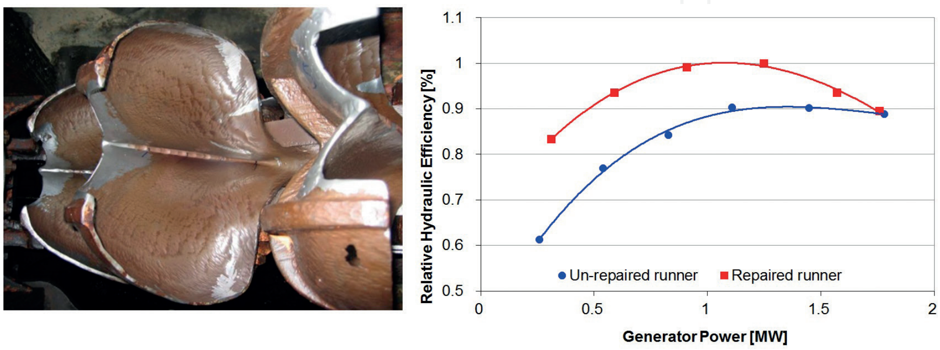

Figure 7. Eroded bucket in Andhi Khola power plant and efficiency drop due to erosion.

4. Erosion in Francis turbines

Francis turbines are also one of the seriously affected turbines due to sand erosion. Brekke [3] has classified the erosion in several components of this turbines which includes (i) the inlet valve system, (ii) the spiral casing, (iii) the pressure relief and/or by pass system, (iv) the guide vane system, (v) the runner and runner seals, (vi) the draft tube and (vii) the shaft seal. In Francis turbines, the highest absolute velocity is found to be at the guide vane outlet. It is the region where the maximum hydraulic energy of water is converted into kinetic energy, producing highly unsteady flow. The velocity distribution inside Francis turbine operating in best efficiency point is shown in **Figure 8a**. It shows that before the flow reaches the inlet of the runner, about 50% of the potential energy is converted into kinetic energy. The kinetic energy increases from about 10% to about 50% from guide vanes inlet to the runner inlet for a high head Francis turbine. This infers that the flow should contain very high acceleration inside the guide vane. Such high acceleration results in secondary flow, aggravating the erosion problems in case of sediment carrying fluid. The erosion removes the material away from the surface, causing more disturbances in the flow. **Figure 8b** shows an example (Lio Power Plant, 45 MW) of efficiency drop due to increase in the clearance gap from erosion [19]. The guide vane span in this case is 230 mm, and only by increasing the clearance gap by 1 mm (~0.5%), the efficiency drops by about 2%. Similarly, highest relative velocity in Francis turbines is found at the runner outlet. This causes erosion at the runner outlet due to high turbulence. The minimization of erosion in Francis turbines, unlike Pelton turbine, significantly depends on the hydraulic design of the vanes other than the turbine material and coating. It can be found from literatures that several investigations have been carried out so far to minimize the erosion by changing the blade angle distribution of the runner blades [20–22].

4.1. Erosion in the stay vanes

In stay vanes, the erosion occurs due to turbulences formed due to high velocity. **Figure 9** shows the eroded surface of a stay vane at Jhimruk HPP. Although the velocity in stay vanes

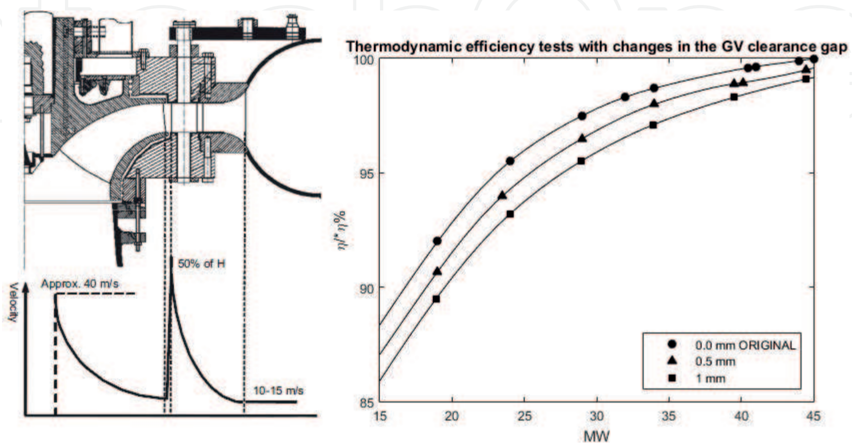

Figure 8. (a) Velocity distribution inside Francis turbine [23] and (b) loss in efficiency due to clearance gap in guide vanes (adapted from [19]).

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Figure 9. Erosion in stay vanes at Jhimruk HPP.

is less than in guide vanes and runner, the detached material from this region after erosion travels downstream, which causes more severe impact on the turbine.

4.2. Erosion in the guide vanes

In guide vanes, the erosion can be classified into four types, based on the flow conditions. They are:

1. *Turbulence erosion*: fine particles can erode the outlet of the guide vane due to high velocity, especially towards the suction side. At this region, the Reynolds number is in the order of 10^8, which is under highly turbulent regime. At such a high turbulence, erosion can be severe on guide vane surfaces as well as on facing plates.
2. *Secondary flow*: guide vanes are accompanied with complex nature of the flow, which gives rise to several forms of vortices. In this case, secondary flow is referring to the accumulation of flow in the corner between facing plates and guide vanes, which give rise to horseshoe vortex. These vortices increases the size of the gap, which brings more consequences as discussed in the next category.
3. *Leakage erosion*: guide vanes are accompanied with a small clearance gap at both ends to adjust the opening angle based on various operating conditions. In the case of sediment affected power plants, the hard fine particles mixed in water erode the connecting ends due to horse-shoe vortices. This erosion together with the head cover deflection due to water pressure increases the size of the gap. Due to the adjacent pressure and suction sides in guide vane, the flow passes through the gap from high pressure side to low pressure or suction side. At high acceleration, when the sediment particles enter in to the gap, it further causes abrasion on the guide vane ends and facing plates. This leakage flow disturbs the main flow in the suction side, which can be observed in the form of a vortex filament.
4. *Acceleration erosion*: the rotation of the water in front of the runner creates acceleration normal to the streamlines, which separates the coarse sand particles from the flow. This impacts the steel surface, which could lead to catastrophic destruction in the guide vane surfaces.

4.3. Erosion in the runner

In runner, the erosion can be classified into four types:

1. *Turbulence erosion*: in the runner, the highest relative velocity occurs at the outlet region. This increases the turbulence erosion due to fine sand particles.
2. *Acceleration erosion*: the highest acceleration is found close to the blade inlet. As discussed in the erosion categories of guide vanes, due to acceleration normal to the streamlines, the coarse sand particles are detached from the flow, which causes erosion of both guide vane and runner inlet surfaces.
3. *Erosion due to incorrect stagnation angle*: the inlet region is sensitive to incorrect pressure distribution and large difference in pressure between pressure and suction sides. The stagnation angle at inlet of the runner may change depending on different guide vane opening angles. It is also seen that the leakage flow through the clearance gap of guide vane mixes with the main flow in suction side, which forms vortex filament that hits the runner blade near hub and shroud. This vortex pushes the stagnation angle, which not only erodes the corners, but also induces cavitation.
4. *Cross flow erosion:* in some cases, cross flow from hub to shroud caused by incorrect blade leaning also increases horseshoe vortex in the blade roots. This may create sand erosion grooves at the blade inlet similar to guide vanes.

4.4. Erosion in the labyrinth sealing

The clearance between the stationary and rotating parts in labyrinth seals is between 0.5 and 1.5 mm depending upon the size of the turbine [3]. The erosion is severe in this region due to a strong turbulence in the flow. **Figure 10** shows the efficiency of Jhimruk HPP in an interval of 2 months. The total hydraulic efficiency loss after erosion in wet season is around 5%. At the same time, the leakage loss through the labyrinths was measured. It can be seen that the total loss due to leakage is between 2 and 4%. Hence, it can be inferred that the losses contributed by the leakage due to erosion of labyrinth seals is significant.

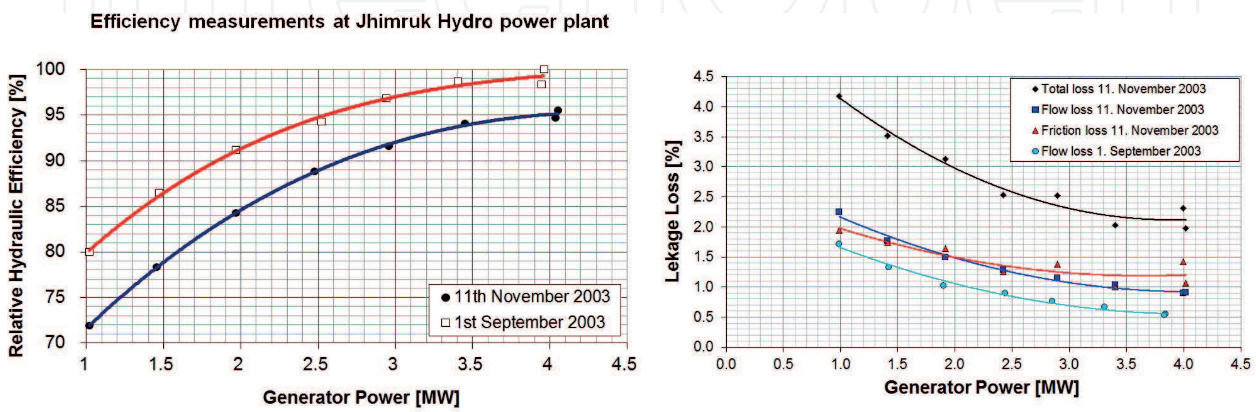

Figure 10. Efficiency measurements at Jhimruk HPP and losses from the leakage through labyrinth seal.

The erosion due to sediment particles not only causes the efficiency loss of the turbine, but also intensifies cavitation. Although the regions of cavitation and erosion is different, as shown in **Figure 11**, sometimes the eroded surface can also enforce pressure gradients in a localized regions, causing a cavitating pressure. Cavitation results in the formation of sharp edges.

Figure 12 shows some eroded components of Francis turbines. From these figures, it can be seen that the erosion in Francis turbine is predominant in particular places. The erosion depends on the nature of the flow in that location. In **Figure 12a**, erosion is shown at the outlet of the runner. It is the region where the flow leaves the runner with high relative velocity. Due to high turbulences, erosion occurs due to turbulence, as explained above. In **Figure 12b**, runner inlet towards the shroud end is seen to be more eroded than other places. This could be due to the incorrect stagnation angle driven by leakage flow through the guide vane's clearance gap. In **Figure 12c** abrasion is occurring at the guide vane ends due to leakage flow. The connecting shafts are also heavily eroded due to flow separation towards the trailing side of the shaft. In **Figure 12d** eroded grooves are formed on the facing plates due to horseshoe vortices. The grooves are formed within the range of some angles towards full GV opening. Hence, it can be inferred that these vortices affect the GV during full flow operation, for example, in the monsoon period, when the concentration of sediment in the flow is high.

Apart from using hard coatings, erosion in Francis turbines can be minimized by using new design philosophies for the design of the turbine blades. The design of runner blades follow complicated algorithms in both axial and radial directions. The blade angle distribution is maintained in such a way that maximum amount of hydraulic energy is converted into kinetic energy in the first half of the blade. However, it is found that the erosion in the blade is sensitive to the blade angle distribution, and by slightly modifying this parameter, the erosion can be minimized significantly [19]. The new design philosophy would reduce the relative velocity at the outlet, which could also affect the efficiency of the runner. However, the most optimized design in terms of erosion and efficiency can be chosen so that the new design can compensate the loss from the losses incurred due to erosion of blades in a period of time.

Similarly, for the design of the guide vanes, it is found that the change in the guide vane's hydrofoil profiles could minimize the erosion in runner and guide vanes at some operating conditions [24]. The hydrofoils are usually standard profiles such as NACA. For zero angle of attack, which is the case for best efficiency, the lift force is zero in a straight passage for profiles symmetrical around the chord. However, due to the circumferential orientation of

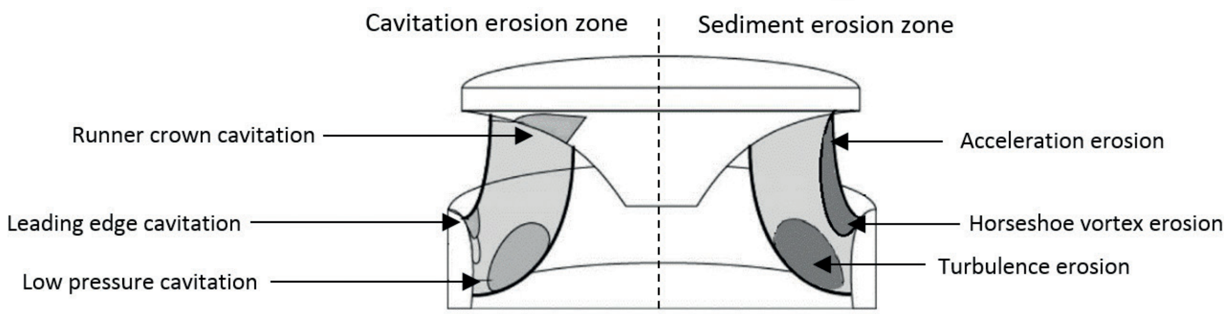

Figure 11. Cavitation and sediment erosion prone zones in Francis runner [5].

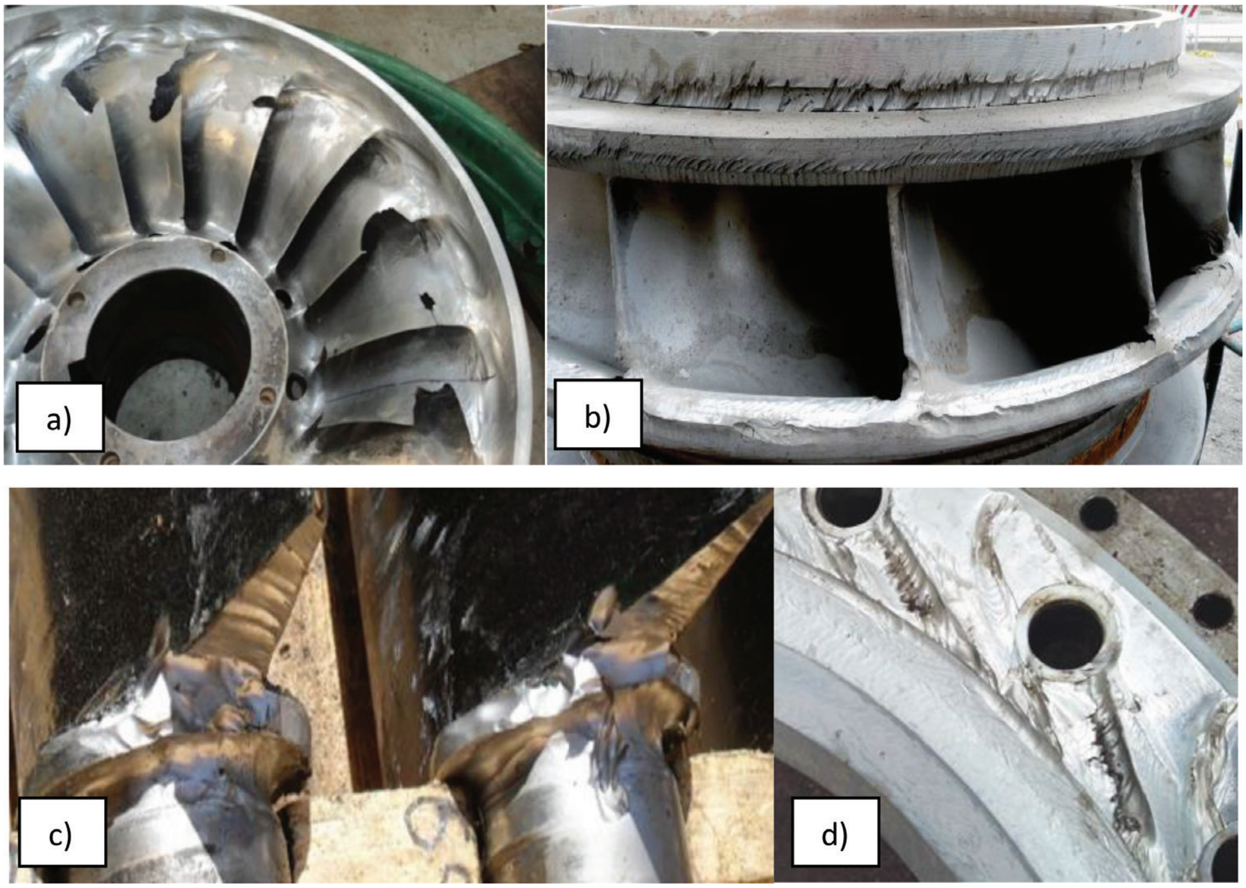

Figure 12. Erosion in Francis turbines at (a) runner outlet, Jhimruk HPP, (b) runner inlet, Cahua HPP, (c) guide vane faces, middle Marsyangdi HPP, (d) facing plates, Jhimruk HPP.

the guide vanes, pressure and suction sides are developed at a same chord-wise position. As a result, when the clearance gap grows in size, the flow from pressure side is driven into suction side. The leakage flow can be minimized by using an asymmetric profile, such that the resultant lift force in zero angle of attack is compensated by the orientation of the guide vanes.

5. Coating techniques and maintenance

The most common hydro-abrasive erosion resistant coating materials in hydraulic machines is thermal sprayed tungsten carbide cobalt chromium, WC-CoCr [12]. These coatings have a Vikers' hardness of 900–1200 HV at 0.3 kg loading [25], which is harder than feldspar (Mohs' hardness 6) and similar hardness as quartz (Mohs' hardness 7). Although it was difficult to apply coating on the surfaces of small and medium sized Francis runners, with recent technologies and use of robots, it has become possible to coat them completely. According to IEC 62364, coatings might initially result in reduction of the efficiency due to increased roughness, but can maintain a higher efficiency compared to the uncoated turbine over time. **Figure 13** shows an example of a coating applied on Francis turbine runner and its effect after 1 year of operation.

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Figure 13. Francis turbines at Nathpa Jhakri HPP in India, without and with coating, both after 1 year of operation [28].

A new production method was developed for Cahua HPP by applying a tungsten carbide based coating to Francis turbine runners and guide vanes [26]. The coated turbine increased the energy production by about 50% compared to the energy generated by the uncoated turbine during the same time period.

However, most of the hydro-abrasive erosion resistant coatings are not effective against cavitation. A high intensity implosion of cavity might locally destroy the coatings. In addition, the dimensional tolerances of the coatings have to be considered before applying the right coating. Hard-coatings are also sensitive to impacts of larger particles, such as gravels and stones. Since the thickness of the coating is around 300–500 μm, it could also hinder the detention of potential cracks in the base material [27].

An effective maintenance strategy must be implemented to get maximum output from a power plant. Frequent maintenance is required to run the turbines with good condition. Although power plants contain spare turbine parts, the downtimes during assembly and disassembly adds to the total cost, due to losses in electricity generation. Hence, design of turbines for easy dismantling, maintenance procedure and overhaul time is being continuously optimized for best production. These days, power plants employ pit stop maintenance for making the maintenance actions more efficient, saving both time and money. This philosophy implies more focus on preparation, planning, follow-up and evaluation than during shutdown maintenance.

IEC 62364 gives some criteria to determine the overhaul time due to hydro-abrasive erosion. Some parameters are listed below:

1. When the efficiency has deteriorated to an extent that it is economically beneficial to restore the unit to its design efficiency.
2. Water outflow through balancing pipes from the head cover indicating erosion of labyrinth seal.
3. Increase in the axial thrust indicating erosion of labyrinth seal.
4. The time taken by the unit to stop after the guide vanes are closed and the inlet valve are kept open, indicating erosion of the guide vanes and covers.

5. Pressure in the spiral casing with closed inlet valve and opened bypass, indicating erosion of the guide vanes and covers.

6. For internal inspection, if runner blade outlet is eroded more than two-thirds of its thickness (for large turbines) or completely abraded (for small turbines), clearance of the runner labyrinth is more than doubled and for coated surface, if the area of coating removed exceeds 5–10% of the total coated area.

6. Some recent findings related to erosion in guide vanes and its consequent effects

As discussed in the previous section, guide vanes are accompanied with highly turbulent flow, which results in a number of unsteady flow phenomena. When sediment particles travel with these unsteady flows, it causes erosion with various mechanisms. Some recent studies have focused on the clearance gaps of guide vanes and how the leakage flow through the gap is affecting the performance of the turbine [29–31]. This section provides a summary of the important findings, which were obtained from these studies.

Guide vanes are connected to shafts which can rotate about its axis so that the closing and opening according to different flow and load conditions are possible. A small dry clearance is present between guide vanes and facing plates to enable movement of the vanes. The leakage losses due to such a small clearance gap can be neglected. However, due to head cover deflection and constant impact of the sand particles, the size of the clearance gap increases. Inside the clearance gap, the flow is driven from high pressure side to low pressure side, disturbing the primary flow in the low pressure or suction side. This leakage flow occurs with a very high acceleration inside the gap, which creates rotational flow component after mixing with the main flow. As a result, vortex filaments are developed downstream at both ends, which eventually hit the runner blades. **Figure 14** shows a guide vane cascade including these mechanisms. These vortices tend to hit the runner inlet towards hub and shroud. Moreover, these vortices also change the stagnation angle at runner inlet towards the edges. Vortex carrying sediments along with improper stagnation angle can have a combined effect of sediment and cavitation erosion. An example of such an effect was shown in **Figure 12b**.

Recently, a one GV cascade rig was built in Waterpower Laboratory of Norwegian University of Science and Technology to study the effect of increasing clearance gap in the flow. The rig consisted of a 1:1 scale guide vane corresponding to a turbine with specific speed of 0.086 and clearance gap on one end. The rig covered angular position of 30°, which is 1/12th of the size of the turbine in angular direction. The cascade's walls were optimized so that the velocity field around the GV gives a close estimation of the actual flow in real turbines. The velocity field around the GV was measured using PIV technique by tracking the particles over a 2D laser sheet flashed at different span locations. Similarly, pressure measurements were taken at points along the surface of the GV.

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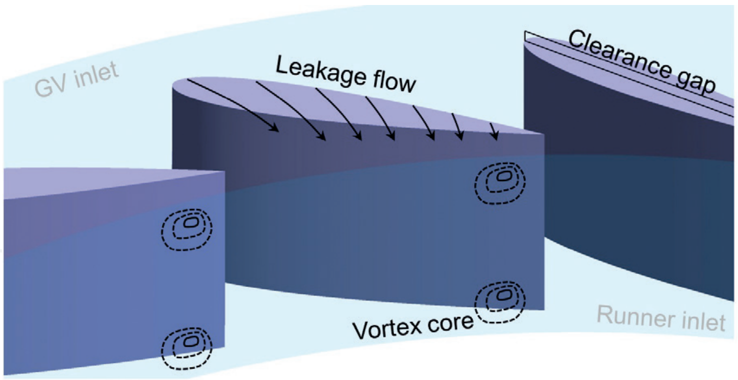

Figure 14. Leakage flow and vortex core through clearance gap.

The rig was used to test several GV profiles of same thickness and chord length, including the reference NACA0012. It was seen that the highest pressure difference between the GV surfaces for the reference design is equivalent to 20% of the net hydraulic head acting on the turbine, and it occurs around 75% of the GV chord. The leakage flow through the clearance gap depends directly on this pressure difference, as the velocity component normal to the chord increases with the similar trend. It is estimated that for a clearance gap of 2 mm (~2% of the span height), the total cross-wise leakage flow is more than 1% of the total flow. Higher leakage produces higher intensity of the vortices, which aggravates efficiency, erosion and structural integrity of the turbine.

It was seen from the experiment that the pressure difference and consequently, the leakage flow can be reduced by using asymmetrical guide vanes in the designed opening angle. **Figure 15** shows the velocity contour in the clearance gap plane taken from PIV in the rig, for three different GV profiles. In the case of NACA0012, the result is significantly affected by cavitation. The intensity of the cavitating vortex is very high, which causes several reflections of the laser sheet inside the plane, producing disturbances in the post-processed contour. However, the intensity of this disturbance is seen (and also heard during the measurement) to be reduced for the asymmetrical profiles. In the case of NACA2412, the vortex originates

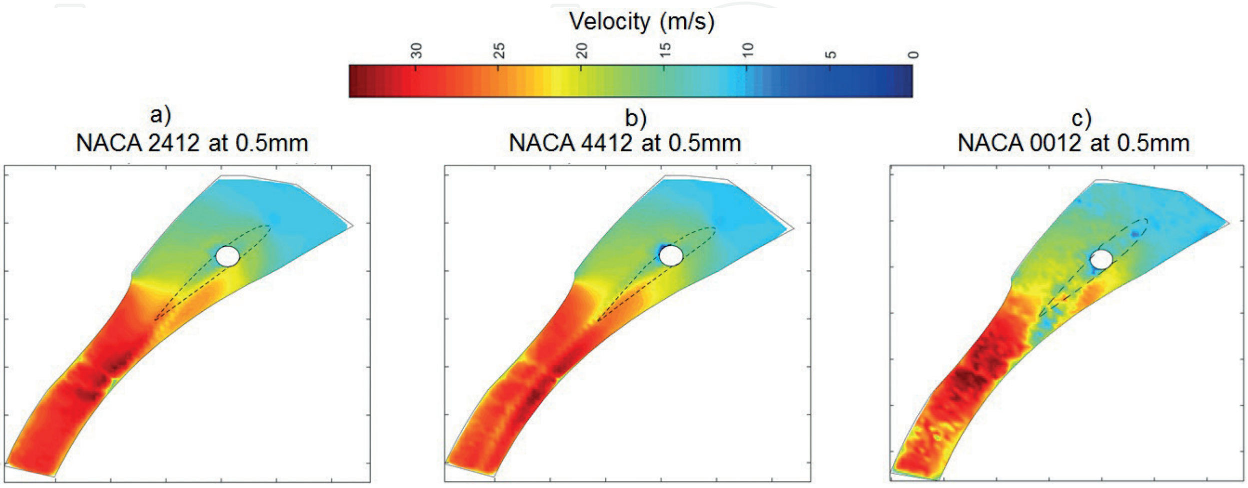

Figure 15. Velocity contours inside the clearance gap from PIV measurement [26].

from the leakage flow and disappears after moving few centimeters downstream of the GV's trailing edge. In the case of NACA4412, the vortex follows straighter path (like the wakes from GV boundary), which infers no cross-flow through the gap. However, the vortex does not disappear and it can be seen up to the end of the rig. It can be explained that the vortices in NACA2412 and NACA0012 have changed their planes while moving downstream. As the image was taken one plane at a time, the out-of-plane movement of the vortices was not captured. The intensity of the vortex in NACA4412 is relatively less, which decreases the rotational velocity component, preventing the out-of-plane movement.

Although the results in the designed opening angle showed reduced leakage flow in NACA4412, some recent results have shown that there could be some negative leakage flow at full load operation. The consequences could be more severe in terms of the erosion, because turbines operate in full load condition during wet season, when the concentration of the sediments in water is maximum. Hence, an intermediate profile between NACA0012 and NACA4412 (e.g., NACA2412) can be one of the optimum solutions, considering all the operating points.

7. Conclusion

Sediment erosion is of the inevitable challenges in the operation of power plants of many countries. Erosion depends on several factors, including materials of the turbine, as well as shape, size and mineral contents of sand. However, because of the highly unsteady flow in turbines, velocity of the fluid plays one of the most important roles for erosion. Depending upon the type of flow phenomena, the components of turbines are eroded at particular locations with particular mechanisms. This chapter explained some erosion mechanisms in Pelton turbine's nozzle systems and buckets and Francis turbine's guide vanes and runner. Some recent findings related to the flow phenomena inside the clearance gap of guide vanes were discussed. The future turbines operating in sediment affected power plants should consider the particles and their erosion potential during the design phase. The repair and maintenance and efficiency loss after erosion imposes heavy loss of capital along with the premature failure of turbines compared to their estimated life expectancy. In order to have reliable investments for hydropower development in a country, the challenges of sediment erosion in turbines need to be addressed with a sustainable solution.

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