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

by

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A THESIS

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The undersigned certify that they have read, and recommended to the Institute of Engineering for acceptance, a thesis entitled **"Fatigue Analysis of Francis Turbine Runner as a Result of Flow-induced Stresses"** submitted by Mandira Adhikari in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Systems Design and Engineering.

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ABSTRACT

Hydroelectricity plays a significant role in energy generation. It is by far the most reliable, sustainable and clean form of energy generation. Another beauty of hydropower plants is that they can operate under various load conditions and can respond well to sudden changes in the power grid. However, while doing so, hydroelectric machines are dragged beyond their operational limitation leading into fatigue deformation thus hidering the structural integrity of mechancial componets of hydraulic turbines. Fatigue deformation incurs huge financial losses, energy losses, national grid imbalance issues and sometimes tragic hazards at plant site. As a preventive measure or as a failure analysis, it is imperative to study about sensitive parts that are susceptible to fatigue failures.

Fatigue study can be carried out experimentally via analysis of real time measurement of pressure, fluid flow, vibration analysis in actual hydraulic turbines or their prototypes. Another method is through numerical analysis which is popularly used by researchers due to complex structures of actual machines and also due to time, cost considerations. Many studies have been carried out to study structural integrity of Francis runner using computer based numerical analysis. Fluid Structure Integration (FSI) is one such numerical method which is widely used for flow induced stress study. This research work has employed the same for fatigue analysis of Francis turbine runner using a commercial software, ANSYS 15.0. Through FSI analysis, it was observed that the maximum pressure lies at the joint between blade and band. It was further observed that Francis turbine runner considered for this study has infinite life and minimum damage combined with maximum factor of safety.

Keywords: Francis turbine runner, Fatigue, Fluid Structure Integration (FSI), Computational Fluid Dynamics (CFD), Finite Element Analysis (FEA), Fluid induced stress, Fatigue Failure, Fatigue Damage, Fatigue Life

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LIST OF SYMBOLS

P	Power	kW
H	Net Head	m
η_t	Turbine Efficiency	%
η_g	Generator Efficiency	%
Q	Design Discharge	m^3/s
n_s	Rotational Speed	rpm
p	Number Of Poles	-
f	Grid Frequency	Hz
V_f	Radial Flow Velocity	m/s
u_1	Tangential Velocity	m/s
K_u	Speed Ratio	-
K_f	Flow Ratio	-
D	Runner diameter	m
b	Runner vane width	m
K	Vane thickness factor/coefficient	-
ρ	Density	kg/m^3
E	Young’s Modulus	GPa
σ	Yield Strength	MPa
UTS	Tensile Strength	MPa

LIST OF ACRONYMS AND ABBREVIATION

APDL	ANSYS Parametric Design Language
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
FEA	Finite Element Analysis
FSI	Fluid Structure Interaction
HCF	High Cycle Fatigue
kPa	Kilo Pascal
kW	Kilo Watt
LCF	Low Cycle Fatigue
m	Meter
MPa	Mega Pascal
rpm	Revolution Per Minute
SST	Shear Stress Transport

CHAPTER ONE: INTRODUCTION

1.1. Background

In Nepalese Power market, hydropower plays a significant role in energy generation. It is by far the most reliable, sustainable and clean form of energy generation. Another beauty of hydropower plants is that they can operate under various load conditions and can respond well to sudden changes in the 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 availabilty in Nepalese rivers and rivulets thus contributing significantly towards flow dischange variability. Hydro-turbines are usually used to operate at variable load due to different climate nature over the whole annum (Mughal et al., 2015). In order to ensure reliability and energy security, these demand fluctuations are required to address 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 (Muntean and Marsavina, 2010).

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 (Flores et al., 2012: Mughal et al., 2015: Muntean and Marsavina, 2010: Negru et al., 2011: Negru et al., 2012: Ramirez et al., 2015: Saeed R. A., 2015: Saeed et al., 2010: Saeed et al., 2010: Storli and Nielsen, 2014: Tanwar et al., 2012: Valkvæ, 2016) have shown hydraulic turbines undergo fatigue deformation when operated at varying load conditions, thus causing wear and tear of turbine and runner lifetime decreases. Fatigue failures in Francis turbine runner frequently occurs in hydropower plants, causing unexpected plant downtime and considerable financial loss.

Decades of operational experience have shown that turbine runners develop fatigue cracks in areas, where stress concentrations and material defects coincide. In Francis turbine runners, cracks tend to propagate from the transition of the welded T-joint between the blade and the band or crown. This type of turbine runner, which operate under a wide range of heads and outputs, are subjected to considerable dynamic forces which can lead to fatigue cracking. The magnitude of these forces is a function of the hydraulic pressure, the water velocity and the geometry of the stationary parts guiding the water into the runner (Saeed R. A., 2017).

As, fatigue deformation incurs huge financial losses, energy losses and national grid imbalance issues, it is necessary to study about the impact of flow induced stresses on the fatigue deformations of turbine in hydropower plant. However, it is quite difficult to analyze the same on actual machines due to complex structures and also cost that would be involved in the process often hinders the study. Also, performing such analysis on actual operating conditons on real hydroelectric machines would be difficult.

Therefore, use of commercial softwares like ANSYS, Solidworks, Catia has become handier for such analysis. It has been observed that ates that stress analysis of hydraulic turbine runner can only be performed by numerical methods due to complexity of these structures (Saeed et al., 2010). An understanding of fluid-structure interaction in turbine have become more essesntial since the different turbine loads are mainly induced by the internal fluid flow (Valkvæ, 2016). With the use of one way FSI, it is therefore possible to study about the impact of flow induced stresses on Francis turbine runner.

1.2. Statement of Problem

The variable demand on the energy market, as well as the limited energy storage capabilities, requires a great flexibility in operating hydraulic turbines. Therefore, turbines are frequently operated over an extended range of regimes (Muntean and Marsavina, 2010). Such balancing or regulating mechanism expose turbine runners to dynamic loads and increases stress. This results into fatigue development. Fatigue deformation is the combination of low cycle and high cycle fatigue. Loads acting on the Francis runner can be classified as steady loading i.e. fluid pressure, centrifugal force and runner’s weight and unsteady loading i.e. high frequency pressure fluctuations due to stator-rotor interaction as well as vortex rope phenomenon (Negru et al., 2011). Fatigue deformation causes wear and tear of turbine thus, incurring huge financial losses, energy losses and national grid imbalance issues. Therefore, it is necessary to study about the impact of flow induced stresses on the fatigue deformations of turbine in hydropower plant.

1.3. Objectives

The main and specific objectives of this research study are as follows:

Main Objective:

- To analyze fatigue deformation of Francis turbine runner due to flow induced stresses as a result of load variations when a hydropower plant is operated in frequency regulation mode.

Specific Objectives:

- To develop a CAD model of Francis turbine runner.
- To perform CFD Simulation using ANSYS Fluent Solver in order to find out pressure distribution in the Francis runner.
- To export the pressure loading to ANSYS Mechanical APDL Solver and perform structural analysis to determine stresses and deformations on the turbine blade.
- To observe stress and deformation development in Francis Turbine Runner so as to estimate runner lifetime.

1.4. Scope of the Work

After the completion of this thesis, fatigue study of Francis runner can be performed to identify sensitive parts of turbine where the impact of flow induced stress is comparatively higher. Fatigue results can help predict the lifetime of hydraulic turbines therefore such data shall be useful in scheduling as well as planning operational and maintenance time/cost in hydropower plant more effectively.

This work shall further help designer and manufacturer to study which material gives better performances and longer lifetime to turbines mostly in the sensitive parts of turbine i.e. runner blade. Additionally, as in this study, FSI can be used as a tool for diagnosing the causes of fatigue failure and study about the effective way to prevent those kinds of failures.

1.5. Limitation of the Study

Limitations of this study are:

- This study is limited to fatigue analysis of Francis runner due to flow induced stresses as a result of load variations. Cavitations related analysis for the Francis turbine runner has been excluded in this study.
- Due to confidentiality issue between manufacturer and hydropower developers, actual data of runner geometry employed in actual hydropower plants was not provided to the researcher. Therefore, the model developed in this study is using arbitrary data.
- No experimental rig has been setup to measure pressure at runner, dimensions, etc. due to cost and time barrier. Study has been performed on the CAD model via CFD and FEA simulations.

CHAPTER TWO: LITERATURE REVIEW

2.1. Previous Research Works

Unsteady flows in Francis turbine have been investigated for over 50 years. Different aspects have been studied, trying to determine what are causing the various load phenomena, how turbine material reacts and how further damage can be prevented through better design and operation restrictions (Valkvæ, 2016). Fatigue development in Francis turbine has increased the operational and maintenance costs of electro-mechanical components of hydropower plants and sometimes has led to tragic situations. It is of major concern and requirement that fatigue failure be studied and a measure to address the same be developed.

Hydraulic turbines are widely used to meet real-time electricity demand at moderate to low cost. Intermittency in the power grid due to high penetration of wind and solar power has raised significant concerns for grid stability and reliability resulting in an increase of start-stop cycles of hydraulic turbines (Trivedi et al., 2017). Because of grid frequency variations, stress oscillations are developed in a Francis turbine runner. Frequency in Nordic grid varies much and relatively fast changing rotational speed on synchronous machines, and production, which is changed inversely to frequency. Thus, torque acting on rotating masses also varies. This torque must be balanced by shear stresses induced in material of rotating masses. Thus, Francis runners in Norway have shown a tendency towards experiencing fatigue to a greater extent (Storli and Nielsen, 2014). Similarly, variable demand on the energy market, as well as the limited energy storage capabilities, requires a great flexibility in operating hydraulic turbines (Muntean and Marsavina, 2010). Therefore, turbines are frequently operated over an extended range of regimes. When run in varying load conditions, each cycle induces fatigue to turbine runner because it experiences unsteady pressure loading of high amplitude. Turbine runner accelerates freely due to instantaneous transition into no load during shutdown. The amplitude of unsteady pressure pulsation increases as the runner accelerates (Trivedi et al., 2017). Hydraulic induced stresses was studied in a complete Francis runner at Derbendikhan power station by using Finite Element Analysis (FEA) in which, it was explained that developments of fatigue cracks in the runner are mainly due to static and dynamic stresses in the runner (Saeed et al., 2010). A greater proportion of intermittent renewable energy sources and more complex structure of power systems brings a problem of increasing wear and tear of turbines. Thus, hydropower turbines experience fatigue due to increasingly more regulation movements of governor actuators (Yang et al., 2016). Similarly, for a Kaplan hydropower station operated in frequency regulating mode, amount of movements in regulating mechanism increases since it can regulate both the guide vanes and runner blades. When the hydropower station changes the produced power there are large servomotor forces applied to the regulating mechanism to open or close the wicket gate and the runner blade. During frequency regulation, large servomotor forces applied to regulating mechanism to open or close wicket gate and runner blade occurs frequently and risk of fatigue failure increases (Tapper, 2016). A research was carried out with an objective to show how dynamic phenomena occurring at various operating conditions may affect lifetime expectancy of different specific speed Francis runners where is was explained that energy market deregulation and arrival of new players, such as solar and wind turbines, has led to an increasing demand for flexible operation of hydraulic turbines. Instead of continuous close to peak operation, nowadays turbines are operated over the whole range, with many start/stops, extensive low load operation, synchronous condenser mode, and power/frequency regulation. Thus, it was concluded that such practice reduces machine life expectancy (Coutu, 2016).

It is necessary to study about the impact of flow induced stresses due to operation at varying load conditions on the fatigue deformations of turbine in hydropower plant. There are various ways to carry out such studies. In a study, runner blade strain gauge and pressure site measurements at various locations was performed, and correlated to CFD results and structural FEA using FSI techniques (Coutu, 2016). In another study, pressure fluctuations in the servomotor were measured along with the angles of guide vanes and runner blades while the plant was operated in frequency regulation mode. These data were inserted in MATLAB model where cyclic forces and stresses were displayed and thus expected lifetime of the turbine was estimated using number of cycles (Tapper, 2016). Similarly, FSI analysis of Francis runner model subjected to variable speed of rotation, as well as a Particle Image Velocimetry (PIV) measurement in the physical model was performed (Valkvæ, 2016). Likewise, numerical simulations of both governor and power plant behavior during primary control was performed, and distance of traversed guide vane movement, number of direction changes and distribution of amplitude between direction changes were extracted (Yang et al., 2016). Another method comprised of measurement of rotational speed; generator power; main servo motor position and grid frequency at a Francis turbine unit. Based on these measurements, simulations including hydraulic domain was performed. However, because of structural difficulties, cost and time considerations, commercial softwares like ANSYS, Solidworks, Catia has become more popular among the researchers (Storli and Nielsen, 2014). It has been explained in research studies that stress analysis of hydraulic turbine runner can only be performed by numerical methods due to complexity of these structures (Saeed et al., 2010). An understanding of fluid-structure interaction in turbine has become more essential since the different turbine loads are mainly induced by the internal fluid flow (Valkvæ, 2016). With the use of one way FSI, it is therefore possible to study about the impact of flow induced stresses on Francis turbine runner.

2.2. Francis Turbine

Francis turbine is an inward flow reaction turbine that uses both axial and radial flow of water to convert potential and kinetic energy of water into mechanical energy through rotation of axel as water enters the turbine radially and leaves axially. Water enters these turbines radially meaning that it enters the turbine perpendicular to the rotational axis. Once entering the turbine, water always flows inwards, towards the center. Once the water has flown through the turbine, it exits axially i.e. parallel to the rotational axis (Boyle, 2012). Therefore, Francis turbine is also called as mixed flow turbine.

Francis turbine is used most frequently in medium or large-scale hydropower plants. These turbines can be used for net head 9 m to 400 m and flow rate 0.45 to 25 m/s. These turbines are beneficial as they work equally well when positioned horizontally as they do when they are oriented vertically (Boyle, 2012). A simple schematic diagram of Francis turbine is shown in figure 1 (below).

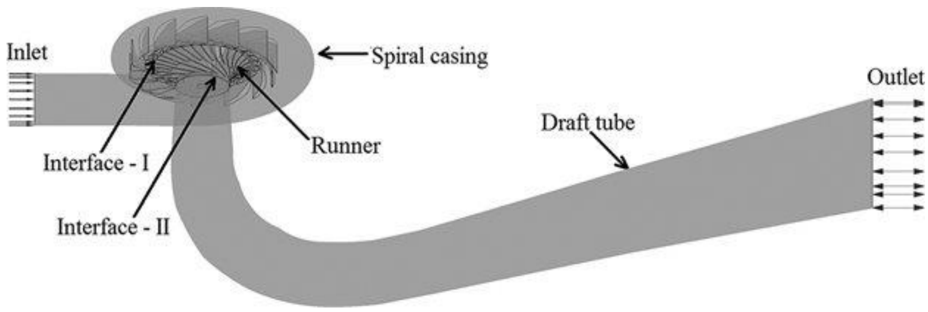


Figure 1. Schematic diagram of Francis Turbine (Turbinesinfo - All About Turbines, 2016)

2.2.1. Major Components

Major components of Francis turbine are as shown in figure 2 below:

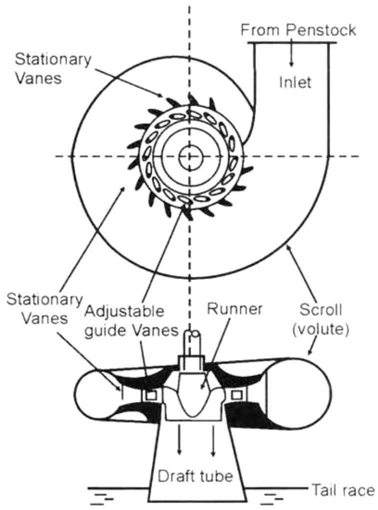


Figure 2. Components of Francis Turbine (Husain et al., 2008)

These components can be further explained as:

i. <u>Spiral casing</u>

It constitutes a closed passage whose cross-sectional area decreases uniformly along the circumference to keep the fluid velocity constant in magnitude along its path towards the guide vane.

ii. Stay vanes

Stay vanes are stationary parts which are provided in order to direct the flow towards the runner blades at designed angle.

iii. Guide vanes

Guide vanes or wicket gates direct the water onto the runner at an angle appropriate to the design. The motion to them is given through a governing mechanism.

iv. Runner

Runner converts energy from water to rotational motion of the main shaft. The driving force on the runner is both due to impulse and reaction effects. Runner comprises of numbers of blades attached to the hub and shroud. Runner blades of Francis turbine remain stationary.

v. Draft tube

It is a closed tube with gradually increasing area, which is used to discharge water to the tailrace. The major function of draft tube is to reduce water velocity as the flow discharges towards tailrace.

vi. Governing mechanism

Francis turbine is governed by varying the flow area in between adjustable guide vanes. Guide vanes are connected to the regulating ring through links. The regulating ring is connected to the regulating lever through two regulating rods. The regulating ring is thus connected to the regulating shaft, which is operated by a servomotor (Rajput, 2015).

2.2.2. Performance Chart

Francis turbine performs well when operated at full load capacity. Figure 3 (below) shows efficiency of different types of turbines at various load flow. It can be seen that efficiency of Francis turbine is almost 90% when the operated at more than 70% of design discharge.

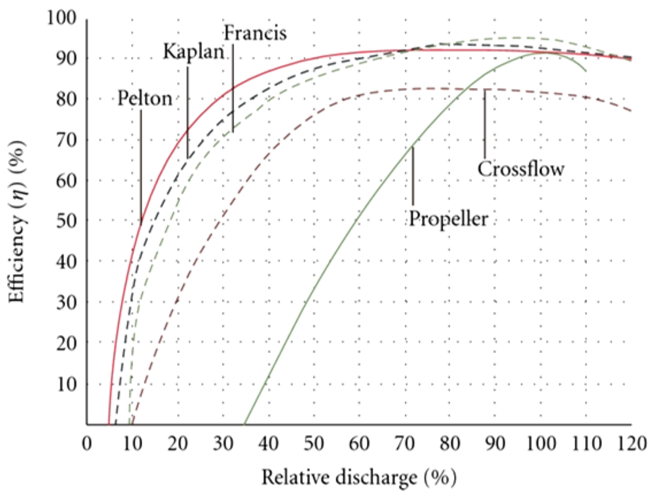


Figure 3. Performance chart for different types of hydraulic turbines (Vinogg and Eldstad, 2003)

2.2.3. Flow Control

The flow rate in Francis turbine is controlled by varying the flow area in between the adjustable guide vanes. The guide vanes are hinged at the center to a circular ring. The area in between the vanes is varied by varying the guide vane angle, α_1.

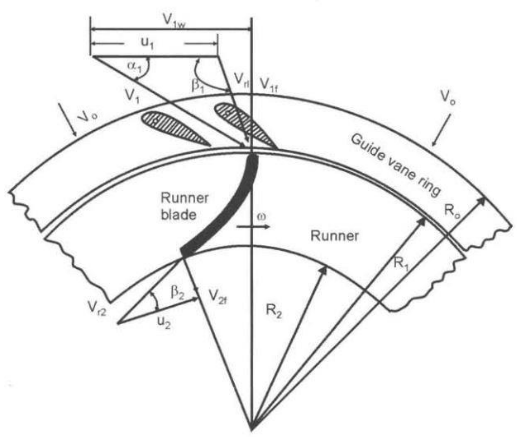


Figure 4. Inlet and exit velocity diagram (Husain et al., 2008)

Referring to velocity diagram above (Figure 4), change in α_1 results in change of whirl velocity and flow velocity components. Such a change also changes inlet angle of the blade β_1 which means deviation from 'no shock entry' conditions. Thus efficiency is reduced at partial loads. Similarly, the exit velocity diagram also changes and whirl component produces vortex motion in draft tube which may cause cavitation phenomena in turbine. The regulation of guide vanes is done by servo mechanism. As load on the turbine decreases the piston of servo mechanism moves to the right and this causes the movement necessary to close the gates (Husain et al., 2008).

2.3. Stress Analysis

Francis type hydraulic turbines are the most common in use today. The turbines are operated at the fixed rotational speed governed by the grid frequency of a country. Traditionally, Francis turbines are designed for a fixed speed and variable discharge characteristics. Hence, the mechanical power from a runner is managed by the guide vane aperture (discharge control). Such turbines work well at the design point, i.e., best efficiency point (BEP), and the turbine efficiency is high. However, when the electricity demand is low/high, the turbines are forced to operate under off-design condition, and the turbines experience fatigue due to high amplitude pressure fluctuations, vortex breakdown and cavitation (Trivedi et al., 2017). During its lifetime in operation, a Francis turbine will go through different operating regimes that will each affect its life expectancy and reliability to a different degree. While it is now well known that some transient and dynamic regimes can be damaging for the runner (JF et al., 2016).

Operating regime of a hydraulic turbine can be divided into steady state operation and transient state operation (Ramirez et al., 2015). The current electrical system is characterized by large variations in the production and consumption of electricity. This results into more balancing services which results into increased stress on mechanical equipment by exerting them to dynamic loads (Valkvæ, 2016). During operation, several steady loading and unsteady loading act on Francis turbine components. Steady loading comprises of fluid pressure, centrifugal force and runner’s weight (Muntean and Marsavina, 2010). During steady state, the unit operates at constant head, speed, load and constant opening of guide vanes. The forces acting during this period are constant in magnitude, direction and frequency. However, due to defects such as excess pressure pulsations generated in the intake pipe, cavitation or misalignment, random non-periodic forces have different directions, amplitudes and frequencies (Ramirez et al., 2015).

Unsteady loading comprises of high frequency pressure fluctuations due to stator-rotor interaction as well as vortex rope phenomenon (Negru et al., 2011). As water flowing towards the runner blades via guide vanes, pressure difference between pressure side and suction side of vanes is created as a result of stator-rotor interaction. Then the pressure difference forms a wake at the tail of guide vanes which can be treated as pressure pulse when it impacts the runner blades. Figure 5 (below) illustrates this phenomenon.

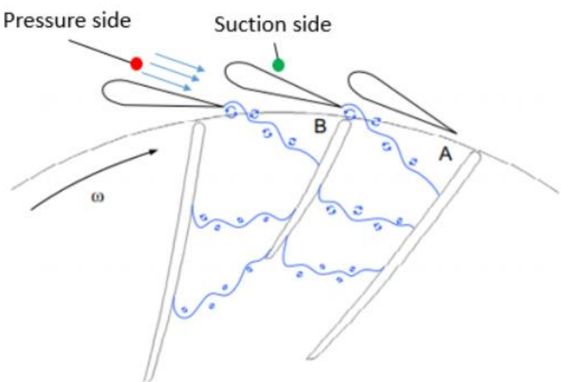


Figure 5. Guide vane pressure difference and the following wakes (Skorpen, 2018)

Transient state operation occurs when there is a change in the head or load or wicket gate opening i.e. starting, synchronizing, changing load, stopping, load rejections, tripping, turbine failures and over speed. During these periods, vibrations do not follow a single pattern but change in magnitude, direction and frequency; their values increase or decrease depending on amount of water that enters. Unsteady forces are superimposed on steady state forces, resulting in vibrations which results in wear and fatigue failure of different components. Although the units are designed with a sufficient margin of safety to withstand normal stresses, high vibrations and dynamic stresses result in gradual development of cracks (Ramirez et al., 2015). Additionally, frequent starts or stops or rapid changes in load pattern can induce stresses in Francis turbine.

2.4. Fatigue

Fatigue is a failure phenomenon that is found in components under fluctuating loads, which are well below the static design loads of the component. With the beginning of the industrial revolution, cast and wrought iron took the place of bricks or wood as construction materials. Thus, components could be designed to withstand tensile forces, which lead to more complex constructions and resulted in an increasing fatigue problem (Huth, 2005). Often, machine members are found to have failed under the action of repeated or fluctuating stresses; yet the most careful analysis reveals that the actual maximum stresses were well below the ultimate strength of the material, and quite frequently even below the yield strength. The most distinguishing characteristic of these failures is that the stresses have been repeated a very large number of times. Hence the failure is called a fatigue failure (Budynas and Nisbett, 2011). Fracture often happens after a long period of cyclic tension. Fatigue is estimated to be the cause of about 90% of all metal failures (Valkvæ, 2016). Fatigue failure is due to crack formation and propagation. A fatigue crack will typically initiate at a discontinuity in the material where the cyclic stress is maximum (Budynas and Nisbett, 2011).

2.4.1. Fatigue-Life Methods

Three major fatigue life methods used in design and analysis are stress-life method, strain-life method and linear-elastic fracture mechanics method. These methods attempt to predict the life in number of cycles to failure for a specific level of loading. The stress-life method, based on stress levels only, is the least accurate approach, especially for low-cycle applications. However, it is the most traditional method, since it is the easiest to implement for a wide range of design applications, has ample supporting data, and represents high-cycle applications adequately. The strain-life method involves more detailed analysis of the plastic deformation at localized regions where the stresses and strains are considered for life estimates. This method is especially good for low-cycle fatigue applications. The fracture mechanics method assumes a crack is already present and detected. It is then employed to predict crack growth with respect to stress intensity. It is most practical when applied to large structures in conjunction with computer codes and a periodic inspection program (Budynas and Nisbett, 2011).

2.4.2. S-N Curves (Wöhler Curves)

S-N Curve is a stress-life approach to fatigue life assessment. In order to specify a safe strength for metallic material under repeated loading, it is necessary to determine a limit below which no failure can be detected after applying a load for a specific number of cycles. By using a testing machine, a series of specimens can each be subjected to a predetermined stress and cycled to failure. The results are plotted as a graph representing the stress S on the vertical axis and the number of cycles to failure N on the horizontal axis. The graph is called an S-N diagram (Valkvæ, 2016).

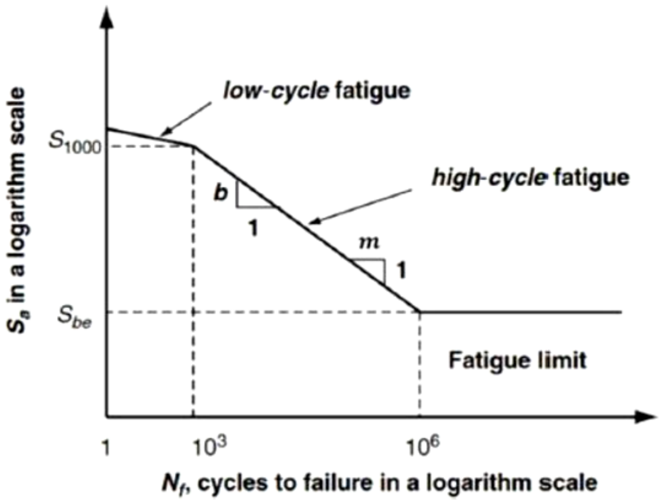


Figure 6. Schematic representation of an S-N curve for steels (Lee et al., 2005)

In Figure 6 (above), lines represent the fatigue limit of steel at particular number of cycle. These curves are either constructed through calculation or tests. From the figure, it can be seen that if the stress amplitude is kept below the fatigue limit where the line has flattened out, the material can withstand infinite number of cycles.

2.4.3. Low-Cycle Fatigue (LCF)

The fatigue stresses that occur in a hydropower plant are divided into two parts, called low cycle fatigue and high-cycle fatigue (Huth, 2005). Low-cycle fatigue is the regime associated with load amplitude high enough to cause the fracture of a part after a limited number of cycles, typically less than 10^5 cycles (Doquet et al., 2019). LCF produces failures after only a low number of cycles with high stresses and plastic deformation (Xin, 2013). In order to analyze the mechanisms which cause low-cycle fatigue fractures, and to predict the associated life, it is necessary to carry out tests specifically developed to control the amplitude of the stress or strain applied. Thus, low-cycle fatigue tests are preferably carried out with controlled plastic strain amplitude, because the level of plastic strain evolves over time in metallic materials for a total deformation amplitude applied (Doquet et al., 2019).

2.4.4. High-Cycle Fatigue (HCF)

High cycle fatigue is a type of fatigue caused by small elastic strains under a high number of load cycles before failure occurs. The stress comes from a combination of mean and alternating stresses. The mean stress is caused by the residual stress, the assembly load, or the strongly non-uniform temperature distribution. The alternating stress is a mechanical or thermal stress at any frequency. HCF requires a high number of loading cycles to reach fatigue failure mainly due to elastic deformation. It has lower stresses than LCF, and the stresses are also lower than the yield strength of the material. HCF usually does not have macroscopic plastic deformation as large as that in LCF. The dominant strain in HCF is mainly elastic. In contrast, the dominant strain in LCF is plastic. Because HCF is governed by elastic deformation, stress is usually a more convenient parameter than strain to be used as the failure criterion. The HCF life of the component is usually characterized by a stress–life curve, where the magnitude of a cyclic stress is plotted versus the logarithmic scale of the number of cycles to failure (Xin, 2013).

For high-pressure Francis turbine runners, high-cycle fatigue may occur due to transverse blade vibration induced by hydraulic load fluctuations over a spectrum of frequencies. The strength of these fluctuations is a function of the hydraulic pressure, the water velocity and the geometry of the stationary apparatus guiding the water into the runner. The spiral casing distributes the water flow through the stay and guide vanes onto the runner equally from all sides. Wakes behind the stay vanes and guide vanes expand to the runner and cause surface pressure pulsations every time a runner leading edge passes one of the guide vanes, the water flow is getting interrupted what consequently leads to fluctuations of forces and torque. The unsteady flow from blades passing the guide vanes is normally assumed to induce the predominant HCF vibration stresses (Huth, 2005). Cracks related to fatigue failure almost always initiate on the surface of a component at some point of stress concentration (Valkvæ, 2016).

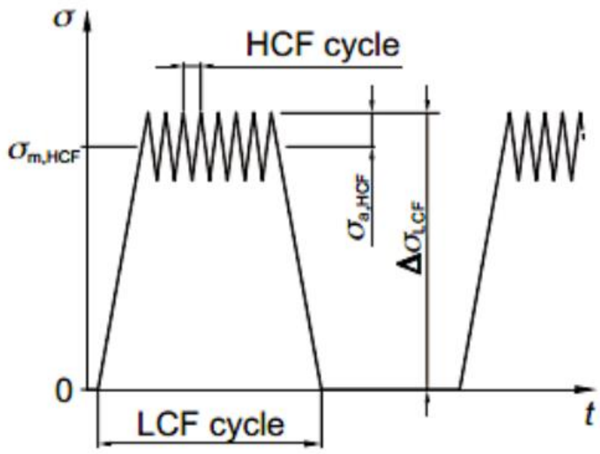


Figure 7. LCF and HCF in a Francis Runner (Huth, 2005)

A typical stress history for a Francis runner (or any type of machine that is exposed to various dynamic stresses) is as shown in Figure 7.

2.4.5. Palmgren-Miner Rule

A generic fatigue model can assess the lifetime of a turbine. The Palmgren-Miner model evaluates the fatigue damage of the turbine, which is a cumulative damage model. The model is also known as the linear damage accumulation rule.

$$D = \sum_{i=1}^{k} D_i = \sum_{i=1}^{k} \frac{n_i}{N_i(\Delta\sigma)} \; [-] \qquad \text{Equation 1}$$

In above equation, D is defined as the cumulative linear summarized damage that results from each load cycle, n_i is the number of load cycles at a given stress level, and N_i is the maximum number of load cycles given by the S-N curve at that particular load. The principle of the model is that when the D reaches one, the component will collapse. The goal of introducing a damage model is to find the impact of cyclic stresses on the functional lifetime of the component. In order to this, a generalized Palmgren-Miner model is utilized. The generalization of the model assumes that both fatigue and static wear is cumulatively summarized. This is a generalization of the model presented in ISO (International Organization for Standardization) 19902, which includes previous damage. Equation below expresses the current remaining fatigue damage.

$$D_2 = 1 - D_1[-] \qquad \text{Equation 2}$$

Here, the subscripts 1 and 2 denote damage at times 1 and 2, respectively (Huth, 2005).

2.5. Literature Review on Francis Runner Fatigue

Intermittency in the power grid due to high penetration of wind and solar power has raised significant concerns for grid stability and reliability resulting in an increase of start-stop cycles of hydraulic turbines. Each cycle induces fatigue to turbine runner because it experiences unsteady pressure loading of high amplitude (Trivedi et al., 2017). In Francis turbine runners, fatigue cracks tend to occur early or after decades of operation. The failure mechanism is considered a combination of low cycle fatigue caused by startups and high cycle fatigue caused or induced by rotor-stator interaction (Ramirez et al., 2015). In a study comprising of pressure and strain measurements performed on the Francis test ring set up at Waterpower Laboratory at NTNU, it was found that highest stress was observed at blade trailing edge towards the shroud, where the blade is at its thinnest (Valkvæ, 2016). In a study comprising of FSI analysis, it was found that maximum stresses caused by hydraulic forces are at transition between blades and crown/band on trailing edge. Thus, this region is regarded as a critical area for fatigue crack initiation in Francis turbine runner (Saeed et al., 2010). A study for life prediction of Francis turbine runner was performed via local strain approach. Static analysis was realized in operation loads obtaining the Von Mises stresses, which showed that the maximum stress was localized in the blade near to the band, close to the runner axis (Flores et al., 2012).

Studies have shown that mostly the stresses appear where the blade is fixed to the hub and shroud near the trailing edge. Deformation and stress distribution results obtained from numerical computations with finite element analysis shows that the fracture occurred at the T-joint stress concentration between crown and trailing edge due to fatigue generated by unsteady loading and started from the deep rills of the welded seam realized in site (Muntean and Marsavina, 2010). In another study carried out by Flores et al. (2012), it was observed that the maximum stress was localized in the blade near to the band, close to the runner axis. If the machine operates under these conditions of resonance and dynamical stress, exceeding the time of crack initiation growth of 23 days, the crack initiation growth will occur.

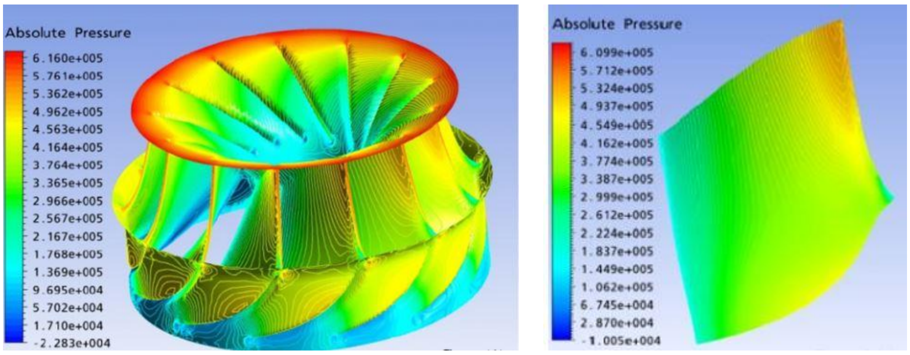


Figure 8. CFD Results showing the maximum stress areas (Saeed et al., 2010)

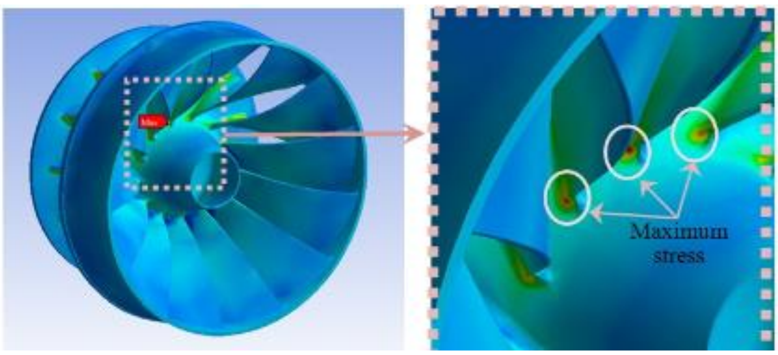


Figure 9. Stress distribution on the pressure sides of the blade for specific cases obtained by the FEM calculations (Saeed et al., 2010)

Additionally, it was found that Francis runner blade failure is a consequence of unsteady loading, stress concentration, inhomogeneity of the structure and improper repair welding (Muntean and Marsavina, 2010). Highest stress is observed at blade trailing edge towards the shroud, where the blade is at its thinnest (Valkvæ, 2016). Since, most of the maximum stresses caused by hydraulic forces were found to exist at transition between blades and crown/band on trailing edge, this region is regarded as a critical area for fatigue crack initiation in Francis turbine runner (Saeed et al., 2010). Thus, such stresses on runner blades result into crack initiation. Once the crack size reaches a critical size, turbines can undergo catastrophic failure long before the design life of turbine.

2.6. CFD Analysis

In later years the advances in computer power combined with powerful graphics and 3D-models have made the process of creating CFD models and analyzing the results much more intuitive and less time consuming. CFD also offers an alternative to expensive laboratory testing. As a result of this, CFD has now been well established as an industrial design tool (ANSYS 15-Help: Zu, 1991). In a CFD analysis the differential equations describing the processes of momentum, heat and mass transfer, known as the Navier-Stokes equations are solved. Also other equations describing processes like combustion and turbulence are included in the solvers (ANSYS 15-Help). There are several different solution methods that are used in CFD codes. The most common is known as the finite volume technique. In this technique the region of interest is divided in to small control volumes. The equations is then discretized and solved iteratively for each control volume. Different solvers use different forms of the equations. ANSYS CFX solves the unsteady Navier-Stokes equations in their conservative form (ANSYS 15-Help).

2.7. FEA Analysis

FEA is a powerful tool used to solve the boundary conditions of continuum. The method is widely used in the analysis of hydraulic turbines. The method is used in stress and strain analysis, Eigen value analysis, dynamic response analysis and flow analysis (Zu, 1991). The FEM discretization process transforms partial differential equations into algebraic equations (Valkvæ, 2016). In FEM the structure of interest is divided in to several small elements. The variables of the governing differential equations and their derivatives are specified at the nodes at the edges of each element. The solution is dependent on the quality of the grid. There is no general rule for division of elements or grid generation, but size and arrangement of the grid is important in practice. It is desirable with fine grids in areas of stress concentration. Elements with very slender proportions should be avoided. When the final element method is applied to three dimensional problems, the number of elements will often become very large and the calculations will require considerable computer time (Zu, 1991).

2.8. FSI Analysis

Fluid Structure Interaction analysis is the coupling of solution fields in fluid and solid domains (Zu, 1991). When solving an FSI problem, there are two options i.e. unidirectional (one-way) FSI and bidirectional (two-way) FSI. A unidirectional (one-way) FSI analysis is performed by running a CFD analysis, extracting the forces acting on a solid surface and then importing them to a structural analysis. In a unidirectional analysis, the response from the structural analysis will not affect the CFD analysis (ANSYS 15-Help).

In a bidirectional (two-way) analysis the structural response will be taken into account and affect the flow simulation (Zu, 1991). One-way FSI have been used by various researchers for fatigue study. Stress analysis of hydraulic turbine runner can only be performed by numerical methods due to the complexity of these structures (Saeed et al., 2010). An understanding of fluid-structure interaction in turbine has become more essential since the different turbine loads are mainly induced by the internal fluid flow (Valkvæ, 2016).

CHAPTER THREE: RESEARCH METHODOLOGY

This study follows the methodology as dipicted in flow chart below:

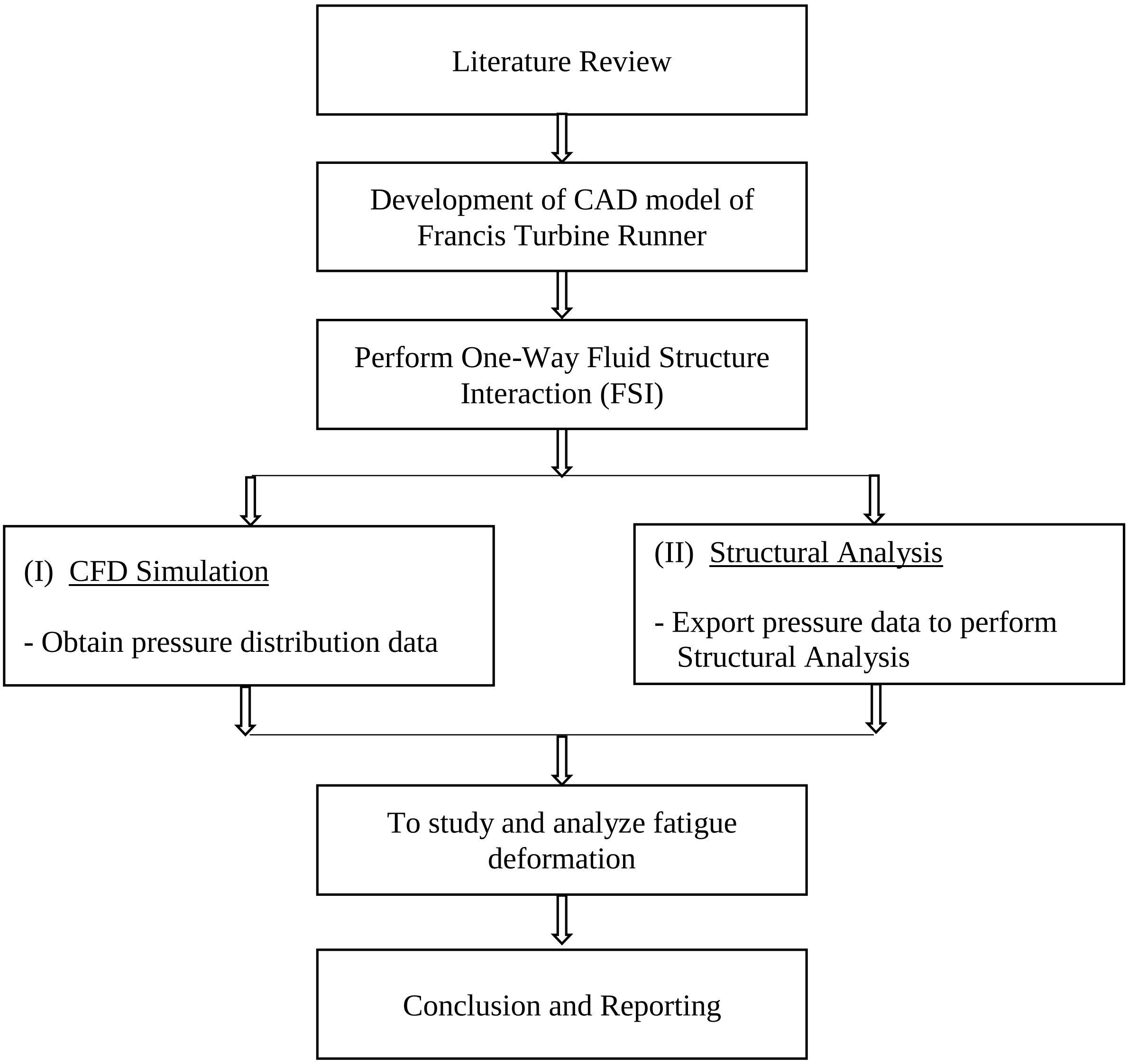


Figure 10. Research Methodology

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. Figure 11 below shows the schematic diagram of FSI analysis process for this study.

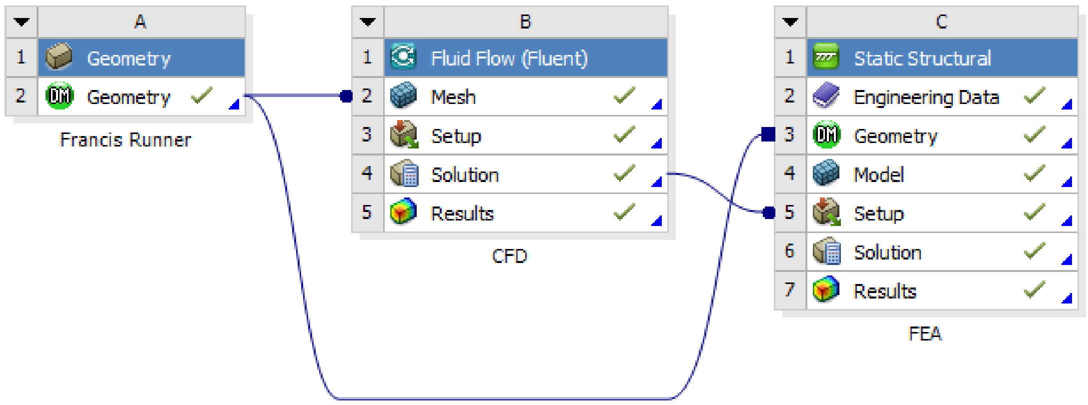


Figure 11. FSI Analysis procedure in ANSYS Software

Methods adopted in the study have been explained below sequentially.

3.1. Geometric Modeling of Francis Turbine Runner

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 3.

$$P = g*\eta_t*\eta_g*H*Q \qquad \text{Equation 3}$$

3.1.1. Rotational Speed

To calculate rotational speed (n) of the runner, following steps were adopted. Firstly, rotational speed was calculated directly using equations 4-6 and then the synchronous rotational speed was calculated using equations 7-8.

$$c_{nq} = min(2600; 2600 - (200000 - P)/365) \qquad \text{Equation 4}$$

$$n_q = \frac{c_{nq}}{H^{0.535}} \qquad \text{Equation 5}$$

$$n = n_q \frac{H^{1.25}}{P^{0.5}} \qquad \text{Equation 6}$$

Here, c_{nq}, n_q and n were calculated as 2054, 253.3 and 1272.91 rpm respectively. Since, n is not a synchronous rotational speed, it is required to calculate the same. For this purpose, number of poles was determined as:

$$number\ of\ pole\ pairs = \frac{f * 60}{n} \qquad \text{Equation 7}$$

The standard value of grid frequency (f) was considered i.e. 50 Hz. Thus, using equation 7, number of pole pairs was found to be 2.35. Taking the rounded value of pole pairs number as 3, synchronous rotational speed was calculated as 1000 rpm using equation 8.

$$n_s = \frac{120 * f}{2 * (number\ of\ pole\ pairs)} \qquad \text{Equation 8}$$

The angular rotational speed of the runner was thus calculated as 104.72 rad/s using equation 9.

$$\omega = \frac{2\pi n_s}{60} \qquad \text{Equation 9}$$

3.1.2. Specific Speed

Similarly, specific speed of turbine was calculated as 67.27 using equations 10.

$$n_{specific\ speed} = n_s \frac{\sqrt{Q}}{H^{0.75}} \qquad \text{Equation 10}$$

3.1.3. Runner Dimensions

To obtain preliminary dimensions for runner, arbitrary values for inlet and outlet diameters were taken and other calculations were done on the basis of these values. Preliminary dimensions of runner are as follows:

Shaft diameter, D_s = 140 mm

Runner inlet diameter, D_1 = 643 mm

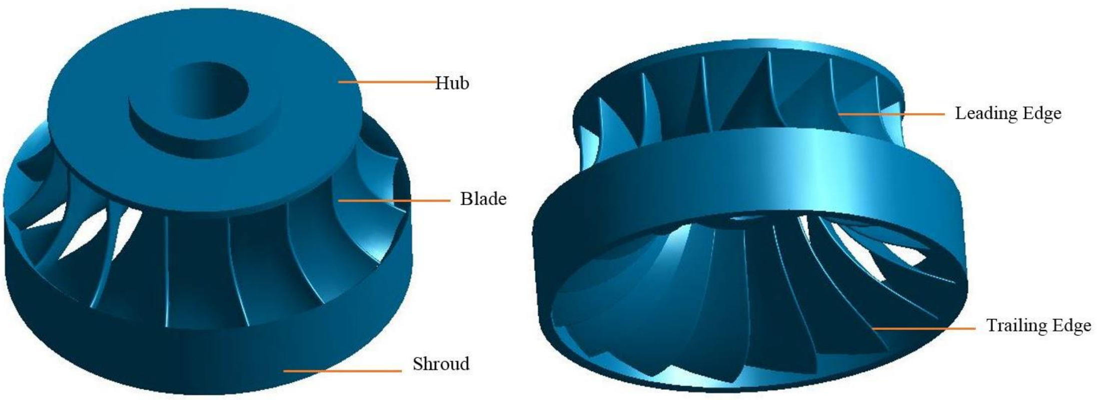
Runner throat diameter, D_2'= 605 mm

Runner outlet diameter, D_2 = 496 mm

3.1.4. Runner blades

Different number of blades are considered while designing Francis runner. For instance, Bergmann, (2012) have considered 17 blades; Skorpen (2018), Okyay (2010), Flores, E. et al. (2012), Trivedi et al. (2017) and Zhou et al. (2017) have studied failure analysis considering 15 blades; Muntean (2010) and Negru et al. (2012) studied failure analysis for runner comprising 14 blades while Mughal et al. (2015), Flores, M. et al. (2012), Ramirez et al. (2015), Saeed (2010) and JF et al. (2016) studied fatigue analysis for runner comprising 13 number of blades. The number of blades are chosen based on the operating head. Runners with higher head will require a higher number of blades. Increasing the number of blades reduce the pressure loading on the blade which will help to avoid cavitation and also prevent separation at the runner inlet during low loads. An increase in the number of blades also leads to more contact surface through the runner and thereby an increase in the friction losses (Bergmann, 2012). The number of blades and blade thickness have been determined based on trail-and-error while designing and literature review.

Francis runner model considered in this study consists of 15 blades, hub, and shroud (as shown in figure 12 below). 3D commercial software, Solidworks 2017 was used to develop the model of Francis Runner.



(i) (ii)

Figure 12. CAD model of Francis Turbine Runner

3.1.5. Characteristic Parameters

Similarly, radial flow velocity (V_f), tangential velocity (u_1), speed ratio (K_u) and flow ratio (K_f) were calculated using equations 11 to 14.

$$V_f = \frac{Q}{\pi DbK} \qquad \text{Equation 11}$$

$$u_1 = \frac{\pi D n_s}{60} \qquad \text{Equation 12}$$

$$K_u = \frac{u_1}{\sqrt{2gH}} \qquad \text{Equation 13}$$

$$K_f = \frac{V_f}{\sqrt{2gH}} \qquad \text{Equation 14}$$

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 (Rajput, 2015). The value of K assumed in this study is 0.95.

The values obtained via calculations for various parameters used in this study are as tabulated below (Table 1).

Table 1. Calculated values of various 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

3.1.6. Fluid Domain

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

3.2. Meshing

Geometry was exported to Meshing tool in ANSYS to generate unstructured mesh optimized for Physics setup in CFD and Fluent as Solver.

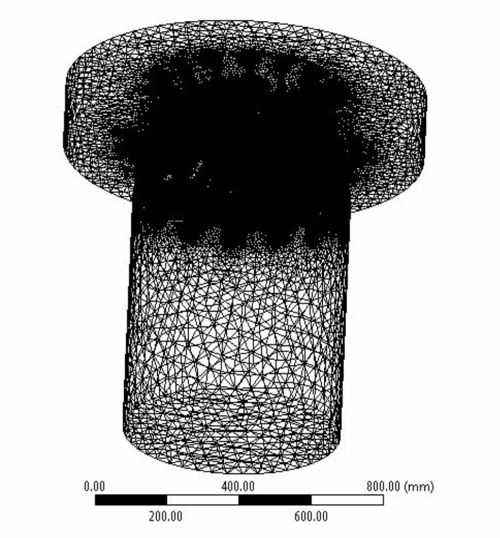


Figure 13. Meshed fluid domain of Francis turbine runner

Program controlled triangular surface mesh was generated consisting of 1527496 and 8427075 number of nodes and elements respectively. Figure 13 (above) shows the meshed fluid domain for Francis runner and figure 14 (below) shows the meshed runner geometry.

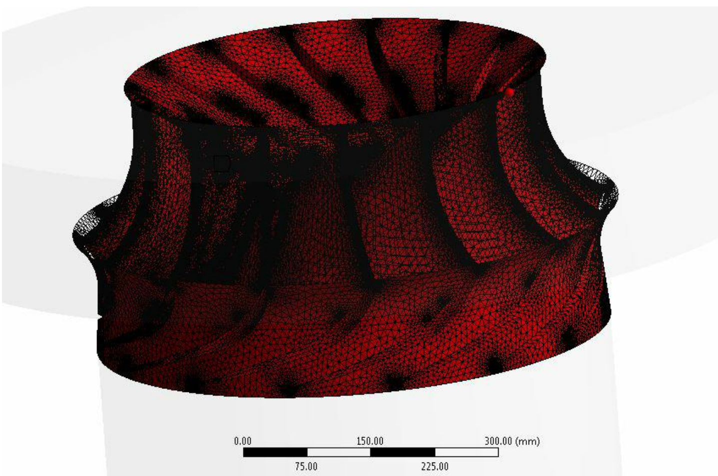


Figure 14. Meshed Francis runner geometry

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.

3.3. Computational Fluid Dynamics (CFD) Analysis

CFD simulation is required to judge the right flow behavior of fluid inside or outside the structure (Tanwar et al., 2012). The meshed CAD model of Francis Runner was exported to ANSYS Fluent Solver, where following boundary conditions listed in Table 2 below were assigned.

3.3.1. 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. Shear Stress Transport (SST) k-omega model was chosen as turbulence model. This mathematical approach is considered as part of the latest generation of turbulence models due to its capability to deal with complex flows and robustness as a numerical tool. 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

3.3.2. Governing Equations

In CFD analysis the differential equations describing the processes of momentum, heat and mass transfer, known as the Navier-stokes equations are solved. The unsteady Navier-stokes equations are solved in their conservative form. Also other equations describing processes like combustion and turbulence are included in the solvers (ANSYS 15-Help). The two general governing equations available for fluid flow simulation in CFD are as represented by equations 15 and 16 below:

Conservation of mass:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho V) = 0 \qquad \text{Equation 15}$$

Conservation of Momentum:

$$\frac{\rho \partial V}{\partial t} + \rho(V.\nabla)V = -\nabla \mathrm{p} + \rho.g + \nabla.\tau_{ij} \qquad \text{Equation 16}$$

3.4. 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 (σ)	250	MPa
4	Tensile Strength (UTS)	460	MPa
5	Poisson's Ratio	0.3	

3.4.1. 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.

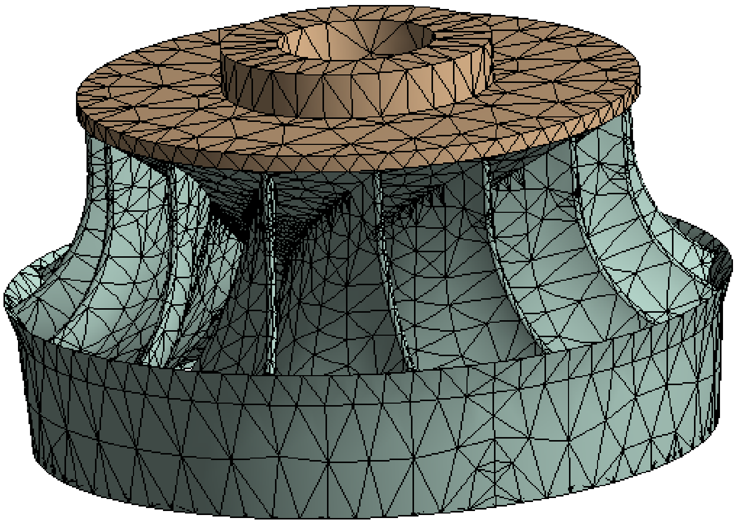


Figure 15. Meshed runner geometry for FEA

Figure 15 (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.

3.4.2. Boundary Conditions and Assumptions

Zero displacement boundary condition was applied at the shaft area. Rotational velocity of 1000 rpm about x-axis was assigned. Pressure loads were used as imported from CFD analysis.

With these boundary conditions, structural analysis was performed. Characteristics stress fields are presented in terms of Von Mises Equivalent stress for the Francis runner.

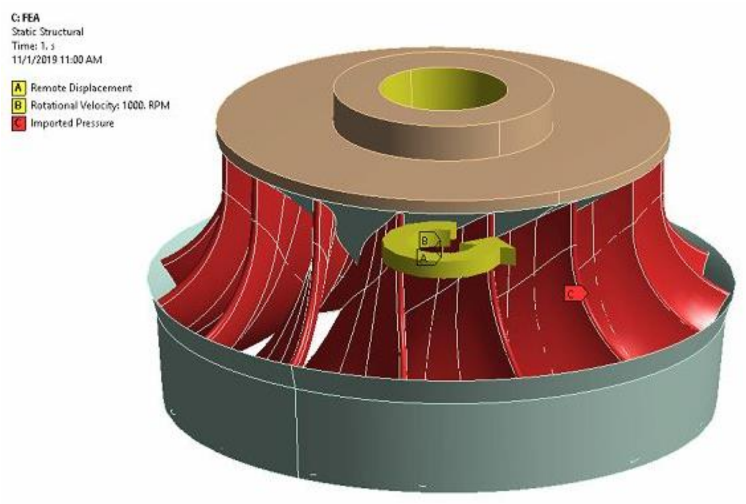


Figure 16. Boundary Conditions for Structural Analysis

Pressure loads obtained from CFD were imported to ANSYS Mechanical. 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 17 (below) shows the pressure distribution on runner blade surface.

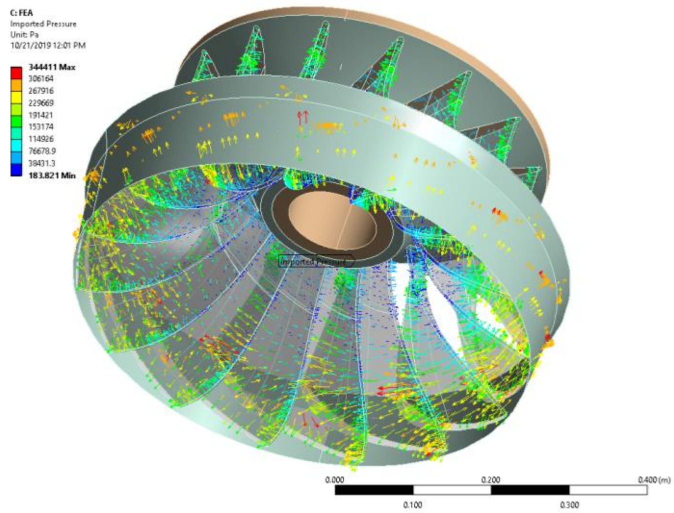


Figure 17. Imported pressure fields

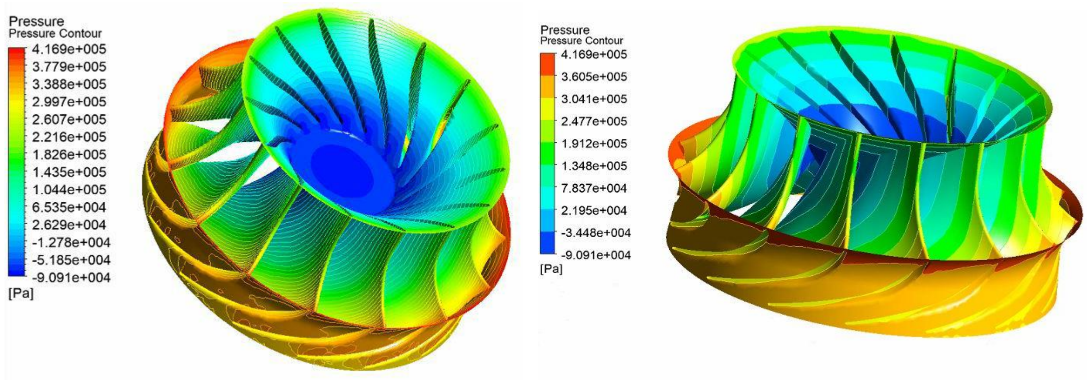
CHAPTER FOUR: RESULTS AND DISCUSSION

The detailed analysis, discussion, and interpretation of the results obtained via FSI analysis have been presented as below:

4.1. CFD Analysis

4.1.1. Pressure Plot

Figure 18 (i and ii) below show the pressure distribution inside Francis turbine runner.



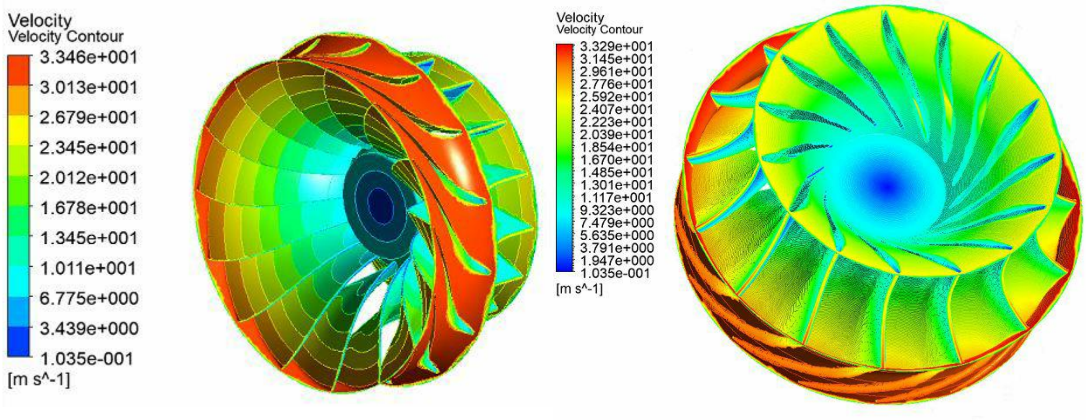
(i) (ii)

Figure 18. Pressure Contour at 1.6m^3/s discharge and 1000 rpm

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. Studies (Flores et al., 2012: Negru et al., 2011: Valkvæ, 2016) have shown that, maximum pressure is observed at the joint between the runner blades and band and/or crown which remained true for this study too.

4.1.2. Velocity Plot

Figure 19 (i and ii) below show the velocity contour generated at 6.13 m/s flow velocity and full load operation condition for the Francis turbine runner.



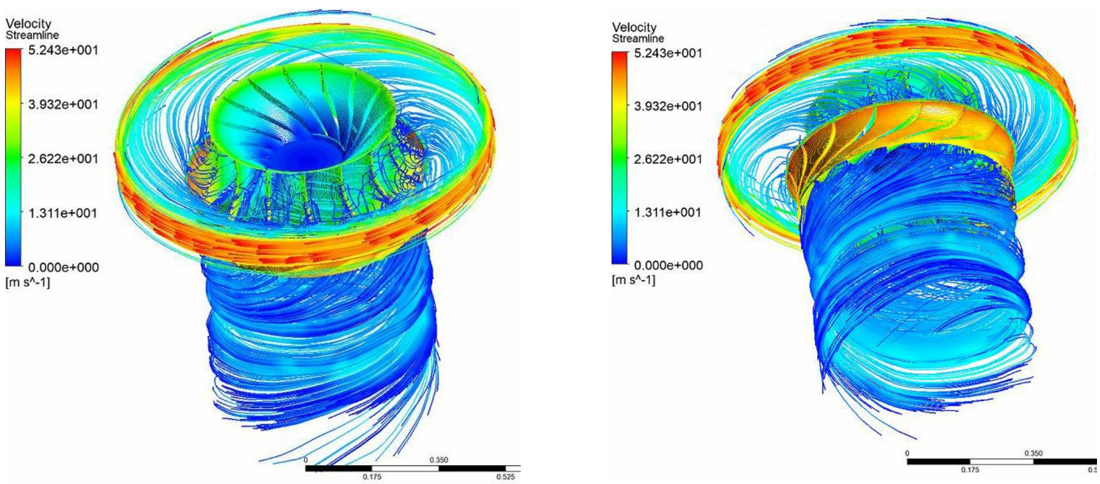
(i) (ii)

Figure 19. 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.

4.1.3. Velocity Streamlines

Figure 20 (i and ii) below show the streamlines of the tangential velocity in the fluid domain as obtained from CFD simulations.



(i) (ii)

Figure 20. Velocity Streamlines

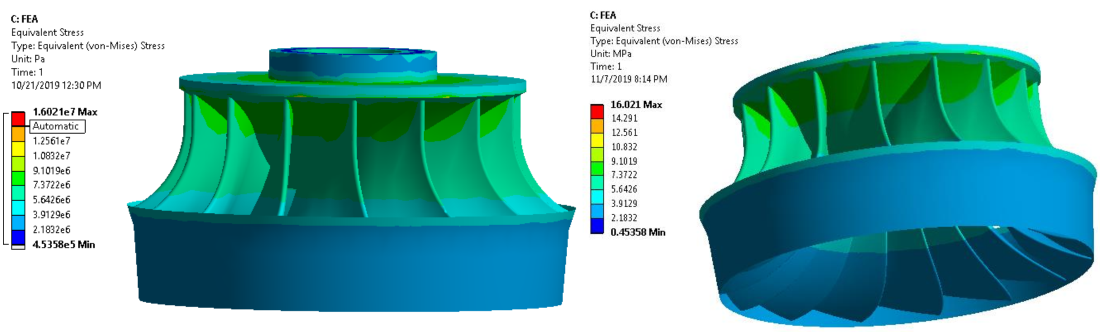
Result shows that maximum velocity observed was 52.43 m/s. It can be observed that the flow velocity is gradually decreasing as the fluid flows towards the outlet.

4.2. Structural Analysis

In structural analysis, results are achieved through analysis of Von Mises stress, total deformation and fatigue tool.

4.2.1. Von Mises Equivalent Stress

Figure 21 shows the Von Mises Stress Distribution in the Francis 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. Therefore, the runner blades shall not undergo cracks propagation.



(i) (ii)

Figure 21. Von Mises Stress Distribution

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 figure 21.

4.2.2. Total Deformation

Figure 22 (below) shows that the total deformation is higher at the suction side of the runner blade.

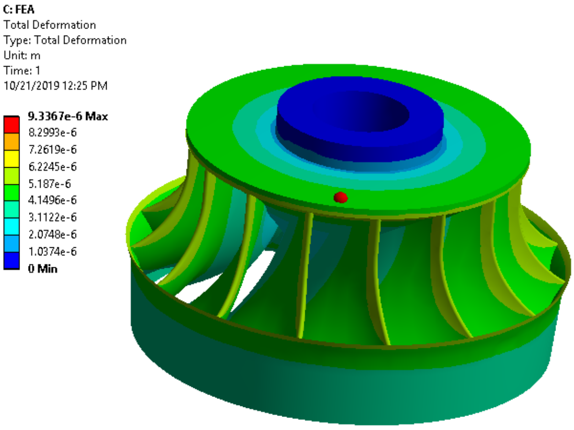


Figure 22. Total Deformation observed in Francis Runner

Maximum deformation observed was 0.00934 mm. Higher value is observed at the pressure side of the blades. However, the value is considerably lower.

4.2.3. Fatigue Life

Figure 23 below shows the contour plot for fatigue life.

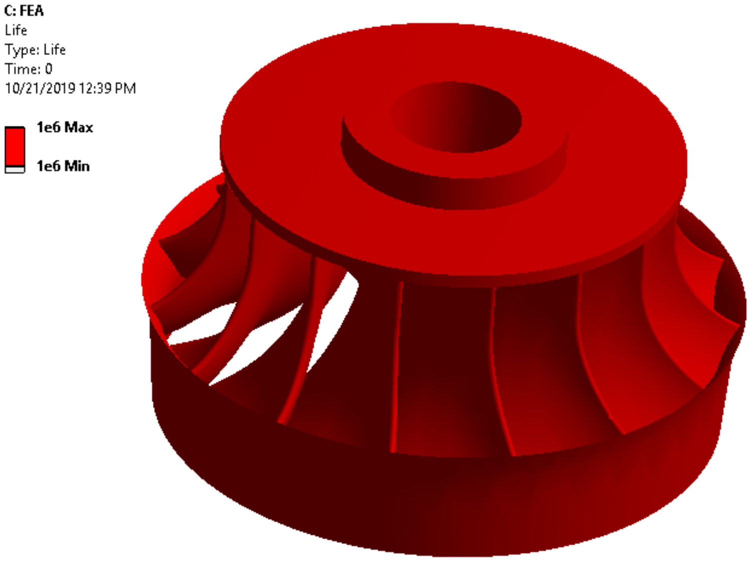
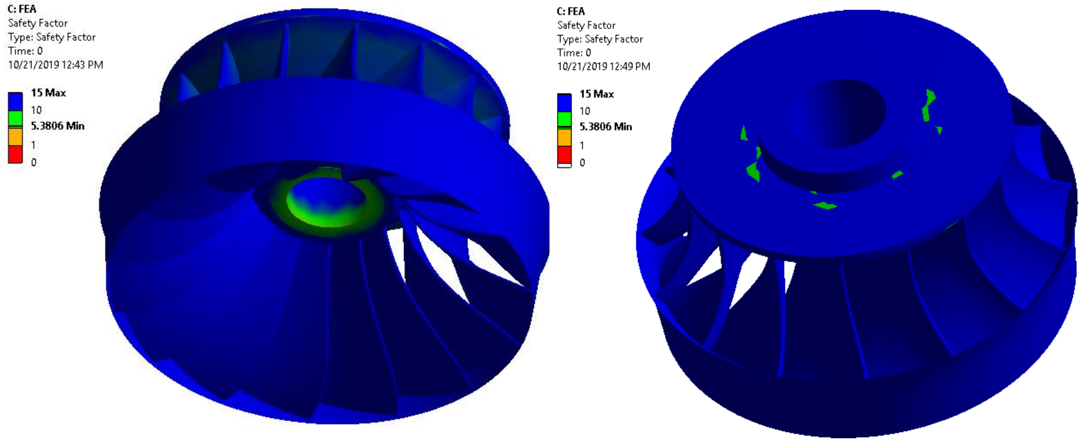


Figure 23. Fatigue Life

Fatigue life plot indicates that if the loading is of constant amplitude type, then the result from life represents the number of cycles till which the structure can withstand until it will fail to fatigue (Tanwar et al., 2012). Observation shows that the runner can withstand minimum 1e+6 number of cycles thus indicating infinite life.

4.2.4. 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 (Tanwar et al., 2012).



(i) (ii)

Figure 24. Fatigue Safety Factor

Figure 24 (above) represent the contour plots of Fatigue safety factor. Observation 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.

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, fatigue damage, 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.

CHAPTER FIVE: CONCLUSIONS AND RECOMMENDATIONS

5.1. Conclusion

From the first part of Fluid Structure Integration Study i.e. CFD analysis, pressure distribution was obtained. Coupled scheme was used for pressure-velocity coupling in which, velocity inlet and pressure outlet conditions were chosen as bounding conditions. Pressure was found to vary from -90.91 KPa to 416.9 KPa. Highest pressure was observed at the trailing edge of the runner blades at the joint between the blade and band. Similarly, velocity streamlines and distribution of velocity of fluid inside the runner were also obtained in which, maximum flow velocity observed was 33.29 m/s.

In the second part of one way-FSI i.e. structural analysis was performed through analysis of the Von Mises stress, total deformation and fatigue tool. Maximum stress observed was 160 MPa which is less than the material yield strength and ultimate strength i.e. 250 MPa and 460 MPa respectively). The analysis of the runner's static stresses field indicates that the maximum Von Mises static stress is far less than the material's ultimate stress, so the runner blade shall not undergo cracks propagation. Total deformation was observed to be higher at the suction side of the runner blade. Analysis shows that the damage values are less than 1, so, the structure will not undergo fatigue damage and the runner can withstand minimum 1e+6 number of cycles before breaking down. Additionally, minimum factor of safety observed was 5.3806 which is more than 1 indicating that the runner will not undergo failure before the design life.

Using the FSI analysis result, 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 are made to operate at varying load conditions and because of load variations, turbine blades undergo fatigue failure. Fatigue failure incurs huge financial and energy losses. To avoid such situation, it is essential to give proper attention during design phase.

5.2. Recommendation

FSI analysis was performed with a purpose to estimate the lifetime of Francis turbine runner through stress field analysis under certain limitations due to assumed data, cost and time considerations. In order to get more refined result, data accuracy is expected to maintain in future. In order to calculate the pressure distribution more accurately, it would be helpful to include the entire flow passage for flow simulation. Furthermore, the procedures adopted in the study can be further used to explore more into following issues:

- As a diagnosing tool to study the causes of fatigue failures, cracks or deformation.
- To schedule maintenance time and allocate cost accordingly.
- To find an effective measure to prevent fatigue failures.
- To analyze the result using various materials and find the right one.

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