Proceedings of IOE Graduate Conference, 2019-Winter
Peer Reviewed
Year: 2019 Month: December Volume: 7
ISSN: 2350-8914 (Online), 2350-8906 (Print)

Fatigue Analysis of Francis Turbine Runner as a Result of Flow-induced Stresses

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Abstract

This study deals with the analysis of fatigue deformation due to flow induced stresses in Francis turbine runner when operated under varying load conditions. One-way steady state Fluid Structure Interaction (FSI) was used for the study on which, pressure loads on the runner obtained via Computational Fluid Dynamics (CFD) analysis was transferred as a mechanical load on blade surface to further carry out structural analysis followed by a simplified fatigue analysis. Characteristics stress fields have been presented in terms of Von Mises equivalent stress for the Francis runner. Through FSI analysis, it was observed that Francis runner considered for this study has infinite life. Additonally, it was found that flow induced stress is maximum at the joint between blade and band. Therefore, this part can be noted as the critical area for cracks to occur thereby making it prone to fatigue deformation if load condition deviates from that of designed condition.

Keywords

Francis turbine runner, Fatigue, Fluid Structure Interaction (FSI), Computational Fluid Dynamics (CFD), Finite Element Analysis (FEA), Fluid Induced Stress, Fatigue Failure, Fatigue Damage, Fatigue Life

1. Introduction

Hydroelectricity is by far the most reliable, sustainable, and clean form of energy generation. Another beauty of hydropower plants is that they can operate under varying load condition and can respond well to sudden changes in power grid. Energy demand is highly unpredictable i.e. demand for energy fluctuates frequently. In addition to this, climate has a huge role in flow availability which contributes significantly towards flow dischange variability. In order to ensure reliability and energy security, these demand fluctuations are required to be addressed accordingly. While doing so, hydroelectric machines are dragged beyond their operational limitation. Therefore, strong vibrations are induced due to varying loads that can produce fatigue failures on the mechanical components of the hydraulic turbines [1]. Hydropower technology has been used for energy generation since long. Numerous researches have been carried out and there are thousands of researches on going for the betterment of hydropower plants and equipment with an aim to optimize the energy generation; operational and manufacturing costs and popularly these days, researches are aimed towards increasing reliability and sustainability of hydroelectricity. Many studies [1–12] have shown that hydraulic turbines undergo fatigue deformation when operated at varying load conditions, thus causing wear and tear of turbine and runner lifetime decreases. Fatigue deformation is the combination of low cycle and high cycle fatigue. Loads acting on the Francis runner can be classified as steady loading (fluid pressure, centrifugal force and runner's weight) and unsteady loading (high frequency pressure fluctuations due to stator-rotor interaction as well as vortex rope phenomenon) [1]. Fatigue deformation incurs huge financial losses, energy losses and national grid imbalance issues. That is why, fatigue study becomes imperative. Fatigue study is quite difficult to perform in actual machineries due to complex structures and also cost that would be involved in the process often hinders the study. Therefore, use of commercial softwares like ANSYS, Solidworks, Catia has become more handy for such analysis. Saeed et al. [7] argue that stress analysis of hydraulic turbine runner can only be performed by numerical methods due to complexity of these structures. Valkvae [12] further argue that an understanding of FSI in turbine have become more essesntial since the different turbine loads are mainly induced by the internal fluid flow. With the use of one way FSI, it is therefore possible to study about the impact of flow induced stresses on Francis turbine runner.

2. Computational Method

One-way steady state Fluid Structure Interaction (FSI) analysis, a coupled solution of Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA), was used to compute fatigue deformation in the Francis runner.

Methods adopted in the study has been explored below sequentially.

2.1 Geometric Modeling

A Francis turbine having net head (H) of 50 m and design discharge (Q) of 1.6 m^3/s has been considered for this study. Assuming turbine efficiency (η_t) and generator efficiency (η_g) as 92% and 97% respectively, power (P) was calculated as 700 kW using equation 1.

$$P = g * \eta_g * \eta_h * H * Q \tag{1}$$

A model of Francis Runner considering shaft diameter as 0.14 m, runner inlet diameter as 0.643 m and outlet diameter as 0.496 m was developed in 3D commercial software, Solidworks 2017. The model consists of 15 blades, hub, and shroud (as shown in figure 1).

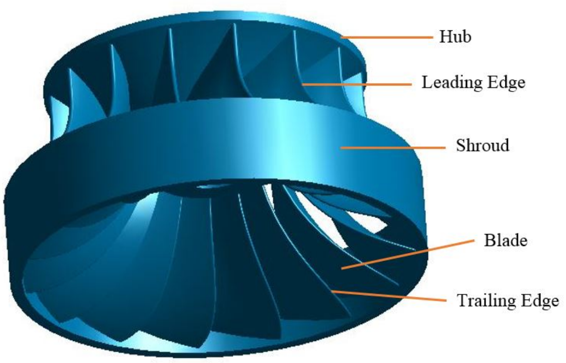

Figure 1: CAD model of Francis Turbine Runner

To calculate synchronous rotational speed (n_s) of the runner, number of poles (p) was calculated as 6 and standard value of grid frequency (f) was considered i.e. 50 Hz. Thus, using equation 2, synchronous rotational speed was calculated as 1000 rpm.

$$n_s = \frac{120 * f}{p} \tag{2}$$

Similarly, radial flow velocity (V_f), tangential velocity (u_1), speed ratio (K_u) and flow ratio (K_f) were calculated using equations 3 to 6.

$$V_f = \frac{Q}{\pi * D * b * K} \tag{3}$$

$$u_1 = \frac{\pi * D * n_s}{60} \tag{4}$$

$$K_u = \frac{u_1}{\sqrt{2gH}} \tag{5}$$

$$K_f = \frac{V_f}{\sqrt{2gH}} \tag{6}$$

Here, D, and b denotes diameter and width of runner vane respectively whereas, K stands for vane thickness factor/coefficient. Its value is always less than unity, usually of the order 0.95 or so [13]. The value of K assumed in this study is 0.95.

Characteristic parameters obtained for this study are as tabulated below (Table 1):

Table 1: Characteristic Parameters

S.N.	Parameters	Value	Unit
1	Speed Ratio, K_u	1.08	
2	Flow Ratio, K_f	0.196	
3	Radial flow velocity, V_f	6.13	m/s
4	Tangential velocity, u_1	33.67	m/s

For the fluid domain, cylindrical region enveloping the runner and the fluid outlet passage was developed in Solidworks 2017. 3D CAD model of Francis Runner was then imported to ANSYS Design Modeler.

2.2 Meshing

Geometry was exported to Meshing tool in ANSYS to generate unstructured mesh optimized for Physics setup in CFD and Fluent as Solver. Program controlled triangular surface mesh was generated consisting of 1527496 and 8427075 number of nodes and elements respectively. Figure 2 shows the meshed fluid domain for Francis runner and figure 3 shows the meshed runner geometry.

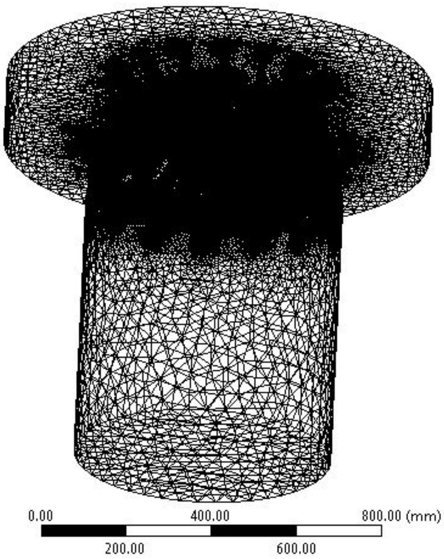

Figure 2: Meshed fluid domain of Francis runner model

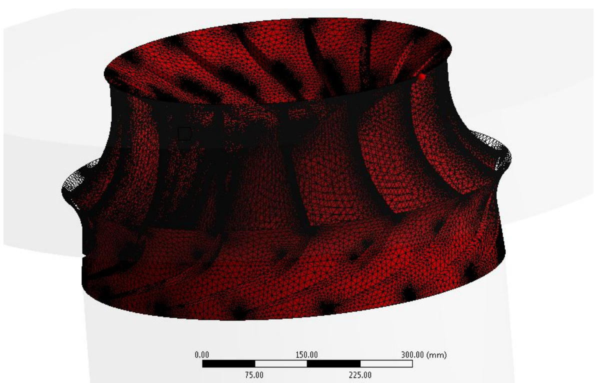

Figure 3: Meshed Francis runner geomentry

Mesh quality was gauged via Skewness and Orthogonality mesh metrics. An average value of 0.24 was observed as a Skewness quality whereas Orthogonality quality was found to be an average of 0.85. These values show a fair indication of mesh quality for CFD study.

2.3 Computational Fluid Dynamics (CFD) Analysis

CFD simulation is required to judge the right flow behavior of fluid inside or outside the structure [11]. The meshed CAD model of Francis Runner was exported to ANSYS Fluent Solver, where following boundary conditions listed in Table 2 in subsection 2.3.1 below were assigned.

2.3.1 Boundary Conditions and Assumptions

Steady state conditions and incompressible fluid flow has been assumed in this study. Coupled scheme was used for pressure-velocity coupling in which, velocity inlet and pressure outlet conditions were chosen.

Similarly, Shear Stress Transport (SST) k-omega model was chosen as turbulence model. Second order upwind discretization scheme was employed to treat the derivatives. Standard solution initialization was opted, 1e-4 as residual was considered and solution was run for 100 iterations.

Table 2: Boundary Conditions

Boundary	Assigned as	Remarks
Inlet	Velocity Inlet	Radial Velocity = 6.13 m/s Tangential Velocity = 33.67 m/s
Inlet top	Wall	No Slip
Inlet bottom	Wall	No Slip
Outlet	Pressure Outlet	Gauge Pressure = 0 Pa

2.3.2 CFD Results

a. Pressure Contour:

Figure 4 below shows the pressure distribution inside the Francis turbine runner. Under given conditions, pressure was found to vary from -90.91 KPa to 416.9 KPa. Highest pressure was observed at the joint between the runner blades and the band with gradual decrease from leading to trailing edge along the runner blade.

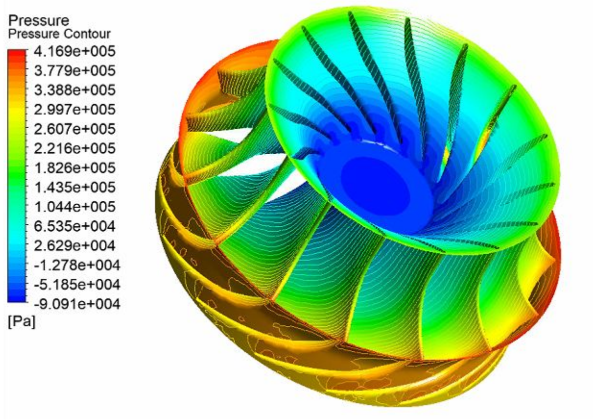

Figure 4: Pressure contour at 1.6 m^3/s discharge and 1000 rpm

Studies ([1], [3], [12]) have shown that, maximum pressure is observed at the joint between runner blades and band and/or crown which remained true for this study too.

b. Velocity Contour:

Figure 5 below shows the velocity contour generated at 6.13 m/s flow velocity and full load operation condition for the runner.

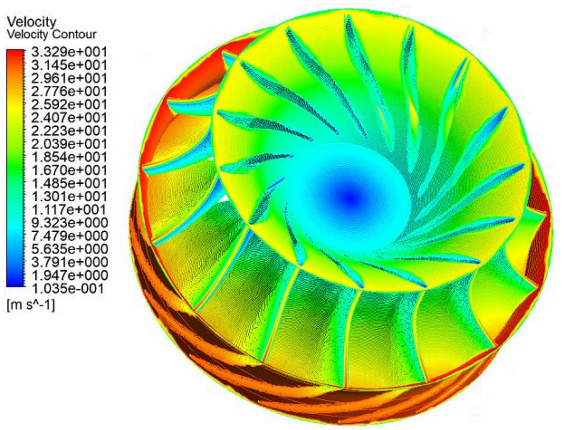

Figure 5: Velocity contour at 6.13 m/s flow velocity and full load operation condition

Maximum flow velocity was observed to be 33.29 m/s. It can be further observed that the velocity is more at the suction side i.e. leading edge of the runner blade as compared to the trailing edge.

c. Velocity Streamlines:

Figure 6 below show the streamlines of tangential velocity in the fluid domain as obtained from CFD simulations.

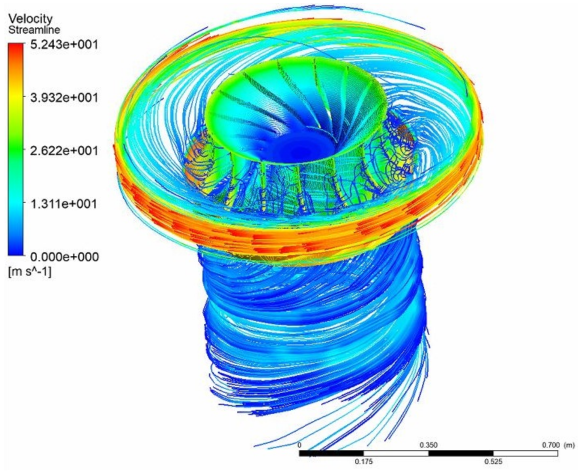

Figure 6: Velocity streamlines

Result shows that maximum velocity observed was 52.43 m/s. It can be observed that the flow velocity is gradually decreasing as the fluid flows towards the outlet.

2.4 Structural Analysis

Structural analysis was conducted to check the structural integrity of the runner. FEA method was adopted for the analysis. For which, Static Structural solver of ANSYS software was used to study stress distribution and deformation of the said turbine runner. Structural material of the turbine runner considered for this study was structural steel having properties as tabulated below in table 3.

Table 3: Properties of Material

S.N.	Properties	Value	Unit
1	Density (ρ)	7850	kg/m^3
2	Young's Modulus (E)	200	GPa
3	Yield Strength	250	MPa
4	Tensile Strength	460	MPa
5	Poisson's Ratio	0.3	

2.4.1 Meshing

Francis runner model was meshed to generate unstructured mesh optimized for Mechanical Solver. Program controlled triangular surface mesh was generated consisting of 129267 and 92096 number of nodes and elements respectively.

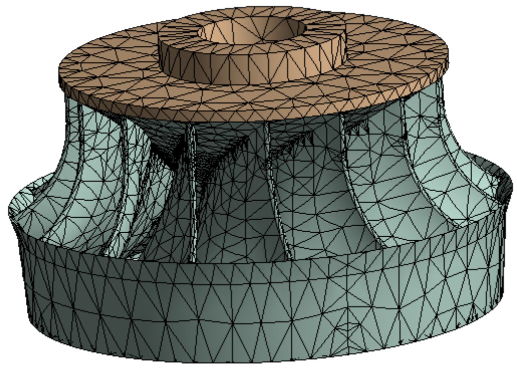

Figure 7: Meshed runner geometry for FEA

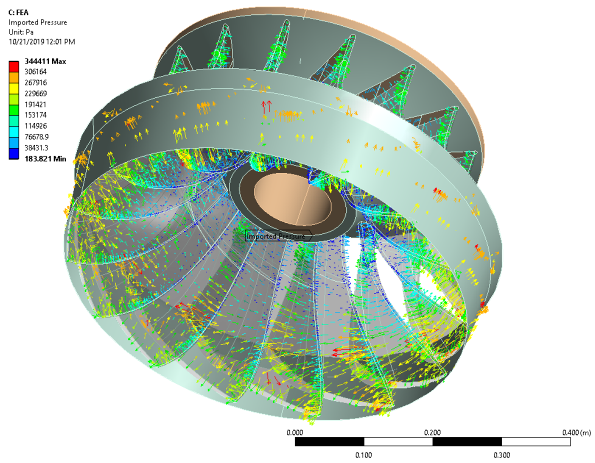

Figure 9: Imported Pressure fields

Figure 7 above shows the meshed runner geometry. Skewness mesh metric was found to be 0.5 in average and Orthogonality mesh metric was found to be 0.7 in average thus, indicating fair mesh quality.

2.4.2 Boundary Conditions and Assumptions

Zero displacement boundary condition was applied at the shaft area. Rotational velocity of 1000 rpm about x-axis was assigned and pressure loads were used as imported from CFD analysis which are as shown in figure 8.

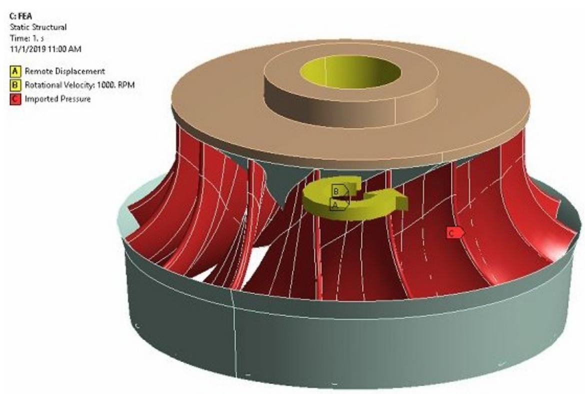

Figure 8: Boundary Conditions for Structural Analysis

To increase the precision of the results, mesh mapping was done while transferring pressure load to the runner geometry which ensured accurate transfer of pressure loads at fluid-solid interface. Figure 9 (below) shows the pressure distribution on runner blade surface.

2.4.3 FEA Results

In structural analysis, results have been achieved through analyzing the Von Mises stress, total deformation and fatigue tool.

a. Von Mises Equivalent Stress:

Figures 10 and 11 show Von Mises Stress Distribution in the runner. Maximum stress observed was 16.021 MPa which is very less than the material yield strength and ultimate strength i.e. 250 MPa and 460 MPa respectively Thus, indicating that the runner blades shall not undergo cracks propagation.

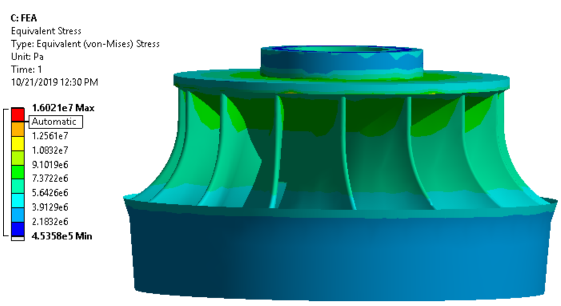

Figure 10: Von Mises Stress Distribution at the leading edge

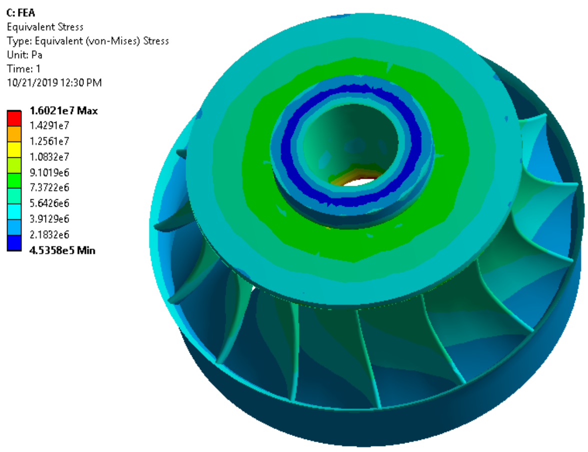

Figure 11: Von Mises Stress Distribution at the trailing edge

Additionally, stress observed is higher at the joint between hub and blade. Similarly, stress at leading edge is higher than trailing edge as observed in figures 10 and 11.

b. Total Deformation:

Figure 12 shows that the total deformation is higher at the suction side of the runner blade. Maximum deformation observed was 0.00934 mm.

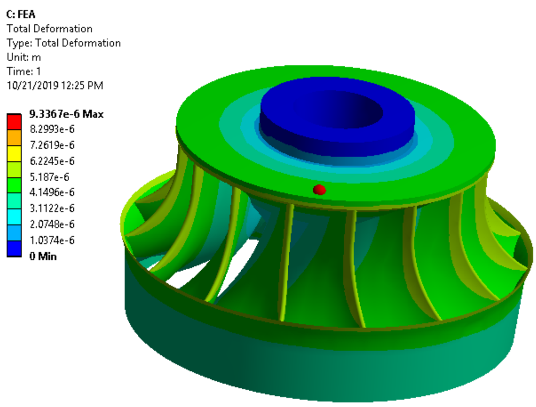

Figure 12: Total Deformation observed in Francis Runner

c. Fatigue Life:

Figure 13 shows the contour plot for fatigue life. Fatigue life plot indicates that if the loading is of constant amplitude type, then the result represents the number of cycles till which the structure can withstand until it will fail to fatigue [11].

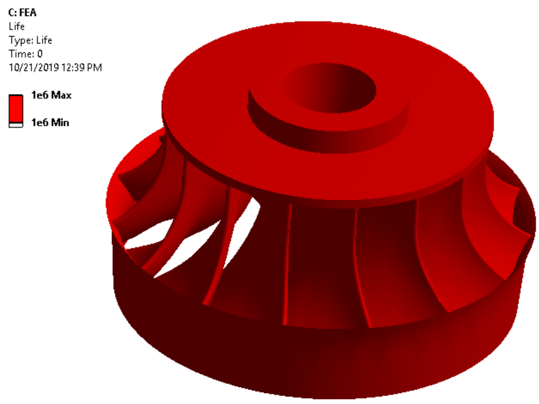

Figure 13: Fatigue life of Francis runner

Observation shows that the runner can withstand minimum 1e+6 number of cycles thus indicating infinite life.

d. Fatigue Safety Factor:

Fatigue Safety factor is a contour plot of the factor of safety with respect to a fatigue failure at a given design life. For fatigue safety factor, values less than 1 indicate failure before the design life is reached [11].

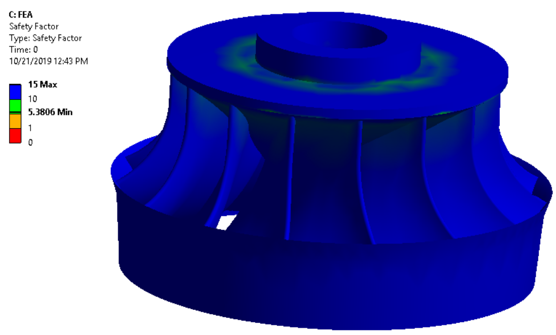

Figure 14: Fatigue Safety Factor

Figure 14 above shows that the minimum factor of safety as observed in the upper part of hub is 5.3806 which is more than 1. Therefore, the runner will not undergo failure before the design life.

3. Result and Discussion

Effect of both CFD and FEA was observed in the Francis runner model. From CFD analysis, pressure distribution, velocity contours and streamlines were obtained. Pressure was found to vary from -90.91 KPa to 416.9 KPa. Highest pressure observed was at the joint between blade and band. Structural analysis was carried out with the imported pressure loading with necessary mesh mapping and Von Misses stress, fatigue life, deformation were computed. It was further observed that the maximum stress lies at the joint between blade and band which is thereby a critical area for cracks to occur due to fatigue loading. Through this FSI analysis, it was found that the Francis turbine runner considered for this study has infinite life and minimum damage combined with maximum factor of safety. Thus, we can conclude that the runner model will not undergo fatigue deformation.

4. Conclusion

From the results obtained in this study, it was observed that the flow induced stress field is maximum at the joint between blade and band. Therefore, it can be noted as the critical area for cracks to occur thereby making it prone to fatigue deformation if load conditions deviate hugely from that of designed condition. Often the hydropower plants face the fatigue situation when the plant is made to frequently operate at the varying load conditions. Fatigue failure incurs huge financial and energy losses. To avoid such situation, it is essential to give proper attention during design phase.

References

[1] Radu Negru, L. Marsavina, and Seby Muntean. Analysis of flow induced stress field in a Francis trubine runner blade. LVII, 2011.

[2] Dr Hameed Ullah Mughal, Muhammad Awais Hamza Mughal, and Muhammad Ibtsam Talha. Fluid-Structure-Interaction (FSI) analysis of Francis turbine for high head operations. 6, 2015.

[3] M. Flores, G. Urquiza, and J. Rodríguez. A fatigue analysis of a hydraulic Francis turbine runner. 2, 2012.

[4] S. Muntean and L. Marsavina. Failure analysis of a francis turbine runner. 12, 2010.

[5] R. Negru, S. Muntean, L. Marsavina, R. Susan Resiga, and N. Pasca. Computation of stress distribution in a Francis turbine runner induced by fluid flow. 64, 2012.

[6] A. L. Ramirez, A. A. Campos, G. O. Dorantes, C. Z. Mazur, and Q. R. Munoz. Failure analysis of runner blades in a Francis hydraulic turbine - case study. 2015.

[7] R. A. Saeed, A. N. Galybin, and V. Popov. FE-analysis of stresses in a Francis turbine runner at Derbendikhan power station. 41, 2010.

[8] R. A. Saeed, A. N. Galybin, and V. Popov. Modelling of flow-induced stresses in a Francis turbine runner. 2010.

[9] R. A. Saeed. Numerical simulation of three-dimensional cavitating turbukent flow in Francis turbine with ANSYS. 9, 2015.

[10] P.-T. Storli and T. K. Nielsen. Dynamic load on a Francis turbine runner from simulations based on measurements. 22, 2014.

[11] T. S. Tanwar, D. Hariyani, and M. Dadhich. Flow simulation (CFD) & Static structural analysis (FEA) of a Radial turbine. 3, 2012.

[12] Ingebjorg Valkvae. Dynamic loads on Francis turbines. Master's thesis, Norwegian University of Science and Technology (NTNU), June 2016.

[13] R.K Rajput. *Basics of Fluid Mechanics*. S. Chand Publishing, 6th edition, 2015.