DESIGN ANALYSIS OF TESLA TURBINE

by

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A thesis submitted to the faculty of
The University of North Carolina at Charlotte
in partial fulfillment of the requirements
for the degree of Master of Science in
Mechanical Engineering

Charlotte

2017

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ABSTRACT

GALAB HARIT KAUSIK. Design Analysis of Tesla Turbine (Under the direction of DR. PETER T. TKACIK and DR. NENAD SARUNAC)

This thesis discusses the “Design Analysis of the Tesla Turbine” or so called “Flat Disc Turbine” using “Compressed Air” as the working fluid. Nikola Tesla invented the turbine in 1906 and received the patent in 1913. Using the word “Turbine”, it might be misleading since turbine generally means a shaft with blades attached to it. Tesla Turbine does not have any blades. It is a series of closely packed parallel discs which are attached to a shaft. This arrangement is packed inside a sealed chamber. Fluid is allowed to enter through an inlet nozzle and it passes onto the discs. This fluid then rotates the discs which in turn rotates the shaft. This rotary motion of the shaft produces torque which can be utilized for various purposes.

In order to check the efficiency by varying different design parameters Flow Simulation was carried out. The design of the turbine was done on Solidworks. The design parameters that were varied during the process were the size of the disc, the spacing of the discs and orientation of the inlet nozzles. The computational method of the turbine was done on Star CCM Plus (Star CD) software package. Three designs were made to carry out the flow simulation. The second and the third designs both having two inlets were taken into account. The inlets of the second design were kept horizontal and the third design were kept at an angle of 20°. Both the turbines had 9 disks and all the flow parameters were kept the same. After the flow simulation, it was observed that the Tesla Turbine design with horizontal inlet nozzles yielded maximum efficiency of 15.8%.

DEDICATION

I dedicate this thesis research to my wife and my son who have encouraged me throughout this journey.

I also dedicate this thesis research to my parents who have constantly supported and motivated me.

ACKNOWLEDGEMENTS

I would like to begin by thanking my advisors, Dr. Peter T. Tkacik and Dr. Nenad Sarunac for guiding me throughout the thesis research work. Their insights from years of experience proved to be valuable for this research work.

I would like to thank Dr. Christopher Vermillion for being on the committee for this research.

I would also like to express my gratitude towards Dr. Mesbah Uddin for helping me get access to StarCCM Plus.

I would also like to thank a fellow student and colleague, Srivatsa Mallapragada for his support in this research work.

Special thanks to Pauras Sawant, Dhiraj Muthyala, Shreyas Joshi and Amol Dwivedi for help and assistance in design for this research work.

Lastly, I would like to thank the Mechanical Engineering Department for their professionalism, support, and encouragement to be involved with on campus projects pursuant with my own career interests.

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CHAPTER 1: INTRODUCTION

1.1 Organization of Thesis

Chapter 1 provides the reader with an introduction to the Tesla Turbine and its history. It also outlines the motivation behind the research work and the advantages of the particular turbine. Finally it provides an in-depth information about the various researches that were carried out on Tesla Turbine.

Chapter 2 describes the experimental setup used for this research work which included designing the turbine and the flow simulation as well as the analytical part corresponding to the flow simulation.

Chapter 3 provides detailed discussions of results from the experiment, conclusions and future work of this research.

1.2 HISTORY OF THE TESLA TURBINE

Nikola Tesla, the brilliant and eccentric engineer from Croatia is revered amongst many for his numerous inventions. But contrary to popular belief, he did not invent the bladeless turbine. It was first patented in Europe in 1832. He worked on the deficiencies on that apparatus that was designed and improved upon it. He worked on it for almost a decade and eventually received three patents related to that machine:

- Patent number 1,061,142, "Fluid Propulsion," filed October 21, 1909, and patented on May 6, 1913

- Patent number 1,061,206, "Turbine," filed January 17, 1911, and patented on May 6, 1913
- Patent number 1,329,559, "Valvular Conduit," filed February 21, 1916, renewed July 18, 1919, and patented on February 3, 1920

Tesla, in the first patent, configured the basic bladeless design as a pump or compressor. He modified the second patent so it would work as a turbine. And in the third he made the changes necessary to operate the turbine as an internal combustion engine.

Out of the three patents, one invention which was very important to him was the Boundary Layer or Flat-Disk Turbine. It is now known as Tesla Turbine [4]. He planned to put forward a useful and efficient way of handling of energy especially on electric generation, fluid power and engines field. Because of the invention of the Flat-Disk Turbine (Tesla Turbine), Nikola Tesla paved the way for other machines operating on the same principle.

Some examples of these are air compressor, an air motor engine, a vacuum exhauster or vacuum pump. These machines use the same principle of “Fluid Propulsion” which is based on two fundamentals of physics of fluids: “adhesion” and “viscosity”.

According to the patents of Nikola Tesla, the Turbine was of high efficiency because of the form of energy transfer. He assumed that for the turbine to reach highest efficiency, the changes in velocity and movement of the fluid should be gradual.

Initially the Tesla Turbine did not achieve much commercial success and eventually other emerging types of turbines took a major share of the market. Research into Tesla turbines has however been conducted since the 1950s [7, 28] and recently there has been a resurgence of interest [8]. This paper will explore the investigations done by various researchers in Section 1.5.

1.2.1 WORKING PARTS OF THE TESLA TURBINE

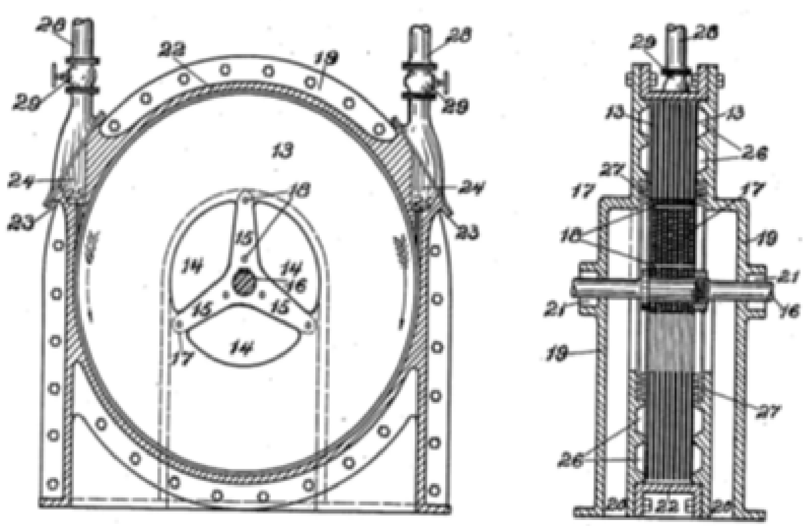

Fig 1.1 - American patent No. 1,061,206 of Tesla turbine [4]

The Tesla Turbine is very simple compared to any other turbines, pistons or engines. Tesla, in an interview with the New York Herald Tribune on Oct 15, 1911, said: “All one needs is some disks mounted on a shaft, spaced a little distance apart and cased so that the fluid can enter at one point and go out at another". He described it in a very simplified manner and it not very far from the truth. This section would describe the parts of the Tesla Turbine and how the turbine works.

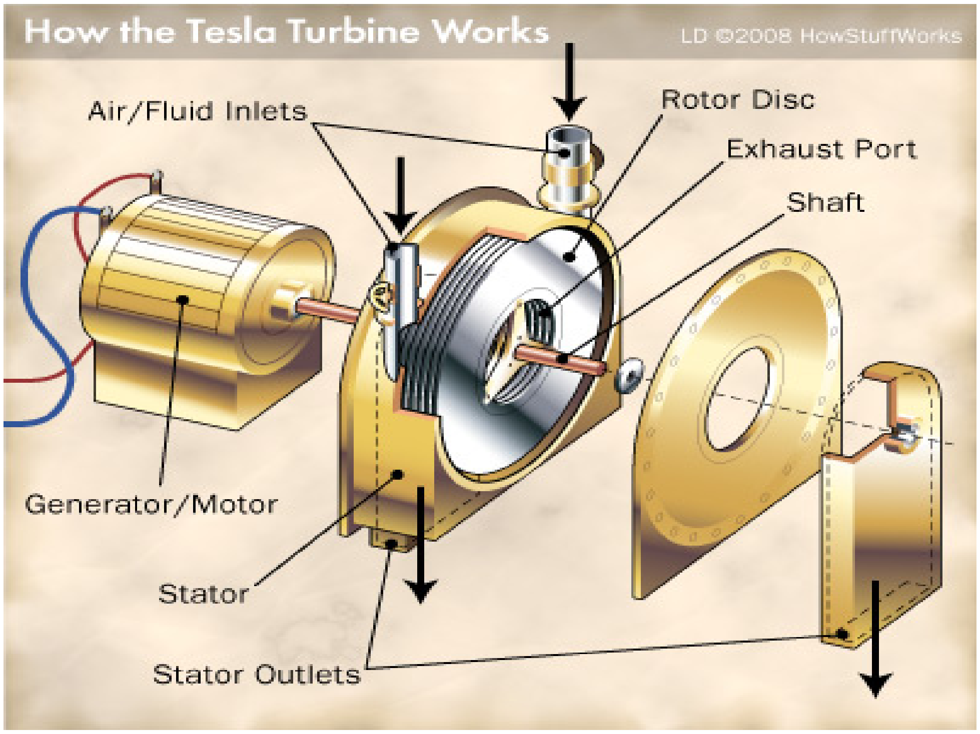

Fig 1.2 – Nomenclature of Tesla Turbine [3]

The two main parts of the Tesla Turbine are the **Rotor** and the **Stator**.

1.2.1.a THE ROTOR

In traditional turbines, Rotors are shafts with blades attached to it. But Tesla did away with blades and instead replaced them with a series of closely packed disks. The number of disks and size would vary depending upon the application. In the patent paperwork, Tesla did not define a specific number but used a more general description. He said that the rotor should contain “plurality” of disks and a “suitable diameter”.

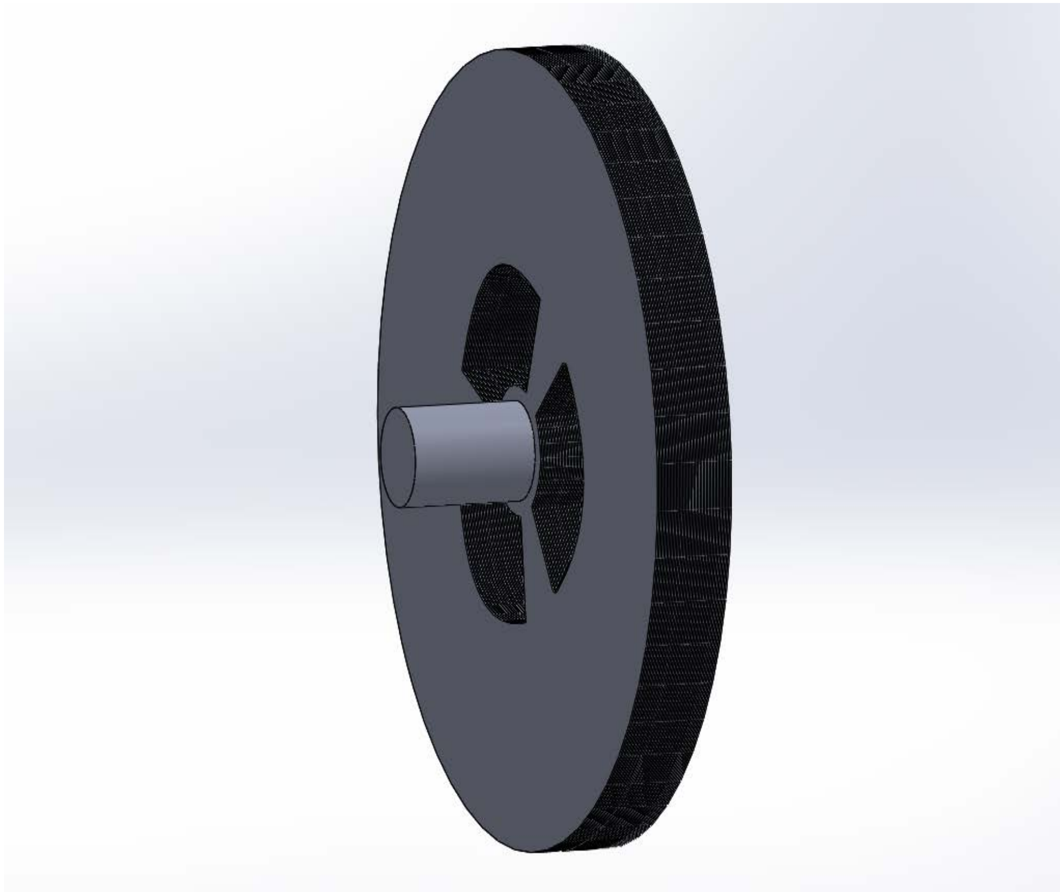

Fig 1.3 – Rotor assembly with closely packed disks attached to shaft

Disks are made with openings around the shaft. These openings act as exhaust ports so that the fluid can pass freely in between. Gaps are provided in between the disks which can differ depending on the application. The particular rotor in the diagram above has 0.02 inches gap in between. Again Tesla did not set a hard and fast rule about the gap in between the disks. The disks are locked on to the shaft so that they do not move. Because the disks are keyed onto the shaft, the rotation achieved by the disks get transferred to the shaft.

1.2.1.b THE STATOR

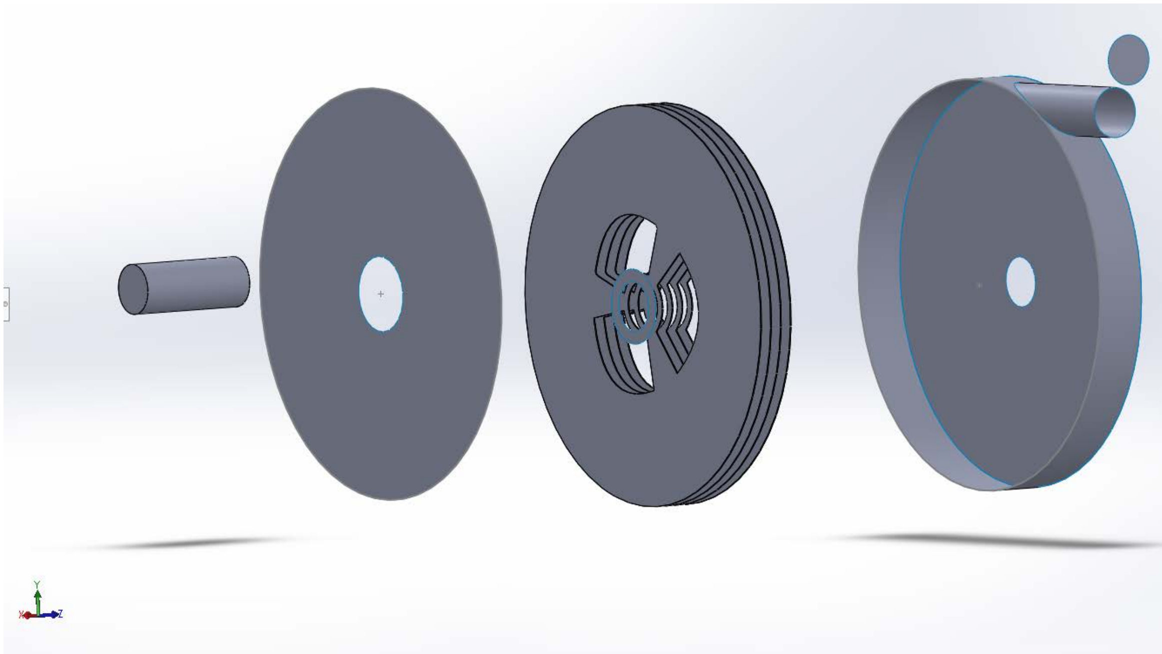

Fig 1.4 – Stator at the rightmost corner

The part in which the rotor assembly is housed is called the Stator. It is cylindrical in shape and is the stationary part of the turbine. The diameter of the interior part of the Stator is slightly larger than the diameter of the disks so as to accommodate the rotor assembly. The Stator also contains one or two inlets and an exhaust port. The original design of Tesla included two inlets for both clockwise and counter clockwise rotation.

1.2.2 WORKING PRINCIPLE OF TESLA TURBINE

The operation of a regular bladed turbine is through the kinetic energy of the moving fluid when it makes contact with the fan blades of the turbine.

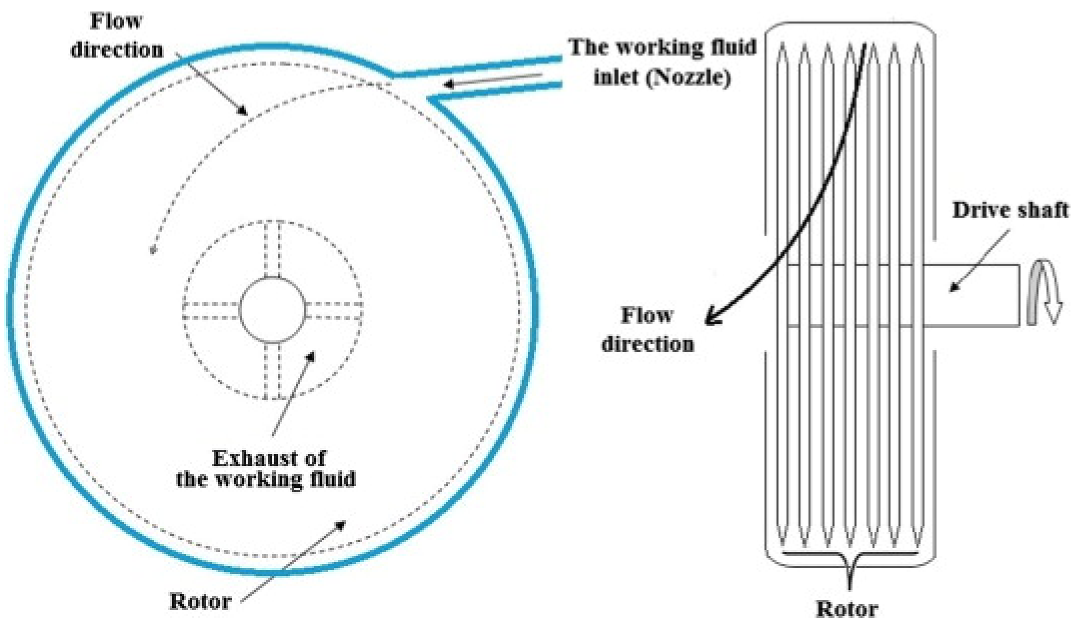

Fig 1.5 – Flow Direction of fluid in Tesla Turbine [23]

But Tesla Turbine does away with blades and instead uses a series of closely packed disks. The kinetic energy transfer of the fluid is very small on the edges of the disks. But instead uses the effect of the boundary layer. It is the adhesion of the solid disks and the moving fluid. The working fluid enters the through the inlet nozzle and it fills up the space between the disks. The fluid enters the turbine in an approximately tangential direction to the periphery of the rotor. And it follows a spiral path in between the disks and finally it gets exhausted through the holes/slots near the shaft. When the fluid passes through the rotor, it exerts shear stress of the fluid onto the disk surfaces. This gives rise to a force on the rotor which results in the development of net torque on the turbine shaft. When the fluid travels through the disks, it gradually loses energy and exhausts from the turbine at an energy state which is lower than when it entered.

If we think logically, a fluid passing through a series of smooth disks, it would simply pass over leaving the disks motionless. But it is not the case in Tesla Turbine. The reason why the disks move is because of two fundamental properties of fluid: adhesion and viscosity. Adhesion is the force of intermolecular attraction which happens between molecules of different phases. The resistance to gradual deformation of fluid due to shear stress is called viscosity. These two properties are the main reason why the disks in the Tesla Turbine move to transfer energy.

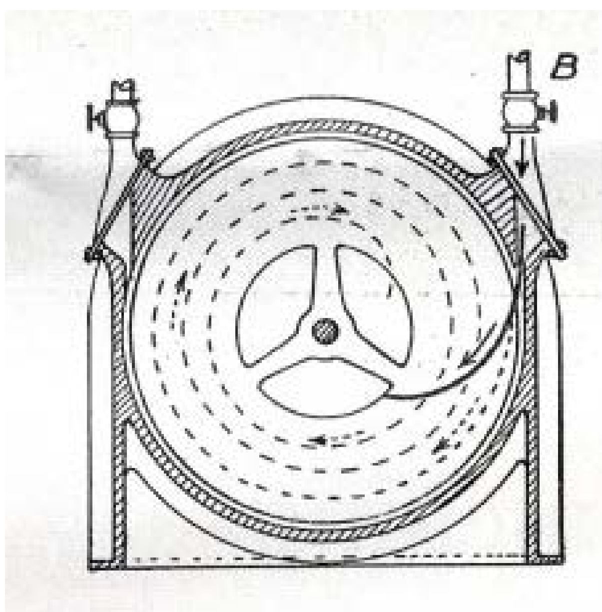

Fig 1.6 – Spiral path of fluid shown by dotted arrow. Solid arrow shows the inlet path of fluid. [29]

The step by step reason why the disks rotate is given as below:

1. As the fluid progresses inside each disk, the adhesive forces of the molecules causes the fluid to slow down and stick just above the metal surface.
2. The next set of molecules collide with the molecules sticking to the surface and finally slows down. This in turn slows the flow above them.
3. The further the molecules from the surface, the fewer collisions they encounter.
4. Parallelly, the viscosity comes into effect causing the fluid molecules to resist separation.
5. A pulling force is generated which in turn is transmitted to the disks causing it to move in the direction of the fluid.

The layer of fluid which is found in the immediate vicinity of the surface of the disks is called the boundary layer. This interaction of the fluid with the solid surface of the disk is called the boundary layer effect. The result of this effect, a rapidly accelerated spiral path is followed by the propelling fluid along the faces of the disks until a suitable exit is reached. Gradual changes in velocity and direction is experienced by the fluid as it follows a path of least resistance unlike normal turbines where disruptive forces are caused by blades or vanes. This way more energy is delivered to the turbine.

Tesla's patented design showed both clockwise and anti-clockwise movement of the fluid and rotor. It would also add a reversal effect on the turbine. Once the fluid after following the spiral path reaches the inner surface of the disks, it escapes through the holes or openings on the center of the disks and ultimately to the exhaust port. The disk diameter varied from six to ten inches in the original experiments conducted by Tesla.

1.3 GOVERNING EQUATIONS

Tesla in his patent file did not specify any governing equations to find Isentropic Efficiency or the Power. But researchers have found that for Isentropic Efficiency, the Mollier Diagram is best suited and can be referred to. As for the Power Output, the Euler Turbine Equation [41] could be used. Both the equations are discussed in this section below.

1.3 (a) Isentropic Efficiency

Most turbines operate in adiabatic conditions and even though they are not truly isentropic, they are considered isentropic from calculation point of view. For this purpose, the Mollier Diagram is taken into consideration.

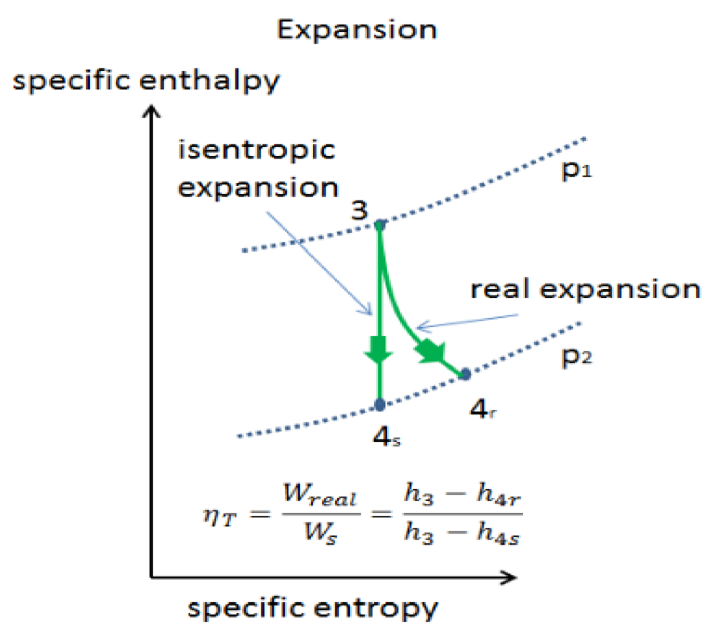

Fig 1.7 – Mollier Diagram [16]

For the calculations the governing equations from the Mollier Diagram are shown below:

Isentropic Turbine Efficiency:

$$\eta_T = \frac{Real\ Turbine\ Work}{Isentropic\ Turbine\ Work} = \frac{W_{real}}{W_s} = \frac{h_1 - h_{2r}}{h_1 - h_{2s}}$$

Isentropic Compressor Efficiency:

$$\eta_C = \frac{Isentropic\ Compressor\ Work}{Real\ Compressor\ Work} = \frac{W_s}{W_{real}} = \frac{h_{2s} - h_1}{h_{2r} - h_1}$$

where

- h_1 is the specific enthalpy of the gas at the entrance
- h_{2r} is the specific enthalpy of the gas at the exit for real process
- h_{2s} is the specific enthalpy of the gas at the exit for isentropic process

Fig 1.8 – Governing Equations from Mollier Diagram [16]

Using the above equations, the Isentropic Efficiency could be calculated from the specified parameters at inlet and outlet conditions.

1.3 (b) Euler Turbine Equation

Power of a turbine could be found out using the Euler Turbine Equation [21] which has been discussed below. For the purpose of generalization, the figure shown below is that of a regular bladed turbine. The equation is based on the Conservation of Angular Momentum and the Conservation of Energy.

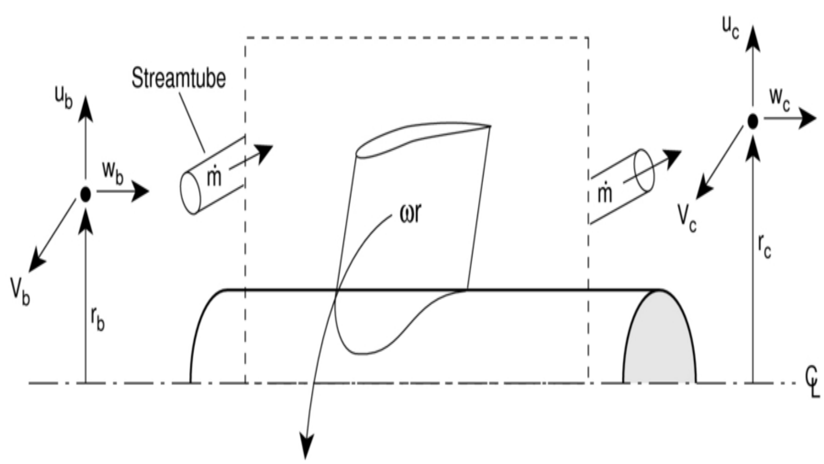

Fig 1.9 - Control volume for Euler Turbine Equation [21]

Where,

$\dot{m}$ - mass flow

V_b - tangential component of the absolute velocity of the fluid at inlet

V_c - tangential component of the absolute velocity of the fluid at exit

r_b – inlet radius

r_c – outlet radius

ω – angular velocity

Applying the Conservation of Angular Momentum, we could calculate the Torque (τ) as follows:

$$\tau = \dot{m}(V_c\, r_c - V_b\, r_b)$$

The work per unit time or Power (P) could be calculated as given below:

$$P = \tau\, \omega = \omega\, \dot{m}(V_c\, r_c - V_b\, r_b)$$

From the steady flow energy equation:

$$P = \dot{m}(h_{Tc} - h_{Tb})$$

Equating both expression of conservation of energy with our expression from conservation of angular momentum, we arrive at:

$$h_{Tc} - h_{Tb} = \omega(V_c\, r_c - V_b\, r_b)$$

For a perfect gas with constant C_p the equation would be:

$$C_p(h_{Tc} - h_{Tb}) = \omega(V_c\, r_c - V_b\, r_b)$$

The above equation is called Euler Turbine Equation.

Another way of finding the Power of a turbine [14] is shown below:

$$P = \frac{1}{2} k C_p \rho A V^3$$

Where,

P = Power output

C_p = Maximum power coefficient, ranging from 0.25 to 0.45, dimension less (theoretical maximum = 0.59)

ρ = Air density

A = Rotor area

V = Inlet Velocity

k = 0.000133

1.4 ADVANTAGES

A lot of experiments have been carried out on the Tesla Turbine since 1950s [7, 28]. Some of the main advantages of the turbine are mentioned in the description below.

If we look from the mechanical standpoint, the turbine is easy to build and the construction costs are economical. Because of the nature of construction, it possesses better durability and low wear and tear than the present day turbines. The internal static pressure is low and so heavy cast housings are not required to build it. It can work both clockwise and anti-clockwise direction in a single machine as shown in the Tesla's patent [4].

Since the machine is made up of disks, it would run at low vibrations and low noise. Because of low vibrations, the overall safety increases. Besides, it has proved to be steady on intermittent applications, rapid load variations and shut downs. If conventional turbines' operations become critical or fails, then there might be explosion which harms the entire machinery including hydraulic lines, control surfaces and even operators. But with the Tesla Turbine, there is very low chances of explosion or danger. If any part goes critical or fails, that part would basically implode into small pieces and would be ejected through the exhaust. The other working parts would still continue to work and provide thrust [30].

Various kinds of fluids with particles, multiphases etc. has been used in the Tesla Turbine without damage. In case of pump, a comprehensive list of fluids and particles that have been pumped through without damage has been provided by Possell [28]. This shows the versatility and ability of the turbine to handle various fluids. It can handle corrosive fluids as well as gases with high ash content or fluids with high viscosity. A company by the name of Discflo [40] has already commercialized friction pumps.

Table 1.1 – List of fluids tested by Possell [28]

Fluid Mixtures tested by *Posell*			
Abrasive solids	Marbles	Fish	Salt
Aggregates	Methane	Flour	Sawdust
Alfalfa	Molasses	Gases	Seawater
Apples	Mud	Geothermal affluent	Seaweed
Ash sump	Oil	Glass	Seeds
Avocados	Oil sludge	Grains	Shrimp
Berries	Ore	Grout	Slimes
Blood [59]	Ozone	Foam (fire fighting)	Sludge
Boiler feed water	Peas	Hot sodium phosphate	Slurries
Boiling liquids	Chip suspensions	Industrial sewage	Sulphuric acid
Cabbages	Clinker	Pellets	Toxic wastes
Carbon	Coal	Potato peeling slurry	Vegetable wastes
Cement	Corn	Raw sewage	Water
Chemicals	Ferrous Chloride	Rice	Wheat

The turbine works for both Newtonian as well as non –Newtonian fluids and the disks do not suffer from cavitation as with the case with conventional bladed turbines. So potentially could be used for geothermal steam or particle laden industrial gases [27]. This also means that the turbine pump could also withstand high temperatures [27].

1.4 MOTIVATION

Primary energy sources are depleting and energy consumption is increasing rapidly with the development of the society. Crucial issues like energy shortage and environmental concerns are coming to the fore. Utilization of geothermal energy, solar energy, biomass energy and waste heat recovery is gaining widespread attention [25]. Traditional expanders like axial flow turbine and radial in-flow turbine would not be suitable for small scale applications as the flow loss would be considerably large [25]. Traditional high speed turbines with small mass flow is not practical. In this case the Tesla Turbine could provide a low cost and reliable alternative.

Researchers are always looking for new and sustainable ways to manage the energy of the world. This leads to analysis of new machines which would be able to deal with new forms of production as well as energy transformation. Tesla turbine is one such machine which is gaining the spotlight slowly. It is still at a very nascent stage. For small scale application, researchers feel that Tesla turbine would be the best option. During the course of research, it has been found that there is a company called Green Turbine [43] and they have constructed a turbine whose output is 15KW. But upon further investigation it has been found that the efficiency of that turbine is between 10% to 15%.

There is lack of technology and understanding of the Tesla turbine which is impeding its growth. There is a huge potential in small scale applications and it is believed that if more research is put towards it, the Tesla turbine would prove to be the best of breed. Because of the scope of applications and the advantages mentioned in the earlier section, more research is necessary and that is what led the researcher to take up this project.

In 2001, Schmidt [41] tested the boundary layer turbine using different fluids and obtained the flowing results:

Table 1.2 – Schmidt's [41] report on boundary layer turbine

Case	Working fluid/fuel	Firing rate [Btu/hr]	Temperature [°C]	Pressure [bar]	RPM [min^{-1}]	Power [kW]	Isentropic Efficiency [%]
1	Compressed air	Not available	Unknown	5.93	8,193	8,650	Unknown
2	Compressed air	Not available	20.5°C	2.27	1,100	447	16%
3	Natural gas flue gas	173,000	444°C	2.41	6,218	3,430	12.25 %
4	Biomass flue gas	192,600	392°C	2.76	6,284	3,206	11 %
5	Saturated Steam	Unknown	170°C	6.89	6,500	9,246	13.7 %

From the table above we can see that the applications most viable for small applications and small power plants. More examples of the Tesla Turbine can be found in reference.

1.5 BACKGROUND RESEARCH

Nikola Tesla received his patent in 1913 and worked on the turbine for a few years but had to give it up for financial constraints. So finally it succumbed to different emerging type of turbines. After almost half a century, research on Tesla Turbine began in 1950 [7]. After that there has been a resurgence of interest for this particular turbine [26]. Current efficiency values of it are lower than conventional turbines which is the main disadvantage. But it is hoped that the current surge of interest would help in improving the turbine as a whole including the efficiency.

James Armstrong [7] in 1952 undertook a research on Tesla Turbine. He tried to find ways to improve upon the original design. He built and tested his design and analyzed it from theoretical standpoint. He used 10 disks made of boiler steel plates with seven inches in diameter. He found out that using four inlet nozzles did not have any effect on the efficiency. But having one inlet nozzle with slightly diverging shape performed better than the straight nozzle. The Rankine Cycle Efficiency he obtained was 14. 59%. And he got 1.108HP at 8300rpm with 118psig inlet pressure. This paved the way for future research.

Table 1.3 – Performance of Tesla Turbine reported by Armstrong [7]

Steam Pressure Lb./Sq .In. Gage	Speed RPM	Torque Output lb-ft	Brake Horsepower H.P.
40	0	0.438	0
40	4000	0.175	0.133
40	4800	0.140	0.128
40	5700	0.105	0.114
40	6500	0.088	0.108
60	0	0.525	0
60	5000	0.350	0.333
60	5300	0.350	0.353
60	5900	0.332	0.374
60	7000	0.263	0.350
80	0	0.700	0
80	4600	0.525	0.460
80	6500	0.437	0.542
100	0	0.954	0
100	3300	0.788	0.495
100	4800	0.700	0.640
100	5000	0.700	0.668
100	6200	0.648	0.765
100	6250	0.612	0.729
100	7500	0.525	0.750
100	7900	0.437	0.659
118	0	1.050	0
118	4500	0.875	0.750
118	5000	0.875	0.833
118	5700	0.875	0.950
118	6200	0.840	0.992
118	6800	0.787	1.020
118	8300	0.700	1.108

Warren Rice [26] conducted experiments on Tesla Turbine in 1965. He established factors which effect performance and efficiency. Initially he constructed a six disk turbine and reported some of the aspects to determine the feasibility of this turbine. He used compressed air as the working fluid. He then constructed a turbine with nine disks. Some improvements were noticed in this turbine when he reduced the gaps between the disks. But the analytical data did not conform to the experiments that he carried out. The highest efficiency that his turbine achieved was approximately 26%.

Table 1.4 – Experimental data from Rice's turbine [26]

Pressure to turbine, psig	Temp to turbine, deg F	Mass rate of flow, lb_m/min	Speed, rpm	Power, hp	Efficiency, percent
40	203	2.82	6300	0.68	21.7
40	203	2.82	8500	0.81	25.4
40	203	2.82	9200	0.85	25.8
60	175	4.05	8000	1.13	21.2
60	175	4.05	10000	1.29	23.8
60	175	4.05	11000	1.32	24.4
60	175	4.05	11500	1.07	20.2
80[a]	182	3.48	8000	1.12	21.7
80	182	3.48	10000	1.24	23.8
80	182	3.48	11000	1.24	23.8
80	182	3.48	11500	1.10	21.1
100	175	4.38	9000	1.49	21.6
100	175	4.38	11000	1.67	24.1
100	175	4.38	12000	1.32	19.0
120	165	5.30	9300	1.88	21.2
120	165	5.30	11000	2.02	22.9
120	165	5.30	12200	1.16	13.5
140	165	6.00	11800	2.41	23.2
140	165	6.00	12500	1.18	11.9

[a] This and following data with flow-limiting inserts in nozzles.

Warren Rice and R. Adams [30] did a research on the laminar flow and incompressible fluid on a turbine with co-rotating disks. A complete problem statement was formulated from Navier Stokes equation using tangential velocity, flow rate and Reynold's number. They considered both laminar and turbulent flow which suggested the torque.

Piotr Lampart [10] in his research presented the results of a design analysis of a Tesla Turbine intended for a micro-turbine of heat capacity 20KW. This power plant operated on Organic Rankine Cycle (ORC). He investigated the flow parameters within the space of the disks. The calculated efficiency of the Tesla Turbine showed competitive efficiency compared to small conventional bladed turbines.

Some other worth mentioning research papers which investigated the Tesla Turbine and presented both analytical and experimental results are as follows:

- 1.8 kW output power, 18 000 rpm, 16% efficiency (Mikielewicz et al., 2008)
- 50W output power, 1000 rpm, 21% efficiency (North, 1969)
- 1.5 kW output power, 12 000 rpm, 23% efficiency (Hicks, 2005)
- 1 kW output power, 12 000 rpm, 24% efficiency (Beans, 1966)
- 3 kW output power, 15 000 rpm, 32% efficiency (Gruber and Earl, 1960)
- 1.5 kW output power, 120 000 rpm, 49% efficiency (Davydov and Sherstyuk, 1980)

The above mentioned research papers were referred to while writing this paper.

Hoya [39] and Guha [39] conducted extensive experiments on sub-sonic and super-sonic nozzles with Tesla turbines. But they focused more on experimental results and not on analytical treatment of the fluid mechanics that drive turbine performance. Krishnan [1] tested several mW-scale turbines, and reported a 36% efficiency for a 2 cc/sec flow rate with a 1 cm diameter rotor.

For most of the experiments carried out by the researchers, it has been found that the efficiency of the rotor can be comparable to conventional bladed turbine. Because of the versatility of the Tesla Turbine in being able to use any kind of fluid as mentioned in the earlier section, there is a lot of scope of development. The research paper is a contribution towards the development of the Tesla Turbine with the hope of it entering into mainstream production.

CHAPTER 2: METHODOLOGY

2.1 COMPUTATIONAL SETUP

This section presents the setup of the computational procedure. The designs were made in Solidworks [38] and the Flow Simulation was done in StarCCM Plus [18] CFD Software package. For the Flow Simulation, the SST K-ω Turbulence has been used.

In many aerodynamic applications, the SST K- ω Turbulence model is widely used. It is a two equation eddy – viscosity model which combines Wilcox K – ω model and the K - ϵ models [37]. The Shear Stress Transport (SST) is the best combination for low R_e turbulence model and free stream behavior. The K – ω is slightly better than K - ϵ model and gives better accuracy and results for boundary layers as well as viscous sublayer. But when the free stream behavior comes into effect then, the K - ϵ model gives better results. So Menter [33] combined both these models and put forward the SST K – ω turbulence model. The model has been researched and validated and gives good results in many applications. It is noteworthy to add that the SST K - ω turbulence model is the most popular in the aerodynamics industry.

2.1.1 Turbine Design and Flow Simulation

The Tesla Turbine consists of rotor, stator, inlet nozzle and outlet. The rotor consists of a shaft on which closely packed disks are attached. The outer diameter of the disks and the shaft depends on the application. Nikola Tesla in his patent [4] did not clearly mention the dimensions of the disks or the shaft. All he mentioned was "*The dimensions of the device as a whole, and the spacing of the disks in any given machine will be determined by* *the conditions and requirements of special cases. It may be stated that the intervening distance should be the greater, the larger the diameter of the disks, the longer the spiral path of the fluid and the greater its viscosity. In general, the spacing should be such that the entire mass of the fluid, before leaving the runner, is accelerated to a nearly uniform velocity, not much below that of the periphery of the disks under normal working conditions and almost equal to it when the outlet is closed and the particles move in concentric circles*".

2.1.1 (a) Design 1 and Flow Simulation

The first design was made using the parameters of Warren Rice's [8] initial design barring a few changes. He used six disks but here four disks have been used. There were two inlets but here only one inlet is used. It is to check the compatibility with StarCCM Plus and to get an initial understanding of the flow simulation. The parameters of the turbine that was designed is given as follows:

Table 2.1 – Design parameters of first turbine

Design 1	
Number of disks	4
Outer Diameter of disk	8 inches
Inner Diameter of disk	1 inch
Shaft diameter	1 inch
Shaft Length	2.5 inches
Inlet Nozzle Diameter	1 inch
Outlet Diameter	1.5 inches
Diameter of Stator	8.2 inches
Width of Stator	1.02 inches
Thickness of Disk	0.02 inches
Inner Diameter of Lid	1.5 inches
Outer Diameter of Lid	8.2 inches
Spacing of disks	0.25 inches

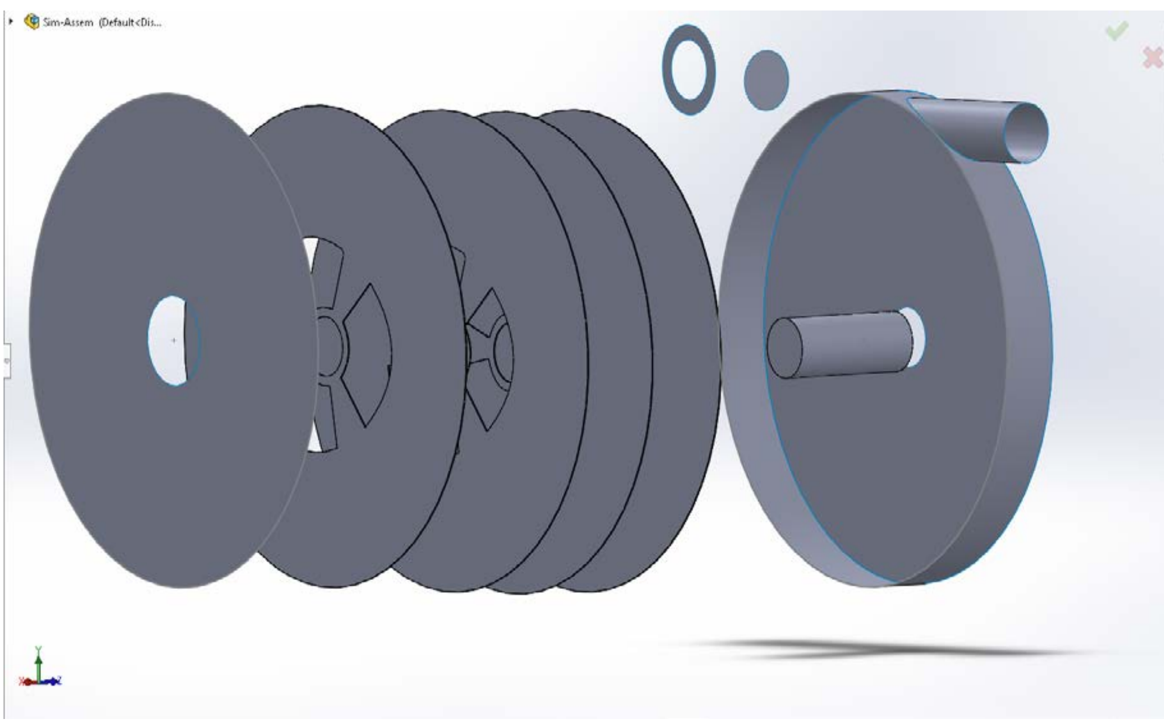

Fig 2.1 – Exploded view of Tesla Turbine with 4 disks and one inlet nozzle

For the Flow Simulation the following assumptions were made to set up the model and run it:

Table 2.2 – Physics Models Assumption for Flow Simulation

Space	Three Dimensional
Time	Steady
Material	Gas
Flow	Segregated Flow
Equation of State	Constant Density
Viscous Regime	Turbulent
Turbulence Model	SST K-Omega Turbulence Model

The Meshers that had been implemented are as follows:

Table 2.3 – Implemented Meshers

Prism Layer Mesher
Trimmed Cell Mesher
Automatic Surface Repair
Surface Remesher

The controls that were used to create the mesh are as follows:

Table 2.4 – Controls used for creating Mesh

Base Size	10mm
Target Surface Size	10mm
Minimum Surface Size	0.5mm
Number of Prism Layers	8
Prism Layer Near Wall Thickness	1.0E-4 mm
Prism Layer Total Thickness	0.75mm
Maximum Cell Size	10mm

Using the above mentioned Controls and Meshers, the following parameters for the turbine were calculated by the software and mesh was created which showed the following:

Table 2.5 – Mesh parameters

Cells	7410683
Faces	22014701
Vertices	7800923

After the processing of the mesh is done, the mesh scene for the turbine is shown below:

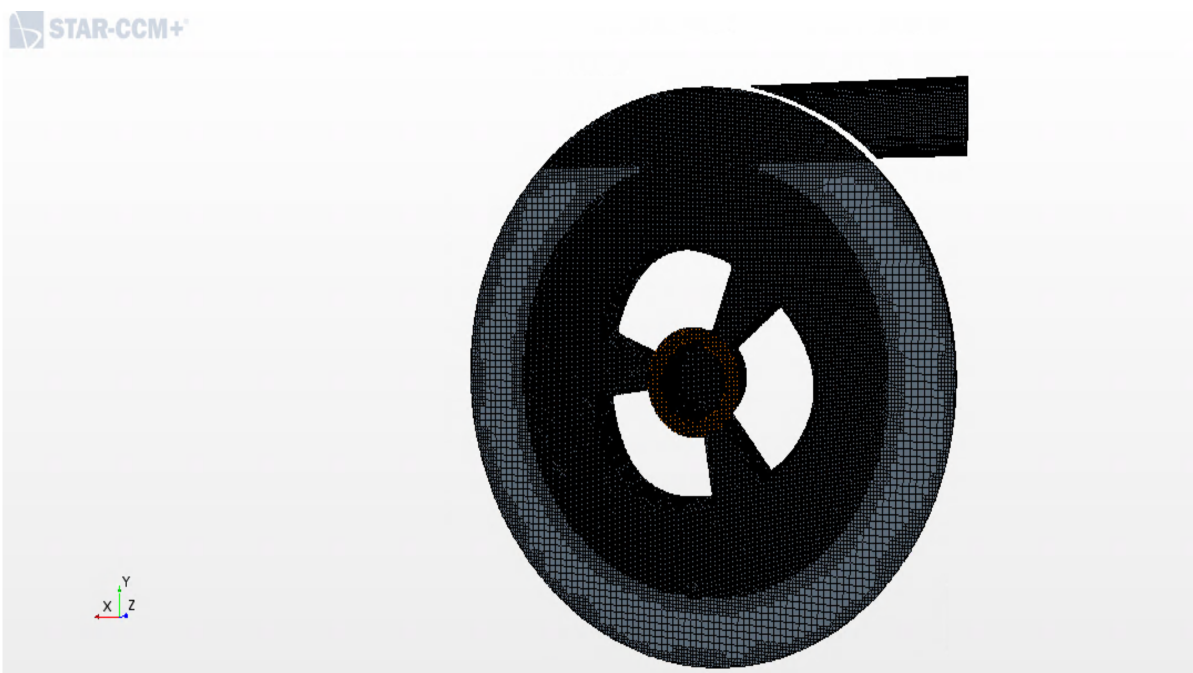

Fig 2.2 – Mesh Scene of the rotor

Fig 2.3 – View of the mesh on the disk

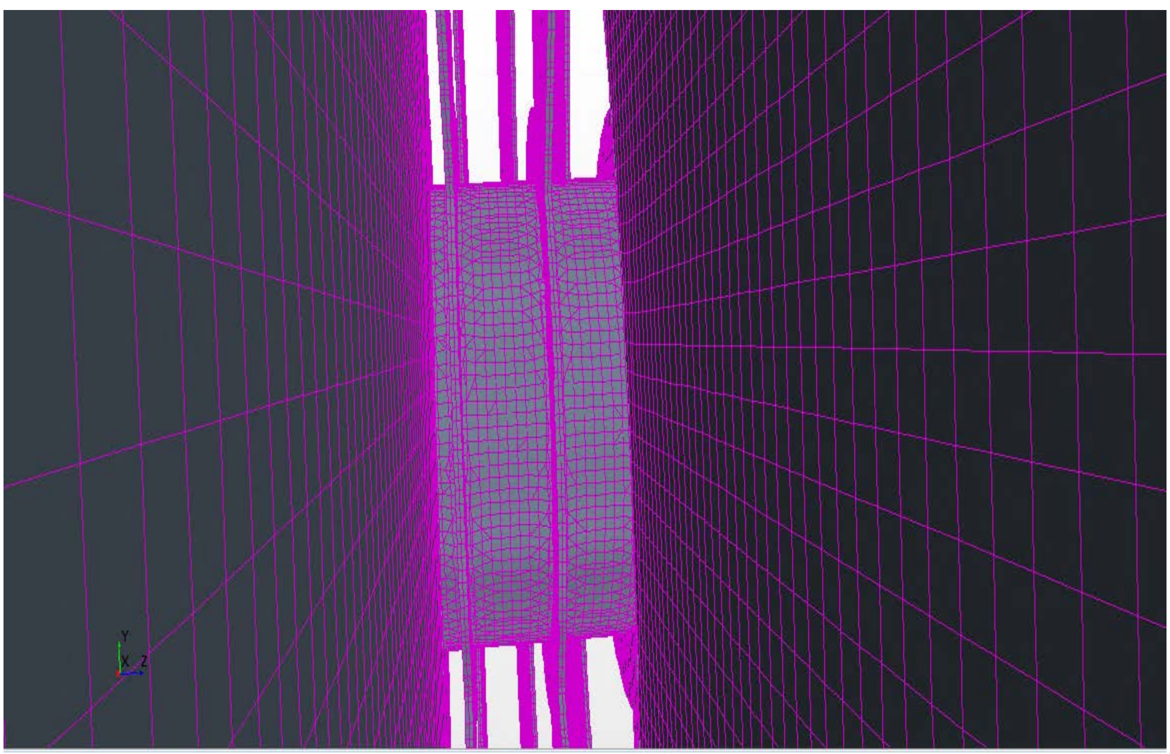

Fig 2.4 – View of the mesh on the shaft and the disks on both sides

The initial conditions were given as follows:

Table 2.6 – Initial Conditions

Inlet Velocity	199.27 m/s
RPM OF Rotor	160000 rad/s

After the mesh and the initial conditions were set up, the simulation was run for 1500 iterations. The Pressure Scene and the Velocity Scene and the path of fluid flow that were created after the run is shown below:

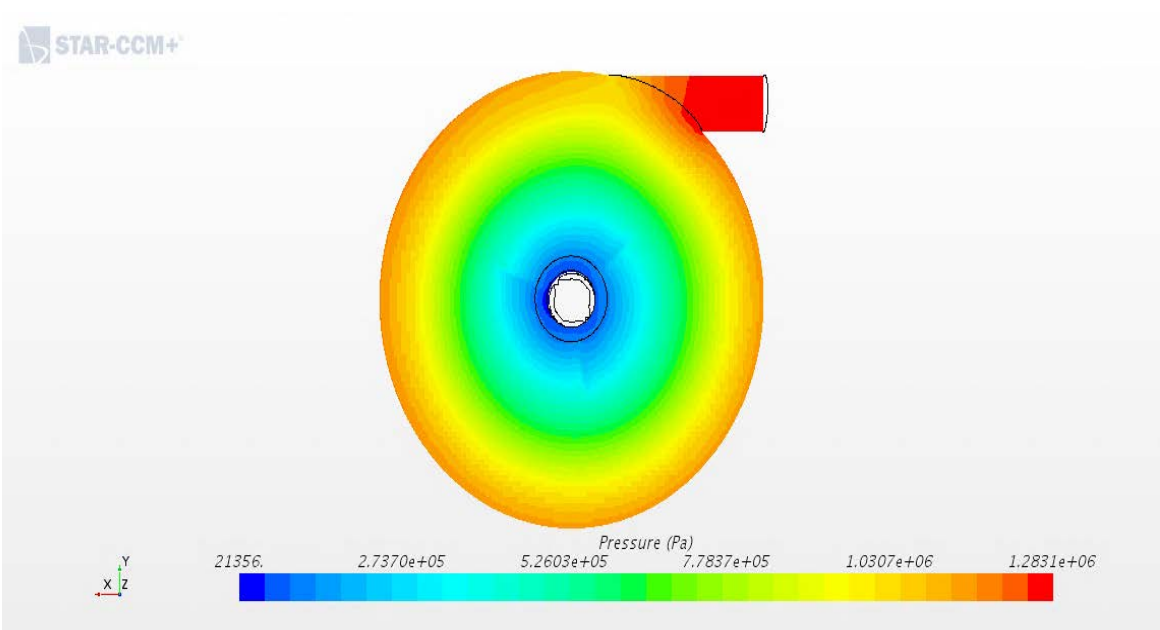

Fig 2.5 – Pressure Scene on the rotor

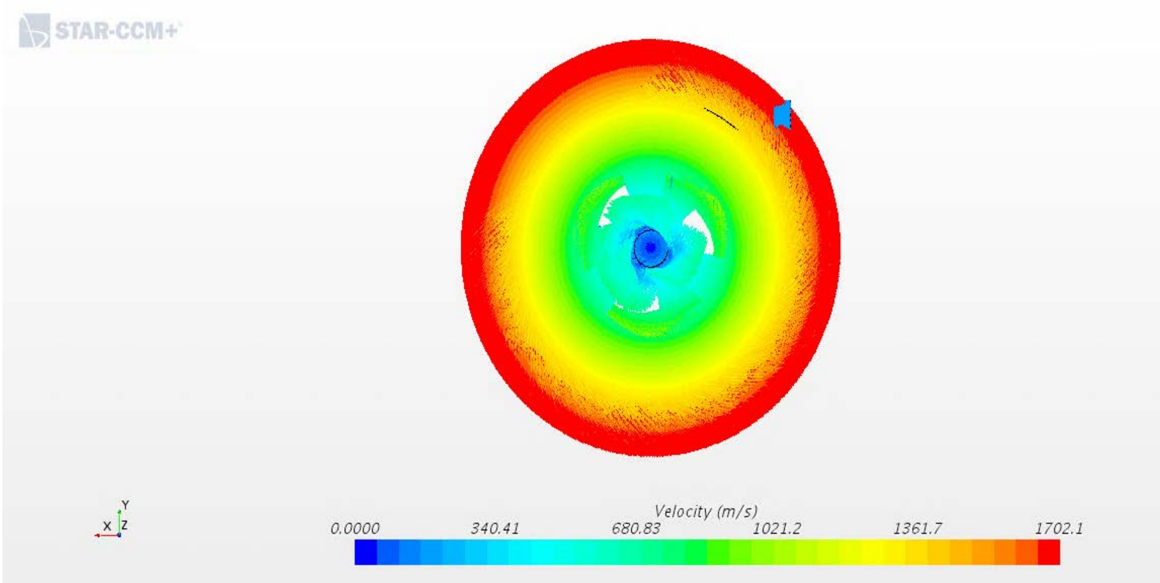

Fig 2.6 – Velocity Scene of the rotor

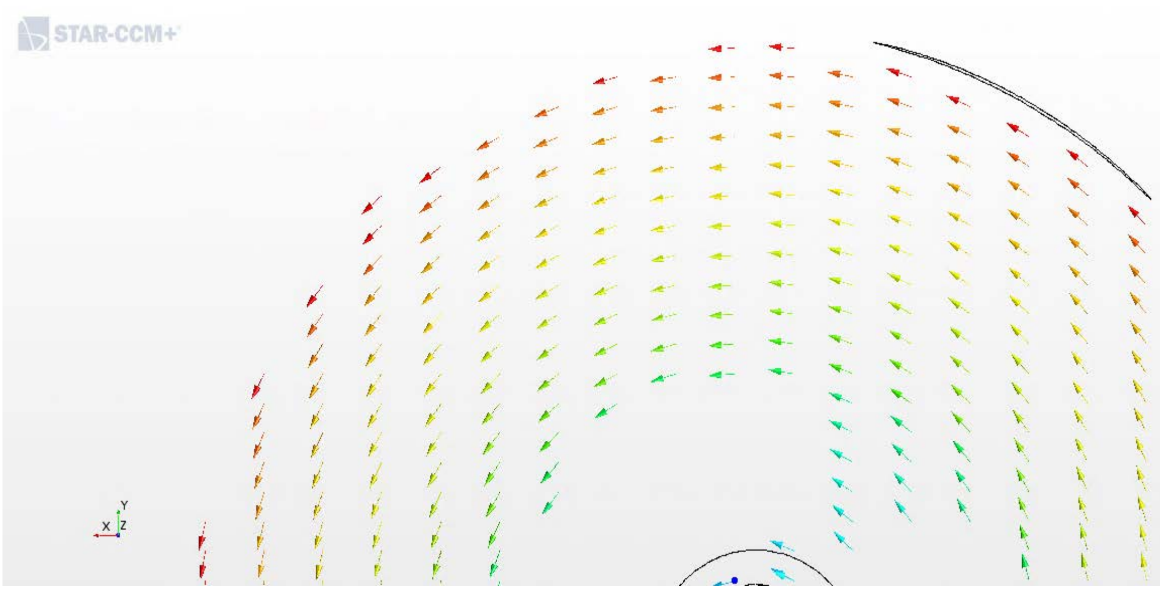

Fig 2.7 – Path of fluid flow

The iterations were carried out and the residuals converged which is shown below:

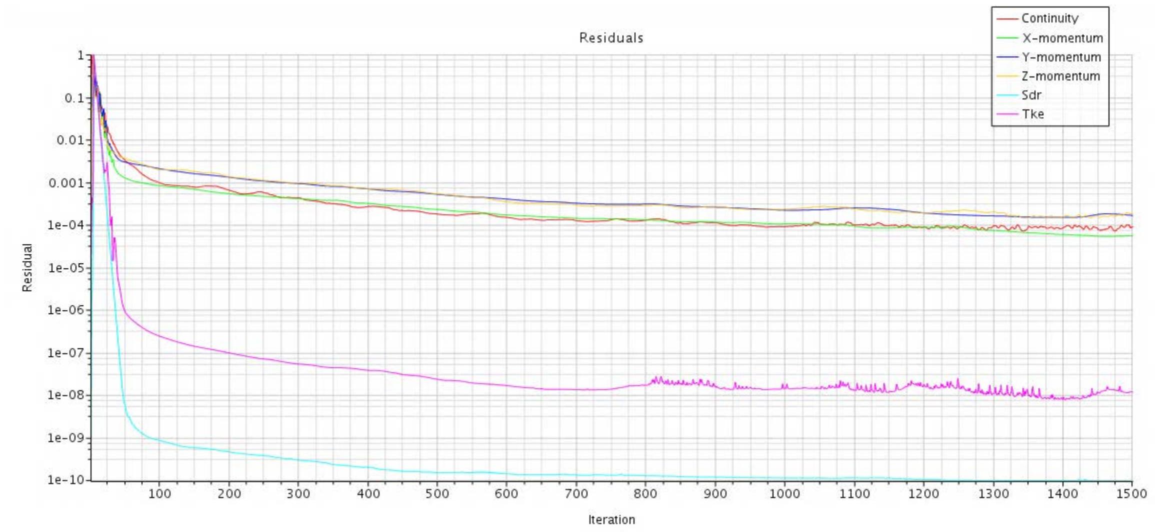

Fig 2.8 – Converged Residuals

This was the base run and the results that were intended out of it are Force on Disk, Pressure Drop, Inlet and Outlet Mass Flow. The values that were obtained from the software are given below:

Table 2.7 – Values obtained from software

Force on Rotor	24.95N
Pressure Drop	1073.2kPa
Inlet Mass Flow	0.0001kg/s
Outlet Mass Flow	0.0001kg/s

From the base run which has been elaborated above, it has been found that modelling a 3D design of the turbine with an acceptable mesh/grid is not feasible due to restrictions in time and resources. Moreover, pressure and frictional losses which will occur in an actual turbine have been ignored in the model. One other restriction is for high velocities, Tesla claimed the turbine achieves higher efficiencies. But then the compressible effects became important but this feature goes beyond the scope of the present work.

Earlier mesh was very coarse and mesh has been refined for the models that have been designed afterwards.

The following models which have been designed subsequently have a variation in the number of disks and the spacing between them. Also the orientation and the number of the nozzles have been varied to check which would give the maximum efficiency. The design and the flow simulation has been discussed in the subsequent sections and the results of the flow simulation has been presented in a tabular format for easy reference.

2.1.1 (b) Design 2 and Flow Simulation

The second design is the extension of the first one and has been designed taking into consideration Rice's [8] second design except for few changes. In his model the inlets were at an angle but in this design the inlets have been kept horizontal. The size of the disks have been reduced as well. The thickness of the disks has been increased and the spacing between the disks have been decreased.

This model contains 9 disks and two inlets. The outlet is a continuous circle on the lid covering the shaft.

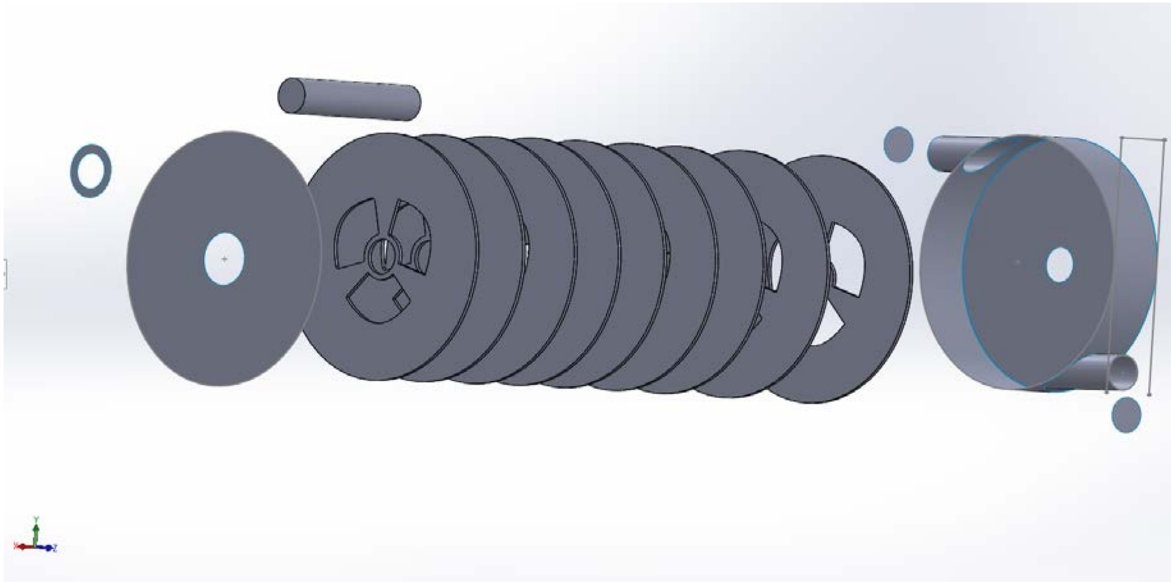

Fig 2.9 – Exploded view of second design

The parameters which were used to design the turbine are given in the table below:

Table 2.8 – Design Parameter of second turbine

Design 2	
Number of disks	9
Outer Diameter of disk	7 inches
Inner Diameter of disk	1 inch
Shaft diameter	1 inch
Shaft Length	4 inches
Inlet Nozzles Diameter	1 inch
Outlet Diameter	1.5 inches
Diameter of Stator	7.2 inches
Width of Stator	1.47 inches
Thickness of Disk	0.095 inches
Inner Diameter of Lid	1.5 inches
Outer Diameter of Lid	7.2 inches
Spacing of disks	0.15 inches

For the Flow Simulation the following assumptions were made to set up the model and run it:

Table 2.9 - Physics Models Assumption for Flow Simulation

Space	Three Dimensional
Time	Steady
Material	Gas
Flow	Segregated Flow
Equation of State	Constant Density
Viscous Regime	Turbulent
Turbulence Model	SST K-Omega Turbulence Model

For the working fluid, compressed air was selected. And the material properties of compressed air that were used are as follows:

Table 2.10 – Compressed Air Properties used for Simulation

Density	17.70 kg/m^3
Dynamic Viscosity	1.84E-5 Pa-s
Specific Heat	1031 J/kg-K
Thermal Conductivity	0.03 W/m-K

Gradients are a function of Constant Density and the Gradient Method selected is Hybrid Gauss-LSQ and the Limiter Method used is Venkatakrishnan [15].

While selecting SST $K - \omega$ Turbulence Model, Reynolds-Averaged Navier Stokes Equation also gets selected.

The Segregated Flow and Segregated Fluid Flow has been used for Convection Heat Transfer of the 2nd Order.

The Mesh that had been implemented are as follows:

Table 2.11 – Implemented Meshers

Prism Layer Mesher
Trimmed Cell Mesher
Automatic Surface Repair
Surface Remesher

Although the Mesh type is the same as used in Design 1, some of the controls that have been used are different.

The controls that were used to create the mesh are as follows:

Table 2.12 – Controls used for creating Mesh

Base Size	10mm
Target Surface Size	10mm
Minimum Surface Size	0.625mm
Number of Prism Layers	9
Prism Layer Near Wall Thickness	1.0E-4 mm
Prism Layer Total Thickness	0.8mm
Maximum Cell Size	10mm

The mesh that was created using the above control features showed the following:

Table 2.13 – Mesh parameters

Cells	11514734
Faces	34024933
Vertices	12140536

There is a marked difference in the number of cells that were created in the new design. The mesh scene shown below shows more uniform cells and lesser amount of errors. Because of the limitations of computer resources, the number of cells could not be increased exponentially. After the processing of the mesh controls, the Mesh Scene that was created is shown below:

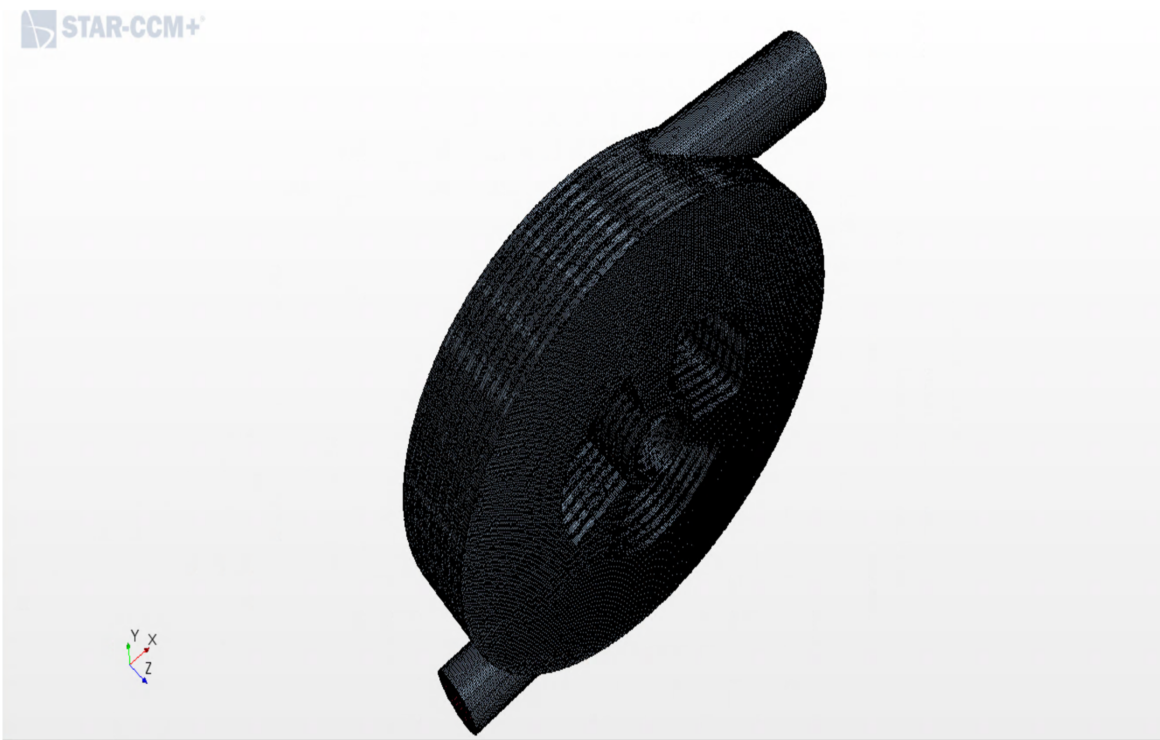

Fig 2.10 – Mesh Scene of the turbine rotor and the inlets

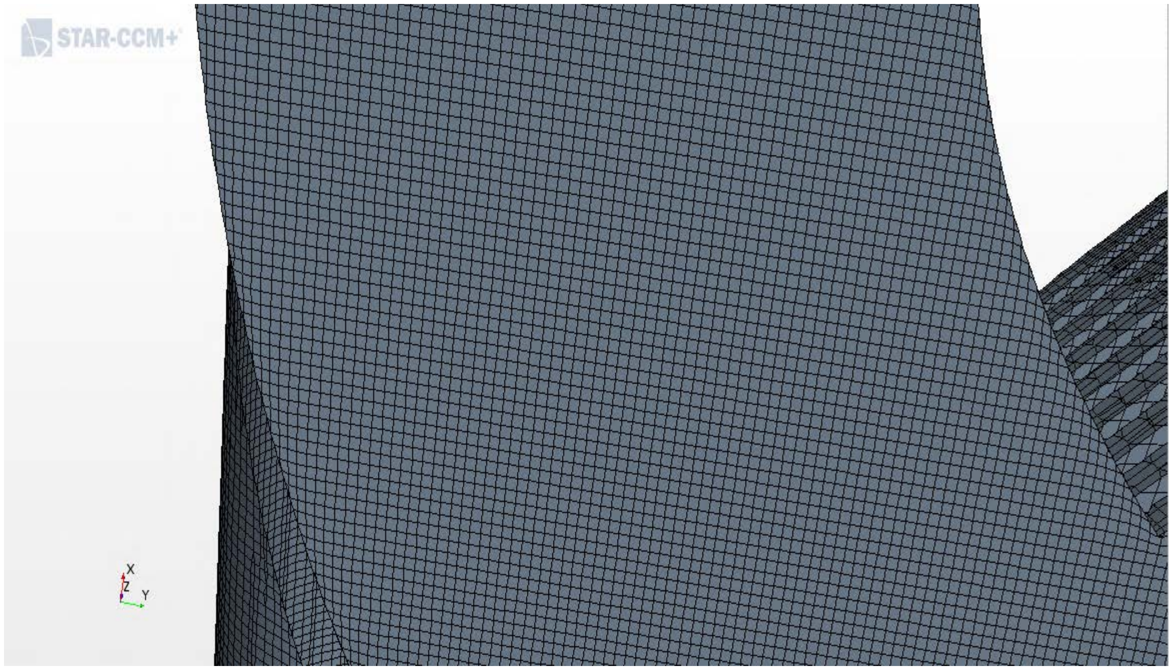

Fig 2.11 – Close up view of mesh on the disk

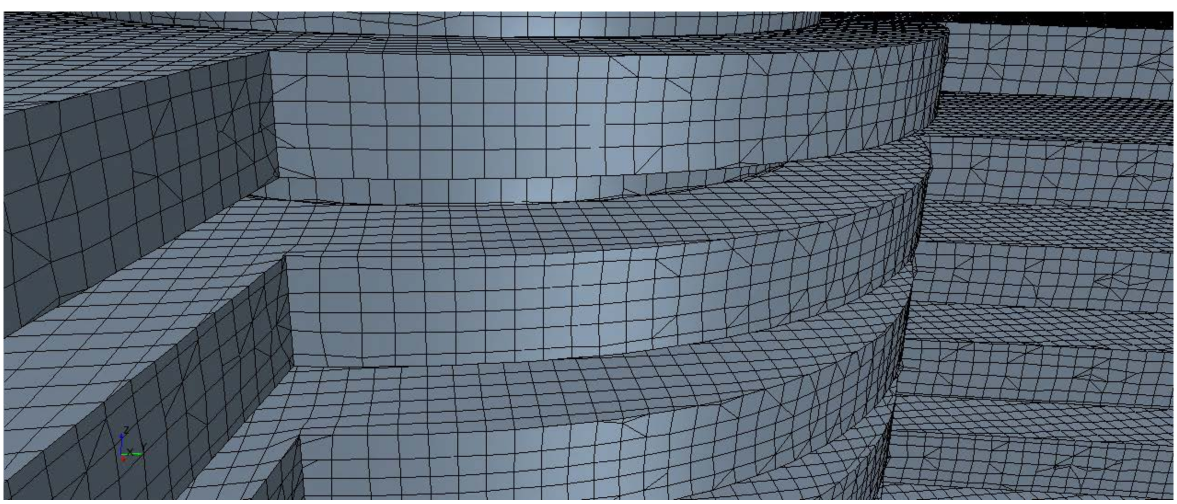

Fig 2.12 – Mesh at the intersection of shaft and disks

After the meshing was done, the inlet conditions were entered to run the simulation. The conditions are given below:

Table 2.14 – Initial Conditions

Inlet Velocity	230 m/s
Pressure	1378.7kPa
Inlet Temperature	293K
RPM OF Rotor	9000 RPM

This simulation was run till 1140 iterations and the convergence of the residuals are shown as below:

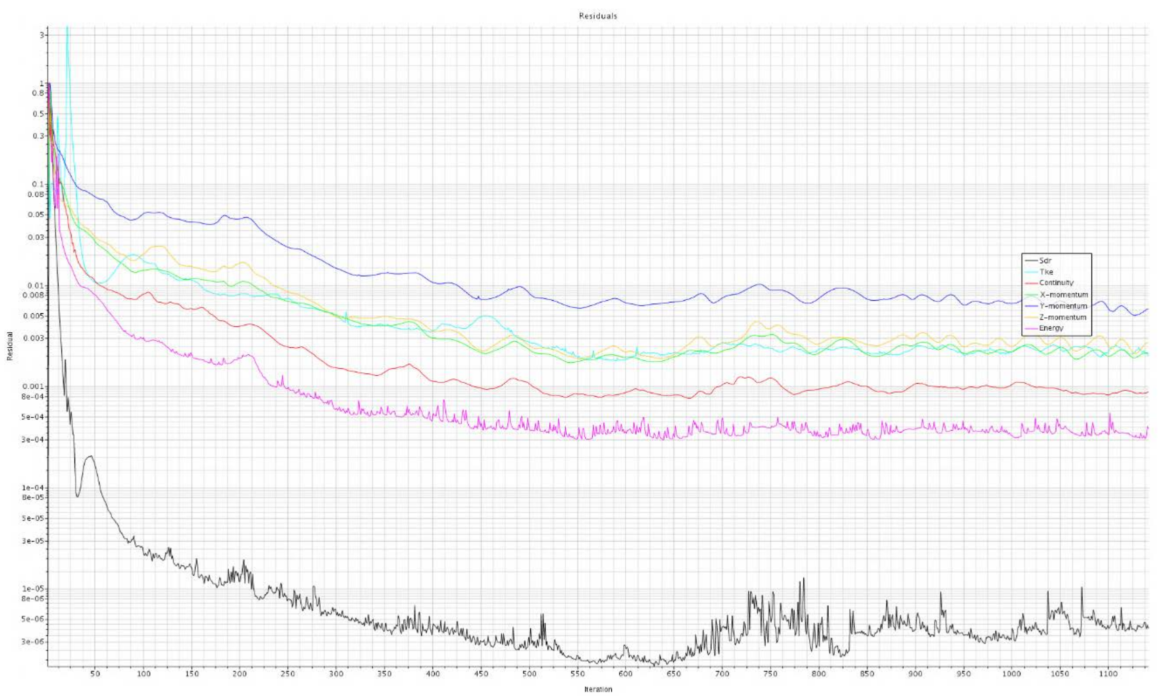

Fig 2.13 – Converged Residuals

From the above screen shot we can see that some of the residuals were increasing initially but after a few iterations, they converged. This also shows that after making a few changes in the initial conditions, the simulation was running as was expected. Although most of the Physics setup was kept the same but there were some variations in some of the models which have been presented on Table 2.9 and Table 2.10 and the explanation as to what modifications were made in the model is mentioned below.

The Pressure Scene after the run is as shown below:

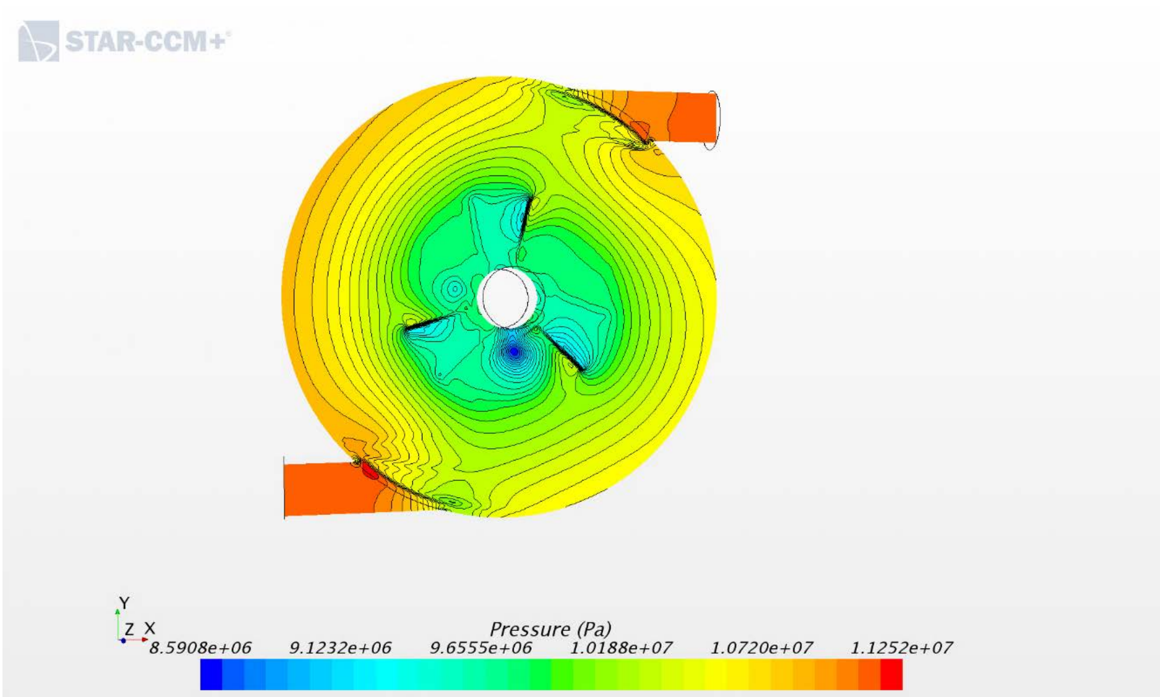

Fig 2.14 – Pressure Scene after the final iteration

The Pressure Scene above shows that maximum pressure of the working fluid is at the inlets and when it hits the disks, the pressure decreases and finally after circulating through the rotor and the casing, it exits at a much lower pressure. This happens because it transfers its energy onto the disks which makes the disks rotate along with the shaft which in turn would help in producing torque.

The inlet velocity which was specified for this simulation was 230m/s and after the final iteration the Velocity Scene is shown below:

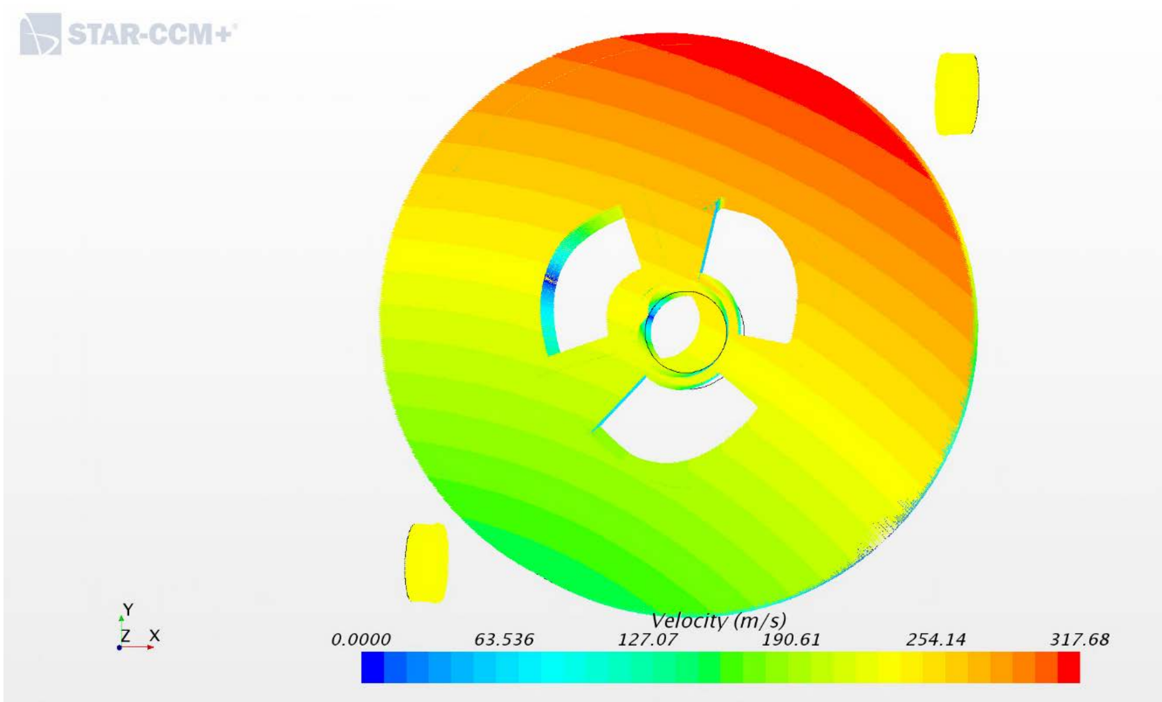

Fig 2.15 – Velocity Scene after the final iteration

The Velocity Scene above shows that the inlet at the upper right corner creates more force than the lower one. This is because of the intersection of the fluid which travels from the inlet above and forces it upon the fluid entering from the lower inlet. But even then the velocity at the lower inlet does not decrease much and stays in the range between 140m/s to 190m/s.

After the final run, the values are shown in the table below:

Table 2.15 – Values obtained after the run

Force on Rotor	6365 N
Inlet 1 Pressure	11245.8 kPa
Inlet 2 Pressure	11181 kPa
Outlet Pressure	173.8 kPa

After the run was completed, we could see that the values have improved significantly. This is attributed to a more refined mesh as well changing some parameters of the Physics Model as well the initial conditions.

2.1.1 (c) Design 3 and Flow Simulation

The third design was made as per the specifications of the previous design. But the orientation of the inlets have been changed. The inlets are placed at an angle of 20°. Piotr Lampart [10] in 2009 carried out several flow simulations by changing the angle of the inlets. His work is also an extension of Warren Rice [8]. He found out that flow transition towards the outlet with small diameters causes velocity increase and pressure drop. This means that high flow losses because of high outlet energy. Such kind of situation is unfavorable if the efficiency is the main concern as it is in this research paper. The outlets should be at a larger diameter. For the all the designs in this research work, the outlet has been designed as a continuous circle on the lid. This would allow free flow of the working fluid at the outlet and create minimum back pressure and lower velocities.

The third design that has been created also contains 9 disks but the angle of the two inlets have been changed to 20°. The design is as shown below:

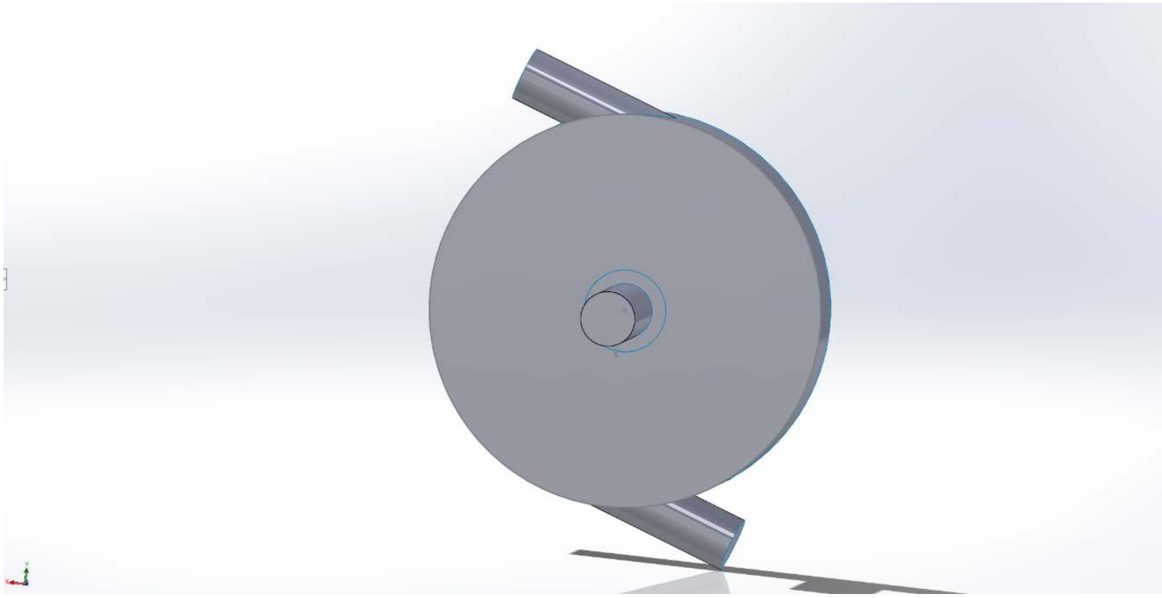

Fig 2.16 – Turbine at 20° inlets

The exploded view of the turbine is shown below:

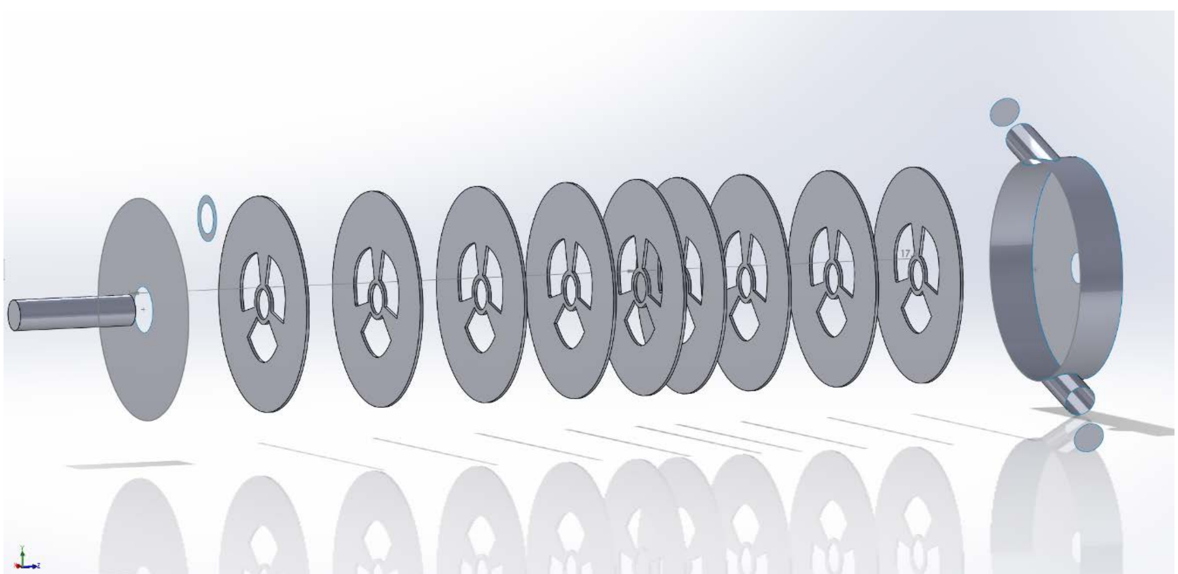

Fig 2.17 – Exploded view of the turbine

The parameters which were used to design the turbine are given in the table below:

Table 2.16 – Design Parameter of third turbine

Design 3	
Number of disks	9
Outer Diameter of disk	7 inches
Inner Diameter of disk	1 inch
Shaft diameter	1 inch
Shaft Length	4 inches
Inlet Nozzles Diameter	1 inch
Outlet Diameter	1.5 inches
Diameter of Stator	7.2 inches
Width of Stator	1.47 inches
Thickness of Disk	0.095 inches
Inner Diameter of Lid	1.5 inches
Outer Diameter of Lid	7.2 inches
Spacing of disks	0.15 inches

For the Flow Simulation, all the parameters that had been used on Design 2 is kept the same for Design 3. The following are the framework that had been used for the simulation:

Table 2.17 - Physics Models Assumption for Flow Simulation

Space	Three Dimensional
Time	Steady
Material	Gas
Flow	Segregated Flow
Equation of State	Constant Density
Viscous Regime	Turbulent
Turbulence Model	SST K-Omega Turbulence Model

Compressed air was selected and the material properties that were used are as follows:

Table 2.18 – Compressed Air Properties used for Simulation

Density	17.70 kg/m^3
Dynamic Viscosity	1.84E-5 Pa-s
Specific Heat	1031 J/kg-K
Thermal Conductivity	0.03 W/m-K

The Gradient Method selected is Hybrid Gauss-LSQ and the Limiter Method used is Venkatakrishnan. K – ω Turbulence Model and Reynolds-Averaged Navier Stokes Equation also selected. Segregated Flow and Segregated Fluid Flow has been used for Convection Heat Transfer of the 2^{nd} Order.

The Mesh that had been implemented for this case are as follows:

Table 2.19 – Implemented Meshers

Prism Layer Mesher
Trimmed Cell Mesher
Automatic Surface Repair
Surface Remesher

Controls used to create the mesh is also kept the same:

Table 2.20 – Controls used for creating Mesh

Base Size	10mm
Target Surface Size	10mm
Minimum Surface Size	0.625mm
Number of Prism Layers	9
Prism Layer Near Wall Thickness	1.0E-4 mm
Prism Layer Total Thickness	0.8mm
Maximum Cell Size	10mm

The mesh that was created after the meshing function was run contained the following volume:

Table 2.21 – Mesh parameters

Cells	11987713
Faces	35403756
Vertices	12668017

There is negligible difference in the number of cells that has been created in the Design 3. The mesh created is still uniform and the errors that might be created is lesser than Design1. Again it is worth mentioning that increasing the number of cells exponentially on the geometry would go out of scope of this research paper. The volume of the mesh is high enough in this case so that the relevant properties that are needed gets captured. On closer look at the mesh, we could see that boundary layer mesh is quite uniform. The mesh that was created after the processing is shown below:

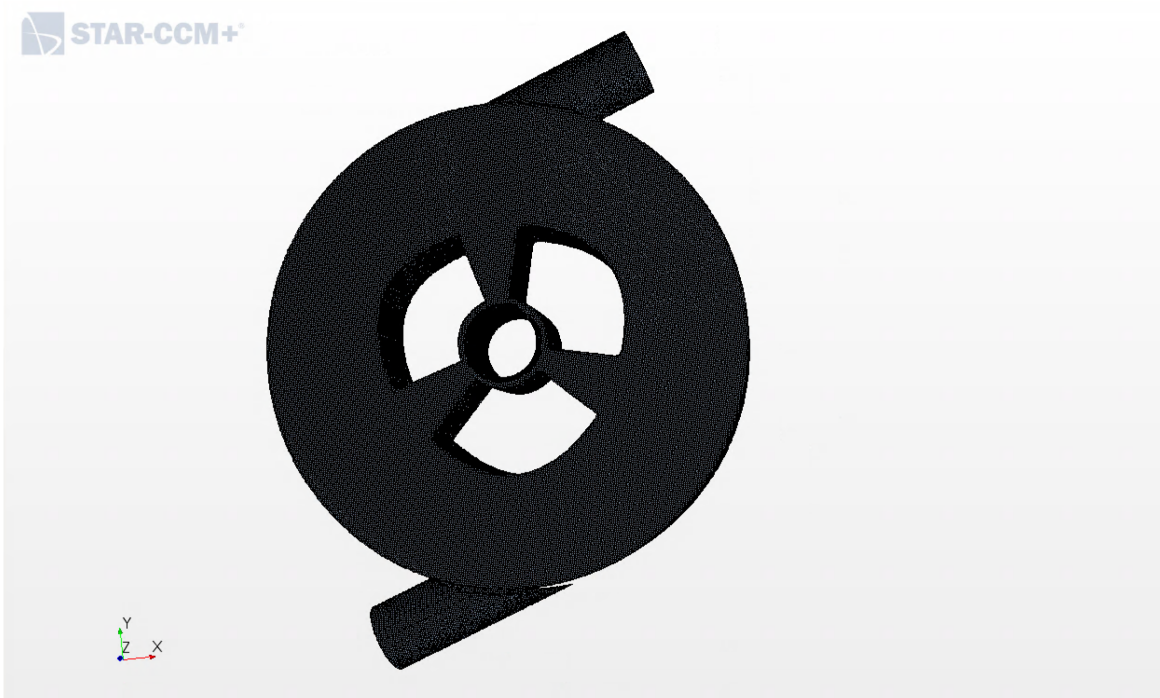

Fig 2.18 – Mesh scene on the turbine

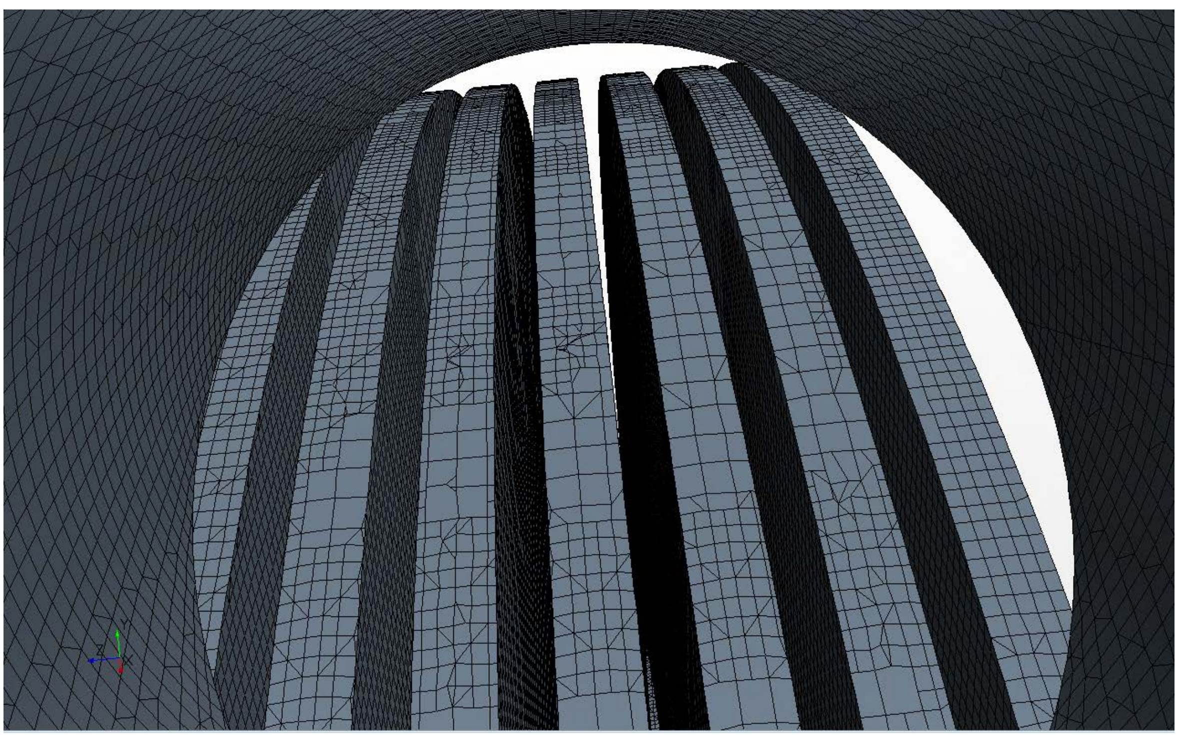

Fig 2.19 – Mesh scene on the inlet and disks

The above scene also shows the prism layer mesh that were created on the surfaces.

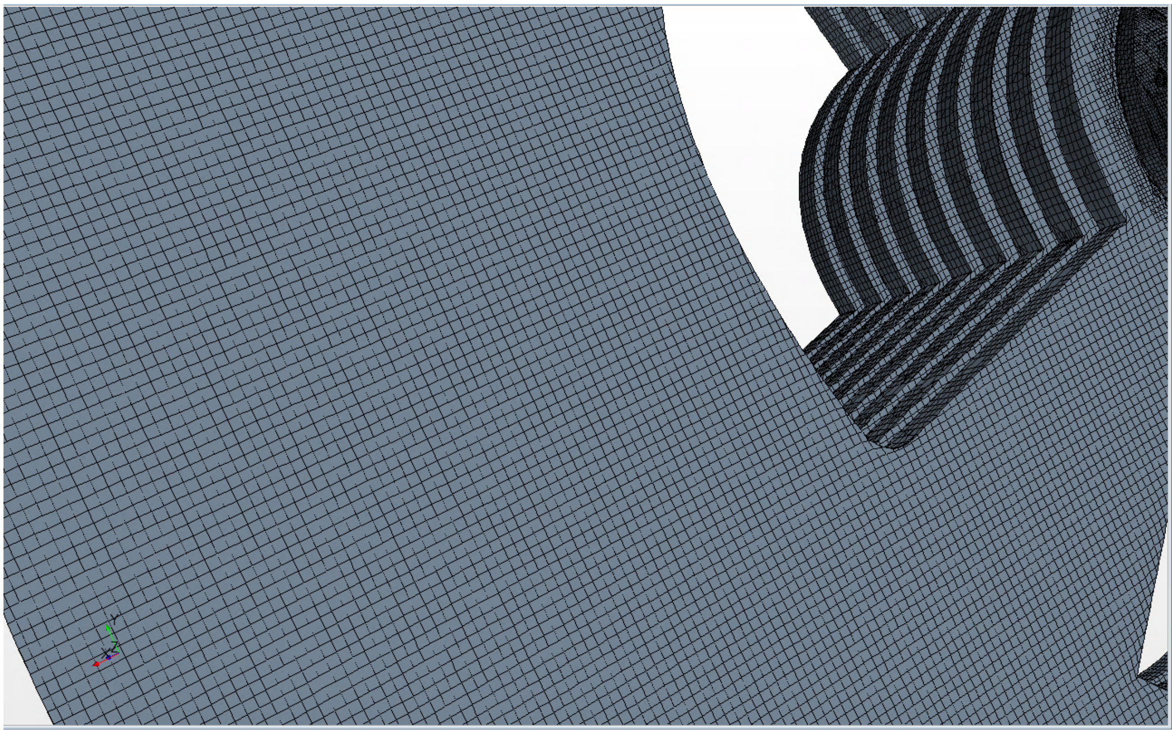

Fig 2.20 – Uniform grid on the rotor

Post the meshing operation, the simulation was run using the same inlet conditions that were used previously.

Table 2.22 – Initial Conditions

Inlet Velocity	230 m/s
Pressure	1378.7kPa
Inlet Temperature	293K
RPM OF Rotor	9000 RPM

The simulation was run for 1500 iterations and the converged residuals are as shown below:

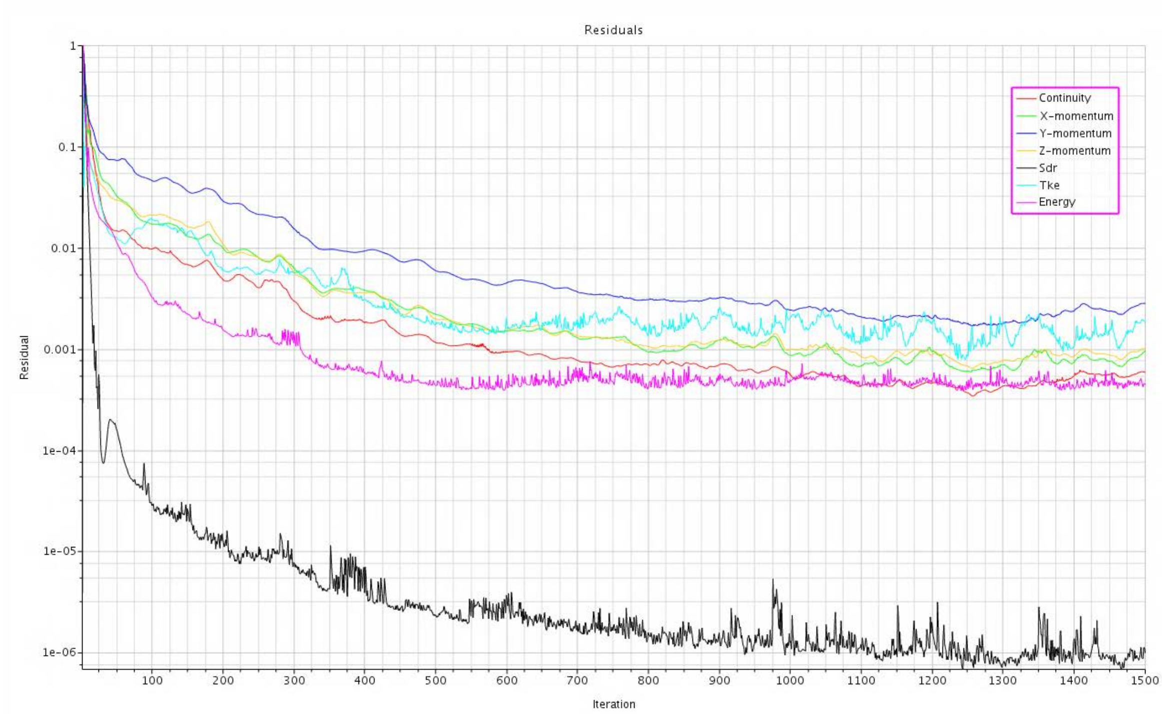

Fig 2.21 – Converged Residuals

The Pressure Scene that was created after the complete run is shown below:

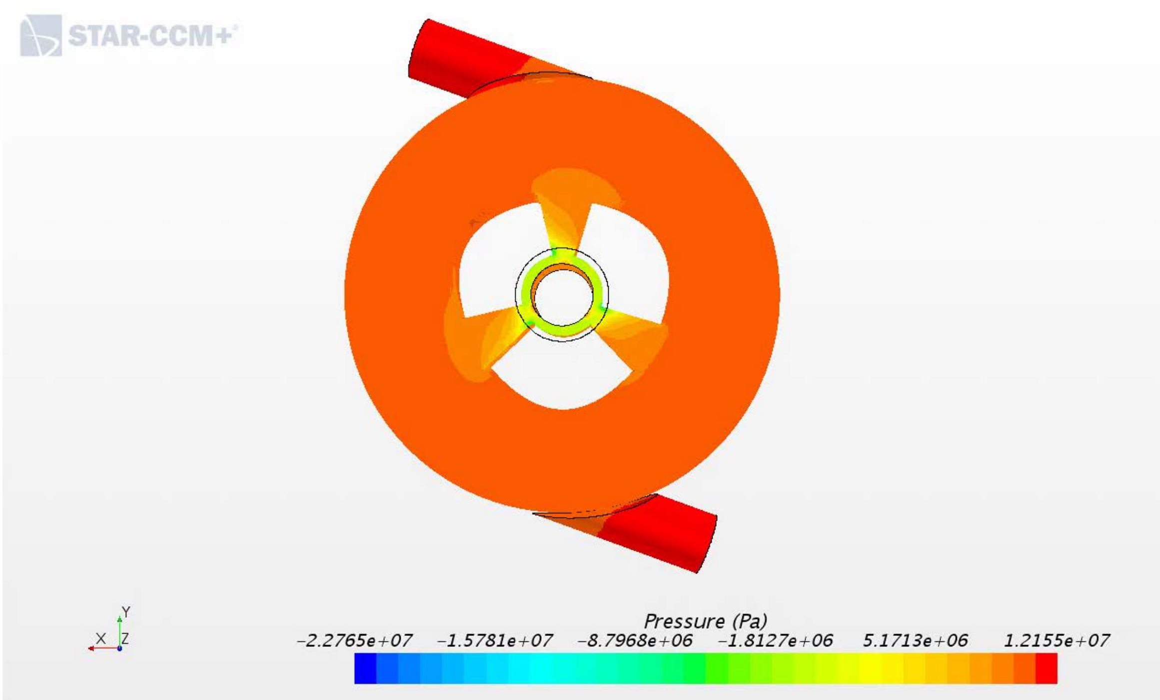

Fig 2.22 – Pressure Scene

The Pressure Scene that was created is a little different from previous one. The pressure of the working fluid is higher throughout the turbine except at the outlet.

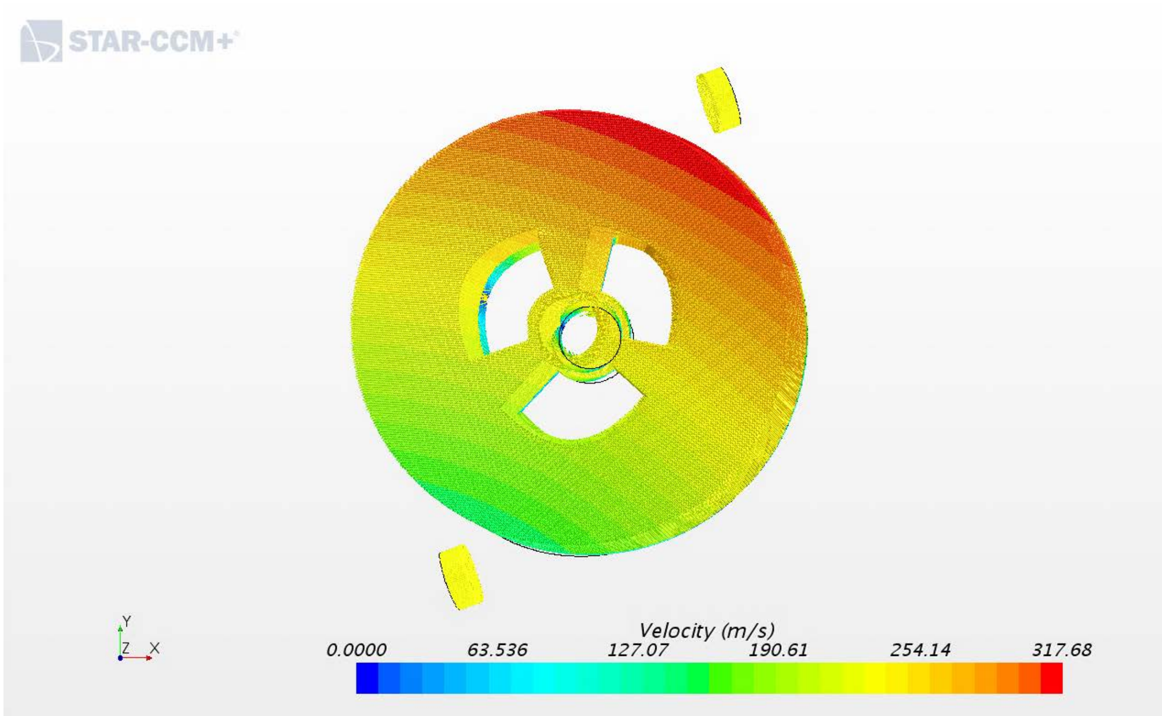

Fig 2.23 – Velocity Scene

The velocity of the fluid is the same as the previous run. There might be negligible difference between the previous and the current one.

The values that were obtained after the complete simulation is shown below:

Table 2.23 – Values obtained after the run

Force on Rotor	6407.3 N
Inlet 1 Pressure	11321.81kPa
Inlet 2 Pressure	11338.2kPa
Outlet Pressure	175.6kPa

2.2 Analytical Procedure

After the simulation was run and all the values that was required of it has been gathered, an analytical procedure is carried out to find the results from the input values that had been used for the simulation.

For the analytical procedure, Engineering Equations Solver (EES) [17] has been used to find out results that could not be processed on StarCCM Plus Software package. The screen shot of the formula that have been used for calculations is shown below:

Fig 2.24 – Formula used for calculations

Since Design 1 was used only to find out the computational capability of the software, therefore, it would be neglected from the further analytical procedure. After the value were input onto EES, the other parameters were calculated and is shown below:

Table 2.24 – Values calculated using EES

Parameters	Design 2	Design 3
Angular Velocity (radians/min)	942.5	942.5
Power of Turbine (kW)	0.10	0.14
Power per Disk (kW)	0.012	0.02
Isentropic Power of Turbine (kW)	0.66	0.66
Isentropic Power per Disk	0.074	0.074
Torque (kN-m)	0.0001	0.0001
Isentropic Efficiency (%)	15.8	15.7

From the figures in Section 2.1, it is seen that there are sonic velocities at certain points in the periphery of the turbine. This might mean that the flow would have turned compressible although for the flow simulation, constant density incompressible flow was considered as most literatures have.

If we assume compressible effects, then the calculations would show the mass flow as below:

Mach Number, M = $\frac{V}{C}$

Where V – local velocity which is specified as 230m/s

C – Speed of sound which is 343m/s at 20°C

Therefore the Mach Number as per the specified values would be:

M = $\frac{230}{343}$ = 0.67

This value is very high. If Mach Number is between 0.3 to 0.8 then it is considered subsonic and compressible. As per the calculation above, we can see that the flow did turn compressible at some point which could also be seen in the figures in Section 1.2.

Since the Mach Number is high, the flow would behave as a compressible flow. To find out the mass flow of the turbine, the following equation [42] was used:

$$\dot{\mathrm{m}} = \frac{AP}{(T)^{\frac{1}{2}}} * \left(\frac{ɣ}{R}\right)^{\frac{1}{2}} * M((1 + \frac{ɣ - 1}{2} * M^2)\wedge(-(\frac{ɣ + 1}{2 * (ɣ - 1)})$$

where A – area where r = 0.09m, therefore A = 0.025m^2

M – Mach Number M = 0.6 which was calculated already

T – Temperature T = 294K

P – Pressure P = 1480kPa

ɣ - Specific Heat Ratio ɣ = 1.401

R – Gas Constant R = 286.9 J/kg-K

Plugging in those values onto the equation it gives us the value of mass flow:

ṁ = 0.094kg/s

This value would be reasonable for a compressible flow.

CHAPTER 3: RESULTS AND CONCLUSION

3.1 Results

The following results were obtained after flow simulation was carried out on all the three designs. The comparison of all the three designs are shown in the tabular format below:

Table 3.1 – Comparison of the three models

	Design 1	Design 2	Design 3
Number of Disks	4	9	9
Working Fluid	Compressed Air	Compressed Air	Compressed Air
Diameter of Disk (inch)	8	7	7
Stator Diameter (inch)	8.2	7.2	7.2
Spacing between Disks (inch)	0.25	0.15	0.15
Thickness of Disk (inch)	0.02	0.095	0.095
Number of Inlets	1	2	2
Inlet Angle	0°	0°	20°
Number of Mesh Cells	7410683	11514734	11987713
Inlet Velocity (m/s)	199.27	230	230
Inlet Temperature (K)	293	293	293
Rotor RPM	16000	9000	9000
Force on Rotor (N)	24.95	6365	6407
Inlet Pressure (kPa)	101.325	11213.4	11330.03
Outlet Pressure (kPa)	97.2	173.8	175.6
Inlet Mass Flow (kg/s)	0.00009	0.0033	0.0035
Outlet Mass Flow (kg/s)	0.000105	0.004	0.004
Isentropic Efficiency (%)	1.66	15.8	15.7

It must be reiterated that the first design was done to gauge the compatibility of the StaCCM Plus [18] Software package. So the isentropic efficiency is very less. After the necessary changes to the model as well the computational method were made, the isentropic efficiency of the turbine can be seen to have increased. There is a difference of approximately 0.1% between Design 2 and Design 3 although the orientation of the inlets have been changed while keeping the other parameters same.

Since the designs that were made were an extension of Warren Rice's experimental design, a table is shown below for comparison:

Table 3.2 – Comparison of Rice and Design2 and Design 3

	Warren Rice	Design 2	Design 3
Outside diameter (inch)	7	7	7
Numbers of disks	9	9	9
Thickness of the disk (inch)	0.094	0.095	0.095
Space between disks (inch)	0.063	0.15	0.15
Inward flow direction	15°	0°	20°
RPM	9400	9000	9000
Working Fluid	Compressed Air	Compressed Air	Compressed Air
Number of Inlets	2	2	2
Power (kW)	0.84	0.66	0.66
Power per Disk (kW)	0.09	0.07	0.07
Efficiency (%)	16.5	15.8	15.7

From the comparison table above, it is seen that the efficiency and power. This might be because the spacing of the disks is different and the RPM of Rice's design is higher.

Also there is a discrepancy in the mass flow rates for both the designs. Upon further investigations it has been found that there are irregularities in the design. This would be addressed in the future works. The irregularities are shown in the figure below. There are gaps between the shaft and the disks on the shaft. So some of the fluid must have escaped without actually being put to use for developing torque.

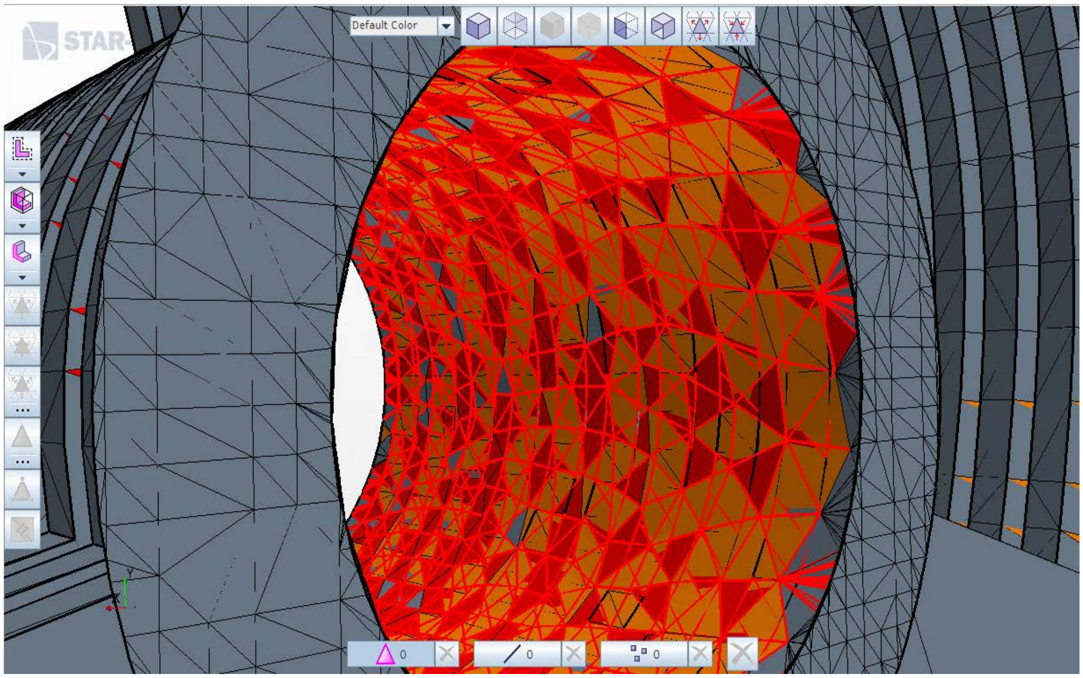

Fig 3.1 – Red color showing design irregularities

3.2 Conclusion

The computational method has been refined to obtain better solution from the software. It has been seen that from Design 2 and Design 3 that when the initial conditions are controlled which includes the inlet velocity, inlet pressure as well as the inlet temperature, then the solution yields better results. Since the topic of this research is design analysis and concerns the isentropic efficiency of the turbine at different inlet angles, the difference in the solution is evident. As discussed above that the losses at the outlet could be greater if the outlet is small, the circular hole was created along the circumference of the shaft so that it would give optimum efficiency.

Since both the designs are almost the same except for the orientation of the inlets, it can be seen that in Design 2, where the inlets are horizontal, the efficiency is better. Even after the input conditions were kept the same, it showed better efficiency. The designs were made keeping in mind the experiments done by Warren Rice. The efficiency investigated in the research paper by means of flow simulation on StarCCM Plus has yielded results which fall within values quoted in the introduction based on the literature reviews. It has been seen in the literatures that reducing the inter-disk spacing also yielded better results.

It would be justifiable to arrive at the conclusion that precise optimization of the Tesla Turbine's design and geometrical properties could obtain results that would be comparable to regular bladed turbines used for small applications. It would provide to be an attractive alternative for small power plants or heat recovery systems.

3.3 Future Work

The Tesla Turbine is a machine which is worthy of future investigation. Although researchers have been working on it since the mid-1960s, there is a lack of consensus between them regarding the standard efficiency of it. Since Nikola Tesla himself did not specify the parameters regarding the building of this turbine, it is safe to say that there is a lack of proper research on it. Some of the features which would interesting to investigate further would be:

- Optimization of the geometry of the turbine to look into the inter-disk spacing, the diameter of the disks, number of disks as well as disk thickness.
- Designing the disks and the shaft as one part rather than different assembled parts.
- Improvement in the computational method by improving the meshing of the parts of the turbine with better computers.
- Evaluation of other turbulent methods for flow simulation.

- Effect of the number of input nozzles on the efficiency and mass flow.
- Effect of different outlets on the efficiency.
- Build experimental Tesla Turbine and find out the various parameters which would affect the efficiency.
- Varying the rotational speed of the rotor for efficiency calculation.

These are some of the work which could be done to contribute towards the development of the Tesla Turbine.

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