

Failure of Francis Water Turbines due to Flow Variations in Papua New Guinea

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Abstract

Essential variations in water flow combined with erosion affect most small Francis Water Turbines installed by small communities across Papua New Guinea. The authors evaluate the effects of harmonically excited vibrations under variable flow conditions and propose design optimization of shafts. The stiffness depreciation of shafts under erosion is also considered and its effects are evaluated by Finite Element Methods.

Keywords

Small Francis Water Turbines, Harmonically Excited Vibrations, Flow Variation, Papua New Guinea.

Historical Background

Many Melanesian societies in the South Pacific are challenged by the need to provide energy to isolated communities. Most of them cannot afford the maintenance and transmission costs required by the connection to the main power plants and depend on thermal engine-driven generators. Pollution and cost tend to drive such communities towards the use of micro-hydro systems for producing electricity.

Around the end of the First World War the Canadian William Park and the Australians Matt Crowe and Arthur Darling penetrated the jungles at the back of Salamaua Isthmus, climbed the rugged mountain ranges and discovered gold at Koranga Creek, near Bulolo River in the territory of Papua New Guinea [01]. The discovery triggered one of the last gold rushes in history and contributed to the early development of hydro-electric plants in the region. Salamaua was destroyed during the Second World War and today only small communities of fisherman and farmers are scattered on long and narrow beaches between Salamaua Isthmus and Lae Port in Morobe. Busama is such a small community located north of Salamaua. Access to the place is possible only by boat and there are no bodies of water, except three waterfalls with hydraulic heads between 20m and 60m and flows of 4000 to 8000 l/min – see figure 1.

In the past twenty years two Francis turbines were installed at the big waterfall north of Busama. Both failed after less than 5000 hours of service and the local users suspect the failure is due to flow variations and erosion. The authors of this paper investigate the causes of failure and propose solutions.



Figure 1. The Narrow Beach of Busama Village. Inlet: Waterfalls of Busama

Challenges of Power Plants with Francis Turbines

Micro-hydro power plants are developed in relationship to the site and Francis turbine systems are no exception. The developers of the hydro-plant at Busama's waterfall took into consideration the low hydraulic head – 40m – combined with a relatively large flow estimated at 6000 l/min and employed Francis turbines. The selection was based on the high theoretical efficiency of Francis design. However, the design is site specific and should be optimized by frequency, harmonic and computational fluid dynamics simulations followed by validation in the field.

Recent developments focused on the optimization of Francis turbine runners and interactions rotor-stator are briefly discussed below:

- Stefan Lais, Quanwei Liang, Urs Henggeler, Thomas Weiss, Xavier Escaler and Eduard Egusquiza simulated the modal behaviour on Francis turbine runner in both air and water and discovered a very good agreement with the experiments. The above authors also analyzed the harmonic response of the runner by applying a time dependent pressure distribution – resulted from unsteady computational fluid dynamics simulation – to the mechanical structure [02].
- Zhenmu Chen, Patrick Singh and Young-Do Choi investigated the effect of the port area of blades on the flow exit angle from runner passage by Computational Fluid Dynamics and predicted that optimized flow exit angle significantly improves the efficiency of Francis turbines [03].
- S Bahrami, C Tribes, S von Fellenberg, T C Vu and F Guibault developed multi-fidelity design algorithms for balancing the cost of computational fluid dynamics in the process of optimization of Francis blade runners [04].
- B. Baidar, S. Chitrakar, R. Koirala and H. Neopane emphasized the importance of site specific design of Francis turbines and proposed the use of Tabakoff – Grant erosion model at the blade level. The authors also evaluated the influence of number of blades by performing Computational Fluid Dynamics simulation in ANSYS CFX with imported 3D geometry and boundary conditions developed in Matlab [05].

The current research tendencies suggest the importance of site related design, interactions runner-stator, sediment erosion, vibration analysis and computational fluid dynamics. Due to the complexity of Francis turbines, the maintenance requires qualified personnel, professional service and quick access to critical spare parts. The authors of this paper are investigating the combined effects of water flow variations and sediment erosion on the vibrational behaviour of small Francis turbines for a specific site. Computational Fluid Dynamics results obtained by various authors [02], [03], [04], [05] are considered in defining the dynamic loads on harmonic analysis.

In Francis turbines the rotor assembly is surrounded by a pressurized spiral casing and completely immersed in water. Fixed or adjustable stay vanes guide the flow at a designed angle to the blade of the rotor – See figure 2.

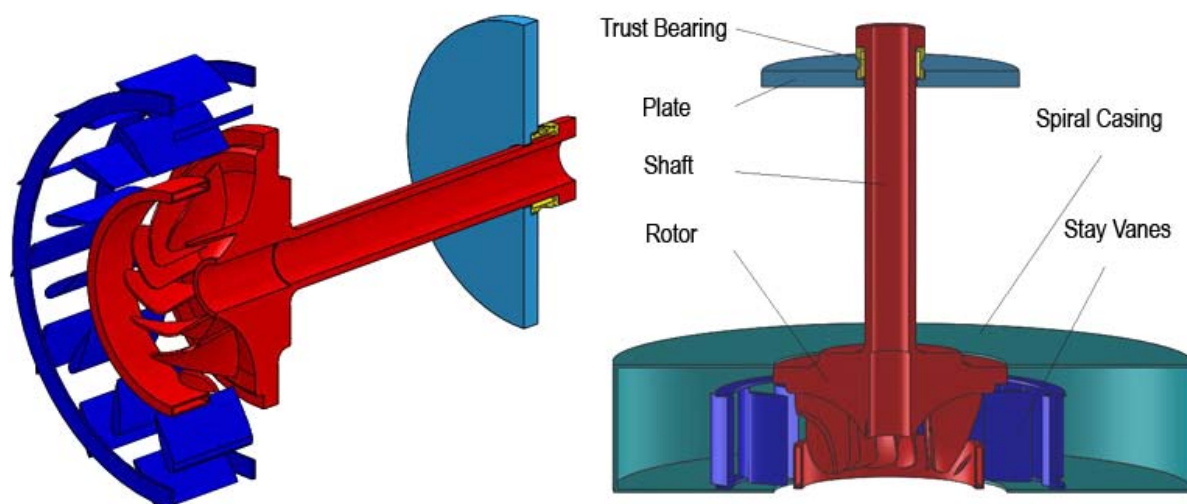


Figure 2. Longitudinal Section on 3D CAD Model of Francis Turbine

The Francis turbine is a radial flow reaction turbine: the water pressure and related kinetic energy are used to rotate the blades and shaft. The fluid radially glides over the blades almost tangent to them and changes the direction inside the rotor assembly and exits parallel with the axis of rotation of the turbine at a reduced velocity.

From a computational fluid dynamics point of view, the following set of differential equations is governing the incompressible flow with constant viscosity in cylindrical coordinates:

- Equation of continuity:

$$\frac{1}{r} \frac{\partial}{\partial r} (rv_r) + \frac{1}{r} \frac{\partial}{\partial \theta} v_\theta + \frac{1}{z} \frac{\partial}{\partial z} v_z = 0 \quad (01)$$

- Navier-Stokes equations:

$$\begin{aligned} \rho \left(\frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_r}{\partial \theta} - \frac{v_\theta^2}{r} + v_z \frac{\partial v_r}{\partial z} \right) &= \rho g_r - \frac{\partial p}{\partial r} + \mu \left[\frac{\partial}{\partial r} \left(\frac{1}{r} \frac{\partial}{\partial r} (rv_r) \right) + \frac{1}{r^2} \frac{\partial^2 v_r}{\partial \theta^2} - \frac{2}{r^2} \frac{\partial v_\theta}{\partial \theta} + \frac{\partial^2 v_r}{\partial z^2} \right] \\ \rho \left(\frac{\partial v_\theta}{\partial t} + v_r \frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_\theta}{\partial \theta} - \frac{v_r v_\theta}{r} + v_z \frac{\partial v_\theta}{\partial z} \right) &= \rho g_\theta - \frac{1}{r} \frac{\partial p}{\partial \theta} + \mu \left[\frac{\partial}{\partial r} \left(\frac{1}{r} \frac{\partial}{\partial r} (rv_\theta) \right) + \frac{1}{r^2} \frac{\partial^2 v_\theta}{\partial \theta^2} + \frac{2}{r^2} \frac{\partial v_r}{\partial \theta} + \frac{\partial^2 v_\theta}{\partial z^2} \right] \\ \rho \left(\frac{\partial v_z}{\partial t} + v_r \frac{\partial v_z}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_z}{\partial \theta} + v_z \frac{\partial v_z}{\partial z} \right) &= \rho g_z - \frac{\partial p}{\partial z} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial v_z}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_z}{\partial \theta^2} + \frac{1}{r^2} \frac{\partial v_z}{\partial \theta} + \frac{\partial^2 v_z}{\partial z^2} \right] \end{aligned} \quad (02)$$

- Shear stress equation:

$$\tau_{r\theta} = \mu \left[r \frac{\partial}{\partial r} \left(\frac{v_\theta}{r} \right) + \frac{1}{r} \frac{\partial v_r}{\partial \theta} \right] \quad (03)$$

In principle, it is possible to solve the above set of equations for the velocity \vec{v} and pressure p fields, provided enough initial and boundary conditions. In practice, there is no analytical solution for the set of differential equations listed above for a number of reasons:

- The differential equations are coupled and nonlinear
- The equations are second-order partial differential equations of a higher order of complexity

Essential differences between solutions predicted by CFD and field evaluations are due to the fact that in many situations the flow is multi-phase and the boundary conditions and design parameters are evolving in time under the combined effects of erosion and flow variations.

Vibration Analysis

The rotor of all Francis turbines is subject of harmonic forced vibrations and the study of harmonic response of the system is important [02]. The maximum amplitude of the rotor due to the rotational unbalance me can be described by the equation [06], [07]:

$$X = \frac{me\omega^2}{\sqrt{(k - M\omega^2)^2 + (\zeta\omega)^2}} \quad (04)$$

The significance of the parameters involved in equation (04) is:

- X - maximum amplitude of the rotor
- me - unbalance, usually expressed in kg mm
- M - mass of the rotor
- ω - rotational speed of the rotor, expressed in rad/s

- ζ - damping coefficient
- k - stiffness of the shaft

The variation of parameter $\frac{MX}{me}$ with the frequency ratio $r = \frac{\omega}{\omega_n}$ for various damping coefficients is plotted in Matlab and shown below in figure 3.

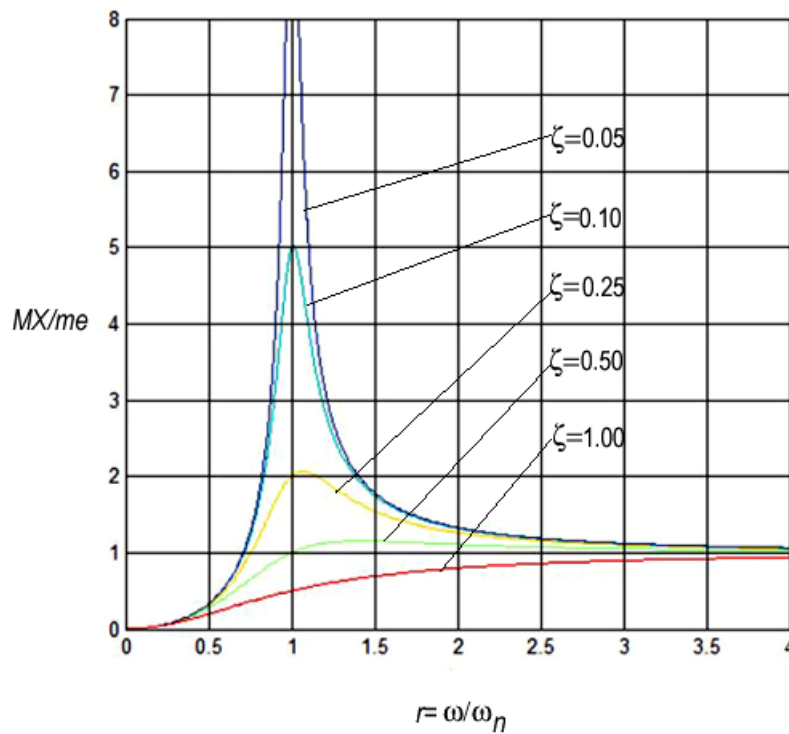


Figure 3. Variation of $\frac{MX}{me}$ with frequency ratio r for different values of damping coefficient ζ

The following observations are important:

- All plots begin at zero amplitude and the amplitude $\omega = \omega_n$ near resonance is severely affected by damping. If the rotor runs near resonance, damping should be considered to avoid dangerous amplitudes.
- At very high rotational speeds ω , $\frac{MX}{me}$ is almost unity, and the effect of damping can be neglected.

- For $0 < \zeta < \frac{\sqrt{2}}{2}$ the maximum of $\frac{MX}{me}$ occurs when $\frac{d}{dr} \left(\frac{MX}{me} \right) = 0$, with the solution $r = \frac{1}{\sqrt{1-2\zeta^2}} > 1$ and a corresponding maximum value of $\frac{MX}{me}$ given by $\left(\frac{MX}{me} \right)_{\max} = \frac{1}{2\zeta\sqrt{1-\zeta^2}}$.
The peaks occur to the right of resonance location at $r = 1$.
- For $\zeta > \frac{\sqrt{2}}{2}$ the parameter $\frac{MX}{me}$ does not attain a maximum. Its value grows from 0 at $r = 0$ to 1 at $r \rightarrow \infty$.
- The force transmitted to the trust bearing due to rotating unbalanced force $F(t)$ is function of the stiffness of the shaft and damping conditions, $F(t) = kx(t) + c\dot{x}(t)$. Its magnitude $F(t)$ can be derived as:

$$|F(t)| = me\omega^2 \sqrt{\frac{1+(2\zeta r)^2}{(1-r^2)^2 + (2\zeta r)^2}}$$

Due to the complex 3D geometry of the shaft, the authors considered numerical approaches for modal analysis. The simulations were performed in CosmosM with a 3D CAD model of the rotor designed in SolidWorks. The effects of erosion on the shaft were considered in its geometry and are based on measured changes of the outside diameter after the failure of the turbine.

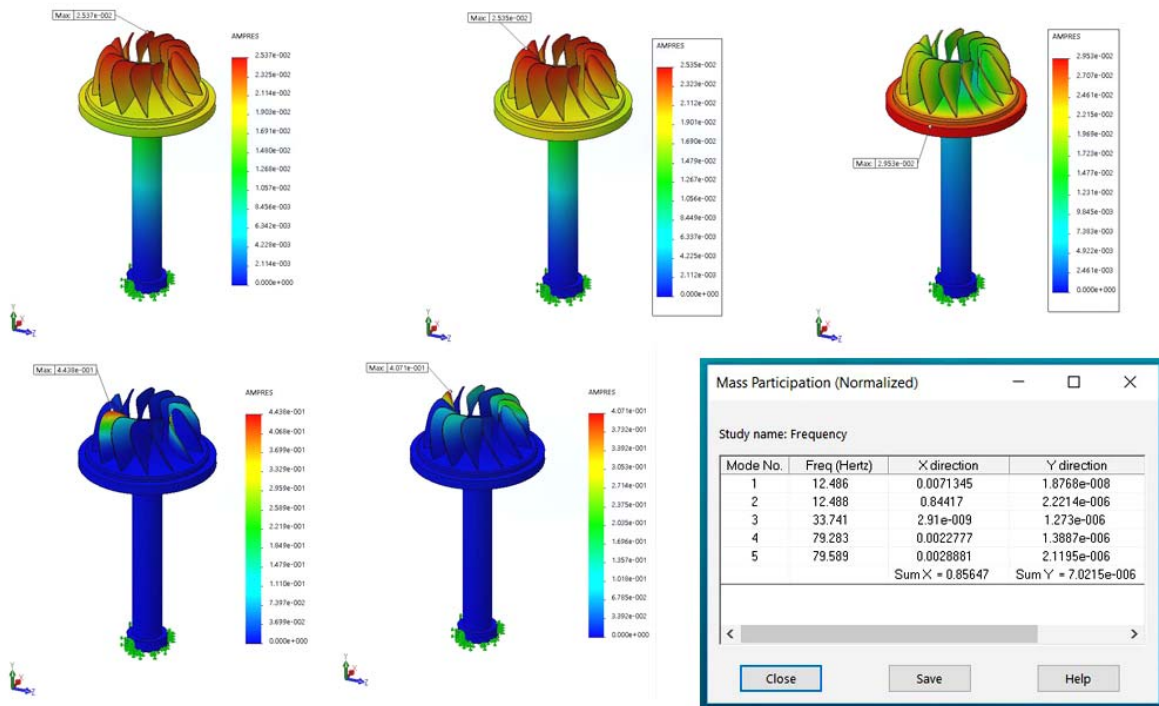


Figure 4. Results of Modal Analysis for a Cast Rotor in Air

It is important to mention that natural frequencies in water are lower than in air, with a natural frequency reduction ratio τ defined as $\tau = \frac{\omega_{n \text{ water}}}{\omega_{n \text{ air}}} < 1$ [08].

The pressure distribution on blade surfaces is maximal at the periphery of the blades. For simplicity we considered it uniformly distributed over their entire surfaces. The results of harmonic analysis are shown in figure 5.

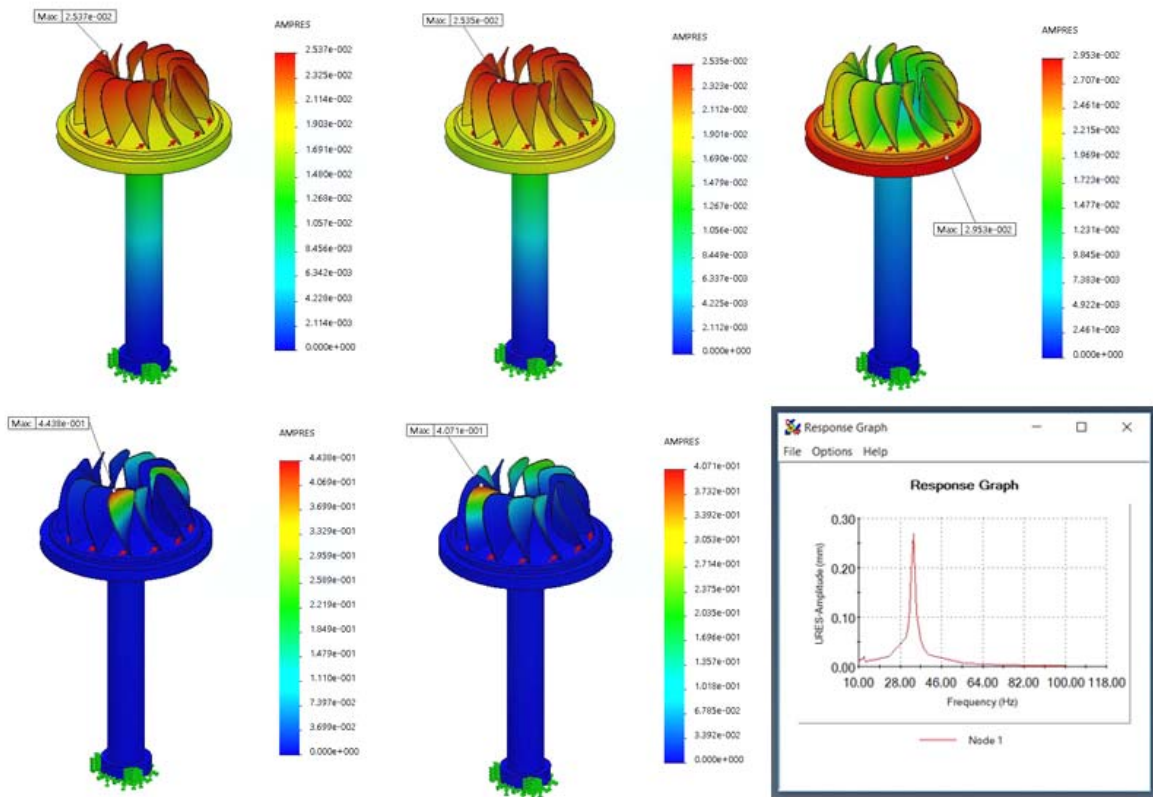


Figure 5. Results of Harmonic Analysis for a Cast Rotor under Uniformly Distributed Pressure over Blades

In relationship with equation (04) and the plots from figure (03), the role of the stiffness of the shaft is of great importance. On one hand, frequency ratios $r = \frac{\omega}{\omega_n}$ away from resonance require small natural frequencies even for the low flow rates at the level of turbine. On the other hand, the natural frequency of the shaft is function of its stiffness and the mass of the rotor:

$$\omega_n = \sqrt{\frac{k}{M}} \quad (05)$$

The minimum possible stiffness of the shaft is evaluated with $k_{\min} = \omega_{\min}^2 \left(\frac{me}{X} + M \right)$, where ω_{\min} is the rotational speed at minimum flow rate. However, the same stiffness of the cantilever beam shaft supporting the rotor's end can be evaluated with $k_{\min} = \frac{3EI}{l^3} = \frac{3E}{l^3} \cdot \frac{\pi(d_o^4 - d_i^4)}{64}$, where d_o and d_i are the outside and inside diameters of the cross section of the shaft. Another option for reducing the natural frequency of the rotor suggested by equation (05) – a classical one – is to increase the mass M of the rotor.

Conclusions

The design, development and maintenance of Francis turbines are site related. Water flow variations and erosion due to the sediment impact influence the vibrational behaviour of the system rotor – stator. The waterfalls at Busama community are fed by fluids crossing mountain ranges rich in quartz particles, due to the vicinity of gold fields. The variations in flow at the waterfalls are related to seasonal variations of the rainfalls and the flow rates in summer months can go down to 60% of flow rates in winter time. Such variations are important for a cast turbine affected by erosion because the vibrational amplitudes reach magnitudes potentially able to annihilate the geometrical clearance between the rotor and the stator and bring failure.

The investigations of the authors of this paper also noted the failure of the trust bearings supporting the rotor. It can be safely concluded that Francis turbines cannot be recommended as micro-hydro plants for small communities due to the high cost of equipment, site-related design complexities, lack of qualified service and long waiting time for spare parts.

References

- [01]. Waterhouse, M., *Not a Poor Man's Field*, Halstead Press, Ultimo, New South Wales, Australia, 2010
- [02]. Lais, St., Liang, Q., Henggeler, U., Weiss, T., Escaler, X., Egusquiza, E., Dynamic Analysis of Francis Runners – Experiment and Numerical Simulation, *International Journal of Fluid Machinery and Systems*, Vol. 2, No. 4, October-December 2009
- [03]. Chen, Z., Patrick Singh, P., Choi, Y., Francis Turbine Blade Design on the Basis of Port Area and Loss Analysis, *Energies* 2016, 9, 164, www.mdpi.com/journal/energies
- [04]. Bahrami, S., Tribes C., von Fellenberg, S., Vu, T., Guibault, F., Multi-Fidelity Design Optimization of Francis Turbine Runner Blades, 27th IAHR Symposium on Hydraulic Machinery and Systems 2014, *IOP Conf. Series: Earth and Environmental Science* 22, 2014, 012029
- [05]. Baidar, B., Chitrakar, S., Koirala, R., Neopane, H., Selection of Optimal Number of Francis Runner Blades for a Sediment Laden Micro Hydropower Plant in Nepal, *International Journal of Fluid Machinery and Systems*, Vol. 8, No. 4, October-December 2015
- [06]. Rao, S. – *Mechanical Vibrations*, 5th Edition, Prentice Hall, New York, 2011
- [07]. Gans, R.- *Mechanical Systems - A Unified Approach to Vibrations and Controls*, Springer, 2015
- [08]. Valentin, D., Presas, A., Egusquiza, E., Valero, C., Influence of Non-Rigid Surfaces on the Dynamic Response of a Submerged and Confined Disk, *6th IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Ljubljana, Slovenia, September 9-11, 2015

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