**CAPITAL UNIVERSITY OF SCIENCE AND TECHNOLOGY, ISLAMABAD**

# Experimental and Structural Evaluation of Cross-Flow Turbine

by

Raheel Ahmad

A thesis submitted in partial fulfillment for the degree of Master of Science

in the

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*This thesis is dedicated to my parents, teachers and mentors for their continuous support, guidance and advice.*

# CERTIFICATE OF APPROVAL

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# *Acknowledgement*

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**(Raheel Ahmad)**

# *Abstract*

Developing nations all across the world are facing challenges in keeping up with the rapid demand for energy consumption in their countries. These countries due to globalization are now able to access technological advancement in all fields of life, these development in fields of manufacturing, agriculture, and other sectors are giving rise to more demand for energy. In this report design and analysis for the development cross-flow turbine is discussed. Design of William cross-flow turbine has been offered as an alternative mean of energy production which contrary to large scale power production projects that utilize large reservoir of water to produce energy, produces hydraulic power using low head and low flow rates of water. The design parameters required for developing a working cross-flow turbine have been discussed in this study along with the materials used for their production. Parameters like nozzle flow velocity, area of the penstock, rotor diameters, etc. were calculated according to the output requirements of 250 kW from the attached generator. The main focus of this study was to locally developed and produce a working prototype of a cross-flow turbine. The design produced is also presented in this study as a guide for future work where design intent of critical components along with the recommended method of their production was done. These designs were also complemented with the development of their CAD models which helped us in visualization and formulation of efficient and robust components and assemblies. To validate our design computation analysis was performed for both structural integrity and flow behavior study of water as it flows through the turbine housing and across the rotor geometry it was observed that our proposed design met the requirements of operating conditions under the given flow rates and RPMs. The forces exerted on the rotor geometry by the flow rate of water during CFD analysis were taken as loads for the statics structural analysis. Secondly modal and Harmonic analyses were performed for our design to validate the safety of our design during operation and a variety of rpm.

Performance of the turbine was evaluated by variation in flow rates and guide vane angles and the parameters were recorded using digital flow meters and tachometers its relationship of flow rates with an output of RPMs was studied keeping in light the results and studies performed future recommendations are also made for further studies in this domain. Stresses in the range of 40 Mpa with deformation of approximately 0.07 mm were produced on the turbine rotor in reaction to the forces applied by incoming flow. The modal analysis for the turbine rotor shows the working frequency of the cross flow turbine as 8.07 Hz which does not overlap with any of the natural frequency of this system. Excitation was observed at the frequency of 425 Hz while performing harmonic analysis of the system. This excitation was negligible when compared to operating and geometric parameters of the turbine under consideration. Experimental results were comparable to proposed design parameters where maximum rpms of 480 were observed at the rotor shaft at the flow rate 0.793 $m^3$ using a head of 25 m .The developed design encompasses ease of manufacturing, and operation while keeping in mind material and technological constrains local manufacturing setups.

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# Symbols

| | |
|---|---|
| $d_s$ | Diameter of shaft |
| $t$ | Blade thickness |
| $\delta$ | Central angle of blade |
| $r_0$ | Radius of pitch circle |
| $\rho$ | Radius of blade curvature |
| $r_2$ | Radius of inner circle |
| $n$ | Number of blades |
| $L$ | Runner length |
| $D_1$ | Runner diameter |
| $\beta$ | Blade angle |
| $\alpha$ | Angle of attack |
| $d_p$ | Diameter of penstock |
| $\eta$ | Expected efficiency |
| $a$ | Radial rim width |
| $g$ | Acceleration from gravity |
| $T$ | Belt tension |
| $p$ | Density |
| $\omega$ | Turbine angular velocity |
| $N_s$ | Specific speed |
| $N$ | Turbine Rpm |
| $t_s$ | Penstock thickness |
| $t_1$ | Blade thickness |
| $Q$ | Flow rate |

# Chapter 1

# Introduction

A power production system to meet the growing electrical energy demands of developing countries, if produced economically by utilizing locally resources can be an effective alternate to most fossil based energy solutions at hand this section of the dissertation discusses the current world scenario of power production and demand. The resources utilized currently in order to meet this ever rising demand of energy are brought under discussion here in order to highlight the need of small hydropower turbines that can utilize the geological sources to minimize the gap between energy production and demand.

Global energy demand has been on an incline as the population and prosperity increased over the past century. All wakes of both economical and domestic environments have experienced an increase in demand for energy from all possible sources. As a result of an increase in these energy provision capacities, a positive impact has been imparted on global living standards. Though these prosperities are welcomed this, in turn, has gradually increased the burden for production and exploration of new and alternative energy sources as the rising consumption in the form of petrochemical and fossil fuels and petrochemicals have vast arrays of adverse on the environmental conditions.

In addition to the fossil fuel sources, other sources of energy have also been developed in the past century which have somewhat eased the pressure from fossil fuel sites and somewhat provided balance to the energy mix of the current world these alternative fuels namely being hydropower, nuclear, wind and solar, etc.

The world's energy ecosystem has evolved at a rapid pace since the industrial revolution and advancement in both industrial and domestic sectors have accelerated tremendously in the last few decades in light of these the future may see a large-scale shift in the energy eco-system in terms of availability of sources and environmental conditions. Global energy demand is expected to follow an upward trend in the futures due to an increase in population and increasing growth in the economy of developing nations [1]. A significant portion of the total population of the world still lacks access to electrical power for their daily activities [2] meanwhile our reliance on fossil fuel sources is not viable due to its adverse effects on our climate and increasing depletion of its supply from available sources.

The world is now thus facing severe energy calamity. Developed countries might able to cope with this calamity due to their predominance over a vast array of resources on the other hand countries still under development like Pakistan which is not able to meet the demand for its current energy needs due to scarcity of currently available sources [3] on other hand countries with better technological development try to consume the available energy sources, in turn, the world is not in the hassle to meet their current energy requirements with a finite number of resources.

As per the above-mentioned argument, researchers are now interested in developing alternate and renewable energy sources as their resources are sustainable and cost-effective on the other hand being environment friendly as well. Enhanced utilization of hydropower as a source of energy is of great potential for developing countries having access to feasible water sources.

Hydropower is a feasible source of power production and its significance in the future cannot be undermined [4] and thus ample efforts are required for efficient utilization of these resources for the economic and environmental prosperity of any country as it is currently the most efficient, reliable and secure way of power generation.

## 1.1 Global Energy Scenario Outlook

Taking a look at the global energy scenarios it becomes clear that the energy demand has risen to ever-increasing heights of the past decade thus forcing the hand for the consumption of the increasing amount of energy sources while mostly relying on petroleum and fossil-based sources. Global energy supply and consumption annually from different sources available is stated below [5,6].

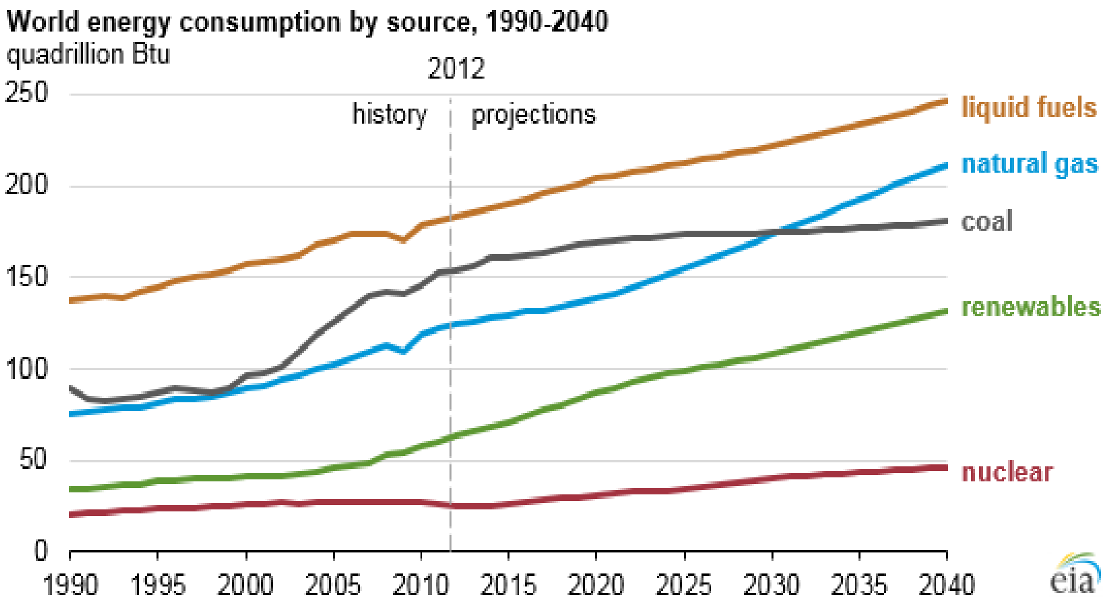


FIGURE 1.1: Global energy consumption by source type [7].

Since the dawn of the industrial age, the appetite for power consumption has increased manifold. Oil is the contributor to supplying energy needs while renewable sources are also being developed at a steady pace as progress is made in research institution all over the world for developing new sources of energy while increasing the efficiency of production of sources that are already underutilization. Shown below is the energy production of the world in million tons of energy.

Owing to increasing cost and ecological constraints alternatives to fossil fuels with clean and cheap commercial power generation capabilities have always been the focus of researchers. Hydropower is a strong contender as an alternative and economical way of power production by the use of water reservoirs providing water at high potential energy to the turbines. These systems if developed economically

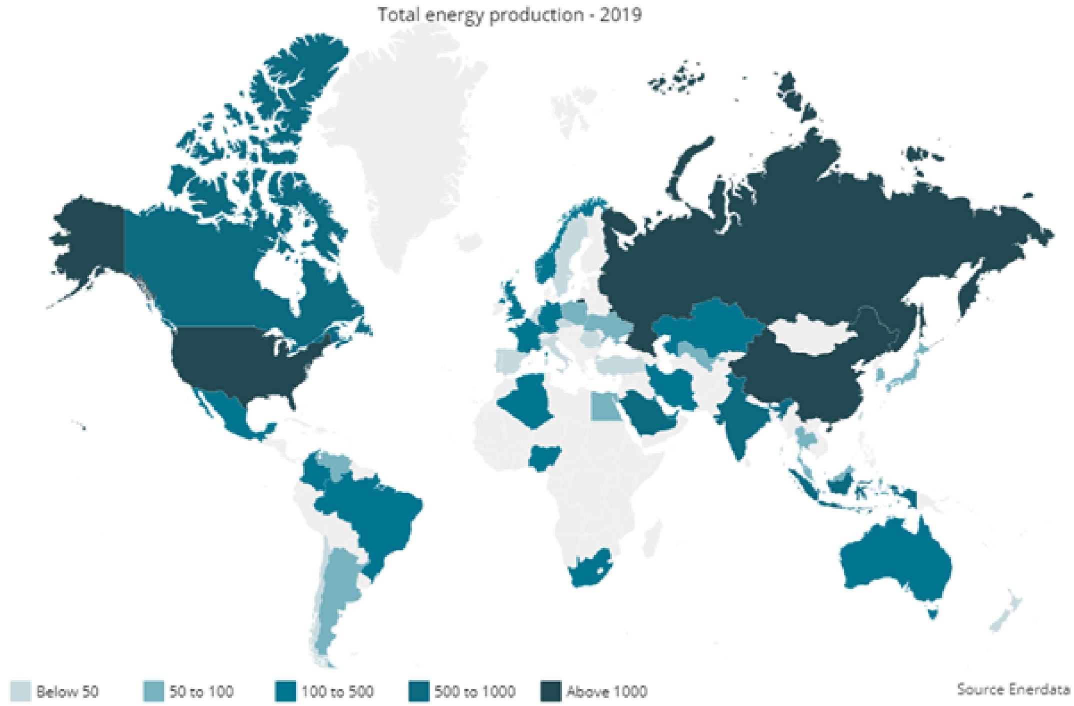


FIGURE 1.2: Total energy production of world in million tons of energy [8].

TABLE 1.1: Global annual energy supply in million tons of energy [7].

| Year | Coal | Natural gas | Nuclear | Hydro | Wind, solar etc | Biofuels | Oil |
|---|---|---|---|---|---|---|---|
| 1990 | 2220587 | 1662187 | 525520 | 184064 | 36571 | 904162 | 3233212 |
| 1995 | 2207669 | 1806624 | 608098 | 212766 | 42391 | 967469 | 3373297 |
| 2000 | 2317134 | 2071233 | 675467 | 224663 | 60262 | 1014659 | 3669477 |
| 2005 | 2990601 | 2360022 | 721706 | 252334 | 70143 | 1088960 | 4010067 |
| 2010 | 3649798 | 2735952 | 718713 | 296474 | 110200 | 1205287 | 4127360 |
| 2015 | 3842742 | 2928795 | 670172 | 334851 | 203821 | 1271235 | 4328233 |

with long working life and ease of maintenance can find a hefty market share in the industry which as of now has few options in these domains.

## 1.2 Small Hydropower in Pakistan

Pakistan due to its geographical location is blessed with bounty full resources in terms of hydropower production, owing to the countries layout including the natural and irrigational flow system the country can harness the potential of hydropower with relative ease. Explicit prospects of small hydropower production are available besides the availability of large hydro sites. Small hydropower is being considered one of the profitable options, these projects are thus far mainly being pursued by provincial governments according to the Alternative Energy Development Board of Pakistan [9] 128 MW of hydropower is being generated currently from these small hydropower resources another 877 MW is currently being developed while 1500 MW of countries natural small hydropower sites are yet to be developed leaving immense openings for development of efficient and economical systems for utilization of these sites.

TABLE 1.2: Pakistan's provincial small hydro power break down [10].

| **Sr. No** | **Area** | **Potential Sites** | **Potential Range (MW)** | **Total Potential (MW)** | **Remarks** |
|---|---|---|---|---|---|
| 1. | KPK | 125 | 0.2 to 32 | 750 | Natural Falls |
| 2. | Punjab | 300 | 0.2 to 40 | 560 | Canals |
| 3. | Gilgit – Baltistan | 200 | 0.1 to 38 | 1300 | Natural Falls |
| 4. | Sindh | 150 | 5 to 40 | 120 | Canal Falls |
| 5. | Azad Jammu & Kashmir | 40 | 0.2 to 40 | 280 | Natural Falls |
| | | **Total** | | **3100** | |

Owing to the capacities of these hydropower systems they can be used for the operational enhancement of lightings, irrigational, educational, and healthcare systems in rural and remote locations of the country these types of the system when used in rural and underdeveloped areas can provide sustainable means for achieving development goals by ensuring the provision of electrical power to the developing economy. To make these systems economical and to provide ease in manufacturing this thesis looks to provide and comprehensive guideline for manufacturing and guideline of such hydropower systems in general and Crossflow hydropower turbine in particular.

This thesis focuses on developing and designing a prototype of a cross-flow turbine which can be easily manufactured while being economically feasible on the same time.. As per requirement, different facilities use different types of hydropower turbines owing to the availability of head and flow rate of water. Based on design some turbines may achieve their optimal operational efficiencies at high heads of waters while others may achieve the same at the low head but the high flow rate of water from the source.

Turbo Machinery used in hydropower can be classified into main categories i.e Reaction turbine and Impulse turbine these classifications are based on the degree of reaction that these types of turbines achieve during their operation.

Different engineering approaches need to be adopted that would enable us to reach a robust and efficient flow system. This would resolve our existing power systems short-comings by using existing hydropower resources, scope of this research aims at providing a framework for ease of development of cross-flow turbines in Pakistan that would further aid in mass production of these systems all over Pakistan we aim to provide technological, design, engineering and manufacturing data that would enable local production using existing skills, capabilities and machinery.

These days development of Cross flow turbines is gaining traction as hydropower is one of the most desirable and environmentally friendly resources of power production throughout the world. We aim to provide technical and practical knowledge developed and used during the design stage and practical production stage that would make it easier for subsequent researchers to develop and improve the existing design and further improve the efficiency of cross flow systems.

This dissertation discusses the overall energy scenario of the world while focusing on the energy scenario and energy mix of South Asia particularly Pakistan. The opportunities of resources available in our geographical localities provide a prime source for commercial development of small hydropower turbines. The development work in rural and remote areas where water resources are available can help in improving the working conditions by using a portable cross flow turbine which can be readily installed as immediate source of electrical power these sources of power can be used for activities which can be used for further development of these areas while introducing the main focus of the thesis engagingly while further focusing on our main topic of research along the way. Currently there are no such option available locally in Pakistan which provide a commercial solution for manufacturing and installation of these types of ready to go hydraulically driven power plants thus missing out on a vast majority of remote and rural areas living near prime water resources which can be easily utilized as a source for operation of these types of turbines. Work can be done to improve the design of these systems and their corresponding assemblies to improve their efficiency and reduce production cost and to make them attractive to government and private sector. The lack of any investment towards development of these types of equipment in Pakistan. A vast gap in the market has thus been left void and needs to be filled with a viable solution that is both theoretically tested and designed by keeping ease of manufacturing and portability in mind this dissertation aims to bridge that gap by making significant contribution that can be used to as a bench mark for further development an standardizing the industry working in this domain.

## 1.3 Thesis Outline

The organization of the thesis is discussed as follows:

**Chapter 2** discusses the existing research on our topic while developing a coherent structure and argument that leads to a clear basis or justification for our approach in developing the design of this cross flow turbine. We further describe how much research has been done into our topic and synthesize key findings from relevant studies and highlight any gaps or limitations in existing research.

**Chapter 3** describes the discusses the methodology used to conduct the overall research from discussing the basis used for calculation of design parameters and techniques that will be employed for simulation the developed structure using structural and computational analysis using simulation tools like Ansys and Solid works.

**Chapter 4** discusses the key design of major components that comprise our developed cross-flow turbine including its support structure, the housing, rotors guide vanes, and the flow directing nozzles and further discusses and provide a guideline of resources used and practical approach adopted for the manufacturing of cross-flow turbine.

**Chapter 5** provides all the detailed results and simulations carried out to support our design these simulations will be based on real working parameters of the available water head and flow rate and our design parameters and will provide us with theoretical rpm of rotors and response of the rotors and housing as we further link this flow rates and RPMs to produce our static structural analysis and vibrational analysis to check the robustness of our system.

**Chapter 6** discusses the experimental setup developed for the cross-flow turbine and compares this experimental and simulated parameter to further strengthen our research and design approaches.

**Chapter 7** tabulated the data obtained from experimental and simulation analysis along with comparison and relationships between variation of input parameters and their collective effect on outcomes parameters and in general overall performance on the cross-flow turbine.

**Chapter 8** discusses the outputs achieved by the prototype developed in terms of data and manufacturing ease and further recommendations are made that would become the baselines for future development work of the provided design and manufacturing practices adopted.

# Chapter 2

# Literature Review

The literature review of our thesis consists of a detailed analysis of past work done in the field of development of Hydropower turbines mainly Cross flow turbine which is mainly a low head turbine. This review of literature will include an in-depth study of a variety of turbines that can be employed in the same environment as a cross-flow turbine and will thus further analyze different types of studies done on cross-flow turbines. Cross flow turbine studies can be divided into different categories which may rely on experimental studies performed or modification concepts applied on an already conceptualized model by varying any existing parameters or components, further studies involve verification of concepts by use of computational analysis using commercial software such as Ansys and Fluent, etc. Turbines can be termed as low head turbines if they employ a water head of 15 meters or less to generate an output of power of less than 5 kW [11]. Conversion of rushing stream of water into energy used to drive generator and other mechanical systems that then convert this energy into electric energy in terms of hydropower this is known as a generating unit and can generate using several different concepts and approaches. Many different types of low head turbines and utilized to achieve this target.

The most commonly used variation of conversion of low head water energy into electrical energy is by employing the use a system known as Francis turbine where the water strikes the edge of the runner's blade which in turn rotates it about its axis the water then runs down along the blade towards the axis of the turbine and exits the system from the bottom of the turbine. This system of where water spirals down a channel and is guided towards the leading edge of the rotor through a guide vane produces up to 60% of total hydropower generated in the world [12].

## 2.1 Basic Classifications of Turbines

Hydropower energy is one of the key contributors to renewable streams of energy. Pakistan is naturally rich in geographical resources and water is one of them the main source of these streams and freshwater rivers which can be utilized to produce electricity [13], still due to having huge resources. Pakistan is still is utilization a small number of hydropower resources which not even enough to meet its existing demand on of the main factors being that Pakistan is still an economically developing country. Countries with limited economic resources available to invest in large water reservoir projects capable of producing enough power to fulfill the country's needs [14].

To promote the production of energy efforts are being done to motivate private and government sector to invest in new and efficient ways of producing energy. Through the deployment of small water-based energy production systems mainly water turbines despite being many alternative technologies the main factors the control the feasibility. These resources can be defined by the availability of resources, capital cost, environmental impact, efficiency, and reliability [15]. These hydropower systems might have a high capital investment and moderate environmental impact these systems have almost negligible running costs associated with them in the light of above small hydropower systems are a definitive way low cost, reliable and efficient way of harnessing the energy and supplying of electrical power to rural and remote areas in Pakistan.

Impulse turbine an evolution of a simple stream wheel, which simply the natural flow of water to operate the rotor and provide power. Contrary to the stream wheel that relies on the natural flow of water an impulse turbine relies on a powerful jet of water that is generated from the high altitude of water [16]. Degree of reaction is the difference in the drop of the amount of pressure between the nozzle and the rotor and is termed as the ratio between static pressure drop in the rotor to the static pressure drop in the stator or nozzle plus the rotor [17]. Impulse turbines operate on the action of the jet of high velocity exerting force on the rotor blades as all of its available potentials are converted into kinetic energy by the nozzle before flow passing through the rotor blades. The momentum of the jet of water is extracted by the blades because of the dynamic pressure difference created by the two surfaces of the blades while keeping the static pressure differential across the surface atmospheric hence providing the impulse turbine with zero reaction . Pelton turbine is an example of an impulse type of turbines that has a set of spoon-shaped buckets constructed at the periphery of a wheel the buckets are formed in such a way that they cause the change in direction of the incoming water and making it exit from the opposite side while transferring its energy to the wheel in the process.

Reaction turbines, on the other hand, have a certain degree of reaction as the work on the principle of reaction forces developed across the surface of the rotor blade. The pressure drops across the impeller blades and static guide vanes in the primary source for extraction of angular momentum. Pressure drop in a reaction based turbine occurs in both rotor and nozzle [18] contrary to impulse-based design fluid that is being discharged from a nozzle that is directly in contact with the impeller, velocity of the discharge medium when exiting the nozzle creates a reactionary force that moves the impeller in the opposite direction from that of the discharged fluid, suction is created through the draft tube in the casing. Reaction turbines often have a spiral type inlet in the casing that contains control barriers to control the flow of water.

In the flow through the nozzle, the potential energy of the water is subsequently extracted by the rotor of the reaction turbine as the water moves through it the relative ratio of the sum of energy that is converted through the rotor blades to the total sum of energy used is the degree of reaction in a reaction turbine, in reality, all turbines run with some degree of reaction but turbines where this fifty percent of the total potential energy available is converted to kinetic energy in the rotor are known as reaction turbines.

One of these types of water production systems is the Cross-Flow turbine which is a type of impulse turbine. This research was driven out of the need to improve and optimize small hydropower systems by designing, manufacturing, and analyzing the design of and development of a cross-flow turbine system using local sources and testing it experimentally through a provided water sources to assure its operational and constructional integrity.

Adjustable guide vanes and blades for efficient use of water resource for the production of electric power are used in a system known as Kaplan turbine and thus is a preferred way to achieve maximum possible efficiency at varying flow rates, being a type of reaction type turbine, it is suitable for low-pressure hydropower facilities and can be utilized with low discharges. Kaplan turbine for its function needs to be completely submerged in water. Radially mounted guide vanes around the turbine receive water that enters through the runner blade. In this system, if the flow rate of water is stable stationary blades are sufficient these guide vanes can also be closed and opened depending upon operation and maintenance needs to protect the runner blades from any damage. Due to this varying configuration of blades and runner change in water flow rate has no influence upon efficiency and power output of the turbine system [19].

A turbine system with adjustable guide vane design but employs the use of stationary runner blades is termed a propeller type of turbine. Being an inward flow reaction turbine similar to a Kaplan turbine mentioned design is similar to the design of the propeller of a ship and is used mainly for large systems at minimal heads thus ensuing large diameter and lower rotational speeds [20]. The geometry of a propeller blade thus allows for the production of airfoil blades by using a forging process rather than casting complex airfoil blades this, in turn, makes this type of turbine considerably low cost in terms of manufacturing and cost. Usually, these types of turbines and four to six bales in which water runs through the runner axially about the shaft. These fixed blade units are less costly than variably configuration blade turbines which in turn reduces the operating ranges. Efficiency losses drop to 2 to 5% at higher heads and fall as high as 15% at lower heads. Conventionally these turbines are mounted using a vertical shaft these vertical shafts are assembled with wicket gate guide vanes that allow the regulation of speed and load and auto shut down of the system [21].

An impulse type of turbine that runs by a jet of water hitting the buckets of the rotor at atmospheric pressure is known as Pelton turbine [22]. A wheel consisting of a series of buckets halves is the main component of this turbine arrangement water directed by the jet hits each bucket independently [23]. The Pelton housing filled with air, such variables created conditions that cause free surface flow which causes splashing and unsteady feeding with centrifugal effects so this causes major challenges in modeling the flow of water in computational modeling. Pelton wheel has no requirements of draft tubes as the runner is located at the top of maximum tail water to permit operation at atmospheric pressure. The flow of water after leaving the buckets of turbines are thus at atmospheric pressure and high velocity [24].

The Cross-flow turbine being the main focus of this study also known as Michell-Banki cross flow turbine in appreciation for original pioneers' hydraulic turbo machinery that can be termed as an impulse-driven machine [25]. It consists of a spout of rectangular cross-sections and a rotor that resembles geometry like a drum that has circular blades aligned axially. The flow of water is converted from kinetic energy to potential energy by the use of a nozzle that is only on a portion of the perimeter of the rotor with its main function being that it directs the direction of water towards the axis of the rotor. The geometrical parameters of a low head cross flow design different from traditional approaches towards turbine design mainly due to the diameter of the runner being larger [26].The geometry of a cross-flow turbine where the water moves across the blades of the turbine twice. Once the initial flow of water is directed towards it through the use of nozzle and secondly due to the geometry of the blades which curve and the flow of water and towards the opposite blade as it flows outside. The mechanical energy produced in the system is due to these two passes of water from the system and kinetic energy in water is absorbed by the rotor in two steps[27]. Although the water passes twice through the rotor to produce this energy but the maximum attainable efficiency of system's using cross-flow arrangements is less than some other conventional turbines but still, it has found vast applications in some situations [28].

Cross flow water turbines make itself feasible since it demands a low production cost due to its simple yet robust design as referenced through the literature and fabrication data analyzed by the researcher's in the past [29]. One of these circumstances is an immediate aftereffect of the reasoned that this turbine is effortlessly fabricated in any facility with welding and metal shaping equipment[30] .Since its edges and spout are manufactured using components with minimum supports, which don't need costly and complex facilities like smelting equipment to be produced. This assembling effortlessness converts into a price for these systems which is lesser as competed to the traditional hydraulic rotor systems. These machines present a profoundly serious option in contrast to the more commercial kinds of turbines in including an exceptionally low force, these types of designs are of key significance where efficiency of the systems is of secondary importance to the cost of the overall design. This type of simplicity in the design ensures this cost-effectiveness for a variety of operating conditions and a large range of flow rates in different situations [31].

A variety of design based on experimental and theoretical approaches have been studied and developed in the past Myint Thein et.al [32] present an approach towards construction of a novel cross-flow turbine which can be easily manufactured and installed in remote areas of underdeveloped vicinities using materials that can be easily arranged and fabricated thus making these design incredibly robust and adding to the advantages of cross-flow turbines selection as it can be easily transported and installed without major construction of allocation of resources and has relatively easy maintenance based on its modular designs which can be easily optimized based on-site parameters that is the flow rate of water and height of water where it flows from and these parameters in tandem with various modification in design and parameters can further improve the efficiency of the system as shown by Rantererung et al. [33].

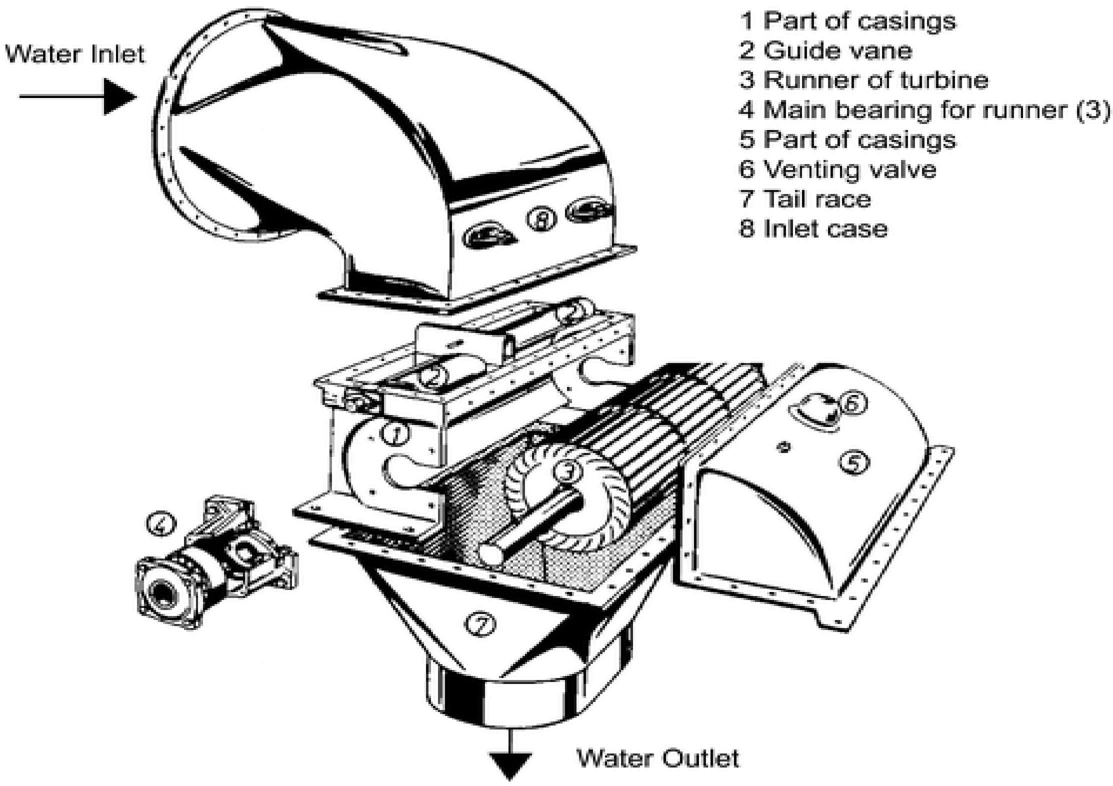


FIGURE 2.1: Generic view and cross-sectional view of cross-flow turbine assembly [34].

One of the major advantages of any impulse turbine is its efficiency at part load conditions thus making cross-flow turbine favorable for performance in different conditions where the load on the rotor is not at its peak. CFWT is particularly at a great advantage when used in locations like a run of rivers or streams because of its performance curve which in comparison with another type of turbines varies very little with a lot of change in its flow rate these characteristics makes it an impeccable choice where the flow rate of water varies due to a variety of reasons seasonal or otherwise these changes can easily be catered without a major change in energy or output produced by the turbine.

Recent studies performed on the performance characteristics of cross-flow turbines make use of its versatility as simple design and take it as a reference frame for further modifications by changing its components by varying their geometrical or material properties their changes are then further elaborated by different approaches including experimental and theoretical approaches which will now further be discussed in this literature view.

## 2.2 Experimental Studies of Cross Flow Turbine

As the investigation was made through previous experimental studies carried on the operation of the cross-flow turbine one of the major contributions were made by Venkappayya and Nadim [35], where an abstract investigation was carried out where the effects of various geometric parameters were studied on the overall efficiency and power output of the turbine. The approach involved changing parameters like the number of blades attached to the rotor, changing the angle of attack of water to the rotor blades, and finally, variations in the inner to out diametric ratios. The clear indication from the experimental campaign was the indication that the efficiency of the system increased with the rise of the number of blades, while the increment in the angle of attack beyond 24 degree does not contribute to the increase in efficiency, furthermore, it was concluded that the combination using the angle of attack as 24 degrees and the diametric ratio of 0.68 yielded maximum efficiency when the runners kept in the configuration were 27 in number. Further investigation concluded that while the increase in the attack angle resulted in a decrease of efficiency as the inner to outer diametric ratio was varied between 0.6 and 0.75.

Djoko Sutikno and Rudy Soenoko [36] at Brawijaya University performed an experiment where they studied the characteristics of cross-flow turbines using three models of turbine nozzle roof curvature. The roof curvature was center around the axis of the shaft of the rotor which was also designed for the same flow rate, diameter on the runner, and speed of rotation of the rotor was kept constant while keeping the parameters like runner's width and entry arc of the nozzle as variables. The nozzle and runner were mapped out as a function of the entry arc of the nozzle i.e., minimal the runner and nozzle combination the large the arc of entry of the nozzle and vice versa. The nozzle entry arcs designed for this particular experiment were designed at 75°, 90°, and 120° respectively while keeping the area of the nozzle entry the same. The potential of water collision on the rotor blades prompts high efficiencies of cross-flow turbines, these experiments made clear that the optimal efficiency regions expanded as the length of entry arc of nozzle decreased consequently increased nozzle entry arc decreased the best efficiency points while confirming that optimal efficiency of cross-flow system was achieved when the entry arc of nozzle was at 75° and 90° degrees respectively and the use of nozzles having the roof curvature center around the axis of the shaft of the rotor was recommended to improve turbine efficiency and the use of nozzle with an arc entry angle of 120° is preferable where there is a fluctuation in load and hydro potential system.

Aidan Hunt and Carl Stringer [37] conducted an experimental study where a cross flow turbine was examined to determine the effects of the aspect ratio of the rotor diameter and the span of the blade. This effect of this aspect ratio was studied while keeping all other geometric parameters constant in a cross-flow turbine where the blades on the rotor were straight-edged. The study demonstrated the significance of studying independently the non-dimensional parameters by isolating them from the geometric parameters under consideration. The study concluded that the operational effectiveness of the turbine relay's on combination of Reynolds and Froude number, the coefficient of effective operation was found to be unalterable, for the values of aspect ratios examined because of minimal interaction of the blade support. The study noticed that when Froude number was allowed to change there was an increase in the coefficient of performance which could accurately be associated with aspect ratio thus an experimental design was proposed by them that fully isolated the non-dimensional variables from the geometric parameters involved in the evaluation thus proving that efficiency of the turbine is insensitive to the aspect ratio of the rotor diameter and span of the blade if all other parameters are kept constant.

Chiyembekezo S. Kaunda et al. [38] focused on studying the behavior of a simplified cross-flow turbine at conditions other than the maximum efficiency position while studying the reaction behavior and torque transfer within the second stage of the cross-flow turbine. The observation was made that the turbine performed well at a degree of reaction thus proving that cross-flow turbine is not fully an impulse turbine this degree of the reaction was achieved at a full open position of the valve which was controlled by a lead screw assembly the study showed that the efficiency decreased with the increase in height of head beyond the maximum efficiency head while showing that the turbine efficiency is not sensitive to the variation of the flow as at different tested head showed that at even 25% of the flow rate the turbine efficiency was upward of 50% except at the head of 3m. The role of the second stage was observed through torque characterization more so when the valve for water flow was fully opened thus indicating the need for a system that would optimally control the angle at which the water flows from the first stage to the second stage so that the torque produced could be utilized efficiently. The simplified turbine design had an efficiency of 79% at a head of 5m and a flow rate of 13.5 m/s making it a low-speed turbine.

Jusuf Harissa and Yudy Surya Irawan [39] designed an experiment to study the quality of rotation in a cross-flow turbine concerning the behavior of turbine flow. Observation made suggested that water after flowing through the first stage of the turbine blades while flowing towards the second stage of turbines became chaotic in its flow because after flowing through the first section of the blades the water flow becomes turbulent and thus while flowing to the second may cause hindrance to the optimal rotation of the rotor there to attempt to rectify this behavior was by installing a guide passage after the first stage with the use of some guide vanes that would make the water flow from the first stage to the second stage at a precise angle thus regulating its flowing behavior and the rotation of the turbine would be more stable which is a key parameter for electric generation quality. It was also observed that by installing a guide plate efficiency of the turbine would increase as much as 11.118% and consequently increasing the power of the turbine by 10% overall.

## 2.3 Computational Study of Cross-Flow Turbine

Building upon previous work done based on experimental analysis where physical modifications and variations in geometric and non-dimensional parameters further investigation and analysis were done by researchers where advanced computational fluid dynamics techniques were used to further understand the flow dynamics and operational characteristics of cross-flow turbines some the research done using these computational techniques were reviewed and is discussed below to further our understanding of these systems.

In their studies in 2013, Vincenzo Sammartano and Costanza Aricò [40] conducted computational studies after reviewing the available criteria for available cross-flow turbine designs and presented a two-step procedure for optimization and selection of key design parameters in the first step where they selected inner and outer blade diameter, the initial and final blade angles and the shape of the nozzle are determined based on the fact that system was greatly simplified and based on these simplification assumptions some key theoretical parameters were estimated and the efficiency of remaining design parameters were analyzed employing computational fluid dynamic testing using these. There 2D computational approach their studies showed that linear nozzles provide an approximately constant angle of attack of water and thus prove their simplification hypothesis that that the amount of energy losses were minimal on impellers inner side. Further results showed that the blade number and diametric ratio has no effect on the characteristics curve but does lead to reducing the efficiency of the turbine.

Work done on the computational study of cross-flow turbine by Sajjan Pokhrel [41] was primarily focused on presenting the plots of shaft rotational speed for twelve and nine-blade water cross-flow turbine versus turbine using primary computational results. These results were dependent on specific input operating conditions selected for the analysis where they demonstrated the effectiveness of the computational tools developed. They used rotational speeds of the rotors at a constant flow rate as the means of simulating load conditions on the turbine shaft in computational simulations, higher rotational speeds represent low load conditions while lower rotor rotational speed represents increased torque load on the turbine shaft. Results presented in this study are in the forms of volume fractions, density, velocity plots, and pressure. These plots of velocity and pressure vectors obtained using the computational tools would be extremely unlikely to be obtained through experimental approaches thus provide better insight in the design of turbines by firstly performing grid and time-step independence studies were performed and secondly experimental case is performed and results are later compared with the computational studies in terms of comparison between power and rotational speeds.

Patel et al. [42] the fundamental information on the Computational Fluid Dynamics mathematical technique for the flow investigation. From the Grid Independent investigation, it is seen that further after 84% the variation in the efficiency is under 1% on expanding the number of grid elements further from 95,40,000. The k-$\epsilon$ model utilized (here with describes optimum model to approach exact outcomes. Here a steady-state CFD evaluation of a 3-D model consisting of forwarding bent 18 edges cross-flow turbine is detailed. The shape and plot of velocity and pressure variation in the flow was discussed.

The working qualities of the turbine are likewise fig used from familiar mathematical outcomes. Thus, through examination, it's seemed to have high-pressure territory on nozzle's inner side and nearer to the primary stage of the rotor edges. At that point, it will diminish upon approaching subsequent stage and turbine's outlet. It tends to be negative in the region where it lacks any cross flow. The flow velocity on the inlet of the guiding nozzle appears consistent and will increment on the nozzle's inner side closer to the primary stage rotor cutting edges and afterward proceed to diminish at the secondary stage of the turbine. Hydraulic efficiency acquired with computational examination and hypothetical methodologies were established as 84% and 88% individually; thus, shows generally excellent understanding among the two methodologies where the efficiency and net head increase in tandem along increase flow of water and the further the efficiency decrease beyond the optimum flow rate of 0.315 $m^3/s$ which is the ideal point with the selected parameters.

Further, there are several other pieces of literature that in detail describe the operational and geometrical characteristics of several different modified and varieties of cross-flow turbines this work performed in this cross flow turbine this dissertation takes account of the pool of knowledge provided by all these literature and further builds on this knowledge to practically manufacture a working prototype of a 250 kW cross-flow turbine while giving the manufacturing procedures and guidelines to link all these theoretical concepts for the actual production of these cross flow turbines which could be then produced commercially and installed locally.

# Chapter 3

# Research Methodology

This chapter discusses the methodology applied for the development of the under discussion cross flow turbines while further elaboration the aims and objectives developed for our thesis in light of the current scenarios and gap heighted in the earlier introductory section from selection of manufacturing methods to computational and experimental parameters are discussed in detail as basis for the design and manufacturing portions in the coming chapters.

Cross-flow turbine discussed here in the dissertation was influenced by the work done by Yi et al. [43] and Kpordze Warnick [44] and was the base for the design development of a cross flow turbine of 250 kW capacity. The basic aim of this research is to develop a prototype of a cross-flow turbine using local fabrication and technical capabilities that would provide a feasible solution to our power production aspirations. Pakistan currently lacks in the sector of locally developed power production solutions that can be utilized on low head and low flow rate water resources of the countries while keeping in mind that the system introduces is robust and economical. Due to power out of these small power production alternatives being marginal as compared to already existing large hydro power plants focus applied for development for them has been minimal.

The lack of industrial interest on the hydraulic system has thus left them without any formal standardizing procedures for manufacturing and design. Significant room is available in this domain to research and develop various components and configuration of these types of plants to optimize them for applications where their contribution to the energy circuit can become prominent. This lack of any formal production capability provides us with intent to introduce in the industry a system for that can produce electrical power using low head and low flow rates readily available at multiple locations in the country.

## 3.1 Research Justification

Recent electricity demand in the country has made the utilization of renewable energy resource such as small hydropower plants necessary. The selection hydropower system vary depending on design and ecological constraints it is of key importance that appropriate selection is made in regards to the type of turbine that needs to be installed. Key importance is given to development of local development of suitable hydropower system. The focus on development of small hydropower project was made due to fact that they are easily transported and deployed at suitable waterfalls and river-off-rivers. This research focuses on providing a numerical analysis of strength of a cross flow turbine rotor under different operating conditions the results produced will be discussed in the light of combination of materials used while comparing the strength and cost estimations while keeping in view the local availability and manufacturing capacity. This type of analysis for where comparison of materials based on local availably and manufacturing capacity will provide a fresh insight towards indigenous development of small hydro power turbines this research proposes the design of a cross flow turbine by justifying the selection of material used using computational fluid dynamics (CFD) approach. Parametric analysis will be performed to for scrutiny of material performance under different working rotational velocities and natural frequencies and stresses produced. This type of system is particular suitable for Lilowanai and Kassbela regions where suitable head of water ranging from 10 to 60 meters are available and with little modification optimum results for power production can be achieved [45].

## 3.2 Specific Objectives

In order to achieve the main aim of our study the key objectives of our research are briefly discussed here with:

i. **Prototype manufacturing of cross-flow turbine.**
   Standardized production procedure and facilities have not yet been developed for manufacturing of cross flow turbines. The first specific object aim to manufacture a working turbine prototype which is manufactured using standardized parts and best industrial practices. These procedures and manufacturing techniques will this be recorded and used as ground work for development activities.

ii. **Design calculations of key parameters.**
   In order to manufacture the proposed cross flow turbine detailed literature review is consulted in order to develop a detailed list of design parameters and their corresponding calculations. Using these calculation parameters for manufacturing of cross flow turbine will be incorporated in the production design and manufacturing stages.

iii. **Simulation of Flow of water through our designed system.**
   The flow interaction of the water coming through the intake nozzle and flowing through the rotor blades needs to be studied in order to visualize the flow characteristics of the fluid through our designed system where velocity field was observed. These phenomena's would be otherwise nearly impossible to observe and predict experimentally.

iv. **Structural analysis of turbine runner.**
   The flow of water hitting the rotor turbine while producing rotational motion will produce stresses on the rotor and rotor shaft. These stresses are to studied using structural analysis where in preset loading condition will be linked to the CFD simulations.

v. **Modal and harmonic analysis.**

To evaluate the vibrational characteristics of the turbine during operation modal and harmonic analysis will be designed to predict safe working frequencies and predict the structural integrity of the design.

In order to develop a working prototype for a cross flow turbine a detailed design methodology was developed that included every aspect of turbine development from conceptualization, design development, simulation and experimentation. In order to justify the design development stage the literature review was consulted to determine key parameters for selection of turbine in terms of flow rate and head of water. Further activities and objectives were based on these primary parameters. As these low head and low flow rate conditions available at our testing site provided us with firm basis for selection of a cross flow turbine for our design development.

Initial stages of our activities were based on deciding the key components of our design from previous and design developed in the same domain. The components which form the basis of for a variety of cross flow turbine design are mainly the turbine housing, guide vane, the rotor and the inlet flow directing nozzle. Assembly based on these four components should be such that the rotor is situated at the heart of the turbine assembly. The flow coming from the guide vane needs to be directed efficiently towards the center of the rotor. For this purpose a flow directing nozzle that provides optimum angle of attack of water towards the rotor should be developed and incorporated within the turbine housing.

The assembly of the turbine housing with the rotor should be kept within close tolerances in order to reduce turbulence and loss of efficiency as the larger clearance will allow water to escape without providing effective contact with the rotor geometry. Positive effects of optimally designed have been discussed in detail over the course of evolution in design of cross flow turbine as the effect of pressure inside the turbine housing has strong influence on operation of these turbines [45]. In order to regulate the internal pressure of the turbine housing air vents have to be merged with the housing design.

After determination of key characteristics of the prototype development further design parameter were developed using relations described in detail in the design section of this dissertation. Turbine parameters such as determination of specific speed and turbine speed were developed based on available site data of head and flow rate. Using recommendation made in literatures reviewed the recommended angle of attack of 16 degree was adopted. The shaft output power was calculated using Eq. (4.2) which use pre define parameters of density and turbine efficiency.

Based on the shaft output power predictions selection for the number of poles of generator was determined using Eq. (4.7). Runner diameter being one of the main parameters for turbine performance was calculated using Eq. (4.8) that relies upon values of coefficient rotational and water velocities while also relaying on head of water and turbine rpm calculated earlier. With available data of runner diameter auxiliary parameters which depend on this primary variable can be calculated such as thickness of water jet that impacts the rotor was calculated based on Eq. (4.9). Selection of length of blade was calculated using the thickness of water jet which is explained using Eq. (4.10). In order to calculate the central angle of the blade $\delta$ the values of outer and inner radius of runner were formulated using the diameter of runner already calculated. With available data runner diameter and runner blade length now the next variable needed to define our rotor geometer is the number of runner blades and the circular pitch between them these were calculated using Eq. (4.15). With calculation of these primary variables torque produced on the rotor shaft and diameter of the shaft were calculated with its length kept accordingly with conceptualized design.

To develop a design of cross flow turbine to utilize low head and low flow cross flow turbine using locally available resources market surveys were performed. Information regarding indigenously available manufacturing facilities and their manufacturing process were analyzed. It was observed that as compared to fabrication facilities there was a lack of casting facilities which made this production method casting. These casting facilities where present lacked in technical skill needed to produce complex patterns needed to cast various components assemblies that would constitute our design. The design developed would need to avoid excessive reliance on casting techniques to keep the project feasible. In order to reduce the cost of manufacturing and to reduce complexity of in production the design would relay mostly on fabrication using welding procedures. The selection of materials used for fabrication were based on past work done and references gathered from literature review. Each component was analyzed for optimum selection of material for its production. Material chosen for the construction of this turbine can be arranged easily from the local markets and can also be easily scavenged from scraps available at any workshop or yard. In order to further justify our material selection the forces produced at the major components would be simulated using commercial computational software. The design developed was according to the available site parameters where the available head from the overhead water tank was of 25 m with the available flow rate from the cross section of an 8 inch pipe being 0.793 $m^3$.

Keeping in view the manufacturing capabilities and material selections and site parameters the design parameters where selected based on work done by Mockmore and Merryfield [40]. After selection of shaft diameter and runner outer and inner diameter the turbine housing was designed which incorporated the design guide nozzle angles. A guide vane was installed at the end of inlet penstock controlled with the help of screw guide system. This guide vane was incorporated using a bearing assembly with rubber seals incorporated for stopping water leakages. The main component of the turbine is its runner which is the heart of the design the dimensional parameters of the runner are based on the available head and flow rate of water.

After the initial draft of drawings the design parameters were used to develop a 3-D cad design using Solid Work and Creo Parametric software packages. Developed 3D models were used for computational, structural and vibrational analysis of our developed system the flow through the system was fully developed before entering the rotor geometry. Further experimental parameters will be calculated during turbine operation where the flow of water and guide vane attack angle will be changed. By changing flow rate and guide vane and angles at certain increments the respective rotor shaft rpm are to be tabulated for plots and graphical representation and operation curves. For every flow rate variation in guide vane angle were made to see the effect of guide vane angle in combination with changing flow rates.

The design developed for this purpose consists of a cylindrical drum which is mounted to a horizontal shaft going through it, this cylinder drum or runner consists of several blades in our case this being calculated to be 30 which are assembled circumferentially on the rotor drum. The blades installed on this rotor drum has sharp edges to facilitate the flow of water across its surface, these blades are joined with the rotor discs by the welding process, these rotor discs which can vary in number following the length of the rotor designed thus making a cage-like structure with a rotor shaft in the center with a spiral cage of rotor discs and blade surrounding it. The flow of water directed towards the turbine first passes from the penstock which sweeps in geometry from a cross-section of circle to a cross-section of rectangular area as its transports water from the inlet pipes towards the turbine. The change in shape from circular to rectangular cross-section ensures that the flow of water is equally divided along the width of the runners. The flow of water needs to be controlled for an optimized flow rate and the angle at which the water is received by the rotor housing, for this purpose a guide vane was designed and installed after the penstock region which controls the amount and angle of water at which it is received at nozzle. The position of this guide vane is controlled by a screw-based flow regulator which can be used to adjust the position of this guide vane.

After the flow of water from the guide vane, the water is received by the flow directing nozzle that directs the flow of water towards the rotor the angle at which the water is diverted by the nozzle. In a cross-flow turbine, the nozzle has critical importance as the angle at which the water enters the rotor has an immense effect on its efficiency the nozzle design was referenced from the work done for efficient nozzle by Adhikari, R.C, and Wood, D.H. [46] their study proposed a nozzle design methodology to design high-efficiency cross flow turbines. An analytical model was formulated for the conversion of the height of water to kinetic energy on the inlet and obtain an optimum flow angle.

Water after being guided from the nozzle designed in the turbine housing is directed to rotor blades at the calculated angle of 16° the kinetic energy of water is converted from water to the blades which start rotating to produce certain rpm's after the water is passed from the blades it still possesses certain potential energy, this potential energy available in the water is again imparted to the blades of the rotor for the second time before it exits through the draft tube. The selection of the generator shall be made in accordance with the rpm at the turbine shafts output so that the output rpms are in correspondence with synchronous rpms of the stator.

# Chapter 4

# Design and Manufacturing

This chapter will discuss the developed design based on all the design parameters and developed in our research methodology, and will give a detailed insight regarding the steps involved in the manufacturing of a prototype. Design of key components based on our formulations will be shared and discussions will be made on their important aspects and manufacturing process used for their production will be discussed. Detailed design of the cross-flow turbine with formulation chosen is discussed below for optimum design and manufacturing of a cross-flow turbine for power production of 250 kW cross-flow turbine. The main components of a cross-flow turbine are also discussed herewith.

## 4.1 Design Parameters of Cross Flow Turbine

The steps involved in the design of a cross-flow turbine followed in the dissertation which is referenced from the literature review are discussed in detail here.

### 4.1.1 Head of Water

In the case of hydropower, it is of key importance to attain the maximum head of water as possible, as the more the head of water the more the output power is attainable, where the head of water is the difference between the height at which the water enters the system and where the water exits the system it is measured in units of meters [47].

### 4.1.2 The Flow Rate of Water

Dynamic Flow rate is known as the volume of liquid under consideration per unit time that is flowing past a certain point across an area in our case it is the amount of water flowing through the inlet section of the turbine per unit time. The formula for a flow rate of water is discussed under as:

$$Q = V \times A \quad (m.s^{-1}) \tag{4.1}$$

### 4.1.3 Calculation of Turbine Output Power

A turbine is a machine that converts the kinetic energy of water available to rotational energy by utilizing the momentum change available in water by deflecting it through a nozzle to its blades of the turbine mounted on the central shaft. The output power of the turbine is equal to the torque provided to the shaft times the rpm produced. The formula for turbine power output is as under.

$$P = \rho \times g \times Q \times H \times \eta \tag{4.2}$$

where,

$\rho$ = Mass density of water in kg/m$^3$

$g$ = Acceleration due to gravity in m/s$^2$

$Q$ = Flow rate of water in m$^3$.s$^{-1}$

$H$ = Head of water

$\eta$ = Efficiency of the turbine

### 4.1.4 Specific Speed of Cross-Flow Turbine

Specific speed of a turbine can be defined in terms of standardization that it is termed as the speed in rotations per minutes at which a model of the similar turbine would operate under 1 m head of water to give an output of 1 kW this constant is an indicator for a specific type of turbine for a specific application. In simple terms, it's the ratio of RPMs of a turbine similar to yours that is producing 1 kW of power under 1 m head of water to the rpm of turbine designed under the specific head of water producing a specific amount of water and can be formulated as.

For cross flow turbines the specific speed ranges is [48] as under

$$40 < N_s < 200 \tag{4.3}$$

Applicable Maximum Specific speed [49],

$$N_s(\text{Max}) \leq 650H^{-0.5} \tag{4.4}$$

So applicable Maximum Specific Speed, $N_s(\text{Max}) = 130$.

According to Kpordze and Warnicks for cross flow turbines

$$N_s = \frac{513,55}{H^{0.505}} \tag{4.5}$$

### 4.1.5 Turbine Speed

Using the formulation presented above the speed of the turbine can be calculated by the equation given below

$$N = \frac{N_s \times HH^{5/4}}{\sqrt{P}} \tag{4.6}$$

So the turbine speed comes out to be 484 rpm.

### 4.1.6 Number of Poles for Generator

The number of synchronous poles needed on the generator stator used for the turbine's rotor running at a speed of turbine speed of 484 rpm can be calculated as under [50].

$$P_0 = 120\frac{f}{N} \tag{4.7}$$

So, the number of synchronous poles is 16.

### 4.1.7 Runner Outer Diameter

Runner outer diameter of the runner can be calculated by the empirical formulation proposed in the work done by Mockmore and F. Merryfield [51] according to them to calculate the outer dia of the runner one needs to know the RPMs at which the run will be operating on, water jet velocity at which the runner will be running and the tangential velocity if the runner. Keeping in view the proposed methodology the runner outer diameter is calculated as:

$$D_1 = \frac{K_{u1} \times 60 \times K_c \left(\sqrt{2gH}\right)}{\pi N} \tag{4.8}$$

where,

Coefficient of rotational velocity, $K_{u1} = 0.46$

Coefficient of water velocity, $K_c = 0.98$

### 4.1.8 Calculation of Thickness of Water Jet

The width of the runner depends on the water inlet flow velocity and the thickness of the jet of water hitting the width of the runner, thickness of the jet per the discussed literature varies from 0.1 to 0.2 times the outer diameter of the runner installed in the turbine system. So I order to accurately calculate the thickness of the jet of water the average is taken and the thickness of the jet of water is calculated as under.

$$S_0 = 0.17D_1 \tag{4.9}$$

### 4.1.9 Length of Runner Blades

The length of blades of the runner needs to account for the number of intermittent disks installed along its path for providing support to the runner blades along its length, the number of the intermittent disk depends on the length of the runner blade these literatures discussed above recommends the following equations for calculation of runner's blades length.

$$L = \frac{Q}{S_0 \times L \times K_c \times \sqrt{2gH}} \tag{4.10}$$

Using this equation, the length of the runner comes out to be 10% of the actual length is 867.

## 4.2 Required Parameters of Blade Curvature

### 4.2.1 Outer and Inner Radius of Runner

The inner and outer radii of the runner can be calculated as:

$$R_1 = \frac{D_1}{2} \tag{4.11}$$

$$R_2 = \frac{D_2}{3} \tag{4.12}$$

### 4.2.2 Radial Rim Width

The radial rim width of the turbine can be calculated as:

$$a = R_1 - R_2 \tag{4.13}$$

### 4.2.3 Central Angle of Blade

The central angle of the blade can be calculated as:

$$\delta = 2\tan^{-1}\left[\frac{\dfrac{\cos\beta_1}{\beta_1}}{\sin\beta_1 + \dfrac{R_2}{R_1}}\right] \tag{4.14}$$

### 4.2.4 Number of Blades for Runner

In the literature, the number of blades (n) for cross flow turbine range from (24 $\leq$ n $\leq$ 30). The number of blades for a cross-flow turbine can be calculated as:

$$n = \pi\frac{D_1}{t_1} \tag{4.15}$$

It is recommended that the number of blades in actual should be 30% more than the theoretically calculated number. The number of blades chosen for this design is 30.

### 4.2.5 Calculation of Shaft Diameter

The shaft of the turbine is used to transmit the rotary motion produced by the rotor to the generator attached to the turbine, the generator can be attached to the turbine either directly or by use of a pulley system. The turbine shaft is under a combination of torsion and bending loads during its operations the diameter and torque on the shaft can be calculated using the relationships given below:

The torque applied on the shaft during rotation can be calculated as:

$$T = \frac{P \times 60}{2\pi N} \tag{4.16}$$

Using shear stress of steel $\tau_s = 42$ Mpa Torque on a shaft can be expressed as:

$$T = \frac{\pi d^3 \tau_s}{16} \tag{4.17}$$

So by solving the above equations the diameter of the shaft can be determined and with a factor of safety is kept as 100 mm.

## 4.3 Determination of Penstock Diameter

The penstock diameter involved in this design can be calculated as:

$$d_p = \sqrt{\frac{Q}{0.785V}} \tag{4.18}$$

Using the relation for $V$ as

$$V = 0.2\sqrt{2gH} \tag{4.19}$$

The diameter of Penstock comes to be 478 mm.

### 4.3.1 Determination of Penstock Thickness

The thickness of penstock can be calculated by the reference of penstock diameter using the following relationship:

$$t_p = \left(\frac{d_p + 508}{400}\right) + 1.2 \tag{4.20}$$

The thickness of penstock is kept to be 5 mm for utilization of available materials.

### 4.3.2 Tangential $V_{u1}$ and Radial $V_{R1}$ Components of Absolute Inlet Velocity

$$V_{u1} = V_1 \cos\alpha_1 = 20.22 \tag{4.21}$$

$$V_{R1} = V_1 \sin\alpha_1 = 5.80 \tag{4.22}$$

### 4.3.3 Tangential Flow Velocity at the Inlet, $U_1$

$$U_1 = \frac{V_{u1}}{\Psi} = 10.11 \tag{4.23}$$

### 4.3.4 Relative Flow Angle at the Inlet, $\beta_1$

$$\beta_1 = 90 + a\tan\left(\frac{V_{u1} - U_1}{V_{R1}}\right) = 150.17 \tag{4.24}$$

### 4.3.5 Relative Flow Velocity at the Inlet, $W_1$

$$W_1 = \frac{V_{u1} - U_1}{\sin(\beta_1 - 90)} = 11.66 \tag{4.25}$$

TABLE 4.1: Summary of design parameters for cross flow turbine.

| **Description** | **Symbol** | **Value** |
|---|---|---|
| Head | $H$ | 25 meter |
| Flow Rate | $Q$ | 0.793 $m^3$/sec |
| Overall Efficiency | $h_0$ | 70% |
| Mass Density | $r$ | 1000 kg/$m^3$ |
| Gravitational Constant | $g$ | 9.81 m/$s^2$ |
| Angle of Attack | $\alpha$ | 16 degree |
| Power | $P$ | 156 kW |
| Specific Speed | $N_s$ | 101 |
| Turbine Speed | $N$ | 452 rotational velocity |
| Number of Poles for Generator | | 16 |
| Runner Outer Diameter | $D_1$ | 500 mm |
| Runner Inner Diameter | $D_2$ | 334 mm |
| Thickness of Jet | $S_0$ | 85 mm |
| Breadth / Length of Runner | $L$ | 473 mm |
| Outer Radius of Runner | $R_1 = D_1/2$ | 250 mm |

| | | |
|---|---|---|
| Inner Radius of Runner | $R_2 = D_1/3$ | 167 mm |
| Radius of Blade Shaped Arc, $r = 0.16$ | $D_1$ | 80 mm |
| Radius of Center Pitch Circle | $R_0$ | 185 mm |
| Blade Spacing | $t_1$ | 87 mm |
| Radial Rim Width | $a$ | 83 mm |
| Number of Blades | $Z$ | 30 |
| Shaft Diameter | $d$ | 100 mm |
| Penstock Diameter | $d_p$ | 478 mm |
| Penstock Thickness | $t_p$ | 5 mm |

## 4.4 Geometric Parameters

A detailed summary of all parameters involved in the design is presented as follows

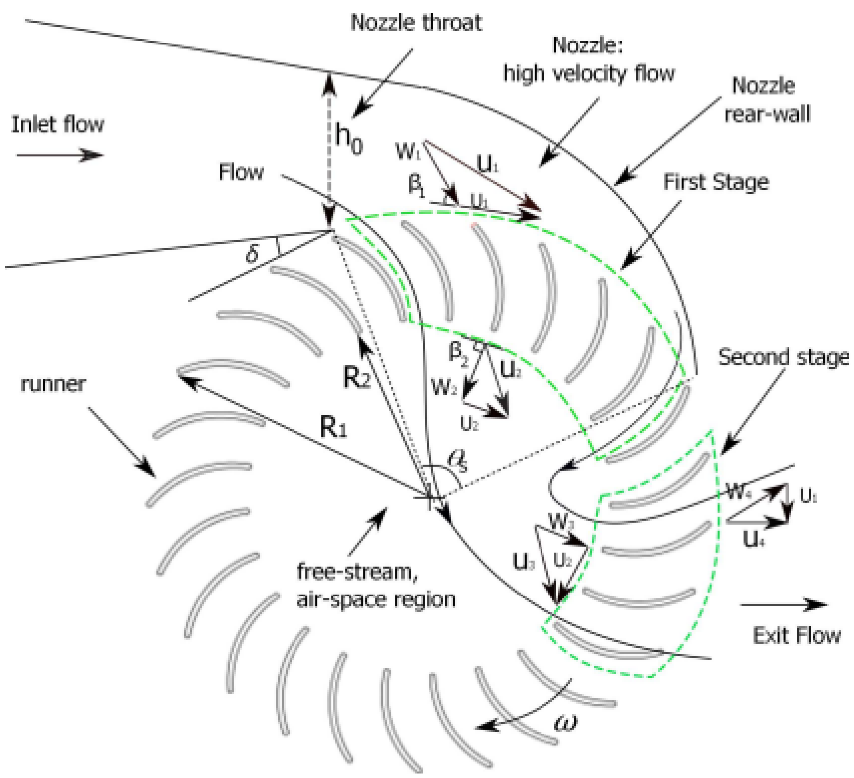


FIGURE 4.1: Geometric parameters for turbine geometry [52].

### 4.4.1 Flow Geometry of Inlet Stage

The detail geometric parameter of the flow at inlet stage calculated from the formulation are tabulated in Table 4.2.

TABLE 4.2: Flow geometry of inlet stage.

| Description | Symbol | Value |
|---|---|---|
| Work Coefficient turbine | $\Psi$ | 2.00 |
| Nozzle hydraulic Efficiency | $\eta_{hn}$ | 0.95 |
| Inlet absolute flow angle | $\alpha$ | 16 |
| The velocity of flow from the nozzle | $V_1$ | 21 |

### 4.4.2 General Rotor Geometry

General Geometry of the designed rotor achieved through parameters calculated in the design stage are tabulate in Table 4.3.

TABLE 4.3: General rotor geometry.

| Description | Symbol | Value |
|---|---|---|
| The angle subtended by nozzle arc | $\alpha$ | 90.00 |
| The outer diameter of the rotor | $D_1$ | 500 |
| The inner diameter of the rotor, | $D_2$ | 334 |
| The radial width of blade annulus | $a$ | 83 |
| Blade pitch | $t_1$ | 85.00 |
| Number of blades | $Z$ | 30 |
| Number of active inlet ducts | $I$ | 7.5 |
| The axial length of the Rotor | $B$ | 383 |
| Aspect Ratio of the rotor | $\frac{B}{D_1}$ | 0.766 |

### 4.4.3 Flow Geometry of Intermediate Rotor Stage

The detail geometric parameter of the intermediate rotor stage calculated from the formulation are tabulated in Table 4.4.

TABLE 4.4: Flow geometry of intermediate rotor stage.

| Description | Symbol | Value |
|---|---|---|
| The angle of absolute fluid discharge from the $1^{st}$ stage | $\alpha_2 = \alpha_3$ | 52.44 degrees |
| The tangential flow velocity of rotor intermediate stage | $U_2 = U_3$ | 6.74 m/s |
| Relative flow angles | $\beta_2 = \beta_3$ | 90 degrees |
| The relative flow velocity of intermediate stage | $W_2 = W_3$ | 8.77 m/s |
| The absolute flow velocity of intermediate stage | $V_2 = V_3$ | 11.07 m/s |

### 4.4.4 Flow Geometry of Rotor Outlet

Geometric parameters calculated for the design of rotor outlet are summarized in Table 4.5.

TABLE 4.5: Flow geometry of rotor outlet.

| Description | Symbol | Value |
|---|---|---|
| Relative outlet flow angle | $\beta_4$ | 29.83 degrees |
| Tangential flow velocity at the outlet | $U_4 = U_1$ | 10.1 m/s |
| Relative flow velocity at the outlet | $W_4 = W_1$ | 11.66 m/s |
| Tangential absolute flow velocity at the outlet | $V_{u4}$ | 0.00 m/s |
| Radial absolute flow velocity at the outlet | $V_{R4}$ | 5.80 m/s |
| Absolute outlet flow angle | $\alpha_4$ | 90 degrees |

### 4.4.5 Blade Geometry

Parameters needed for optimized design of blade geometry calculated with the help calculations referenced in the design section are presented in Table 4.6.

Table 4.6: Summary of blade geometry.

| **Description** | **Symbol** | **Value** |
|---|---|---|
| Blade spacing (pitch) at the inner radius | $t_2$ | 34.98 mm |
| Blade pitch arch angle | $F$ | 12 degree |
| The radius of Blade curvature | $r$ | 80 degree |
| Blade curvature arch angle | $f$ | 73.37 degree |
| Chord length across the blade | $L$ | 96 mm |
| Rotor solidity at the inner diameter | $s$ | 2.75 mm |

### 4.4.6 Nozzle Shape Parameters

The summary of nozzle shape parameters is given in Table 4.7.

Table 4.7: Nozzle shape parameters.

| **Description** | **Symbol** | **Value** |
|---|---|---|
| Nozzle throat width or thickness of jet | $S_0$ | 108.24 |
| Chord length of nozzle outer casing | $C$ | 436.65 |
| The angle between nozzle entry arch and entry chord | $t$ | 45.00 |
| Chord length of the nozzle entry arch | $f$ | 354.00 |
| The radius of the nozzle outer casing | $R_0$ | 281.00 |

### 4.4.7 The Angle between Nozzle Entry Chord and Nozzle Outer Casing Chord, $\mu$

$$\mu = a\cos\left(\frac{f^2 + C^2 - S_0^2}{2fC}\right) \tag{4.26}$$

$$\mu = 10.98 \text{ degree}$$

### 4.4.8 The Angle between Nozzle Entry Arch and Nozzle Outer Casing Arch, $\alpha_0$

$$\alpha_0 = \alpha_1 = 16.00 \tag{4.27}$$

### 4.4.9 The Angle between Nozzle Entry Chord and Nozzle Outer Casing Arch, $\phi$

$$\begin{aligned} \phi &= \alpha_0 + \tau - \mu \\ \phi &= 50.92 \text{ degree} \end{aligned} \tag{4.28}$$

### 4.4.10 The Radius of Nozzle Outer Casing, $R_0$

$$\begin{aligned} R_0 &= \frac{C}{2\sin\phi} \\ R_0 &= 281 \text{ mm} \end{aligned} \tag{4.29}$$

## 4.5 Generator Design Parameters

The design parameters of the generator used in our design are given below:

Table 4.8: Generator design parameters.

| **Parameter** | **Value** |
|---|---|
| Generator RPM ($N_g$) | 1800 |
| Generator KW | 285 |
| Generator Pulley Diameter ($d_g$) | 250 |
| Generator operation time (hrs) | 24 |

## 4.6 Turbine Pulley Design

In calculating the turbine-generator speed ratio the turbine and generator speed and diameter ratios need to be considered as follows:

TABLE 4.9: Turbine pulley design parameters.

| Parameter | Value |
|---|---|
| Turbine RPM ($N_t$) | 484 |
| Turbine Pulley Diameter ($d_t$) | 1000 |

## 4.7 Sizing of the Belt Length

The length of the belt of the pulley can be determined using the following relationship.

$$L = 2C + \frac{p\,(dt + dg)}{2} + \frac{(dt - dg)\,2}{4C} \tag{4.30}$$

where, $C$ is the centre distance.

$$\begin{gathered} C = (dt + dg) = 1250 \\ \text{Belt length } (L) = 4576 \end{gathered} \tag{4.31}$$

After complete formulation of primary and auxiliary design parameter, these parameters were used as basis for development of manufacturing drawings and manufacturing activities for these production activates we present here the design and manufacturing approaches for major components of a cross flow turbine as under.

1. Turbine Housing
2. Guide Vane
3. Guide Vane Regulating Assembly
4. Flow Directing Nozzle
5. Rotor and Rotor Disc
6. Rotor Shaft

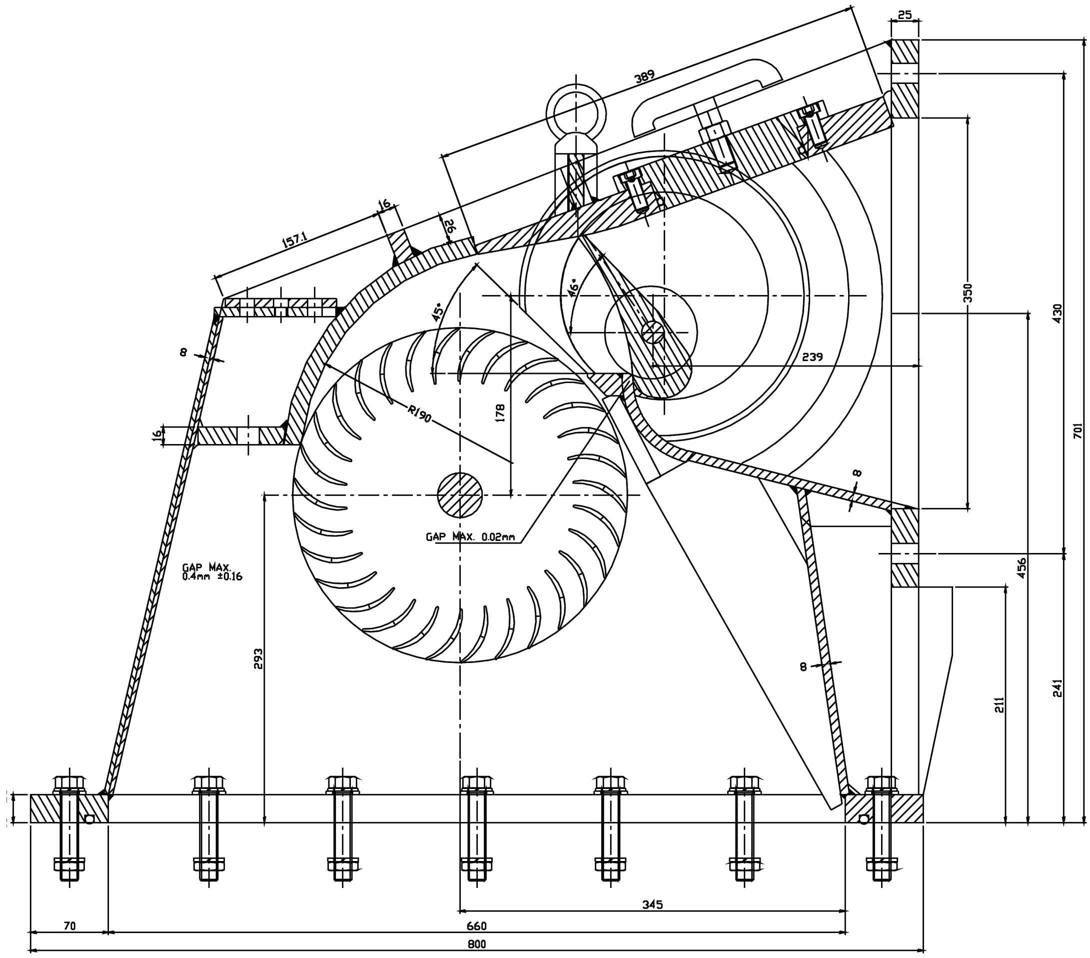


FIGURE 4.2: General view of cross-flow turbine housing.

## 4.8 Turbine Housing

Turbine Housing of the manufactured cross-flow turbine measures 701×800×1200 in length height, and width respectively. The material chosen for manufacturing this turbine housing is SS41 with the average thickness of plates used is 16 mm in thickness, material was chosen based on their structural properties and cost feasibility. The comparison of materials was done using Key to a steel database where several aspects from strength, weld ability, and corrosion resistance were compared. This chosen grade of steel has excellent weld ability and machinability and can be subjected to various types of heat treatments as and when required during manufacturing of the turbine housing the two sides of the turbine were welded using bracings in the center as two counter the effects of metal shrinking and distorting the center distance between the two sides of the panels.

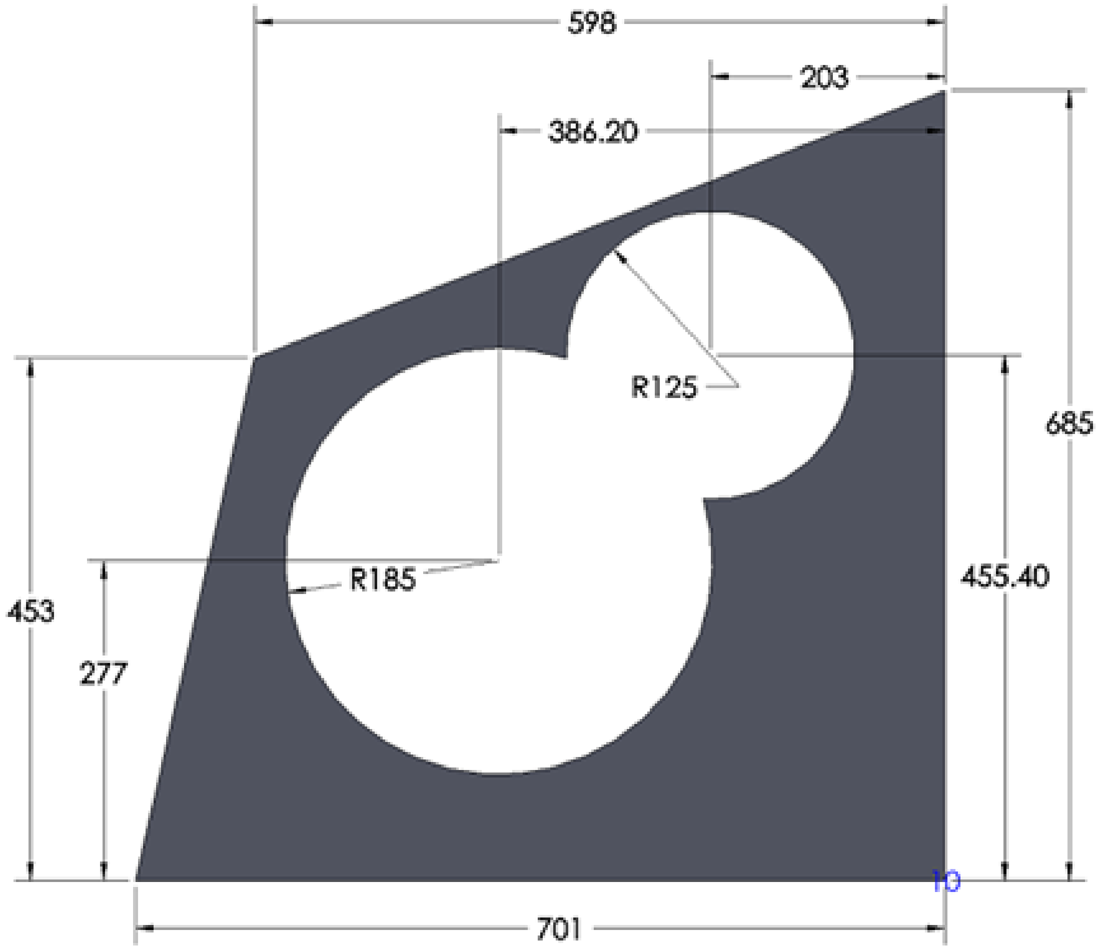


FIGURE 4.3: Side plate of cross-flow turbine.

FIGURE 4.4: Turbine housing after machining and arrangement of components inside.

The turbine housing is the main component of the structure as it houses all the key components required for the operational configuration of a cross-flow turbine that is the rotor, guide vane, etc. The side panel of cross-flow turbines was designed for manufacturing feasibility. All side panels of this turbine housing were blanked from the parent plate using a CNC Flame Cutting machine where AutoCAD drawings were used to guide the flame cutting troche of the machine After fabrication of individual panels welding procedure developed was followed were bevel of 30 degrees with 2 mm of welding penetration gap was made for strong welding joints. Three sides of the housing will be welding while the fourth is to be welded after assembly of internal components.

## 4.9 Guide Vane and Regulator

Water flowing from the penstock is directed towards the guide vane before it reaches the directing nozzle that guides the water on the rotors. The guide vane controls the amount and direction of water that enters the turbine rotor. The material used for manufacturing was A36 a variety of carbon steel with good machinability and ductile properties. A blank of this material was first marked using a 1:1 template drawn with the help of autocad. Blank was machined using manual horizontal milling machines to achieve the final shape. The final profile was then achieved by the craftsmanship of experienced fitters and fabricators.

The guide vane assembly can be controlled using the regulator valve, the screw-based mechanism was used for the operation of this assembly. The main component of this valve is the screw shaft which employees' square threads for precise movement of guide vane. The guide vane can be operated under the following basic conditions:

1. Maximum Position
2. Rated Position
3. Closed Position

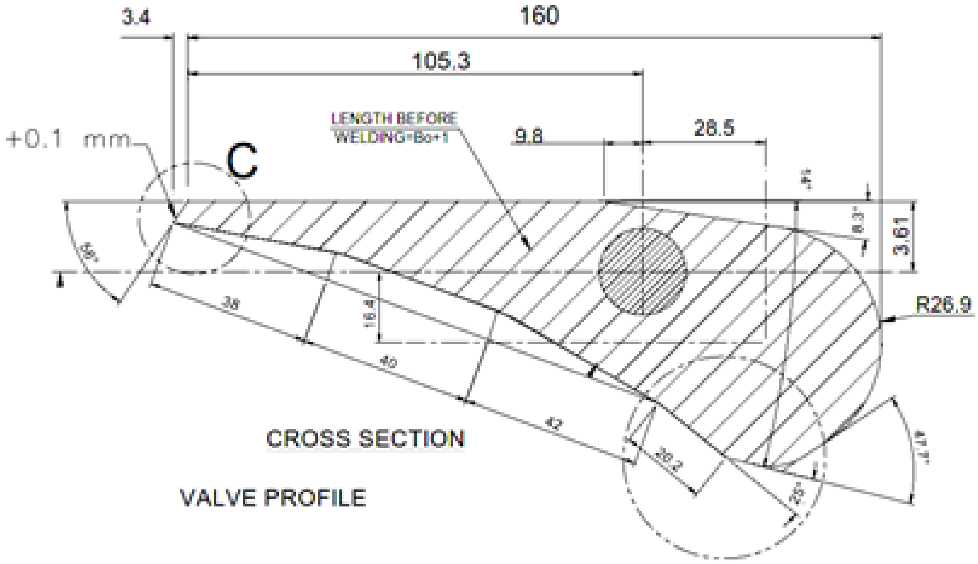


FIGURE 4.5: Cross-sectional view of guide vane.

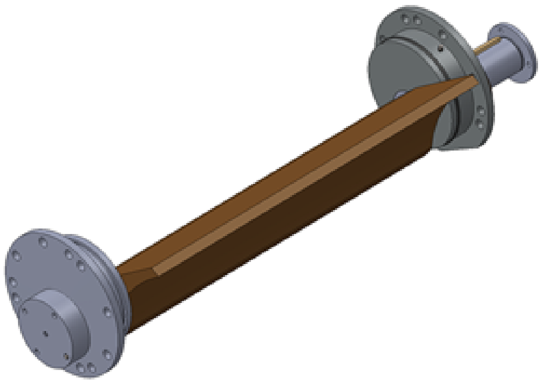

FIGURE 4.6: Guide vane assembly.

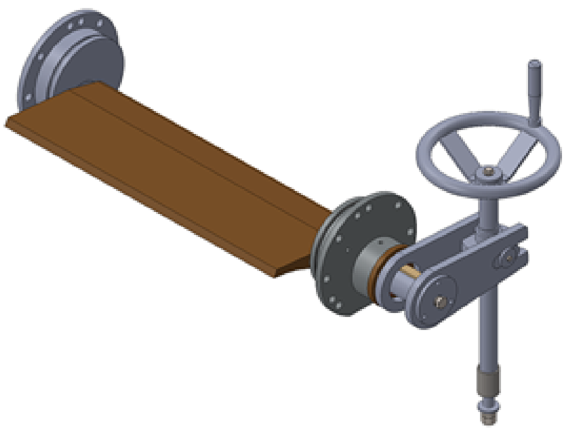

FIGURE 4.7: Guide vane regulator assembly.

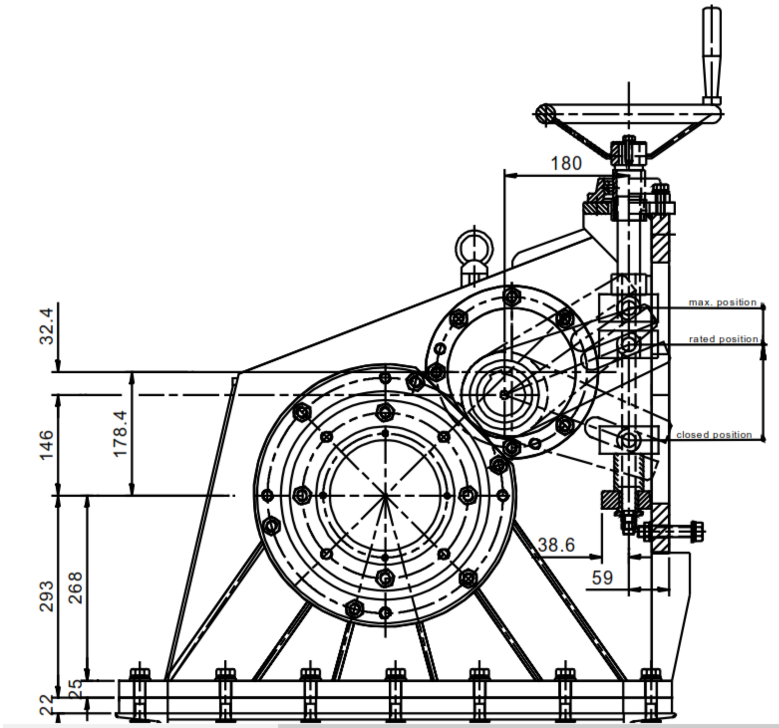


FIGURE 4.8: Diagram Showing operating positions of the guide vane.

In our arrangement the max position allows the maximum amount of flow of water to flow through the guide vane assembly. The rated position is the position which is at which the amount of water flowing through the guide vane is equal to the flow rate of water for which the turbine is designed to give maximum efficiency at the given head and operating conditions. The closed position is used to cut off the water supply to the rotor in case of maintenance or when the operation of the turbine is no needed.

## 4.10 Guide Nozzle

The guide nozzle fabrication in the turbine housing helps in direction of water coming from the guide vane to the rotor blades. The installation of guide vane has positive effects on the performance and internal flow of water through the turbine [50] thus improving the overall efficiency of the turbine. Manufacturing of the inlet of the guide nozzle to a smooth shape helps in directing water towards the rotor and in doing so also reduces the flow separation effect during the flow of water.

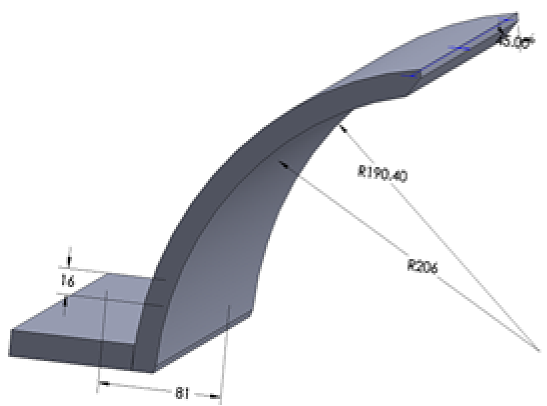


FIGURE 4.9: Solid works model of guide nozzle.

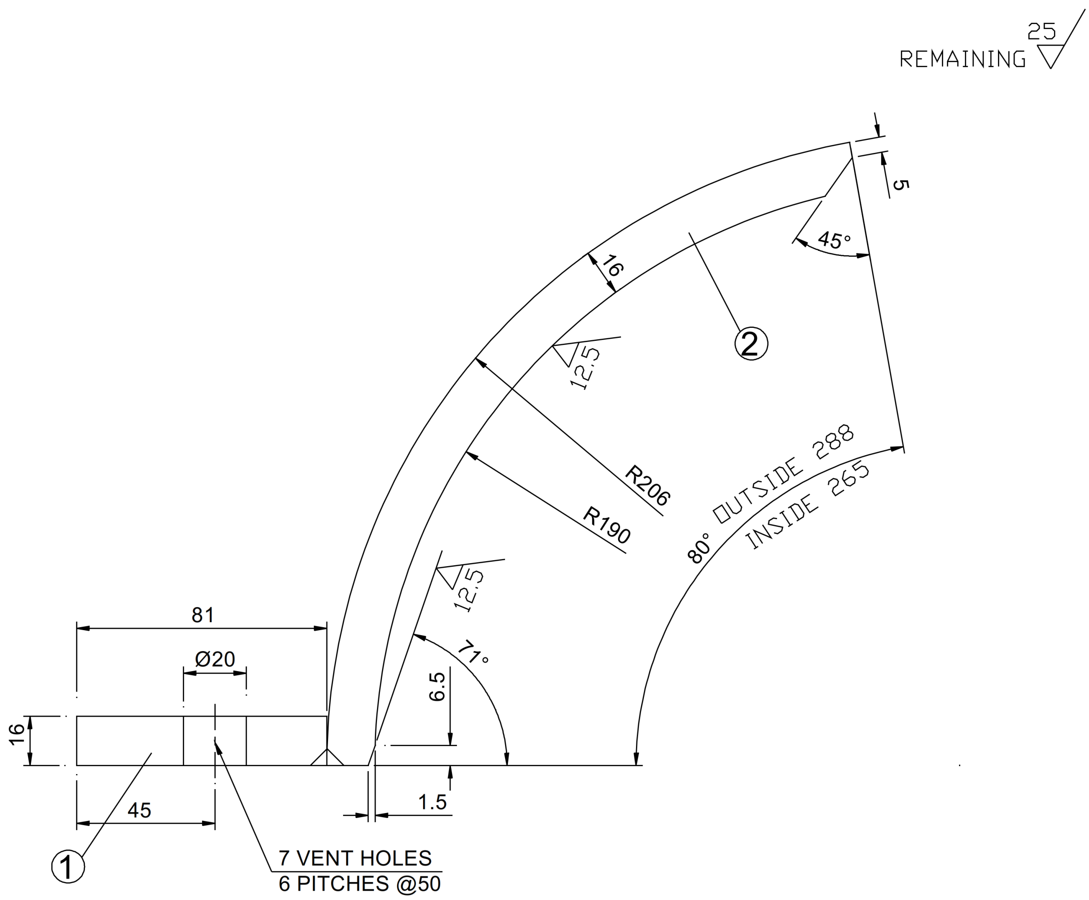


FIGURE 4.10: Auto cad section of guide vane.

The guide vane also has a set of vent holes that help maintain a suction pressure inside the turbine casing that helps inflow water from the inlet nozzle towards the turbine rotor. The suction pressure can be controlled using a damper system that allows us to limit the amount of air flow across the housing of the turbine this valve was simply designed as a flange with a number of holes that can be covered using a plate slide across its surface.

## 4.11 Turbine Rotor Assembly

The turbine rotor assembly is the heart of a cross-flow turbine as it produces the rotary motion that is then transmitted to the generator for power generation. The rotor assembly of the turbine consists is shown in detail below and thus discussed.

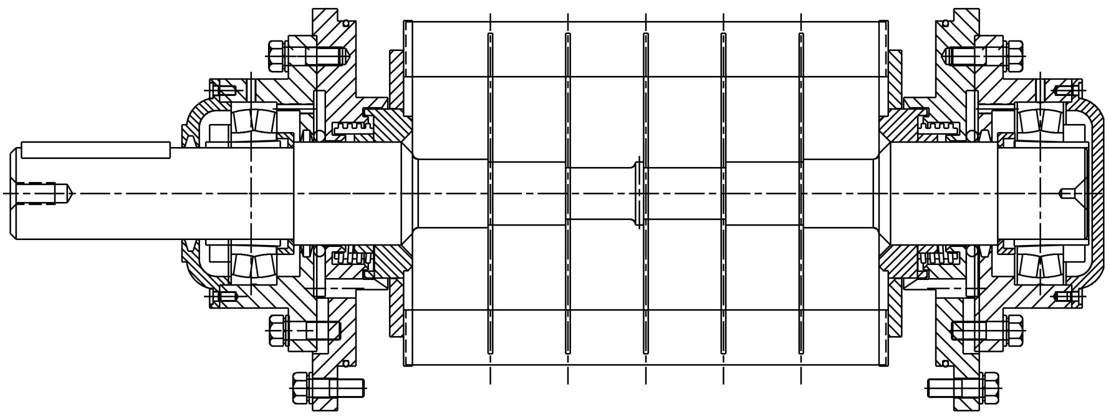

FIGURE 4.11: Turbine rotor assembly.

FIGURE 4.12: Turbine rotor assembly solid works model.

The main components of the rotor are as follows:

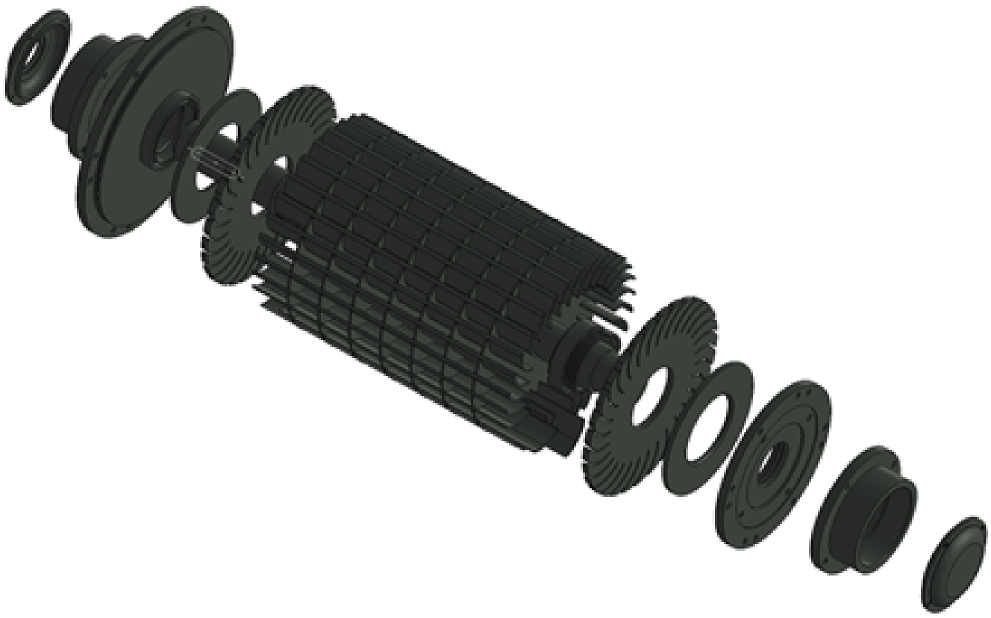

Figure 4.13: Exploded view of turbine rotor assembly.

1. Rotor Shaft
2. Bearing Housing
3. Side and Intermediate Disks
4. Runner Blades

## 4.12 Rotor Shaft

The rotor runs across axially through the center of the turbine. The diameter of the rotor shaft was previously calculated in our formulation and the following manufacturing drawing was drafted for production while choosing the material SS41 owing to its good machining and durability while being cost-effective. The ends of this shaft are housed in the bearing housings on both sides thus the final dimensional sized are of high accuracy. The steps were machined for welding of intermediate disks in the middle while chamfers were made for welding of side disk on the shaft. All bearing sizes were machined after the disks were welded on the shaft to cater for material distortion due to welding on the shaft.

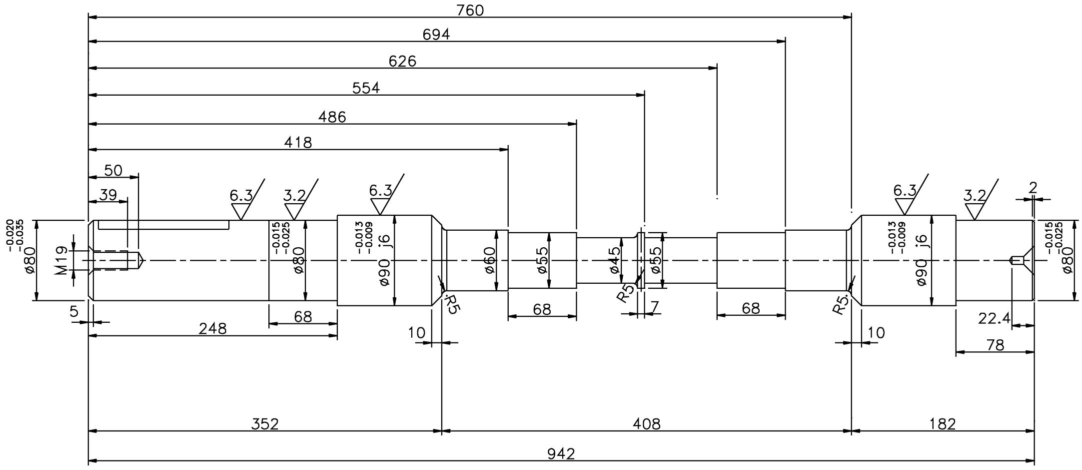


Figure 4.14: Detailed drawing of rotor shafts.

Figure 4.15: Solid works model of rotor shafts.

## 4.13 Bearing Housing

Due to a combination of torsional and axial thrust on the rotor shaft an assembly was designed which would provide ample support to the shaft during turbine rotation and will also provide axial and torsional stability after calculation loads on the shaft, bearing calculations were done, and bearing chosen were axial bearings SKF22218EK and the following bearing housing of suitably designed around it. The bearing housing consists of a flange assembly that connects the turbine shaft from the turbine housing with the use of rubber based seals and metallic spacers. These components allow for an efficient sealing system to be developed that allow the bearing housing to connect to the turbine assembly without causing any water leakages.

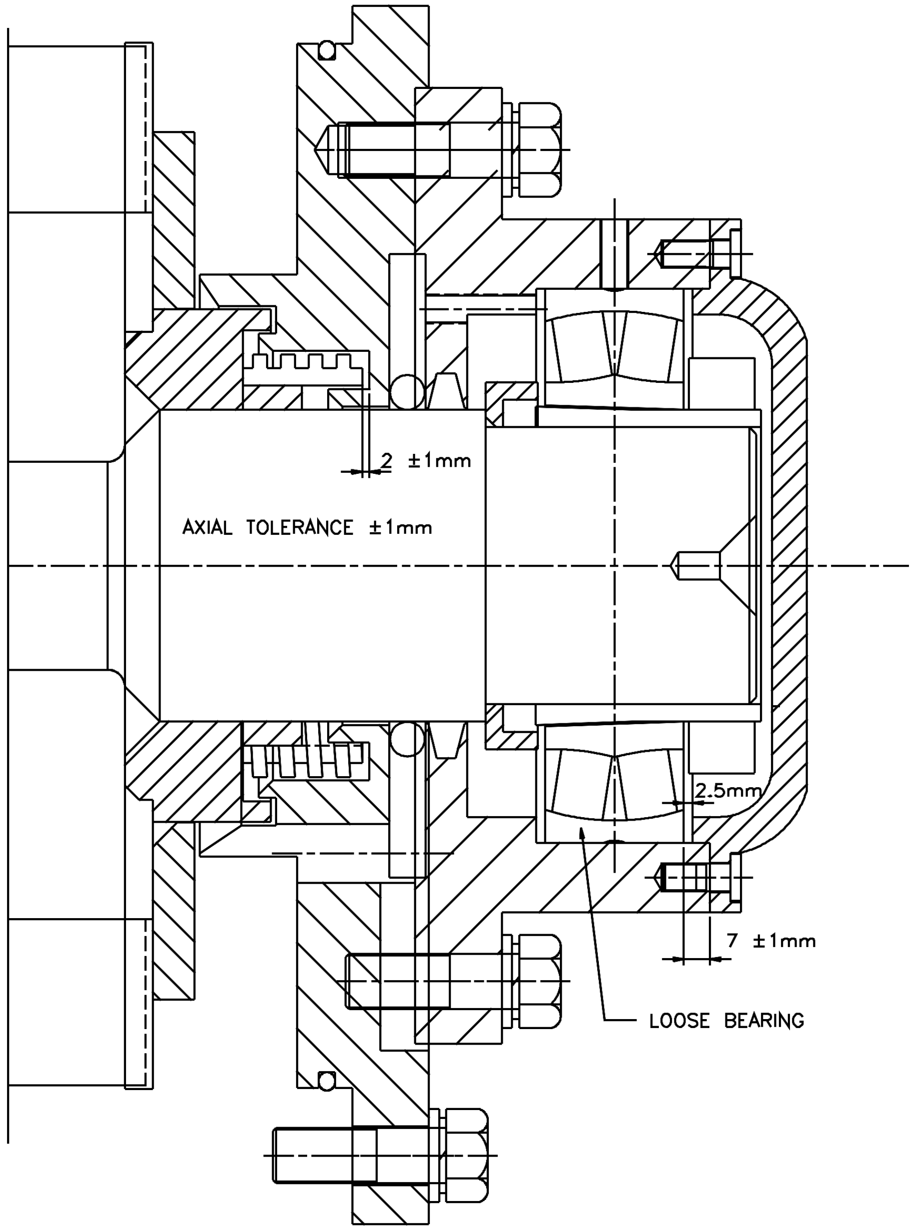


FIGURE 4.16: Bearing housing cross flow turbine.

## 4.14 Side and Intermediate Rotor Disk

The rotor used in the developed prototype consists of 2 No's of side disks that have 12 mm thickness that act as the main support where 5 No's of intermediate side disks were installed along the axial length of the shaft. These side disks manufactured from SS41 were later coated with anti-corrosion enamel paint for prolonged life and protection against pitting. Steps were machined along the length

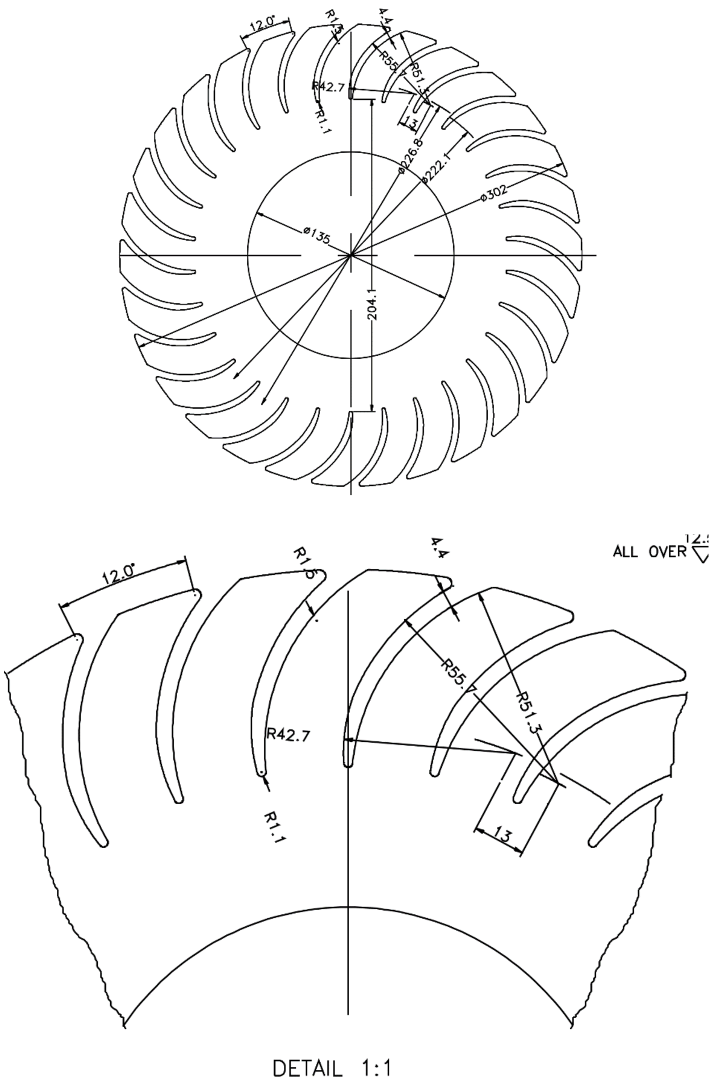


FIGURE 4.17: Side disk 2 No's welded on shaft.

of the shaft to provide support for the intermediate and side disk after welding. To ease assembly these disks for manufactured in two halves like 2 pieces of a whole part. The machining was achieved to a degree of high accuracy using an Electric Discharge Wire Cutting machine.

As the width of the turbine increases the number of the intermediate disk can be installed to provide support the blades running across the width of the turbine as large width means the larger flow of water interacting with the surface of the rotor which could impart a significant amount of force upon these blades and this intermediate disk provide much-needed support to the rotor assembly.

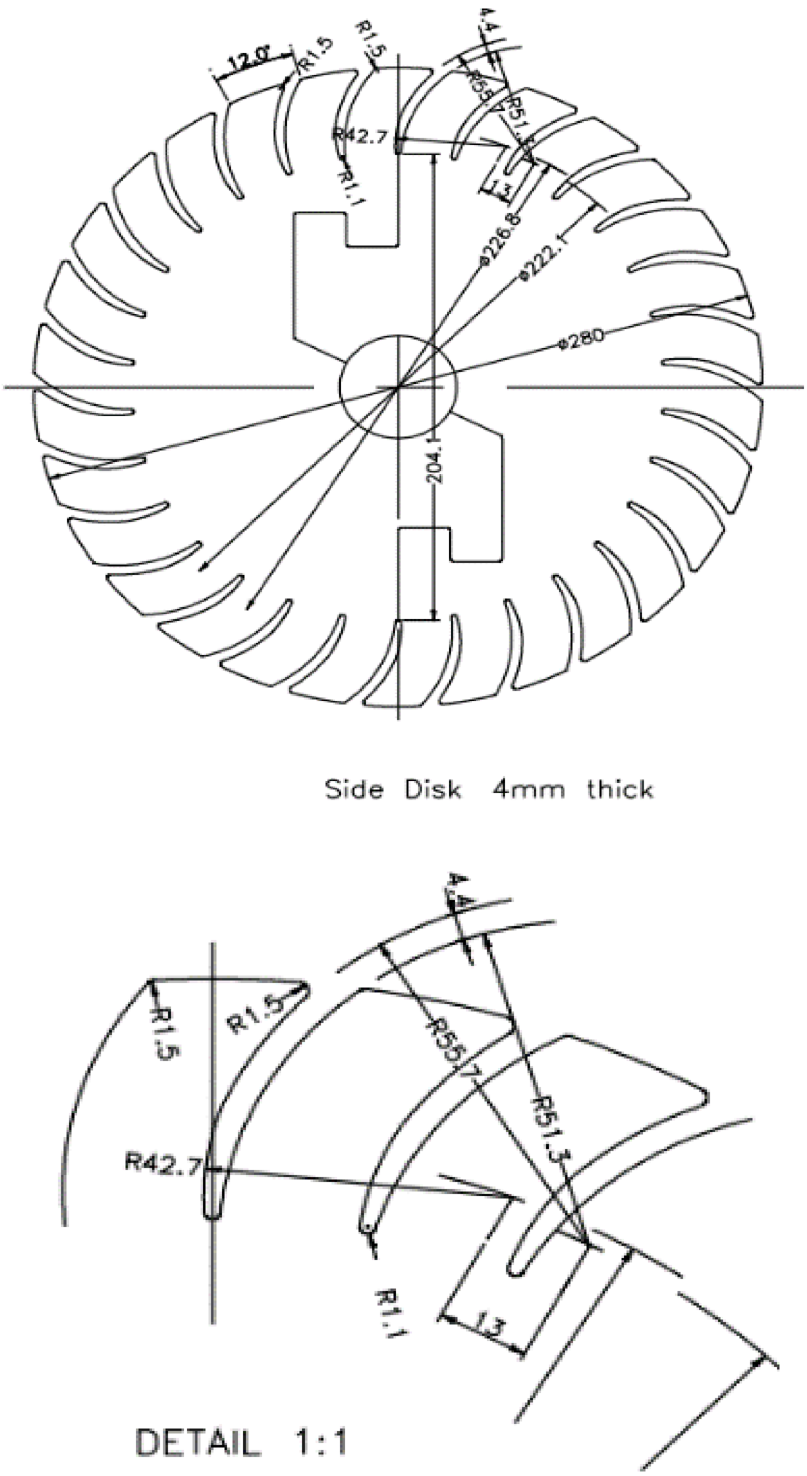


FIGURE 4.18: Intermediate disk for rotor shaft 5 No's.

## 4.15 Runner Blades

The force of the water is transferred to the runner through the interaction of water with the curved surface of the runner blades, the geometry of the rotor blades has been discussed in detail in the methodology of the thesis and the number of blades selected for this particular design has also been discussed. The material chosen for manufacturing of these rotor blades is A106-Grade B which is a seamless pipe with high corrosion resistance and is mostly used in pressure vessel applications where the long life of the equipment is of key importance. The runner blades were machined from a single pipe that was parted in 4 sections and milling machines were used to achieve the final profile of the blade. Seamless steel pipe of size DIN 2448 with an outside diameter of 114.3 mm, inside diameter, being 102.26 with a wall thickness of 6.02. So the selected length of the blade is 424 mm. For 30 No's of blades the total length of the required piper was calculated to be 2560 mm.

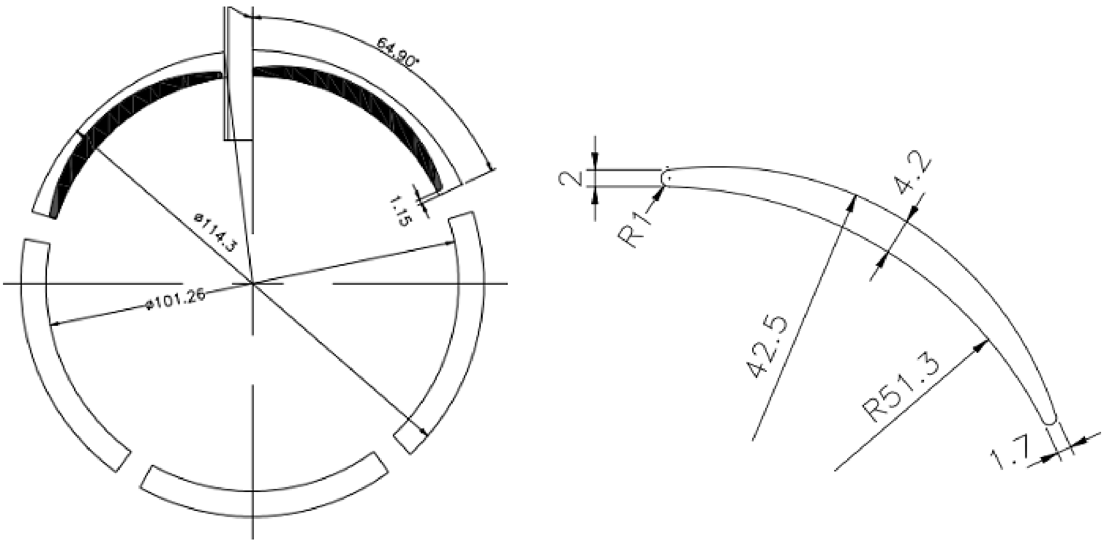


FIGURE 4.19: Runner blade cross-section profile and CAD model.

## 4.16 Welding Procedure for Turbine Housing

After machining of all key components that comprise the turbine assembly a welding procedure was developed. The procedure and sequences adopted for welding were designed to keep material distortion to a minimum in order to maintain ease of assembly and reduce significant rework on machining equipment. The welding used to assemble the turbine was accomplished using Tungsten insert gas welding equipment, which in comparison to arc welding produces a much stable and highly


FIGURE 4.20: Components of rotor assembly after machining.

controllable electric arc with minimum distortion and high strength of the weld joint [53]. In order to accomplish manufacturing of turbine housing with minimum distortion the following sequence of procedures were adopted and are explained with reference to the welding procedure diagram.

1. Align side plates of the turbine housing using level gauges and tighten with the help of connection pipe that extends between the two side plates in and holds them in position in order to minimize distortion in straightness.

2. Tack weld the side plates with the base frame flange. This flange is already screw with the base frame structure.

3. Now in order to perform final inspection of turbine housing design insert all the housing parts to check if they attain their desirable positions as expected.

4. Now welding of housing parts using the following sequence for both left and right sides.

    i. Check clearance of runner gap with feeler gauge it should be at maximum 0.1 mm.

    ii. Tack weld outside at the position referenced.

    iii. Check clearance using the 0.5 mm feeler gauge strip.

iv. Tack weld outside.

v. Check Clearance of lower valve gap with feeler gauge strip of 0.1 mm.

vi. Tack weld outside.

vii. Check clearance with feeler gauge strip of 0.5 mm.

viii. Tack weld outside.

ix. Check Clearance of upper valve gap with feeler gauge strip of 0.1 mm.

x. Tack weld outside.

xi. Check clearance 1 mm.

xii. Tack weld the lower nozzle plate with the side plates and tack back side plate from inside as accessible.

xiii. Fully weld upper nozzle and rear runner then re-check the clearances 1, 5 and 9.If they are within the range of 0.15 mm then tack all parts properly.

xiv. Tack weld the from side plates of the housing

xv. Tack weld the back plates of the housing.

xvi. Now tack weld the reinforcement ribs for the housing structure.

xvii. Tack weld the penstock flange at the inlet of penstock.

xviii. Now fully weld all accessible parts of the housing assembly.

xix. Fully weld all the reinforcement's ribs of the structure and weld rest of the housing.

## 4.17 Welding Procedure for Turbine Runner Assembly

The runner assembly for the cross flow turbine consists of the rotor shaft, intermediate and the side support disks. Runner blades design accordingly run across the length of these runner disk which bear the full load of incoming water flow. During

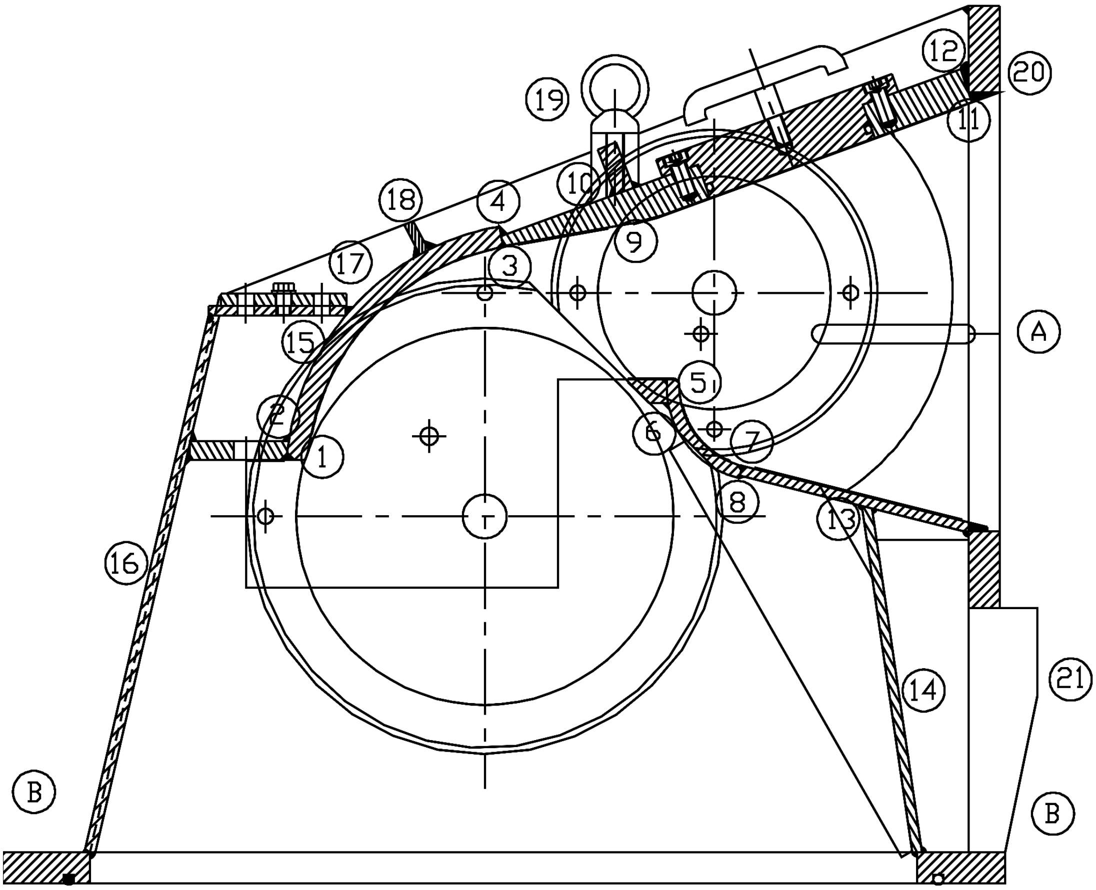


FIGURE 4.21: Welding procedure diagram.

manufacturing stage many deviations from the original designed dimensions can occur both due to machine or welding procedure selection errors. In order to control close dimensional and geometrical tolerance during the fabrication of runner assembly steps were taken to minimize these deviation from design. The rotor shaft was initially rough machined for all bearing sizes, these bearing sizes which were in the ranges of 1 thousands of a millimetres would face severe distortion due to heat zone of welding throughout the shaft. Several steps were designed in the shaft that would provide support to the intermediate disk and align them in position for ease of assembly. Intermediate disk were manufacture in two halves which after alignment at their position were tack welded initially with its respective other half and with their point of contact with the rotor shaft. In order to maintain straightness and proper pitches between all rotor disks screw jacks were positioned between them before welding. After placement of screw jacks welding was completed using TIG welding were equal passes were made on opposites in equal increments to significantly reduce distortion in diametric direction. The diameters of rotor disk were kept lager than the final dimension for the purpose that the deviation produced at the free end of the structure can be later machined out using lathe tools. Bearing surfaces and key ways were machined on both ends of the rotor shaft after welding after complete welding of the turbine assembly and all support assemblies were removed.

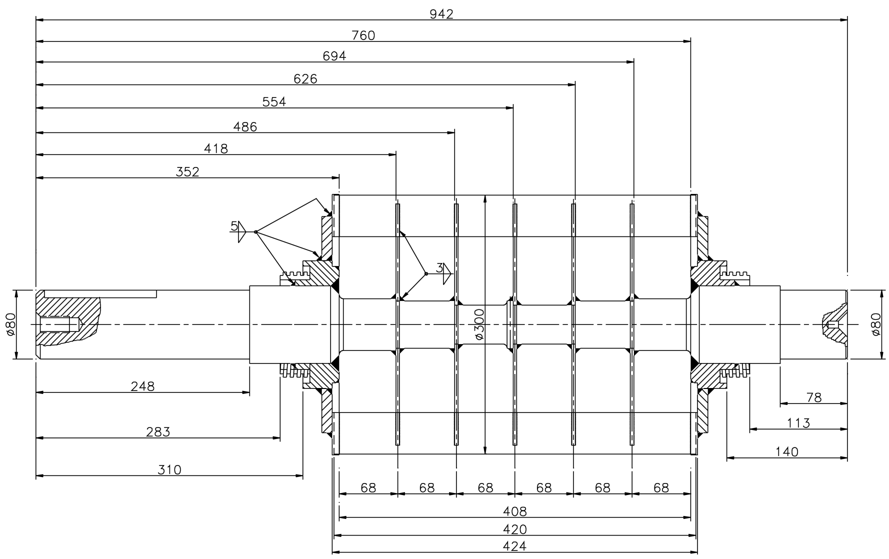


Figure 4.22: Welded runner assembly CAD model.

Figure 4.23: Turbine rotor after fabrication.

# Chapter 5

# Numerical Analysis

This chapter discusses the analysis of cross flow turbine design using simulation tools commercially available. Ansys 19.0 was selected to simulate the design for CFD, static structural and vibrational domains. The boundary conditions and input parameters discussed earlier are the basis for the analysis performed in this section of the thesis presented. This chapter is divided into four categories where CFD analysis was performed to determine the loads applicable on the designed rotor. These loads were later applied for structural and vibrational analysis where harmonic analysis was applied to define safety of design under cyclic loading.

The design of the cross-flow turbine was complimented by the use of computational analysis techniques available. Ansys version 19.0 is a commercial-level software used for an accurate depiction of fluid and structural characteristics using well-defined boundary conditions and material properties. In the current study, a combination of structural and fluid dynamic analysis was performed using finite elemental methods. The use of FEA proves to be of key importance in scenarios where a comprehensive understanding of any physical phenomena based on its mathematical formulations. Cases, where structural, thermal, or fluid behavior needs to be studied using the applications of FEA, have been of wide importance, the majority of these behaviors can be predicted by solving partial differential equation approaches. The objective of applying simulations applied in this study was to study the static structural behavior of the rotor, analysis of the natural frequency of the system under the rpm produced by the flow of water, and to study the harmonic response of the system under operating conditions.

Computational Fluid Dynamic simulations were carried to study the characteristics and behaviors of the cross-flow turbine under the flow rate and head of water as the input conditions the fluid dynamics under focus under this analysis where the velocity distribution profile and pressure distribution within the turbine housing while the flow pattern exhibited by the fluid was also under observation. The simulation was designed for the given inputs of flow rate and RPMs of the turbine rotor.

## 5.1 Design of Analysis

The basic design of the analysis will be discussed in this section. The cad model of the cross-flow turbine was developed using drawings already discussed in the manufacturing stage. The cad models were developed using Solid Works 2019 using a scale of 1:1 and assemblies were compiled to verify the designs. The rotor and Turbine Housing were assembled as per drawings where each component was assembled in Solid Works assembly modeler using relevant mates and fits.

The modeled geometries of rotor and turbine were converted in step format so that they can be imported into Ansys Design modeler without loss of any parametric detail relevant to the design. The rotor and housing models were imported in separated design modeler modules where separate parameters and named selections were defined based on the different surface to surface and surface to fluid interactions expected in the simulation. The turbine was defined by the two different surface interactions. The imported models were linked to mesh, modeler, where optimum meshes were produced. After the model definition and meshing the model was imported in CFD Pre where input parameters and boundary conditions were defined and simulations were solved for the given conditions. After a successful solution, the outputs of the CFX solve were applied as inputs to the static structural module of our simulations. The load applied on the rotor and rotor shaft was simulated to check the structural integrity of our rotor and shaft at the given rpm and loads applied by fluid during rotation of the rotor under the flow rate of fluid in the turbine housing. The loads were applied to the blades were calculated using the CFX solver module. The outputs of this static structural module used as the framework for doing modal and harmonic analysis of our system.

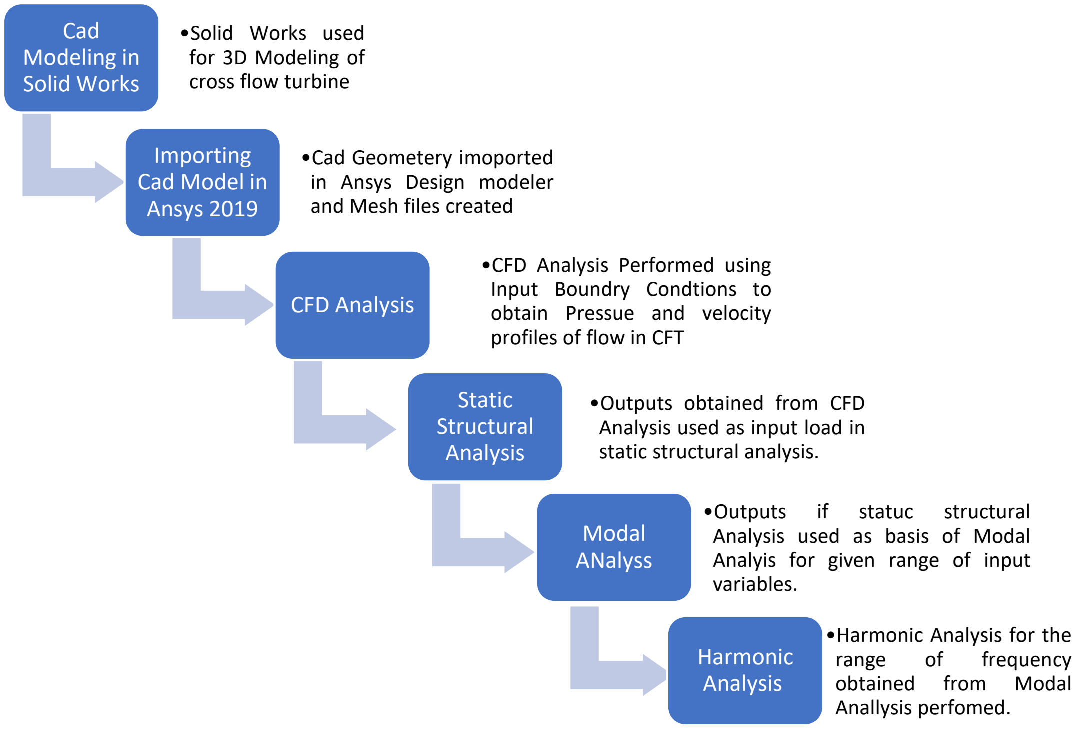


FIGURE 5.1: Computational analysis topology.

## 5.2 Preparation of CAD Model

Parametric modeling for the cross-flow design analysis was done using Solid Works software. To perform the analysis, the model consisted of two components namely the turbine housing with the nozzle and the turbine rotor. The models after preparation were converted into .step file format so that could be imported in Ansys. The figure below shows the cad model of the cross-flow turbine created to perform CFD and structural analysis using Ansys. The design was made for easier assembly using actually dimension clearance as per produced cad drawings.

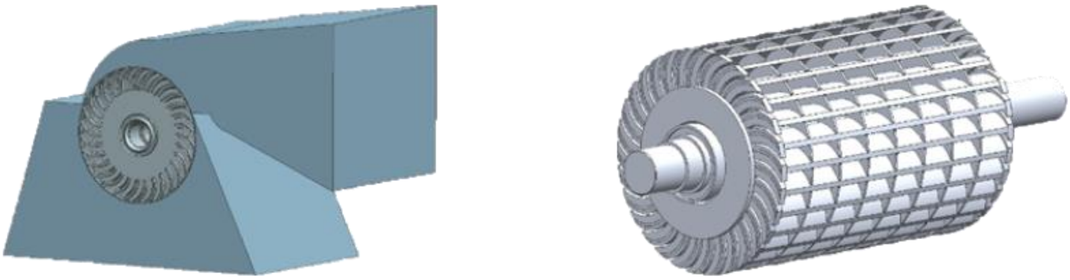

FIGURE 5.2: CAD models prepared for CFD analysis.

The model for rotor and turbine housing were imported into two separate design modeler modules of Ansys to separate assign their geometric and meshed parameters. The nozzle and turbine housing are the considered stationary domain for this analysis while the rotor is defined as a rotating domain that would be rotating with a defined number of RPMs.

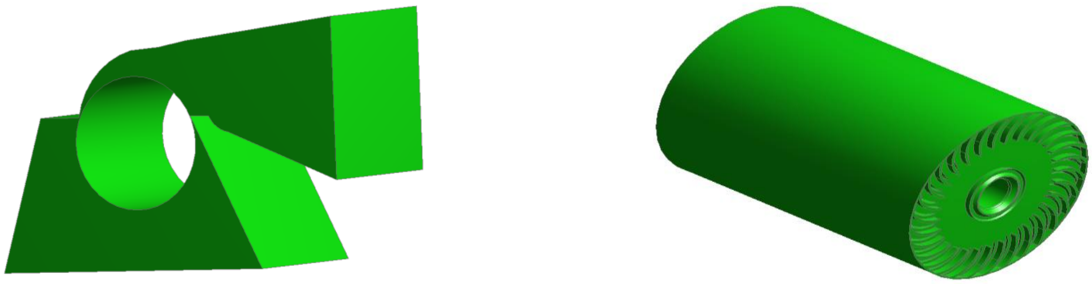

FIGURE 5.3: Models imported in design module Ansys.

Different regions of the imported models were defined based on their characteristics. The inlet of the nozzle was defined as the inlet region and vice versa the outlet was defined as the outlet region. The region between the rotor and the involute of the turbine housing was defined as an interacting surface between the rotor and the turbine housing and was termed Rotor_volute_interface. Material properties were assigned using engineering data for the respective components.

## 5.3 Meshing of Geometries

After defining geometric properties of the cad models in the design modeler the both separated design modeler modules of turbine housing and rotor were linked with their separate meshing modules where individual meshing characteristics were defined for both components. Hexahedra mesh nodes were chosen for the analysis and mesh element size was kept as 6 mm to achieve refined accuracy.

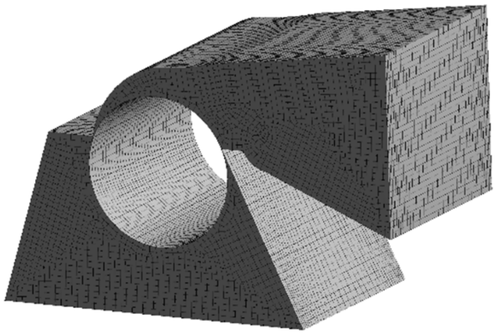

FIGURE 5.4: Turbine housing meshed.

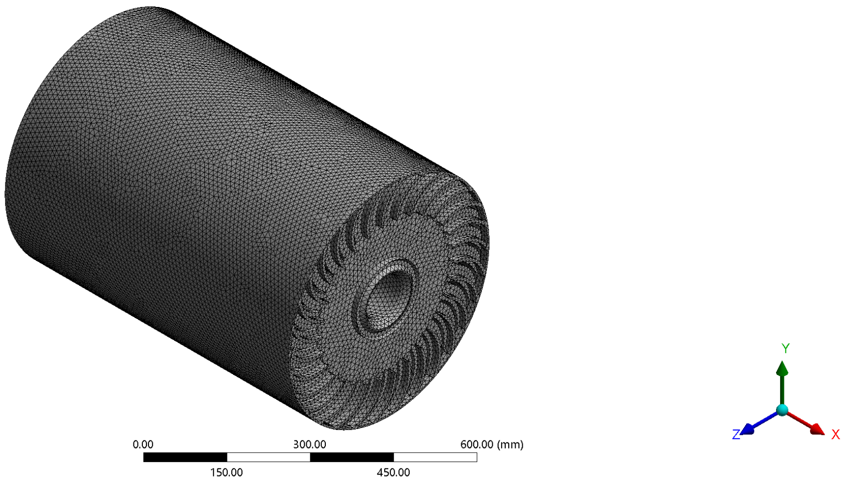


FIGURE 5.5: Rotor being meshed.

## 5.4 Computation Fluid Dynamic – CFX Physical Setup and Analysis

After the meshes were defined for both individual components i.e turbine housing and rotor geometry, both of these geometries were linked with the CFX module of Ansys. For this analysis, the k-epsilon turbulent model with scalable wall function was chosen.

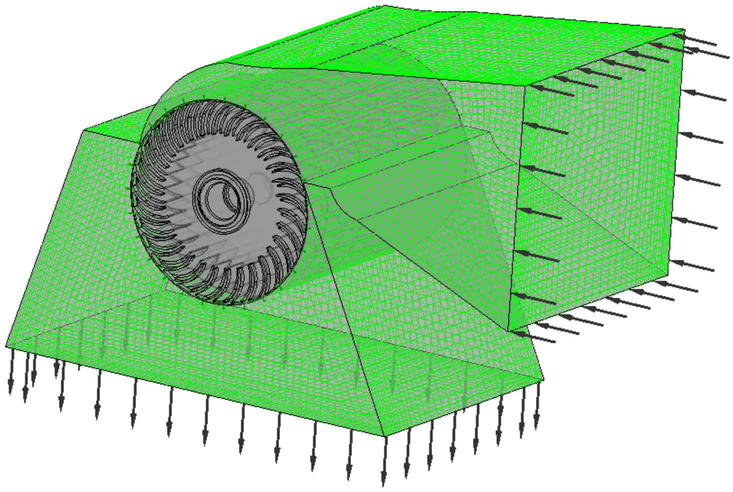

FIGURE 5.6: CFD setup for analysis.

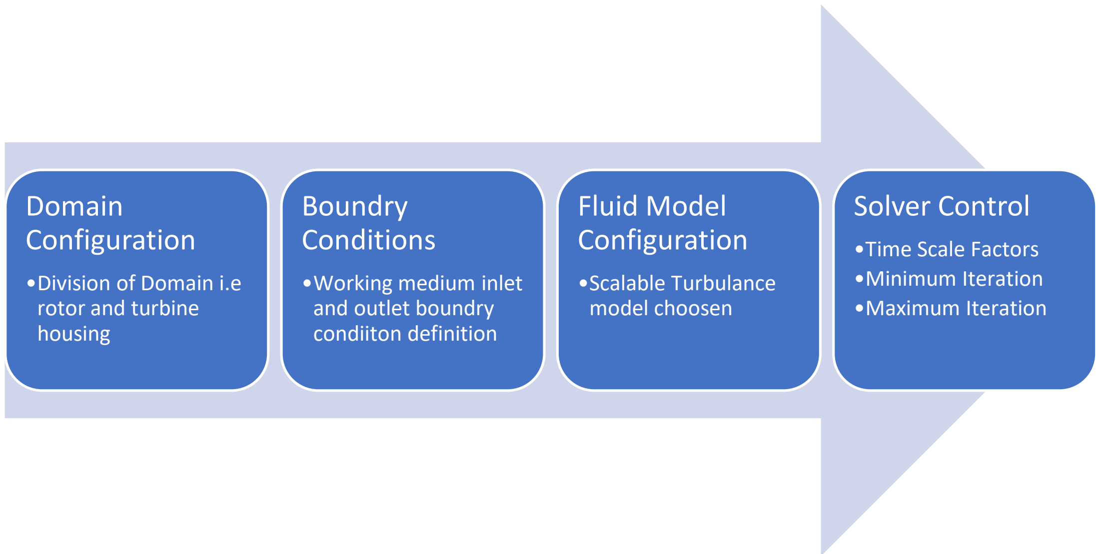


FIGURE 5.7: Process outline for CFD analysis.

The domains were divided into two categories based on their nature, turbine housing and nozzle were defined as a stationary domain while the rotor was defined as the rotary domain. The working medium of the model was chosen as water. The reference pressure is set as zero and boundary conditions are specified as absolute values. At the boundary inlet relative pressure of 345 kPa equivalent calculated from the head of water was applied and the ambient working temperature was set. The outlet boundary condition was set to 1 bar as atmospheric pressure. Now for the rotor domain, the rotor geometry was defined as a rotary domain using

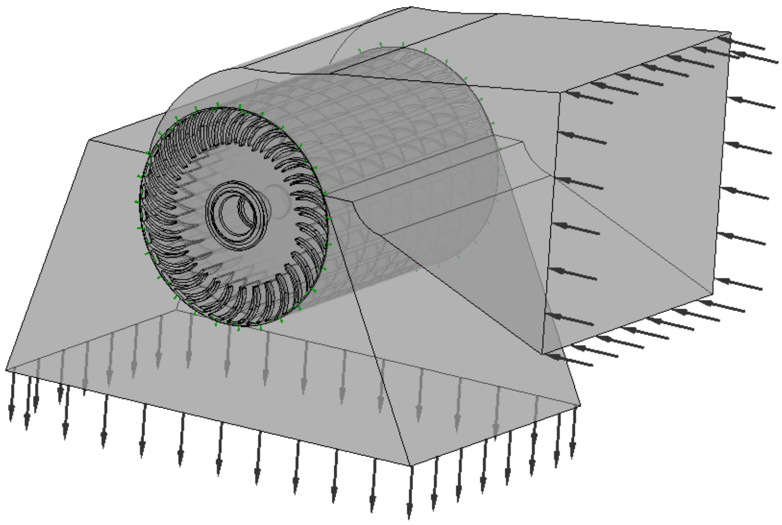

Figure 5.8: Inlet and outlet boundary condition as per CFD setup.

domain motion option in analysis setting and angular velocity of 484 rpm was defined as input angular velocity and the no-slip smooth wall boundary condition was defined for all interacting interface was set.

The interface between the rotor and the casing namely Rotor_volute_interface was defined as the frozen rotor interface model. Expressions were defined for the analysis were mass flow rate at the inlet and mass flow rate at the outlet of the casing. The solver controls were defined using the solver analysis tab and were defined as high resolution. The maximum iterations for the simulation were kept at 400 with a timescale factor of 1.00 and the convergence criteria were targeted at 1e-06. The output control was set to monitor the expression of mass in the system and mass going out of the system.

## 5.5 CFX Results

After checking the convergence of the given model and its boundary condition the simulation was solved and the outputs targeted were the velocity profiles of the flow in the rotor and the pressure distribution on the rotor surface during its revolutions. The following outputs were collected after the simulation.

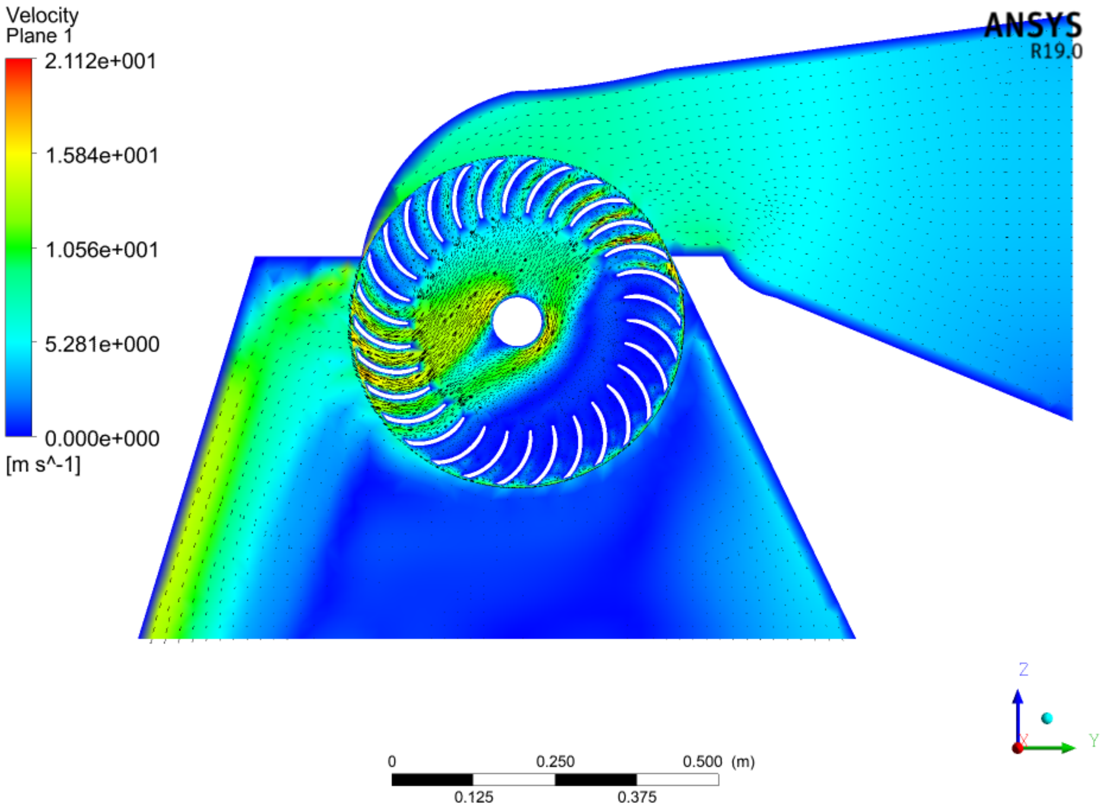


FIGURE 5.9: Velocity profile of water flow showing $1^{st}$ and $2^{nd}$ stage interaction of water with the turbine rotor.

From the output, monitor expressions produced it is clear that the mass flow inside the turbine is equal to the mass going out of the turbine housing thus conclusion was made that the solution has converged very well and our boundary conditions had no contradiction with our defined solution models. The figure for the velocity profile of flow show that water enters in the rotor and the $1^{st}$ stage passes through one set of blades and exerts its momentum to the system for rotation of the rotor and after passing through the $1^{st}$ stage again delivers the regain momentum to the second region of the blades during $2^{nd}$ stage interaction with the rotor.

The following pressure plot on the surface of the rotor was obtained under the defined boundary condition. This is the pressure acting on the rotor of the turbine due to driving water that flows from the guide nozzle towards the turbine. This pressure will be set as the basis for static structural analysis where the deformation behaviour of the rotor will be studied using finite element analysis techniques and structural integrity of rotor geometry and material will be discussed.

## 5.6 Static Structural Analysis

After completion of CFD analysis, the CFD Post Module was linked with the static structural analysis module to directly couple the output conditions of the CFD module as input load parameters for the static structural analysis. The same mesh was also imported was imported in the structural analysis domain and material properties were described for the chosen material.

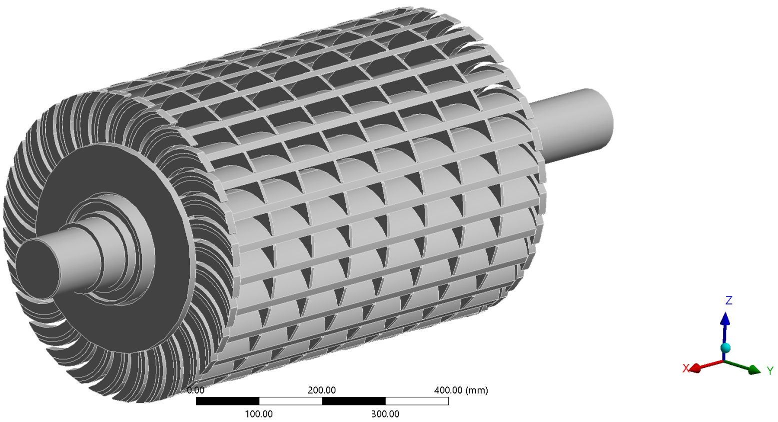


FIGURE 5.10: Meshed model for structural analysis.

Name selection was created for the rotary vane surfaces to apply the load on the surface of the vanes by selecting only cross-flow rotor vanes. Rotational velocity on the component of the x-axis with the value of 484 rpm.

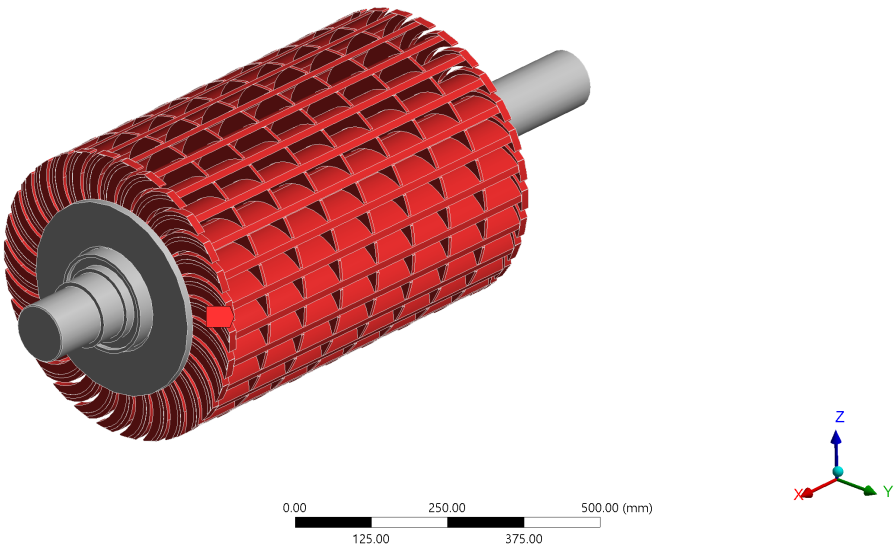


FIGURE 5.11: Rotational velocities applied at the face of rotor vanes.

Cylindrical supports were applied in the below-shown positions. The axial direction was set as free while all others were set as fixed.

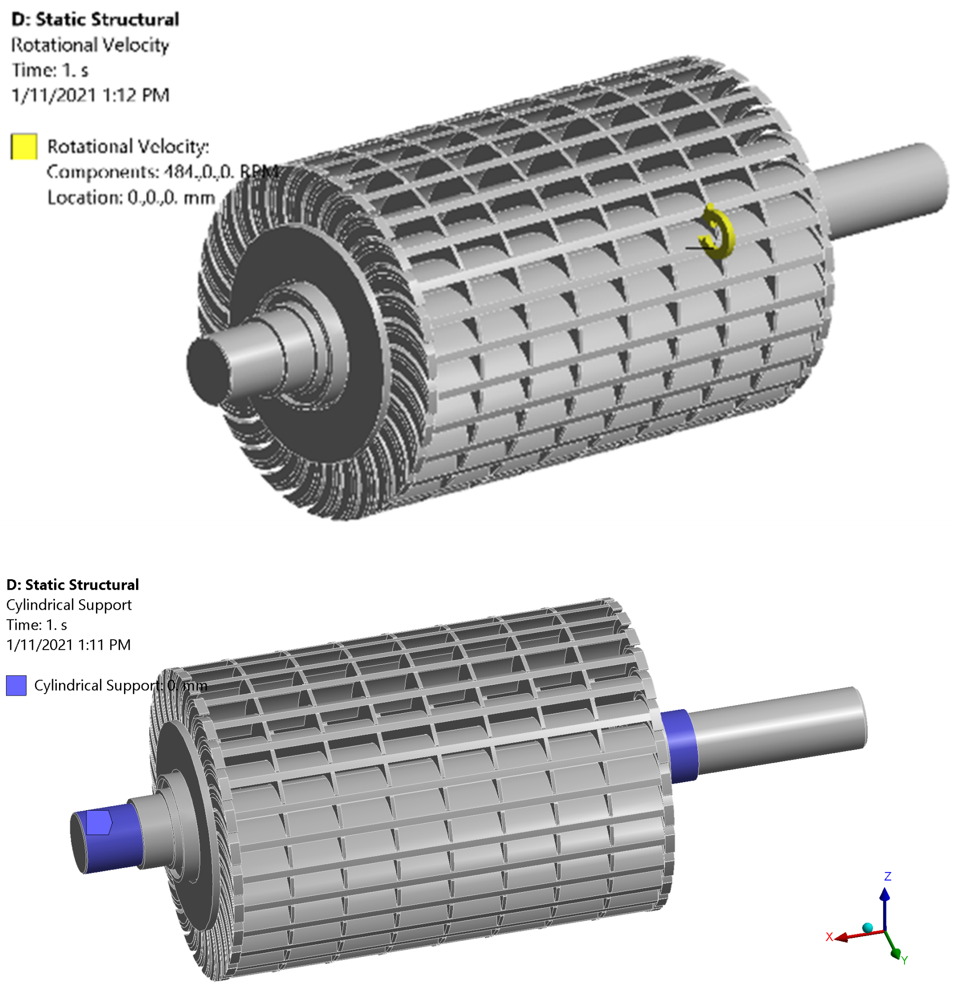


FIGURE 5.12: Constraints defined on CFT rotor.

Now for application of the loads imported from the computational fluid dynamics module of Ansys to the static structural module we first define the module with boundary conditions for fixed support using edge selection option. Pressure load was applied to the rotors face by selecting load feature description. The boundary conditions of structures support and load after being defined successfully were used as the basis to develop solutions for total deformation of the system and directional deformations produced in the system. These deformations were later used to produce the outputs for total deformation reports and plots by using Ansys.

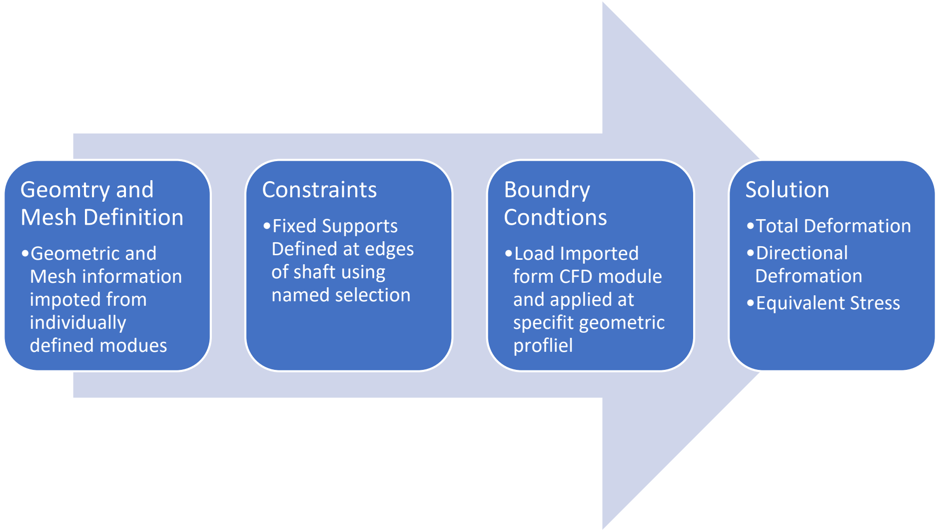


FIGURE 5.13: Process outline for structural analysis.

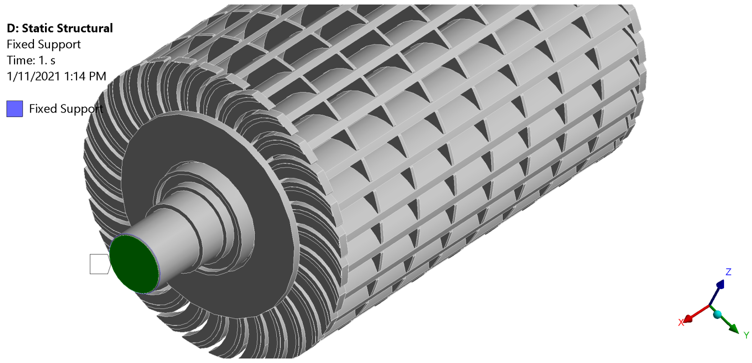


FIGURE 5.14: Load coupling from CFD to static structural.

The pressure distribution from the imported pressure is shown here in the figure below.

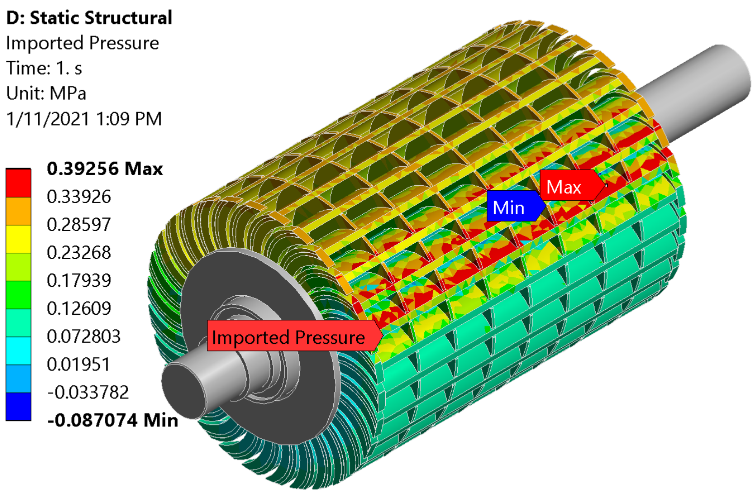


FIGURE 5.15: Imported pressure distribution on the rotor.

## 5.7 Static Structural Results

The following results were obtained after solving for structural analysis under the defined load conditions. The parameters allocated for output were Von-Misses stress and total deformation of the rotor and the results are discussed and total deflection and equivalent stress plots across the length of the turbine shaft under a pre stress condition while rotating at 484 rpm's are shown here.

- The maximum deflection observed on the rotor geometry is 0.07 mm for SS41,0.20 AL1050 and 0.09 mm for Ni-Cr Alloy.
- The maximum stress produced by the applied loads is 40 MPa for SS41, 39 MPa for AL1050 and 40.3 Mpa for Ni-Cr Alloy.
- The produced deflection and stress are quite low as compared to the rotor geometry and material chosen, hence indicating that the design is safe from a structural point of view.

Numerical analysis was performed for a number of candidate materials in order to compare them on the basis of produced equivalent stresses and accompanying deflections produced with their respective yield strength values while keeping in mind their affordability and local availability.

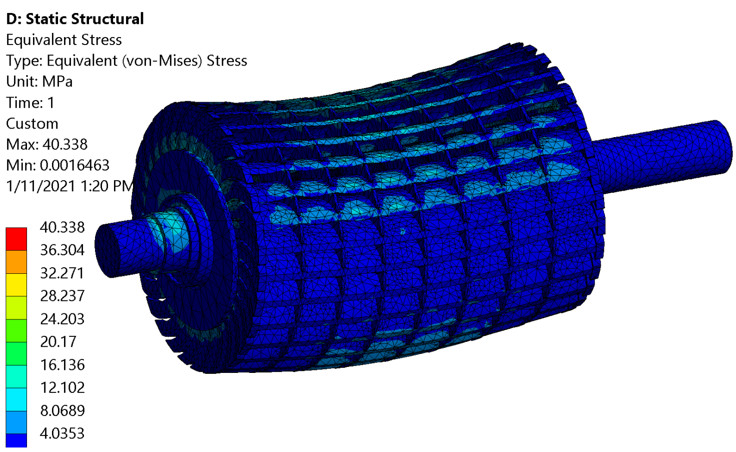


FIGURE 5.16: Von-Misses stress (40.3 MPa) for SS-41.

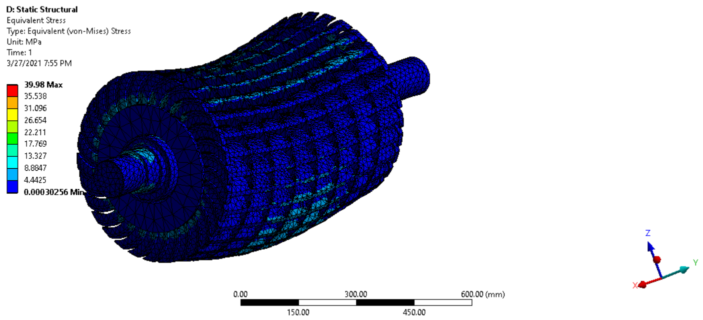


FIGURE 5.17: Von-Misses Stress (39.98 MPa) for Al-1050.

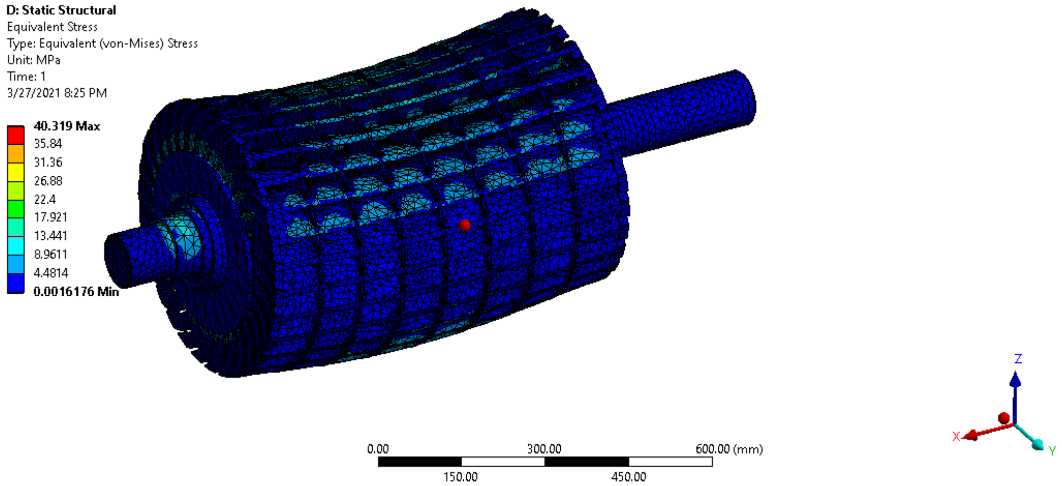


FIGURE 5.18: Von-Misses Stress (40.319 MPa) for Ni-Cr.

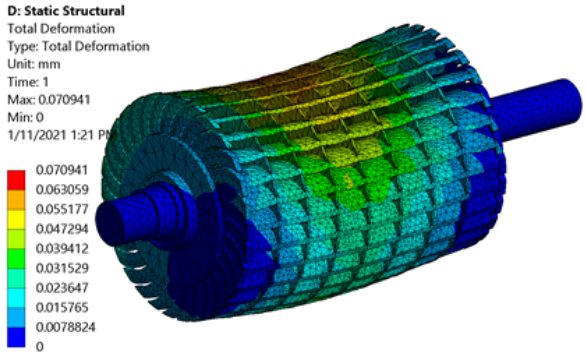


FIGURE 5.19: Total Deformation (0.071 mm) SS-41.

## 5.8 Modal and Harmonic Analysis

Modal analysis is the analysis of the limits of the system, in other words, it studies the limits of a system under certain conditions such as amplitude or vibrations, etc. The model analysis deals with the phenomena of the natural frequency of the system, these natural frequencies of the system are the points at which are present for every object at these frequencies the system can transfer energy from one form to another with minimal losses as the frequency of the system approaches the natural or resonant frequency of the system the amplitude of the system reaches to infinity [54]. The modal analysis helps us in identifying at what frequency the given system will exhibit the maximum amplitude.

The outputs of the static structural analysis were imported to the modal analysis module of Ansys software and the Pre-Stress static structural option was selected as the input conditions were already linked from the static structural module.

The number of max modes under modal analysis settings was set to 4 and under the solutions tab total deformation was selected as the outcome criteria. After defining the criteria, the system was solved and the following results were obtained to find the natural mode shapes and corresponding frequencies of our system during free vibration when the rpm for the system were defined at 484. As per the input parameter, the natural frequency of our system comes out to be 8.07 Hz, and correspondingly the following 4 modal shapes were obtained.

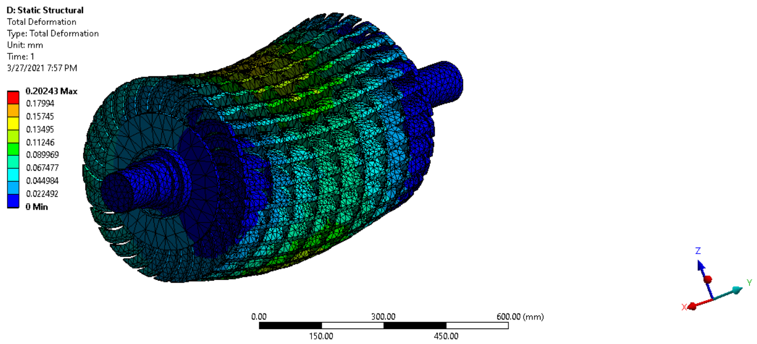


FIGURE 5.20: Total Deformation (0.202 mm) Al-1050.

From the modal analysis, the following natural frequencies were obtained. Our turbine design operates at 484 rpm with its respective frequency being 8.07 Hz. It can be seen here that there is no cross-over resonance at low order of excitation levels and it is clear that there is no overlapping of the runner's resonant frequency with the operating frequency of the runner thus making our design the same for operation.

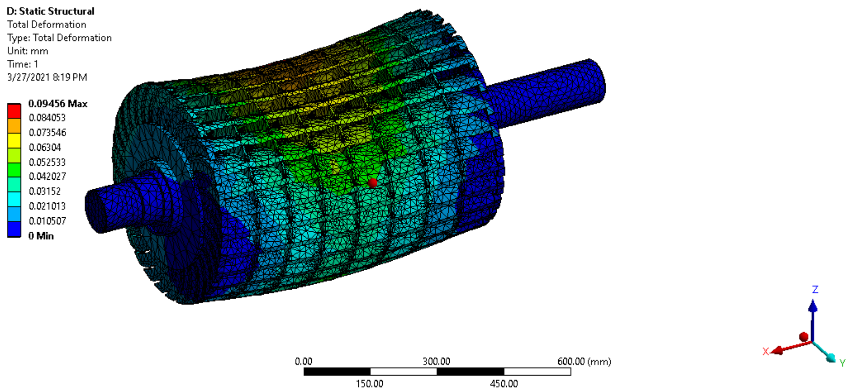


FIGURE 5.21: Total Deformation (0.09 mm) Ni-Cr.

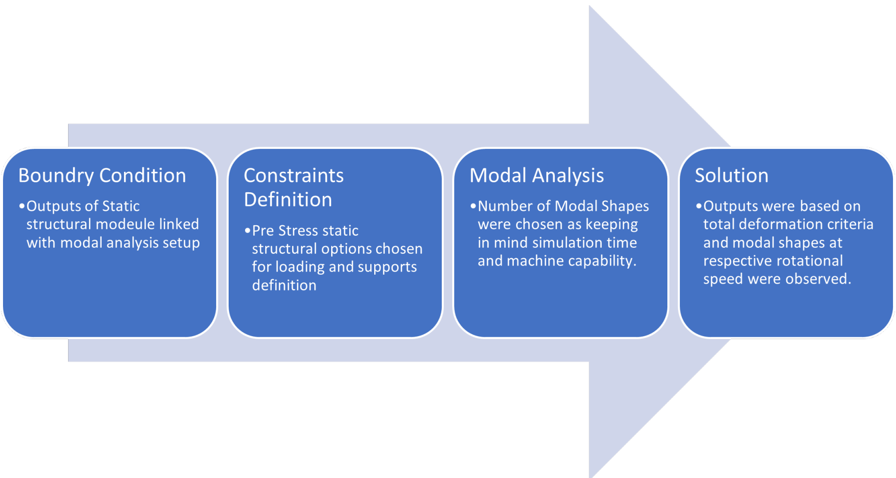


FIGURE 5.22: Process outline for modal analysis.

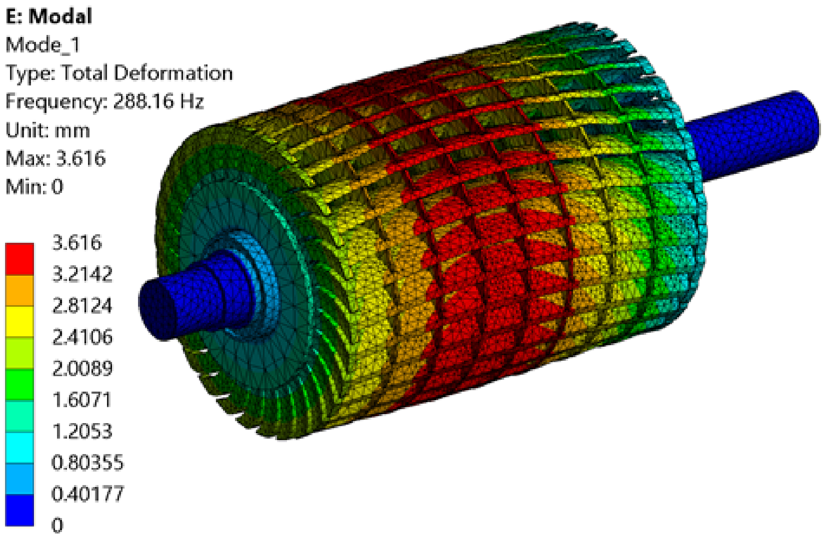


FIGURE 5.23: Mode 1 shape at frequency of 288.16 Hz (SS41).

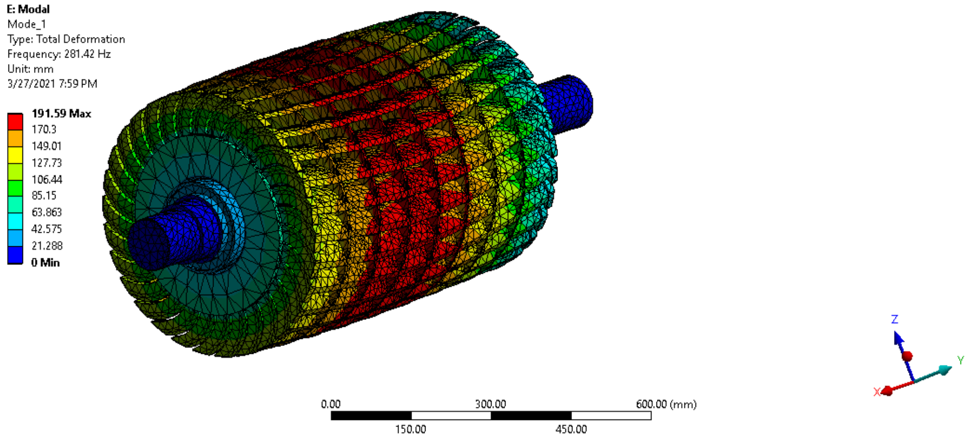


FIGURE 5.24: Mode 1 shape at frequency of 281.42 Hz (Al-1050).

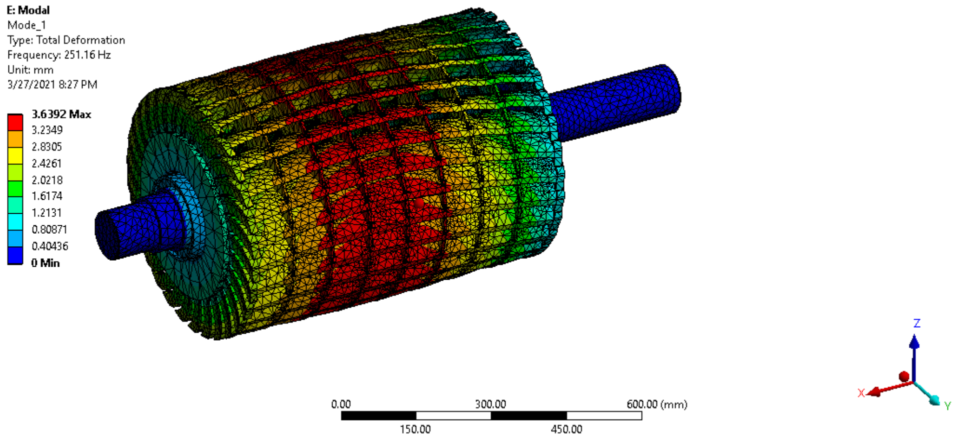


FIGURE 5.25: Mode 1 shape at frequency of 251.16 Hz (Ni-Cr).

## 5.9 Parametric Analysis of Modal Shapes

The previous modal analysis provides us with natural frequency for 4 modal shapes for the 484 RPMs. Now for requirements of modal shapes for a range of different rpm parametric analysis is to be performed. Parametric Analysis in Ansys provides an excellent tool to study the outcomes of an objective function with a given range of input variables [55]. Outcomes with different modifications of the input parameters can be quickly analyzed by employing this analysis.

For evaluation of our design based on harmonic response static structural were linked with the input definition using the parameter option where the output parameter were obtained by defining frequency range from the modal analysis as

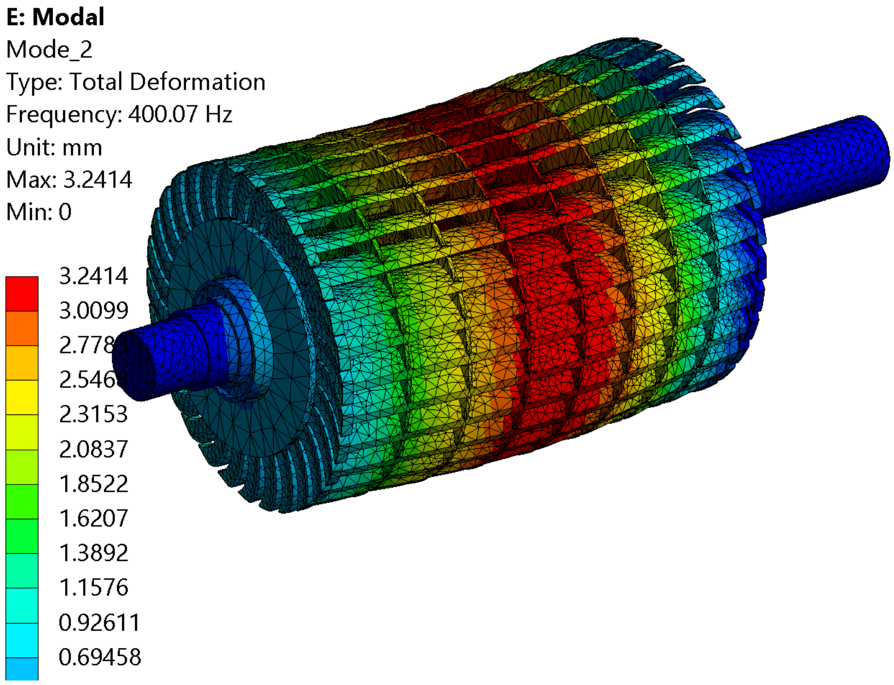


FIGURE 5.26: Mode 2 shape at frequency of 400.07 Hz (SS-41).

the initiator for the parametric analysis. Parametric module was used to define the set of design points in terms of rotational velocities for which respective modal frequency were to produce as an output.

## 5.10 Harmonic Analysis

Now the outputs of the Modal Analysis were linked with the harmonic analysis module. Harmonic response analysis uses the Frequency Response Plot (FRF) where it plots the stress against a different range of frequencies. Modal analysis and harmonic response analysis to predict the behavior of a given structure part or assembly under vibrations and we can also evaluate the critical frequencies where our structure or part may sustain damage. In other, we can use harmonic response analysis to study under sinusoidal changing load the dynamic response of the system. In our case, the cyclic load was applied as momentum about the x-axis component of rotation for the range of frequencies observed in the modal analysis of the system.

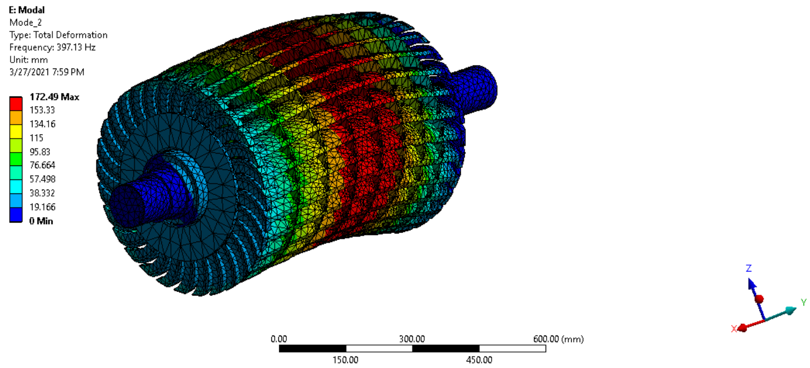


Figure 5.27: Mode 2 shape at frequency of 397.13 Hz (Al-1050).

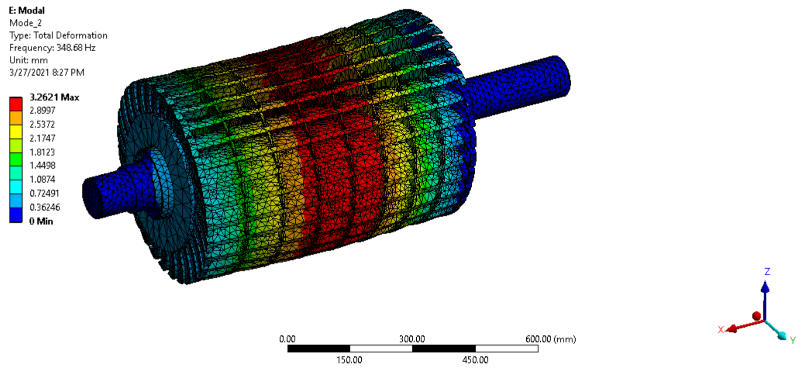


Figure 5.28: Mode 2 shape at frequency of 348.68 Hz (Ni-Cr).

To harmonic analysis was based on the outcomes of modal analysis solution where these outcomes were transferred to the harmonic response module. After successfully coupling modal outcomes with harmonic module the frequency range of 250 Hz ∼500 Hz was defined for harmonic analysis with solution interval was kept as 10 this range was deduced based on results of modal solutions. Momentum offset was applied in order to produce harmonic response under loaded conditions and was applied at the named selection defined as Rotor_Vane_Faces. Offset of 1000 N.mm in direction of X-Component applied as boundary condition and stress and deformation plots obtained.

After execution of harmonic analysis for the frequency range of 250-500 Hz the amplitude response in both mm and Mpa has been plotted. And it can be seen that excitation in the system was only observed near 425 Hz which is way far off from our operating frequency, thus making our design safe for operation in these operating conditions.

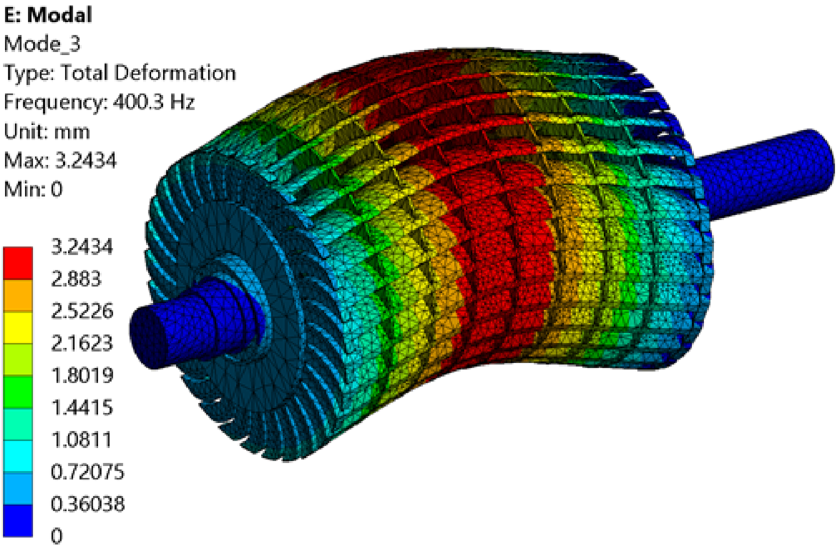


FIGURE 5.29: Mode 3 shape at frequency of 400.3 Hz (SS-41).

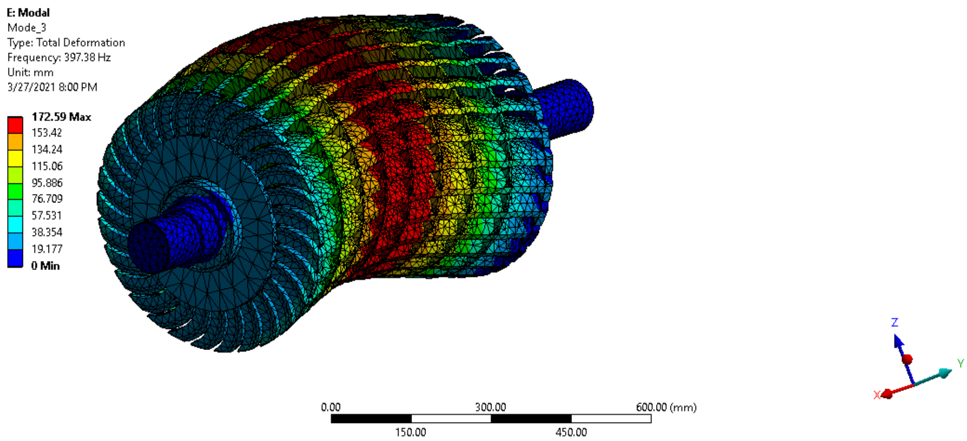


FIGURE 5.30: Mode 3 shape at frequency of 397.13 Hz (Al-1050).

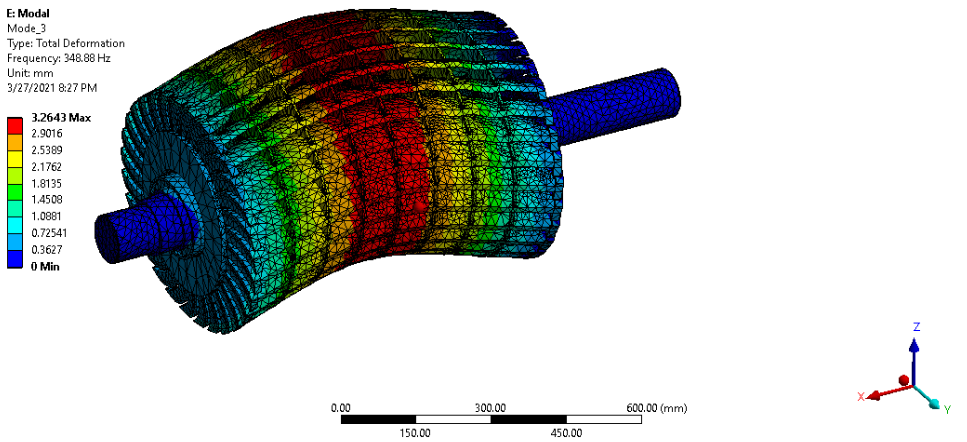


FIGURE 5.31: Mode 3 shape at frequency of 348.88 Hz (Ni-Cr).

Table 5.1: Natural frequencies comparison at 3 modal Shapes at 484 rotational velocity.

| **Modal Shape** | | **1** | **2** | **3** |
|---|---|---|---|---|
| Natural Frequency | SS41 | 288.16 | 400.7 | 400.3 |
| | AL1050 | 281.42 | 397.13 | 397.38 |
| | Ni-Cr | 251.16 | 346.68 | 348.88 |

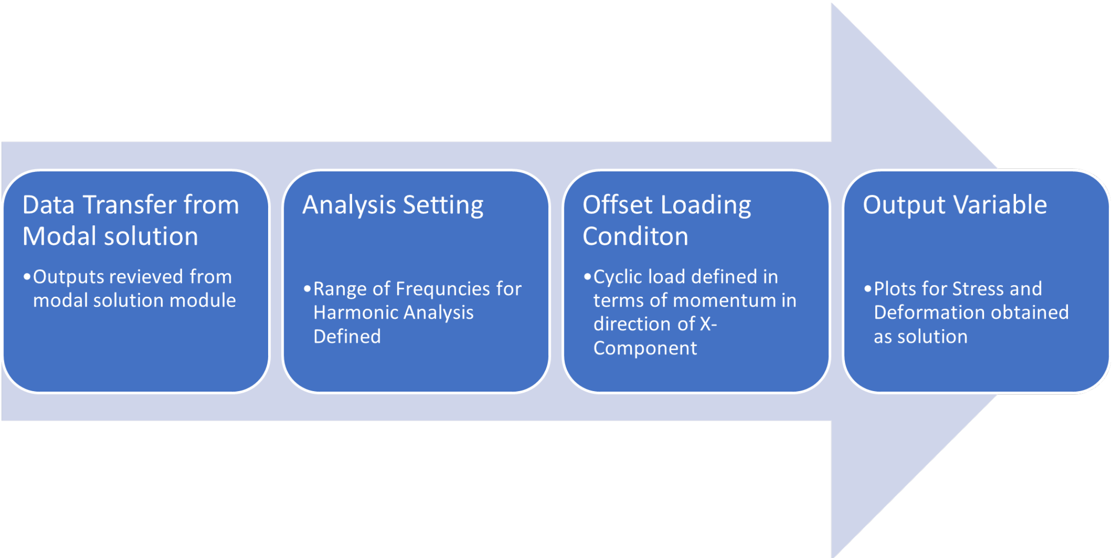


Figure 5.32: Process outline for harmonic analysis.

The materials were selected based on strength and economic impact on manufacturing of cross flow turbine. The material was chosen for the design keeping in view that minimal stresses and deformation will be produced during the operation of the turbine as it faces pressures loads of jets of water striking on the surface of the rotor blades. The choice of materials was made from categories of metal keeping in mind the practicality of minimal mass while keeping in mind the strength of the blade material. The below tables share the summary of chosen materials for the manufacturing of cross flow turbine rotor's where comparison is made based on numerical simulations where different materials where subjected to same set of input condition's and the results produced are tabulated as under. The choice of SS41 was made based on its ease of machining and manufacturing while keeping in minds its readable availability and cost. Material Al1050 though light weight in

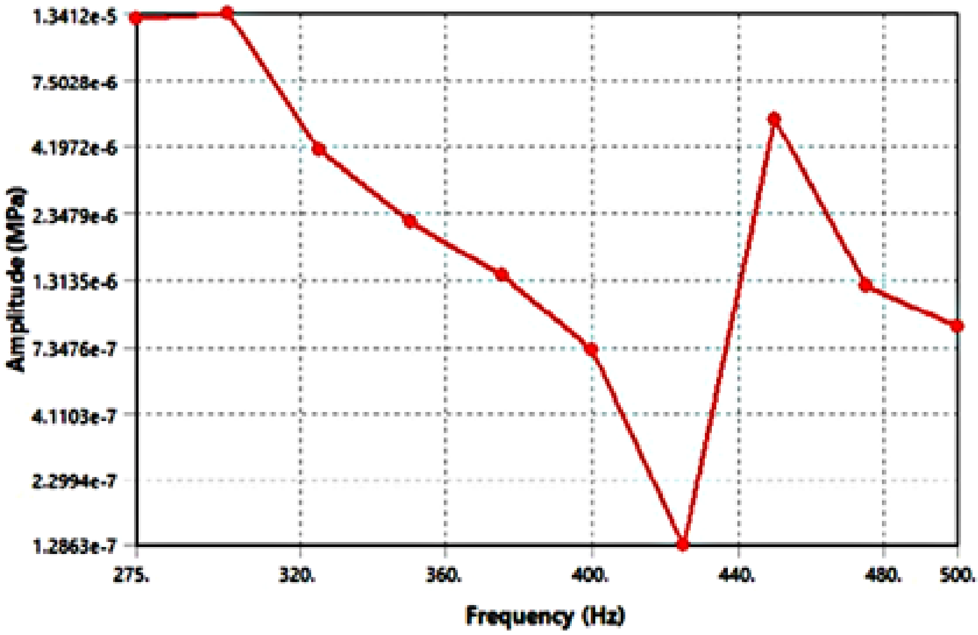


FIGURE 5.33: Frequency response i.e., frequency vs. pressure distribution (SS-41).

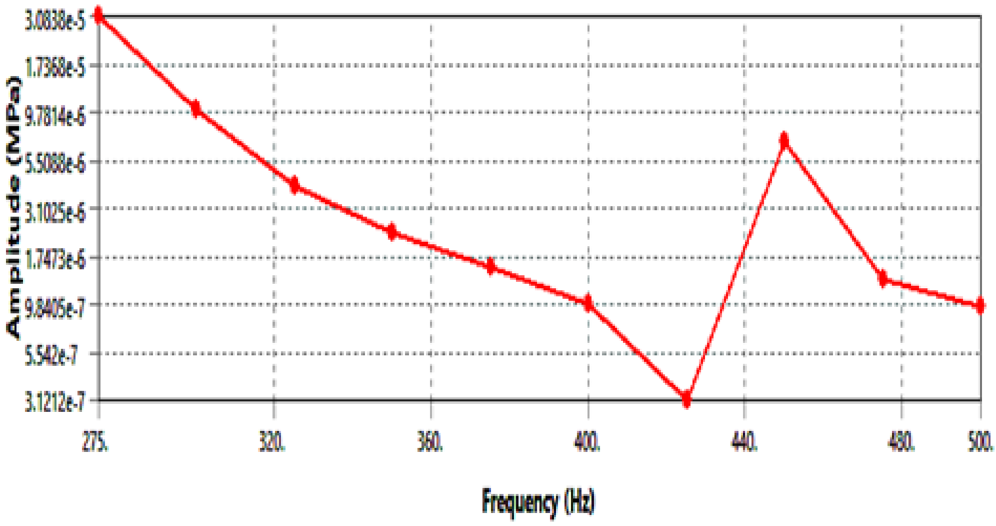


FIGURE 5.34: Frequency response i.e., frequency vs. pressure distribution (Al-1050).

construction while still maintain its mechanical strength proves to be hard to handle while manufacturing and fabricating and its cost and availability also hinders its readable use in local manufacturing setups. The Ni-Cr alloy proves difficult to source and its economic costs renders it unfeasible. Thus the manufacturing of cross flow turbine was carried out using stainless carbon steel SS41.

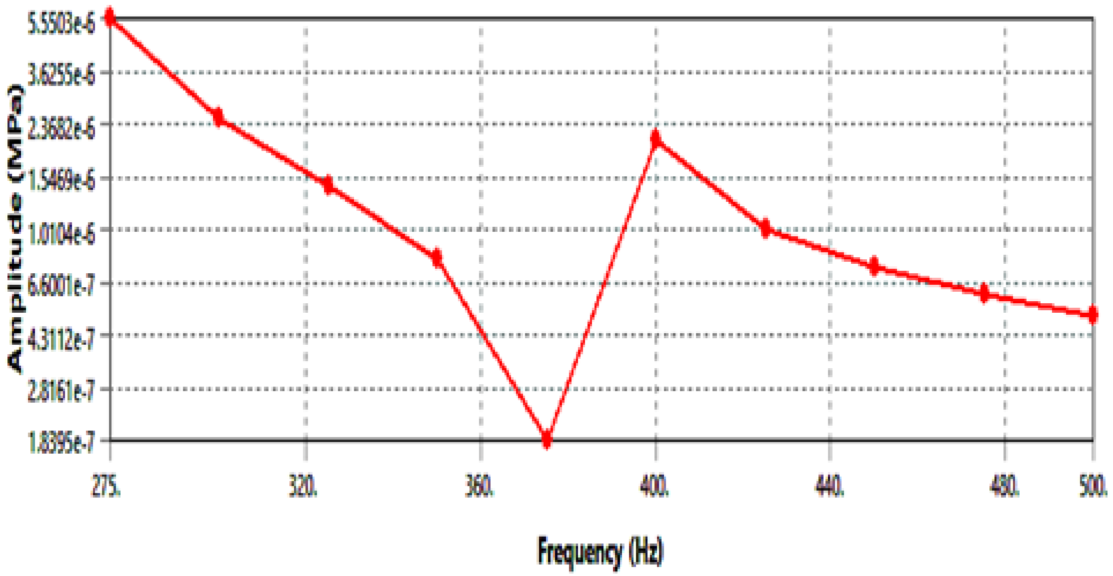


FIGURE 5.35: Frequency response i.e., frequency vs. pressure distribution (Ni-Cr).

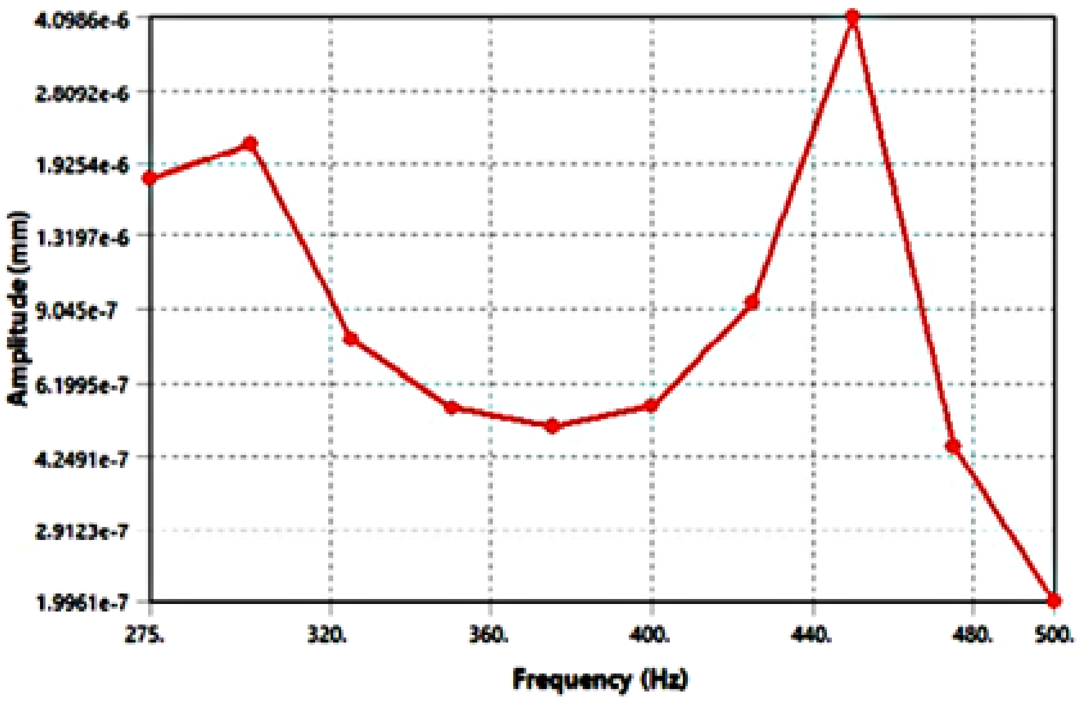


FIGURE 5.36: Frequency response i.e., frequency vs. deformation (SS-41).

Summary of numerical analysis and their results for candidate materials is presented below using parametric analysis to where natural frequencies at respective model shapes is compared for different number of rotational velocities.

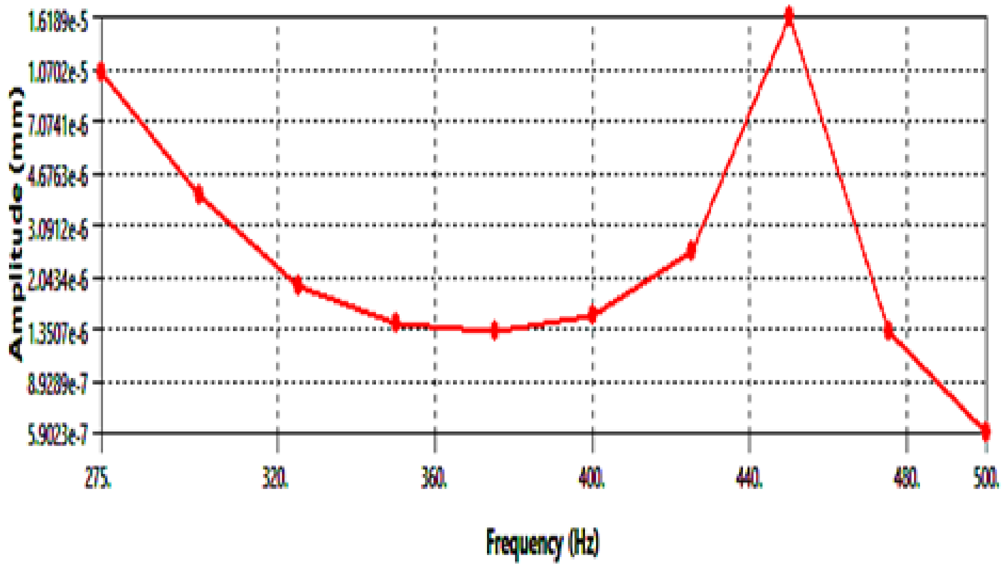


FIGURE 5.37: Frequency response i.e., frequency vs. deformation (Al-1050).

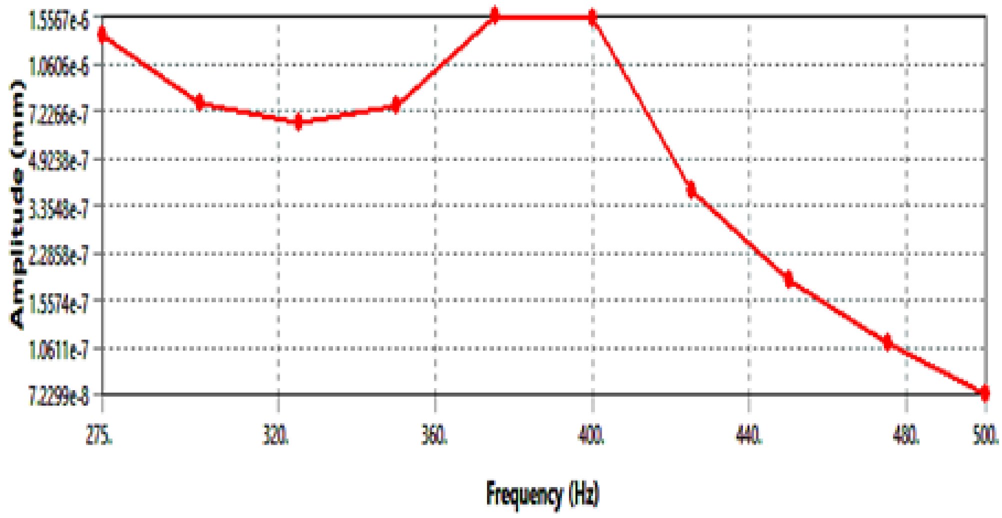


FIGURE 5.38: Frequency response i.e., frequency vs. deformation (Ni-Cr).

TABLE 5.2: Comparison of range of rotational velocity with their modal natural frequencies and for different materials.

| Sr. | RPM | Natural Frequencies Mode 1 | | | Natural Frequencies Mode 2 | | | Natural Frequencies Mode 3 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Material | | SS41 | AL1050 | Ni-Cr | SS41 | AL1050 | Ni-Cr | SS41 | AL1050 | Ni-Cr |
| 1 | 484 | 288.16 | 281.42 | 251.15 | 400.07 | 397.12 | 348.67 | 400.3 | 397.38 | 348.8 |
| 2 | 400 | 288.16 | 281.42 | 251.15 | 400.09 | 397.14 | 348.70 | 400.32 | 397.40 | 348.90 |
| 3 | 500 | 288.16 | 281.42 | 251.15 | 400.06 | 397.12 | 348.67 | 400.29 | 397.38 | 348.87 |
| 4 | 550 | 288.16 | 281.42 | 251.15 | 400.05 | 397.10 | 348.65 | 400.28 | 397.36 | 348.85 |
| 5 | 600 | 288.16 | 281.42 | 251.15 | 400.03 | 397.09 | 348.63 | 400.26 | 397.34 | 348.8 |
| 6 | 800 | 288.16 | 281.42 | 251.16 | 399.95 | 397.01 | 348.54 | 400.18 | 397.26 | 348.74 |

Table 5.3: Comparing deflection and stress of different candidate materials with their mode frequencies and Cost/Kg.

| **Sr. No** | **Material** | **Deflection (mm)** | **Stress (Mpa)** | **Yield Strength (Mpa)** | **Mode 1** | **Mode 2** | **Mode 3** | **Cost/Kg ($)** |
|---|---|---|---|---|---|---|---|---|
| 1 | SS41 | 0.07 | 40 | 250 | 288.16 | 400.7 | 400.3 | 0.67 |
| 2 | Al1050 | 0.2 | 39 | 85 | 281.42 | 397.13 | 397.38 | 8.50 |
| 3 | Ni-Cr | 0.09 | 40.3 | 330 | 251.16 | 346.68 | 348.88 | 30.70 |

# Chapter 6

# Experimental Setup and Evaluation

In this Chapter, the manufactured design of the cross-flow turbine was set up to evaluate its performance parameters and gather experimental data. The experiment is designed to evaluate the design of the cross flow turbine along a range of flow rates coming from the reservoir sources. The incoming flow rate is to be varied by use of a gate valve and out coming rpm's are to be tabulated at the turbine rotor.

To install the cross-flow turbine for experimental evaluation the following arrangements were made at the test site.

- For the water source, the water supply tank with a height of 25 m was selected.
- The 8-inch pipeline was connected to the intake penstock of the cross-flow turbine.
- To control the flow of water 8-inch gate valve was installed on the flow pipe.
- Foundation was laid to secure during operation using the base frame installed at the bottom of the support structure of the turbine.

- Water Channel to take the water away from the exit of the turbine housing was laid which returned the water to the underwater tank where it would get recirculated to the overhead supply tank using pumps.

The Gate valve is used to control the flow of water from the outlet of the overhead water storage tank, the inlet pipeline was connected to the inlet of the penstock using a special design adapter that would gradually increase the area on the inlet pipe to match the inlet size of the penstock nozzle, both of these connections were made using slip-on flanges made of Steel plates.

FIGURE 6.1: Cross-flow turbine setup for performance analysis.

For this experiment, the rotor shaft was a couple with a pulley and belt system. The weight of the pulley was kept as such that it also acted as a flywheel of the assembly thus regulating and stabilizing the output of power provided to the generator from the cross-flow turbine. The rotor shaft of the power was coupled with a flywheel pulley with the help of flexible coupling which is optimized for use in locations where axial and torsional stress is applied in the system.

FIGURE 6.2: Turbine pulley and rotor shaft coupling.

The test was based on the calculation of the flow rate of water coming from the inlet pipe to the cross-flow turbine, the measurement of revolution per minute of the turbine and generator pulley, and the adjustment of guide vane positions with the help of the guide vane regulator valve. The guide vane was adjusted in three different positions by keeping the flow rate constant. Then the same experiment was repeated and different values of RPMs were obtained for different values of flow rates received to the turbine rotor.

TABLE 6.1: Guide vane positions at flow of 0.793 cubic meter.

| **Sr. No.** | **Head (m)** | **Guide Vane Position** | **Turbine RPM** | **Generator RPM** |
|---|---|---|---|---|
| 1 | 25 | + 45 degree | 440 | 1660 |
| 2 | 25 | Mean Position | 480 | 1920 |
| 3 | 25 | -45 degrees | 450 | 1772 |

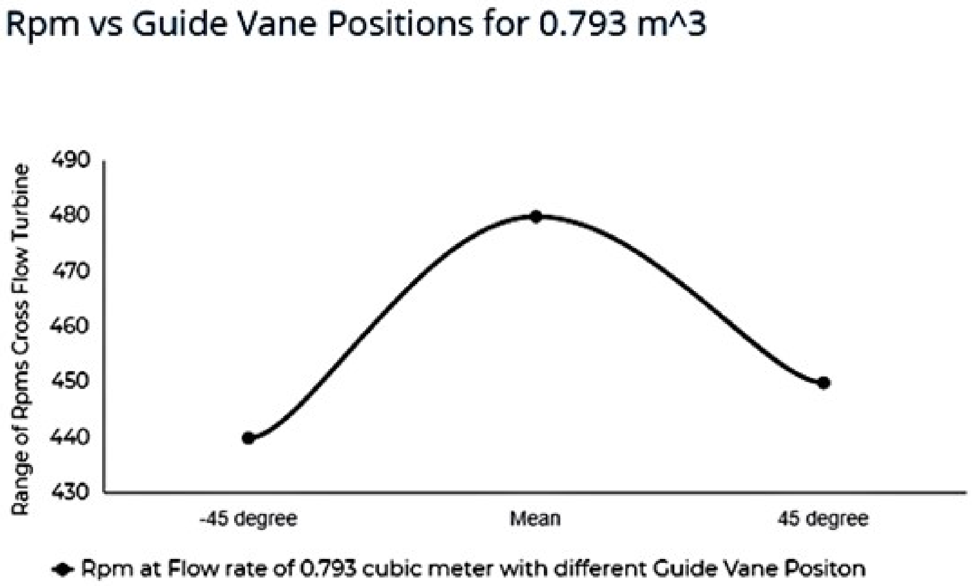


FIGURE 6.3: Graph showing rpm against 3 positions of guide vane at 0.793 cubic meter.

TABLE 6.2: Guide vane positions at flow of 0.61 cubic meter.

| Sr. No. | Head (m) | Guide Vane Position | Turbine RPM | Generator RPM |
|---|---|---|---|---|
| 1 | 25 | + 45 degree | 320 | 1660 |
| 2 | 25 | Mean Position | 370 | 1920 |
| 3 | 25 | -45 degrees | 335 | 1772 |

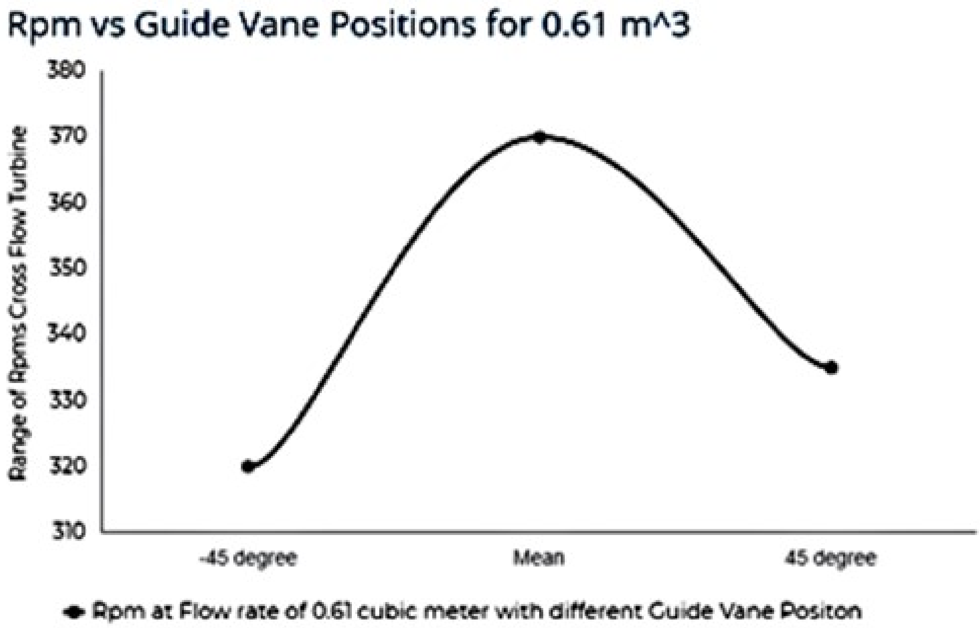


FIGURE 6.4: Graph showing rpm against 3 positions of guide vane at 0.61 cubic meter.

Table 6.3: Guide vane positions at flow of 0.44 cubic meter.

| Sr. No. | Head (m) | Guide Vane Position | Turbine RPM | Generator RPM |
|---|---|---|---|---|
| 1 | 25 | + 45 degree | 205 | 820 |
| 2 | 25 | Mean Position | 254 | 1016 |
| 3 | 25 | -45 degrees | 215 | 860 |

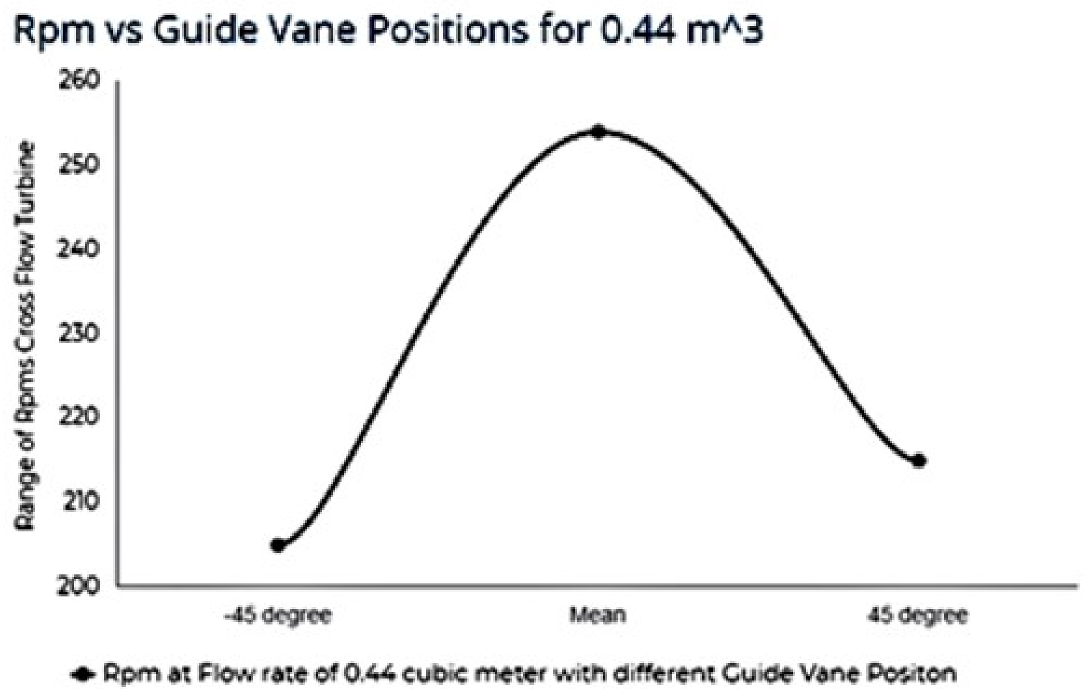


Figure 6.5: Graph showing rpm against 3 positions of guide vane at 0.44 cubic meter.

Table 6.4: Guide vane positions at flow of 0.37 cubic meter.

| Sr. No. | Head (m) | Guide Vane Position | Turbine RPM | Generator RPM |
|---|---|---|---|---|
| 1 | 25 | + 45 degree | 185 | 740 |
| 2 | 25 | Mean Position | 230 | 920 |
| 3 | 25 | -45 degrees | 194 | 776 |

The above-tabulated data shows the variation in rpm at the turbine shaft and transmitted rpm towards the generator shaft for the given range of flow rates and varying positions of guide vane.

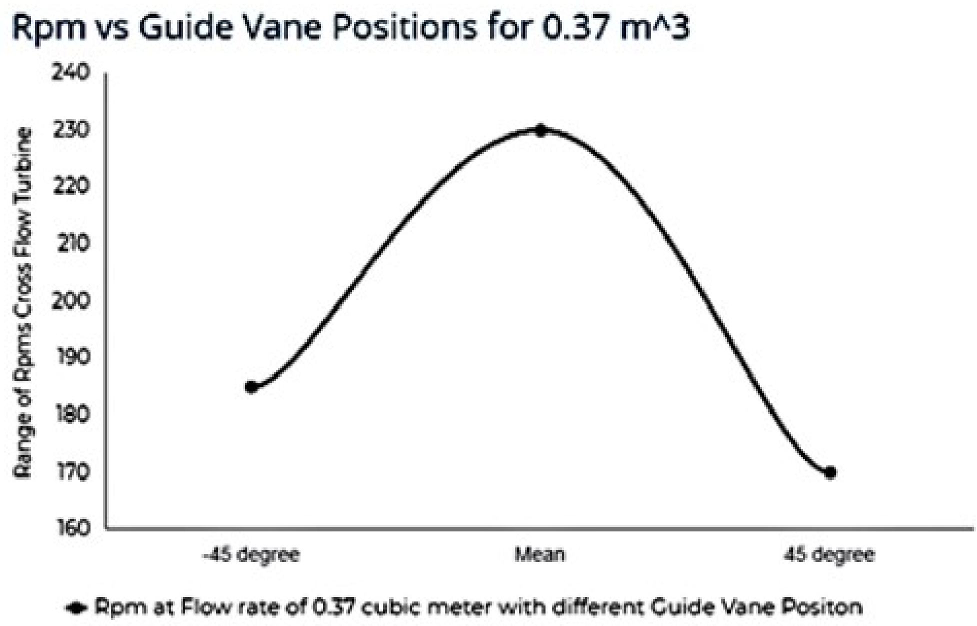


FIGURE 6.6: Graph showing rpm against 3 positions of guide vane at 0.37 cubic meter.

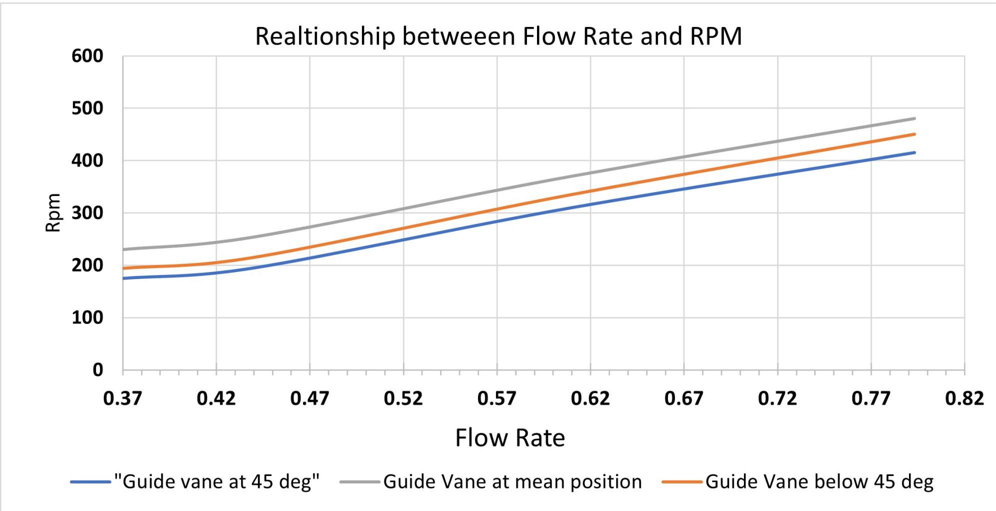


FIGURE 6.7: Relationship between flow rate and RPM at different guide vane angles.

# Chapter 7

# Results and Discussion

This section discussed the results obtained against the experiments and simulation performed for the developed design of our cross flow turbine. These discussion are made from the contours and plots obtained from computational and structural evaluation of the rotor turbine under different flow conditions. The observation tabulated during experimentation at test sites are translated and summarized to justify design expectations and results achieved.

In this dissertation, the complete design of a cross-flow turbine has been presented by keeping in view ease of manufacturing and transportability in mind. The design made use of materials and technology readily available in local manufacturing concerns in Pakistan. Details were provided for complete formulations of turbine variables i.e number of blades, runner outer and inner diameter, the velocity of flow from nozzle, angle subtended by nozzle arc, the calculation for specific rotational velocity for the rotor, load on the turbine shaft, and diameter of the penstock, etc. The material used for manufacturing most elements of the turbine was chosen as SS41 mild steel because of its mechanical, machining, and welding properties. The design developed was first evaluated in Ansys 19 before its manufacturing was initiated the material properties were defined and static structural analysis was performed on our design and designated Rotational velocity the maximum produced load on our rotor was recorded to be 40 Mpa which in the safe zone and is much lower than the yield strength of SS41 which is recorded to be 240

Mpa. The maximum deflection observed on the rotor was observed to be 0.07 mm this amount of deflection seems minimal in comparison to the size of our turbine thus indicating the structural stability of our turbine during operation. The same computational study was performed for AL1050 and Ni-Cr alloy where the maximum stress observed at the rotor was recorded to be 39 Mpa at 0.2 mm deflection where the yield strength of the material is 69 Mpa for Al-1050 and 40.3 Mpa at 0.09 mm deflection was recorder for Ni-Cr where the yield strength is 330 Mpa. Computational fluid dynamics analysis was performed for the same design by using the CFX module of Ansys.

Flow behavior was observed which matched with our theoretical prediction and the produced load pressure with the flow rate of incoming water. This was used as the loading condition for the static structural analysis which made our loading conditions for static structural analysis realistic. The conversion of kinetic energy to potential energy was achieved twice in the presented design of a cross flow turbine first through direct contact at the entry and secondly where the opening is available between the blades of the turbine runner. As per the recommendation from literature reviews the selection of number of blades was made to be at 30 No's this selection for higher number of blades resulted in a smooth velocity profile for the rotor blades and provided with optimum dissipation of kinetic energy towards production of power through the turbine shaft .The contours plotted for the fluid dynamic analysis for its velocity and pressure profiles. It was clear from the flow simulation that the velocity of the stream increased within the inlet nozzle and before making contact with the rotor blade. The velocity started decreasing after interaction with the rotor and the turbine casing before its beings its lowest at the turbine exit. Similarly it can be seen that the pressure dropped in the turbine in two stages the first drop in pressure was during the first stage of interaction between the flowing water and the turbine blades where it's transferred some of its kinetic energy to drive the blades of the turbine. The second loss of pressure was when the water again interacted with the turbine blades. At the second stage where it transferred its remaining energy and decreased in pressure contours shows the same.

In the second stage of our analysis modal and harmonic analysis was performed on our design which would predict the operation stability of our turbine at different values of RPM's. The modal analysis of the design provided us with 4 modal shapes of our design at frequencies of 288.16, 400.7, 400.3 and 446.18 Hz these modal frequencies were way far off from the operating frequency of our system validating our design. After modal analysis, the services of Parametric analysis were utilized to further evaluate the natural frequency and modal shapes of our system at a range of 484, 400, 500, 550, 600, and 800 RPM's the obtained data were tabulated in the respective section and pointed towards the safety of our design. Harmonic analysis of our system under cyclic loading of 1000 kN was performed in the frequency range of 250-500 Hz the amplitude response in both mm and Mpa has been plotted. And it can be seen that excitation in the system was only observed near 446 Hz which is way far off our operating frequency, thus making our design safe for operation in these operating conditions.

The experimental evaluation was conducted at the suitable installation site where arrangements for water supply and flow rate controls were established. Evaluation of our design was performed by connecting it with an overhead water tank the experiment was designed to record output RPMs of the turbine and generator pulley by variations of the flow rate of water coming from the water to tank in combination with change of angles of guide vanes by use of guide vane regulator. The data gathered during our experiments was recorded and tabulated and is presented in our work with the combination of pulleys designed for the system was able to achieve the rpm's required for the system at operating rpm and flow rate of the turbine. Rpm value of 480 was achieved at flow rates of 0.793 contrary to the designed value of 484 rpm this variation can be attributed to losses that might arise due to mechanical our variation in manufacturing from design. The minute difference that might arise between the calculated values and CFD may be due to the assumption of losses is not usually considered for flow analysis further refinement of boundary conditions that incorporated much more detailed conversion of energy by considering mechanical behaviors of components will provide much more accurate depiction of these types of flows in turbines.

## 7.1 Cost Analysis

In manufacturing of the discussed cross flow turbine using indigenous resources one of the major initial investment is incurred for electromechanical equipment. The cost of these electromechanical systems depends on the proposed capacity and head of selected site particularly for turbines that relay on low heads of water for their operation. Among these systems selection and manufacturing of turbine rotor is one of the main components that establishes the cost. For low head projects construction and erection of power house structure and its auxiliary are major cost driver and is a key factor in total break up of cost that is required for manufacturing. In these low head sites the cost required for development of penstock at water inlet and tail race at the exit is minimal as compared to other high head turbines installation projects. In order to make our choice of turbine cost effective and durable it should be kept in mind that a balance between manufacturing expenditures and expected returns must be managed. Since hydropower system usually have good working efficiencies so innovation in control system such as electronic governors are utilized for efficient use of available flow rates.

For a small hydro power turbine manufacturing and installation venture the planned cost per kW for a cross flow turbine is mostly dependent upon the scope of civil works that is required for site establishment. Since the electro mechanical system consists of control systems, governors, generators and turbines, these components and their costs do not depend on the conditions of the selected site since the contribute a small variation to the total planned cost of the hydropower project. The breakup of material utilized in the manufacturing of presented and summarized below for cost estimation.

### 7.1.1 Estimated Cost Summary of Cross-Flow Turbine

Bill of quantity for major components utilized for the manufacturing of 250 kW cross flow turbine are given along with their unit and total weights respectively.

Table 7.1: Bill of quantity for cross flow turbine.

| P. No. | Drawing Title | QTY | MAT. | U/WT.(Kg) | T/WT.(Kg) |
|---|---|---|---|---|---|
| 1 | BASE FRAME FOR HOUSING | 2 | SS41 | 13.38 | 26.76 |
| 2 | LEVER BOSS | 1 | SS41 | 4.38 | 4.38 |
| 3 | HUB FOR HAND WHEEL | 1 | SS41 | 0.30 | 0.30 |
| 4 | BEARING FLANGE | 1 | SS41 | 1.06 | 1.06 |
| 5 | ADAPTER | 1 | SS41 | 18.18 | 18.18 |
| 6 | BASE FRAME FOR HOUSING | 8 | SS41 | 1.39 | 11.15 |
| 7 | LOCK NUT | 2 | SS41 | 0.81 | 1.62 |
| 8 | LOCKING WASHER | 1 | SS41 | 0.02 | 0.02 |
| 9 | PLATE WASHER | 1 | SS41 | 0.06 | 0.06 |
| 10 | PLATE WASHER | 1 | SS41 | 0.06 | 0.06 |
| 11 | WASHER | 1 | SS41 | 0.06 | 0.06 |
| 12 | INTERMEDIATE DISK | 7 | SS41 | 8.52 | 59.66 |
| 13 | ADAPTER | 1 | SS41 | 5.18 | 5.18 |
| 14 | SPOKES | 4 | SS41 | 0.21 | 0.84 |
| 15 | ADAPTER | 2 | SS41 | 38.00 | 76.01 |
| 16 | ADAPTER | 2 | SS41 | 2.45 | 4.89 |
| 17 | ADAPTER | 2 | SS41 | 1.65 | 3.30 |
| 18 | DRAFT TUBE | 2 | SS41 | 30.05 | 60.10 |
| 19 | DRAFT TUBE | 2 | SS41 | 30.35 | 60.71 |
| 20 | BASE FRAME FOR HOUSING | 12 | SS41 | 0.71 | 8.57 |

| | | | | | |
|---|---|---|---|---|---|
| 21 | DRAFT TUBE | 2 | SS41 | 4.85 | 9.69 |
| 22 | DRAFT TUBE | 2 | SS41 | 5.88 | 11.76 |
| 23 | REINFORCEMENT FLANGE | 2 | SS41 | 12.77 | 25.54 |
| 24 | SIDE DISK | 2 | SS41 | 24.86 | 49.72 |
| 25 | LEVER ARM | 2 | SS41 | 4.79 | 9.58 |
| 26 | BASE FRAME FOR HOUSING | 2 | SS41 | 8.06 | 16.13 |
| 27 | ADAPTER | 2 | SS41 | 9.69 | 19.39 |
| 28 | ADAPTER | 2 | SS41 | 5.50 | 10.99 |
| 29 | BASE FRAME FOR HOUSING | 2 | SS41 | 12.94 | 25.87 |
| 30 | BASE FRAME FOR HOUSING | 2 | SS41 | 8.95 | 17.89 |
| 31 | STOPPOER NUT | 1 | SS41 | 0.59 | 0.59 |
| 32 | BASE FRAME FOR HOUSING | 2 | SS41 | 14.56 | 29.12 |
| 33 | SPINDLE NUT | 1 | SS41 | 2.95 | 2.95 |
| 34 | SHAFT REINFORCEMENT FLANGE | 2 | SS41 | 5.83 | 11.65 |
| 35 | ROTOR FLANGE | 2 | SS41 | 94.16 | 188.32 |
| 36 | DISTANCE RING | 1 | SS41 | 0.05 | 0.05 |
| 37 | DISTANCE RING | 2 | SS41 | 0.63 | 1.26 |
| 38 | DISTANCE RING | 1 | SS41 | 0.17 | 0.17 |
| 39 | HAND | 1 | SS41 | 0.54 | 0.54 |

| | | | | | |
|---|---|---|---|---|---|
| 40 | HAND WHEEL | 1 | SS41 | 3.39 | 3.39 |
| 41 | STOPPER PIPE | 1 | A106-B | 0.81 | 0.81 |
| 42 | ROTOR BLADE | 1 | A106-B | 242.59 | 242.59 |
| 43 | BEARING SLEEVE | 1 | BRONZE | 0.20 | 0.20 |
| 44 | BEARING SLEEVE | 1 | BRONZE | 0.16 | 0.16 |
| 45 | SLIDE BEARING | 2 | BRASS | 0.03 | 0.07 |
| 46 | SPINDLE | 1 | C45 | 5.57 | 5.57 |
| 47 | ROTOR SHAFT | 1 | C45 | 95.86 | 95.86 |
| 48 | GUIDE VANE | 1 | C45 | 164.39 | 164.39 |
| 49 | TUBINE ROTOR ASSY | 1 | C45 | 0.40 | 0.40 |
| 50 | VANE ASSY | 1 | C45 | 0.13 | 0.13 |
| 51 | KEY FOR SPINDLE | 1 | C45 | 0.02 | 0.02 |
| 52 | BEARING COVER DRIVE END | 1 | HT 20-40 | 23.51 | 23.51 |
| 53 | BEARING COVER DRIVEN END | 1 | HT 20-40 | 20.80 | 20.80 |
| 54 | BEARING HOUSING | 2 | HT 20-40 | 78.39 | 156.77 |
| 55 | VALVE FLANGE LEVER SIDE | 1 | HT 20-40 | 44.70 | 44.70 |

| | | | | | |
|---|---|---|---|---|---|
| 56 | VALVE FLANGE BLIND SIDE | 1 | HT 20-40 | 25.79 | 25.79 |

- Turbine Manufacturing Cost = 2.4 Millions
  (All Material arranged utilizing existing stock)
- Estimated Cost of mini HPP per kW = 0.22 Million
  (Inclusive of feasibility cost + fore bay + Penstock + Installation + commissioning)
- Production Cost of 250 Kw @ 0.22 mil per kW = 55 Million
- Annual Operational Hours @ 16 Hours = 5840 Hrs.
- Total Units produced in 5840 hrs = 14,60,000 kWh
- Proposed Total Cost per unit = 10 /-
- Total payback per year = 14.6 Millions
- Pay Back period = 3.8 Years

# Chapter 8

# Conclusion

This chapter summarizes the work done in this study and provides an oversight of the overall intent of design and relates them to the archived results. The decisions taken during the study are evaluated and recommendations are made further improving upon certain design criteria's based on observations made during theoretical and simulation studies. Based on the simulated design manufacturing procedures followed are also evaluated by high lighting room for improvement where applicable.

The future of this technology depends on providing market awareness where environmentally friendly alternatives can be in reach of individuals having low purchasing power. In this presented thesis report a Computational fluid dynamic based simulation is presented which can be used as a basis for design evaluation of any variation of a cross flow turbine. The presented method for turbine analysis showed good results for flow prediction of a turbine rotor witch its interaction with flow of water in a turbine housing. The main drive for this study was based on the idea that in developing regions like Pakistan hydropower power has been historically the cheapest for of energy production and since the focus on renewable energy has been gaining traction as of late and the work done in field of optimizing production and design activities holds key importance.

The flow based analysis using Ansys is a solid argument that flow based prediction for efficiency and structural strength of the turbine can be predicted using this computational models without the need to manufacture costly models and prototypes. One of the main hurdles observed during production work was a lack of skilled power that was familiar with these types of production work in terms of both preparation of assembly, maintenance, and planning activities. High focus is to be given to the production of components of these types of the system through local means and technology.

The objective of developing a design that would be economical and would be practical for production of energy at low head and flow rates was achieved. In the presented design the cross flow turbine was installed with a water head of 25 m and the maximum flow rate of 0.793 $m^3$ to produce power 250 kW at the generator with the runner diameter of 500 mm was utilized. The length of the runner was supported by use of 5 intermediate supporting discs that provided our design with enough strength to withstand the incoming flow of water at 484 rpm.The material used in turbine manufacturing proved safe and the stress produced where in the safe range of operation. A generator consisting of 16 number of synchronous poles was used for the production of energy from the rpm's produced at the rotor shaft. The rpm's produced at the rotor shaft were directly related to the increase of flow rates of water up to the maximum flow rate achievable at site available and were related to the change of the guide vane position at different angles.

Once these types of systems can be efficiently developed, modernized, and produced locally the need for localized standards for quality assurance of these developed systems will arise otherwise uniformity in quality of produced parts, components and systems cannot be assured.

The quality of the turbine construction was improved by selection of materials based on their ability to withstand the environmental and structural impact during their operations. Best industrial practices were adopted to manufacture components where close tolerances were required. This keen significance provided to accuracy of manufactured parts resulted in ease of assembly and re-assembly of turbine components. The use of jig and fixtures for drilling and welding where necessary resulted in accurate assembly of the turbine housing. The use of fixtures for welding significantly reduced the deformation and resulting deviation in the structure. The available manufacturing capacities of workshops and machine tools available throughout Pakistan can modified by developing use of jigs and fixture to be able to produce these turbines on production scale while maintaining the quality standards.

## 8.1 Future Recommendations

In the light of the dissertation presented and the knowledge gathered through literature reviews, manufacturing activates and simulations performed the following recommendations are made for scope for improvement exists.

- Furthermore, modification in the design of cross-flow turbines is the scope of future work where modification can be made to alter the behavior of the flow of water as it interacts with the rotor of the turbine during its second stage. The second stage interaction made smooth can influence the overall efficiency of the system.
- The blade shape can be optimized using different objective function-based approaches in Ansys, Solid work, and Creo Parametric which will further improve efficiency.
- Further research work can be done in the manufacturing of intermediate disks and Blades can be performed in the field of development of materials used to manufacture these components, i.e. the use of composites or Un-Plasticized Polyvinyl Chloride technology.
- The use of a governing system at the guide vane is also proposed in the future where the guide vane angle can is altered as per the incoming flow rate to maximize the utilization of flow rate.

- Further work can be done in the field of production of these systems at smaller scales using reusable materials which can be of great use to the agriculture industry and remote regions.

- Studies need to be performed at arrangements where these types of systems can be installed in combinations with another type of hydraulic power production systems.

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# Appendices

# A-1: Bearing Calculations

In most applications, the load on a radial bearing is composed of axial as well as radial forces. The equivalent bearing load, $P$ is obtained from the following general equation.

$P = XVF_r + YF_a$

Where X and Y are Radial load factor and Axial or Thrust Load actors.

$V$ = Rotation factor = 1.0 For all types of bearing when inner race is rotatong.

Force of jet of water

Gravitational constant, $g$ = 9.81 m/s$^2$

Weight Density, $w$ = 9.81 kN/m$^3$

Force of jet on turbine, $F$ = waV(V-u)/g

Jet Area, $a = L \times S_0$ = 40,205 mm$^2$

Jet Velocity, $V$ = C.(2gH)0.5

Jet Velocity, $V = 21.70$ m/sec

Periphery velocity of runner, $u = 0.5$ VCosa

Periphery velocity of runner, $u = 10.4$ m/sec

Force of jet on turbine, $F = 9.8$ kN

Constant Radial Load, $F_r = 9{,}836.65$ N

Thrust Load, $F_a = 5{,}901.99$ N

$F_a/F_r = 0.60$

$X = 0.56$

$Y = 1.00$

$P = XVF_r + YF_a = 11{,}411$ N

Service factor, $K_s = 1.5$

$P = 17{,}116$ N

$N$ (rpm) = 452.00

Life Years = 5 years

Hours per day = 10

Life in working hours, $L_h = 15{,}000$ hours

The relationship between the nominal life in millions of revolutions, $L$ and the nominal life in working hours, $L_h$ is:

$L = 60NL_h \times 10^{-6} = 407$ millions of rev

The relationship between the basic dynamic capacities, $C$ of a bearing load and its life, $L$ is expressed by:

$L = (C/P)q$

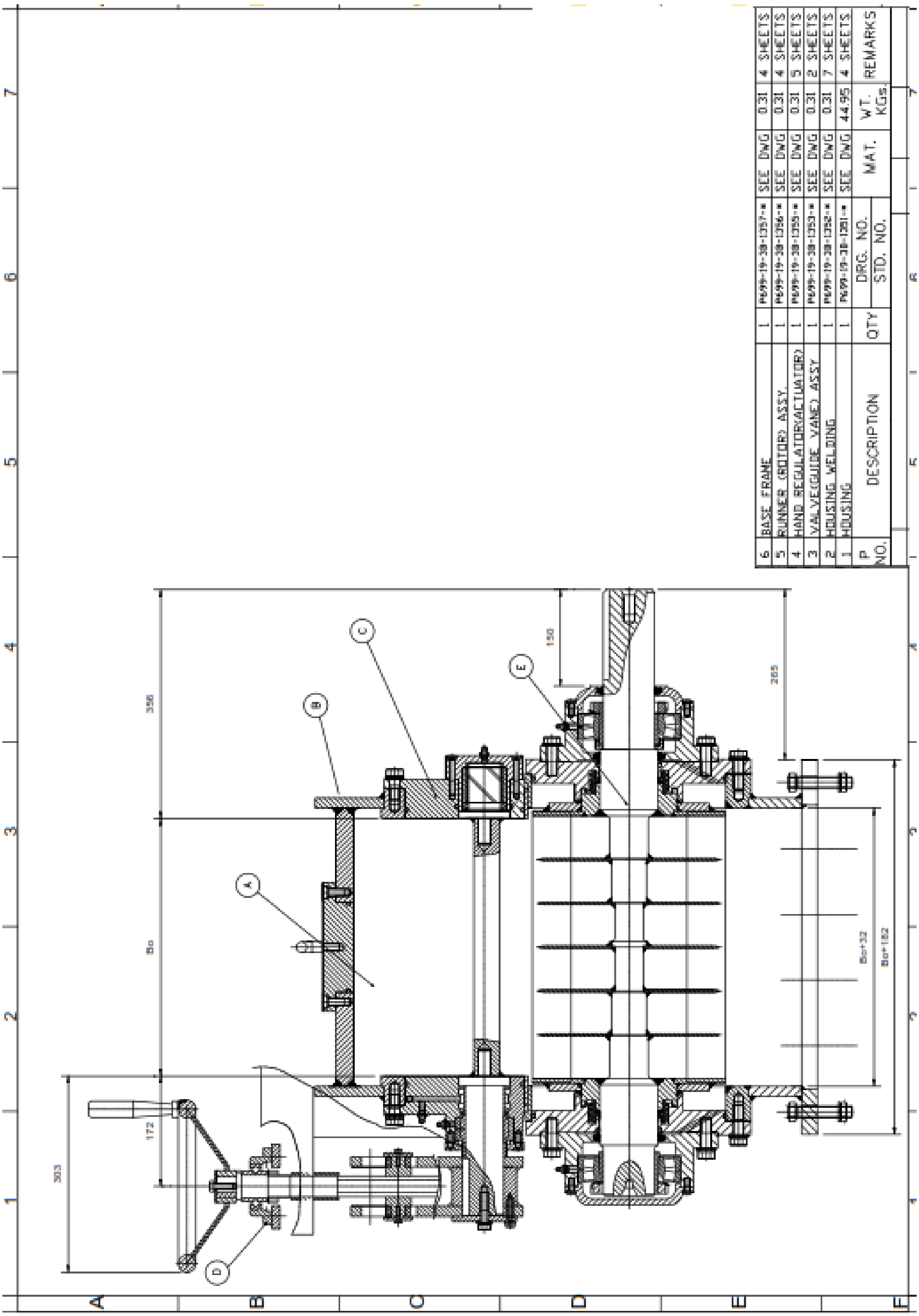
$q = 3$ For ball bearings

Basic dynamic load rating, $C = P \times (L/106)1/q$

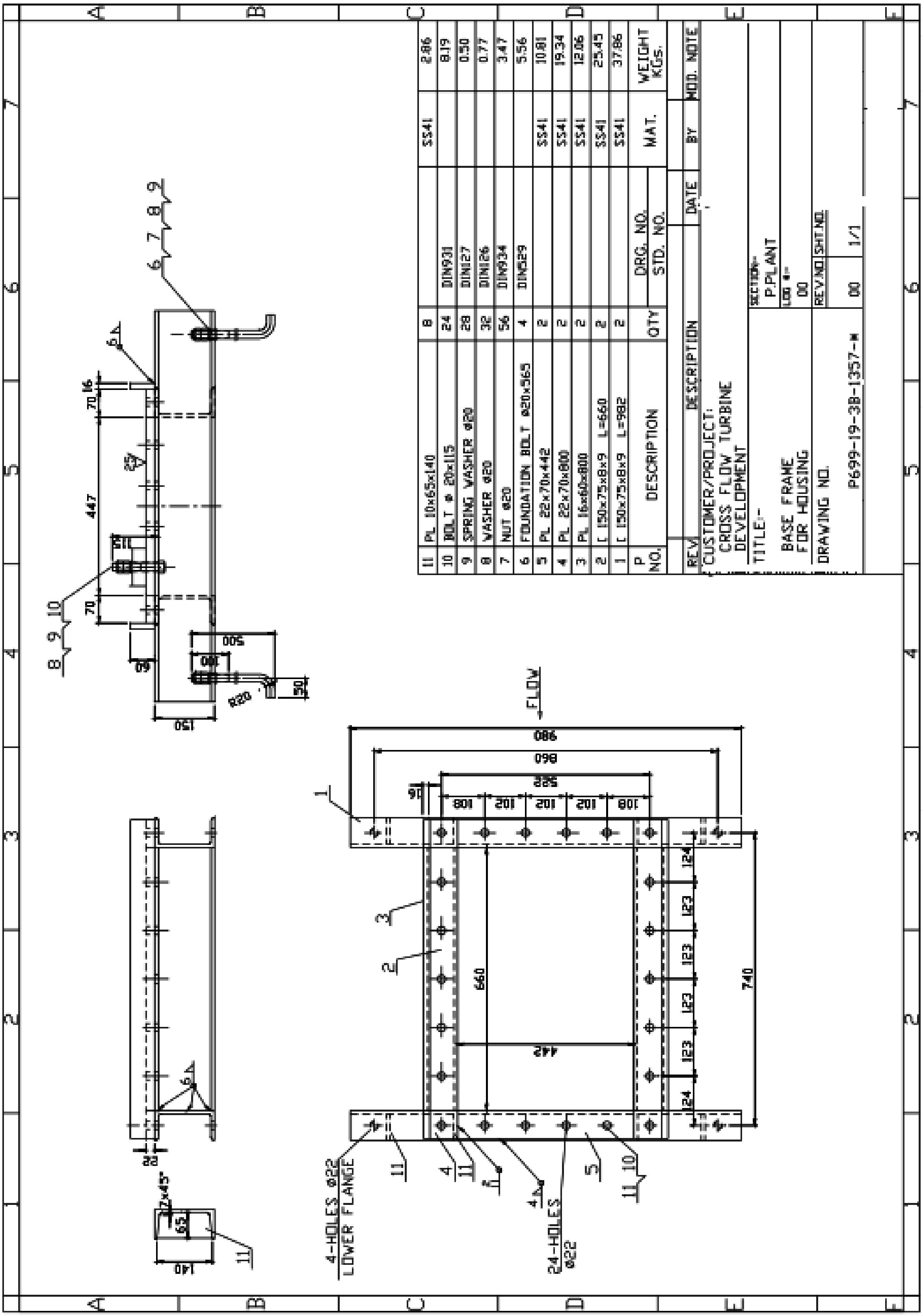
Basic dynamic load rating, $C = 1268.21$ N

Basic dynamic load rating, $C = 1.27$ kN

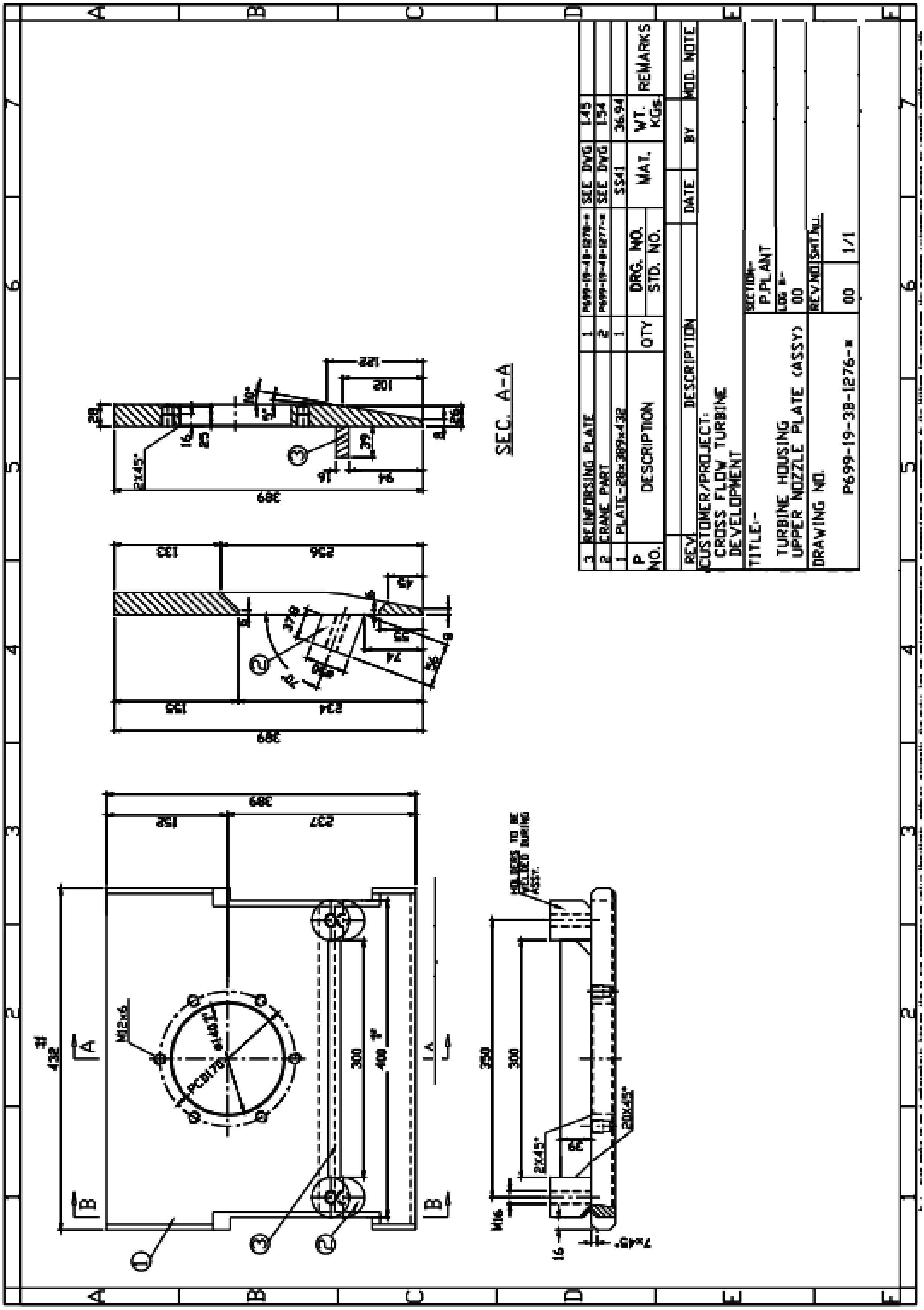
# A-2: General Arrangement Cross Flow turbine



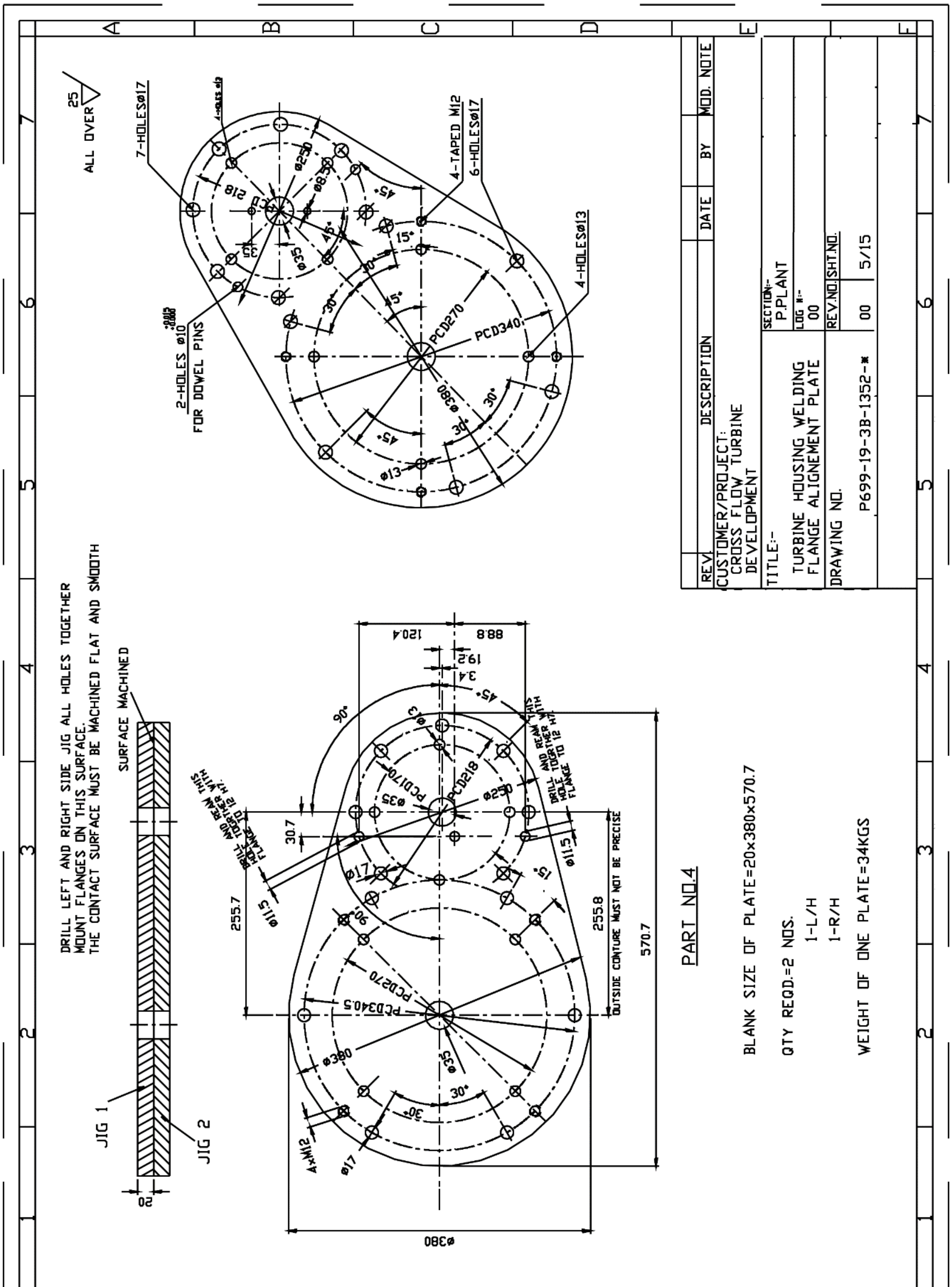
# A-3: Base Frame Cross Flow turbine



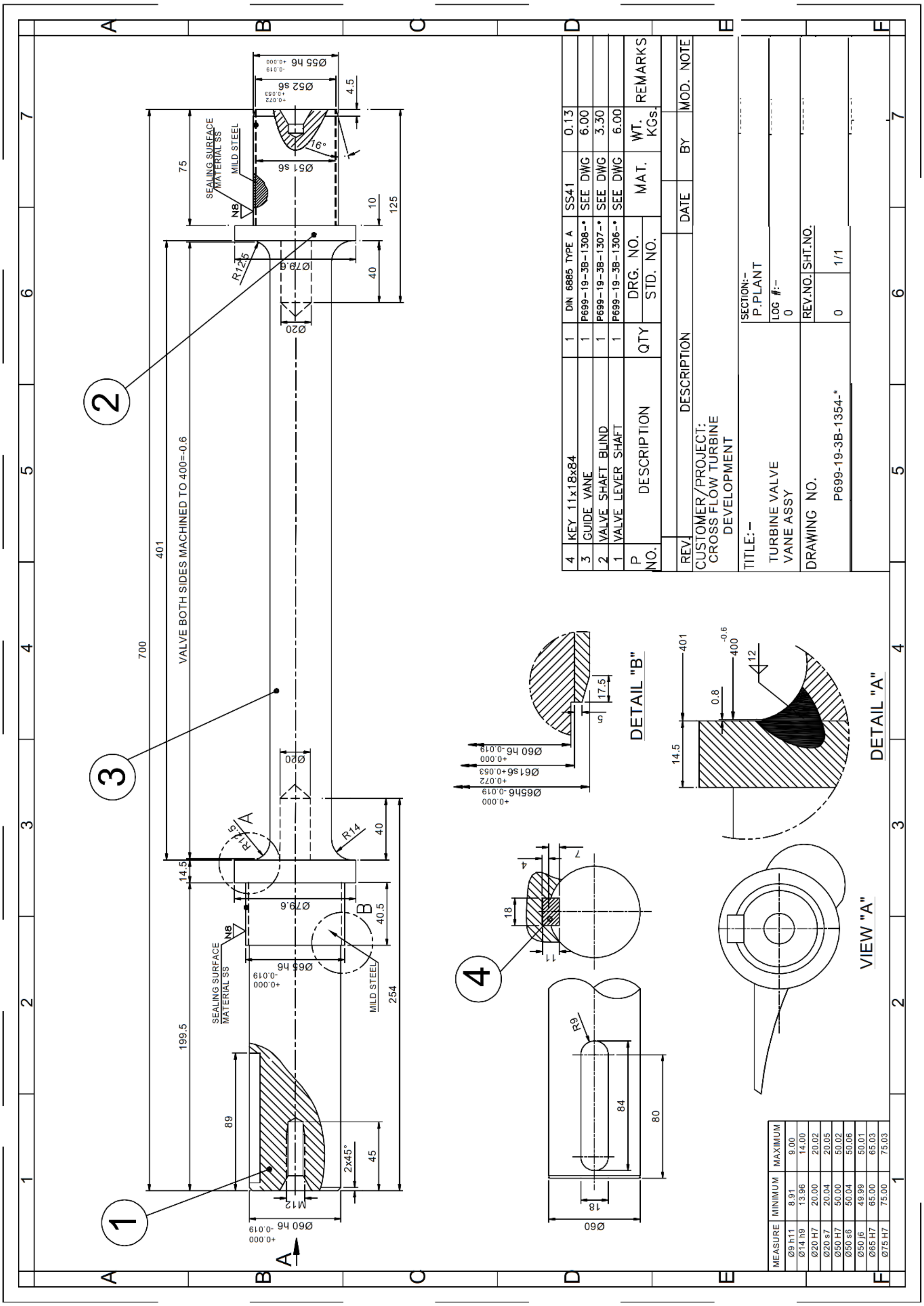
# A-4: Upper Nozzle Plate



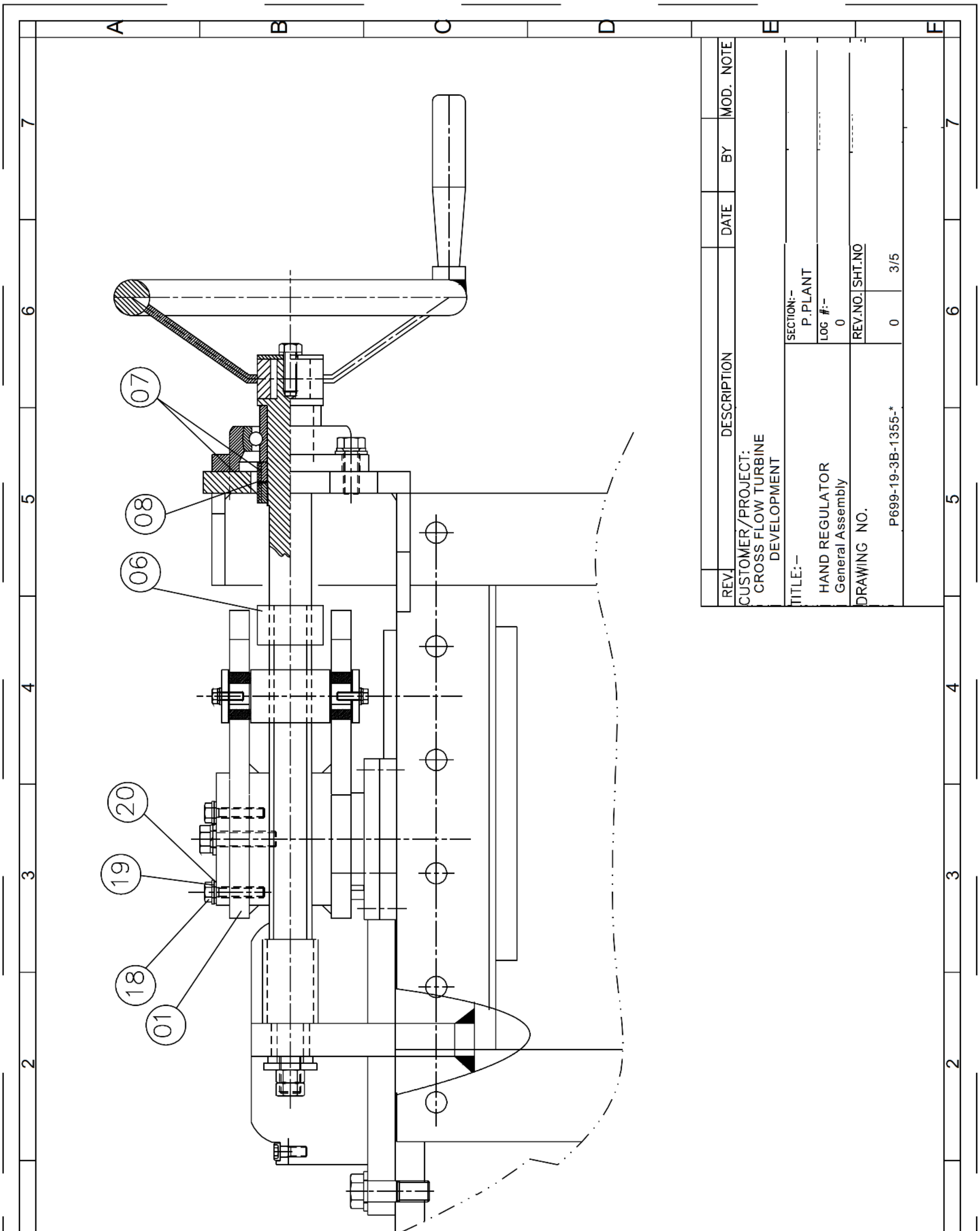
# A-5: Turbine Housing Alignment Plate



# A-6: Turbine Valve Vane Assembly



# A-7: Hand Regulator General Assembly



# A-8: Hand Regulator Arrangement