NEW HORIZON
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Mini Project Report
On

"Optimization of Centrifugal Compressor for Miniature Gas-Turbine by using Inlet Pre-Whirl"

Submitted by

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In partial fulfillment of

**BACHELOR OF ENGINEERING
IN
MECHANICAL ENGINEERING**

Under the guidance of
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CERTIFICATE

It is certified that the Mini project (Phase-II) entitled **"**Optimization of Centrifugal Compressor for Miniature Gas-Turbine by using Inlet Pre-Whirl**"** is a bonafide work carried out by **R Ganesh Nag, Roshan Suhail, Shashank N, Vishnu Tej S (1NH15ME734, 1NH15ME741, 1NH15ME746, 1NH15ME758)** for the partial fulfillment for award of degree of **Bachelor of Engineering in Mechanical Engineering** of **New Horizon College of Engineering**, Bangalore during the year 2017-2018. It is further certified that all corrections/suggestions indicated for internal assessment has been incorporated in the report deposited in the department library. The Mini Project (Phase-II) has been approved as it satisfies the academic requirements in respect of Mini Project Work prescribed for the **Bachelor of Engineering** degree.

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DECLARATION

We hereby declare that the entire work embodied in this dissertation has been carried out by us and no part of it has been submitted for any degree of any institution previously.

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ABSTRACT

Gas Turbine is a continuous internal combustion engine which creates thrust by burning fuel in its combustion chamber and using the fast flowing gases to operate the Turbine which in turn powers the Compressor at the inlet. As a general rule it is difficult to control Gas Turbines because the work at high rotational speed. It is much more difficult to control miniature Gas Turbine because they are supposed to be simple and light in weight. One way to control them is by using Variable Inlet Guide Vanes.

Variable Inlet Guide Vanes (VIGV) are fixed vanes which are found in different applications and help in directing water, gas or air in to a flow which creates a maximum efficiency or working conditions. As impellers increase or decrease the mass flow rate in a Gas Turbine these VIGV ensure that the mass flow rate is passed evenly and as smoothly as possible.

A gas turbine engine in which the angle of the Inlet Guide Vanes can be changed to meet the requirements of the engine operating conditions. The Inlet Guide Vanes are designed based on the dimensions of the Compressor. The air flows through the VIGV which creates the swirl and the swirling air hits the Compressor blades with a certain angle of attack which makes the air to get accurately pressurised at the given rotational speed and the efficiency of the Gas Turbine also increases

A high speed compressor derived from turbo-charger for use in miniature gas turbine applications was tested with Inlet Guide Vanes with different configuration as effective means of controlling surge and choke. The flow characteristics were compared with and without Inlet Guide Vanes. It was seen that the flow characteristics had improved with the inclusion of Guide Vanes than by just the Compressor. A virtual model of the Turbo-Compressor with Casing and Inlet Guide Vanes were designed and appropriate working conditions were devised for CFD simulation by using ANSYS Fluent. Here we wanted to demonstrate the role of Inlet Guide Vanes in controlling surge and choke.

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LIST OF SYMBOLS

- $\frac{\partial}{\partial t}$ - Time variable
- ρ - Density
- κ - Turbulent Kinetic Energy
- μ - Viscosity
- D_κ - Diffusion coefficient of Turbulent Kinetic Energy
- ∇ - Nabla/Del
- G - Generation of Turbulent Kinetic Energy
- u - Fluid Velocity
- ϵ - Dissipation rate
- S - Source term
- $D\epsilon$ - Diffusion coefficient of dissipation
- C_1 - Characteristic flow constant
- C_2 - Characteristic flow constant
- ν - Velocity vector
- ν_t - Turbulent Viscosity
- C_μ - Model Constant (0.09)
- A - Cell Surface Area
- U - Internal Energy per unit mass of fluid

CHAPTER-1

INTRODUCTION

1.1 Introduction

Gas turbine which is also known as Jet engine is an internal combustion reaction type engine, which produces a fast moving jet, which in-turn provides the thrust. This is known as jet propulsion. A typical gas turbine is a highly technical, geometrically large and a very expensive machine which is engineered, manufactured and produced with the highest possible precision techniques. [1] The leading manufacturers of the gas turbines like GE, ROLLS ROYCE and PRATT&WHITNEY etc.

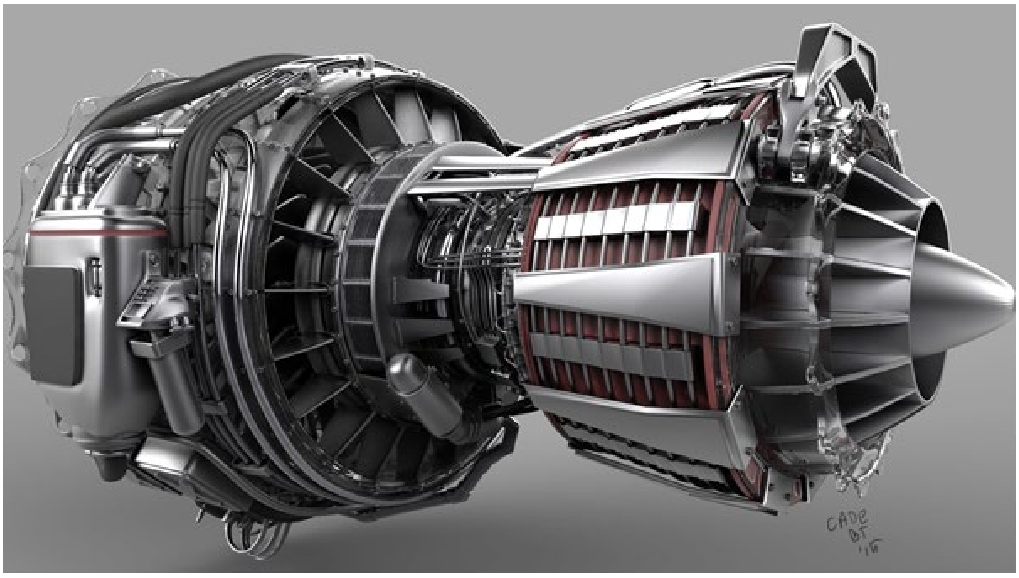


FIG 1: Typical large scale Gas Turbine from PRATT & WHITNEY

Concentrate in manufacturing engines for large scale applications. Hence research towards small scale applications of gas turbine is often neglected. A miniaturized gas turbine finds its applications mainly in militaristic devices such as drones, weather forecast, agriculture etc. As there is no in depth research done in this field, the manufacturing of these mini gas turbines turns out to be an extremely expensive affair. The small gas turbines which are made with the required reliability end up being extremely expensive partly because they are made in such small numbers and also because their configurations change for each application. There are many acceptable configurations of the miniature gas turbines, the most commonly accepted design is the use of single stage compressor and turbine, with the compressor being centrifugal in nature. This project focuses primarily towards the compressors used in gas turbine. The Compressor stage of a gas turbine has a primary function to supply air in sufficient quantity to enable the smooth functioning of downstream stages. To fulfil its exact function, the Compressor must increase the pressure of the air received and to discharge it at the pressures required for smooth functioning of the entire Engine and also to maintain the pressure gradient.

FIG 2: Centrifugal Compressor derived from Garrett GT28 Turbo-Charger

1.2 Types of Compressors

There are only two types of compressors used in gas turbines [2]

a) Axial flow compressors: Used in large scale applications.

b) Radial flow compressors: Used in small scale applications.

These radial flow compressors are also known as centrifugal compressors. In Gas Turbines Radial flow Compressor consists of an impeller and a diffuser. Centrifugal Compressors can achieve high pressure rise per stage than other forms of Compressors this is why they are preferred in miniature applications. The impeller is usually made up of forced aluminium alloys which are machined, heat treated and smoothed for maximum flow and minimum turbulence.

Through this project we have oriented our research towards the centrifugal compressor. There are a wide variety of centrifugal compressors that are currently used in the market. One such application in which centrifugal compressors are pre-eminently used, are the turbochargers of automobiles. As this research is focused on miniaturization of gas turbine, it can be taken into account that the Turbo-compressor is geometrically small and it can also be observed that the characteristics of the Turbo-compressors were very similar to that of the required compressor at its working range, relative to the inlet and outlet requirements.

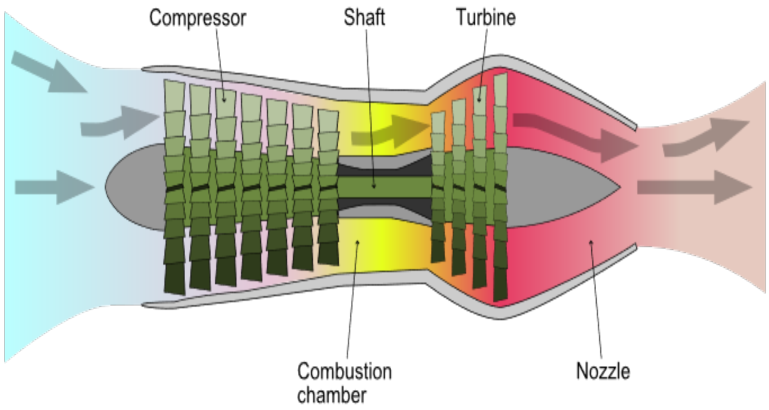


FIG 3: Gas Turbine with Axial flow Compressor

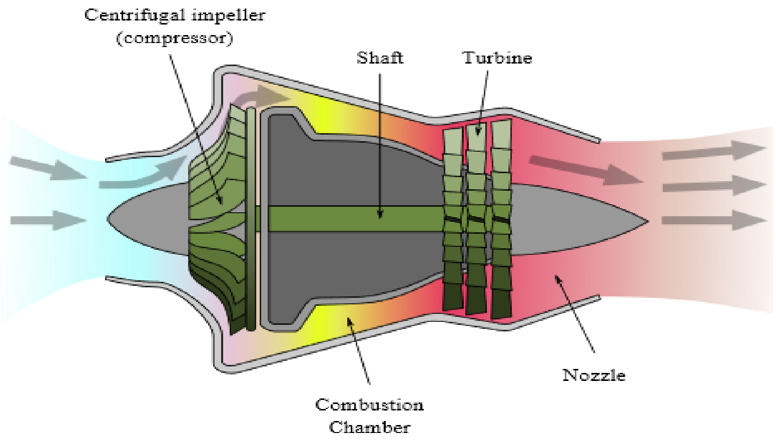


FIG 4: Gas Turbine with Radial flow Compressor

Hence it was theoretically possible by the equation that governs all centrifugal compressors that the pressure ratio created by the turbo-compressor is in accordance with the required conditions at the working range of the gas turbine.[3] The working range of the turbocharger is at a speed range of 60000-120000 rpm at a pressure ratio of approximately 3, when this working range is taken in comparison with the working range of a gas turbine, the speed range of miniature gas turbine is around 150000-200000 rpm at a pressure ratio of 9.

Variable Inlet Guide Vanes (VIGV) are fixed vanes which are found in different applications and help in directing air to flow which creates a maximum efficiency or working conditions. As impellers increase or decrease the mass flow rate in a Gas Turbine these VIGV ensures that the mass flow rate is passed evenly and as smoothly as possible.

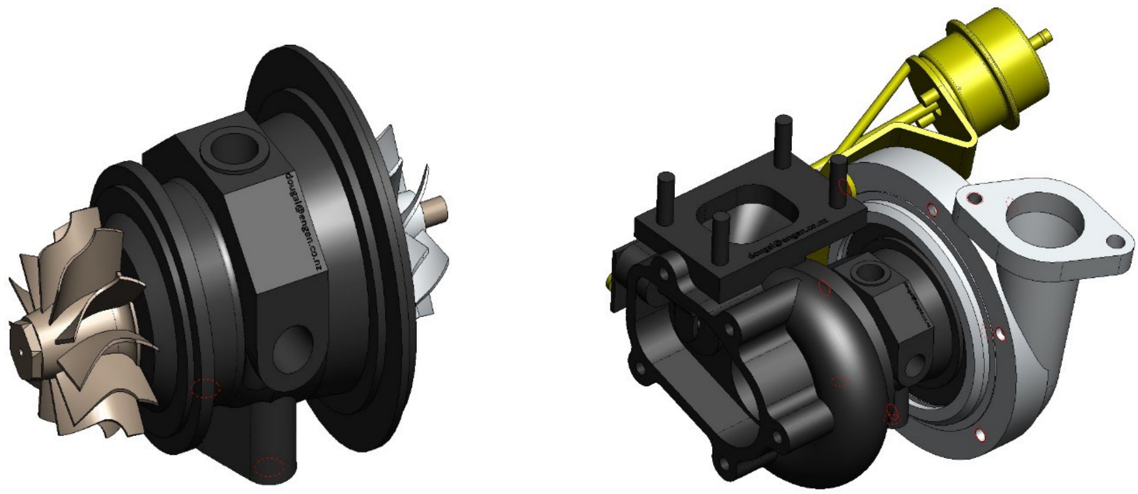


FIG 5: CAD model of Garrett GT28 Turbo-Charger

Inlet Guide Vanes are stationary and guides the air to the compressor by creating a swirl and prevents the Compressor blade damage by increasing or decreasing at the angle of attack.

1.3 Types of Inlet Guide Vanes

Variable Inlet Guide Vanes are of three types

1. Straight Blades.
2. Parabolic Blades.
3. Straight and Parabolic Blades.
4. Reverse Blades

In Straight type of Guide Vanes the blades are designed mostly by using standard NACA ducts. In this Guide Vanes the air flow is simple and not much of a swirl is generated.

The Parabolic type of Guide Vanes are basically curved in structured and have different angles at root and the hub of Inlet Guide Vanes. The swirl generated is much better than the straight blades.

The straight and parabolic Guide Vanes (mixed profile) are a combination of both the blades (straight, parabolic) where the blades are parallel to the flow direction initially with straight profile and then gradually curve parabolically to the required angle. In this, the blade angle varies at different lengths of the blade. It starts at about 15° at the hub and ends at 45° at the root.

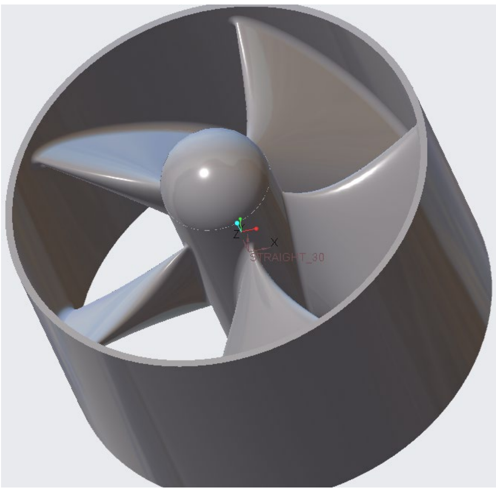


FIG 6: Inlet Guide Vanes of straight blade profile

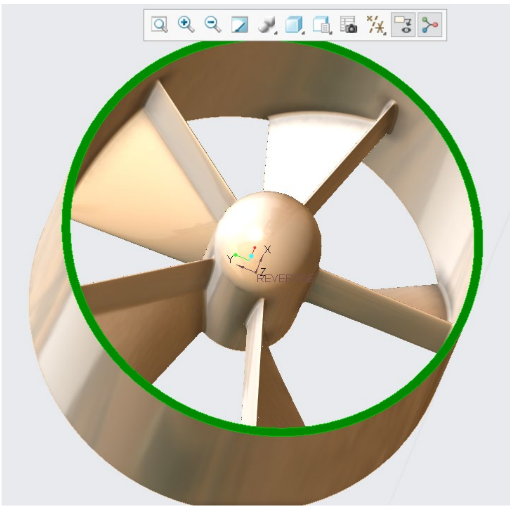


FIG 7: Forward blade with mixed profile

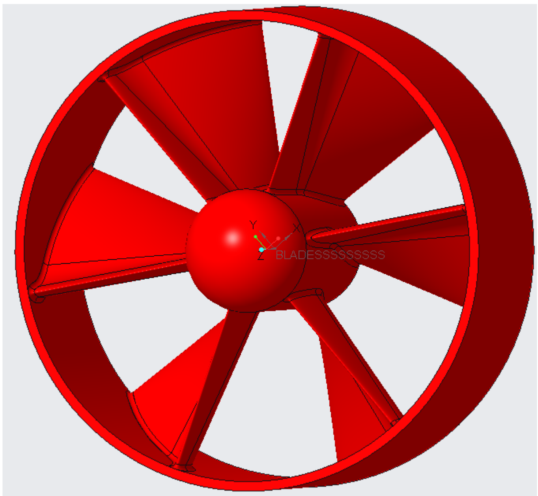


FIG 8: Reversed blades with mixed profile

CHAPTER-2

HYPOTHESIS

2.1 Introduction

Even though the pressure ratio increases due to the rapid increase in the speed, this pressure ratio cannot be sustained due to the phenomena surge and choke. So we through this project suggest the adoption of the below mentioned methods in order to sustain the pressure ratio created.

- Inlet guide vanes[6]
- Diffuser vanes
- Use of vortex generator

This project exclusively deals with the changes that might occur on the account of inclusion of guide vanes at inlet of the turbo-compressor. These guide vanes induces pre-whirl. Pre-whirl is the phenomenon of adding a circular component to the inlet air. The circular component maybe towards or away from the inlet blade angle of the compressor. Pre-whirl reduces the angle of attack and increases the mass flow rate and hence the surging is prevented. [7] By accommodating the guide vanes with a positive angle of attack, a positive pre-whirl can be generated, which in-turn increases the speed of the compressor blades. By accommodating the guide vanes with a negative angle of attack, a negative pre-whirl is generated, which in-turn decreases the speed of the compressor blades. If surge is detected, a positive angle is given to the inlet guide vanes in-turn increasing the flow rate of air. If choke is detected a negative angle of attack is established which in-turn produces a negative pre-whirl this reduces chocking.

CHAPTER-3

BACKGROUND RESEARCH

3.1 Introduction

The pressure ratio in the gas turbines is in the range of 9:1 to the atmosphere. By the journals, we understood that by using Inlet Guide Vanes the pressure ratio, mass flow rate of air can be increased or decreased. When Inlet Guide Vanes are attached in front of the compressor without any angle the flow does not change and if any discrepancies are found, the flow characteristics can then be altered by giving appropriate angles.

3.2 Literature

3.2.1 Components of Gas Turbine: [1] the components of a gas turbine are divided into two sections

i. Cold section.
ii. Hot section.

In cold section Air intake (inlet), compressor or fan, bypass ducts, shaft and diffuser section. In Hot section Combustor (combustion chamber), Turbine, Afterburner (reheat), Exhaust (nozzle).

3.2.2 Centrifugal Compressor: [2] Centrifugal Compressor is also termed as Radial Compressor. The idealized compressive dynamic turbo-machine achieves a pressure rise by adding kinetic energy to a continuous flow of fluid through the rotor or impeller. This kinetic energy is then converted to an increase in potential energy/static pressure by slowing the flow through a diffuser. The pressure rise in impeller is in most cases almost equal to the rise in the diffuser section.

3.2.3 Shuai Guo, Fei Duan and Hui Tang: [3] this authors did a theoretical analysis using a Mat lab program and they also conducted a 3-D CFD model by using a software-Axcent. They concluded that the experimental test for optimal compressor shows a 7.5% increase in pressure ratio and change in the compressor efficiency.

3.2.4 C. Rodgers: [4]paper describes the results of compressor rig testing with a moderately high specific speed, high inducer Mack number, single-stage centrifugal compressor, with a vaned diffuser, and adjustable inlet guide vanes (IGVs). The results showed that the high-speed surge margin was considerably extended by the regulation of the IGVs, even though the vaned diffuser was apparently operating stalled. Simplified one-dimensional analysis of the impeller and diffuser performances indicated that at inducer tip Mach numbers approaching and exceeding unity, the high-speed surge line was triggered by inducer stall. Also, IGV regulation increased impeller stability. This permitted the diffuser to operate stalled, providing the net compression system stability remained on a negative slope.

3.2.5 Compressor choke or stone wall: [5] **it** is an abnormal operating condition for centrifugal compressor. Choking of centrifugal compressor occurs when the compressor is operating at low discharge pressure and very high flowrates. These high flowrates at compressor choke point are actually the maximum that the compressor can push through. Any further decrease in the outlet resistance will not lead to increase in compressor output. This operating condition is also known as stonewalling of a centrifugal compressor.

3.2.6 Michele Becciani, Alessandro Biachini, Giovanni Ferrara: [6] the onset of aerodynamic instabilities in proximity of the left margin of the operating curve represents one of the main limitations for centrifugal compressors in turbocharging applications. An anticipated stall/surge onset is indeed particularly detrimental at those high boost pressures that are typical of engine downsizing applications using a turbocharger. Several stabilization techniques have been investigated so far to increase the range ability of the compressor without excessively reducing the efficiency. One of the most exploited solutions is represented by the use of upstream axial variable inlet guide vanes (VIGV) to impart a pre-whirl angle to the inlet flow. In the pre-design phase of a new stage or when selecting, for example, an existing unit from an industrial catalogue, it is however not easy to get a prompt estimation of the attended modifications induced by the VIGV on the performance map of the compressor. A simplified model to this end is presented in the study. Figuring out a typical industrial pre-design phase, the model assumes the availability of the original performance data of the compressor without pre-whirl and only very few geometrical parameters. Based on fluid dynamic considerations and some additional models and correlations, a procedure is defined to correct the attended stage pressure ratio and efficiency as a function of the pre-whirl angle imposed by the VIGV. The model has been successfully validated using an experimental literature case study and is thought to represent a new useful preliminary tool for turbocharger designers.

3.2.7 Xinqian Zheng and Anxiong Liu: [7] they mentioned that Centrifugal compressors usually include a broad operating range between surges and choke. This becomes increasingly difficult to achieve as increased pressure ratio is demanded. In order to suppress the tendency to surge and extend the operating range at low flow rates, inlet swirl is often considered through the application of inlet guide vanes. To generate high inlet swirl angles efficiently, an inlet volute has been applied as the swirl generator, and a variable geometry design developed in order to provide zero swirl. The variable geometry approach can be applied to increase the swirl progressively or to switch rapidly from zero swirl to maximum swirl. The variable geometry volute and the swirl conditions generated are described. The performance of a small centrifugal compressor is presented for a wide range of inlet swirl angles. In addition to the basic performance characteristics of the compressor, the onsets of flow reversals at impeller inlet are presented, together with the development of pressure pulsations, in the inlet and discharge ducts, through to full surge. The flow rate at which surge occurred was shown, by the shift of the peak pressure condition and by the measurement of the pressure pulsations, to be reduced by over 40 percent.

3.2.8 Xinqian Zheng, Anxiong Liu: [8] they tested using a pre-whirl which is an effective way to supress compressor instability. Compressors usually employ a high degree of positive inlet pre-whirl to shift the surge line in the performance map to a lower mass flow region. The efficiency of a compressor at high inlet pre-whirl is far lower than that at zero or low pre-whirl. They investigated the performance of a centrifugal compressor with different pre-whirl and develops flow control methods to improve efficiency at high inlet pre-whirl the pre-whirl is design based on velocity condition at the inlet boundary. This paper lays a theoretical basis for overcoming the efficiency drop induced by high inlet pre-whirl and for developing compressors with high inlet pre-whirl.

CHAPTER-4

CHALLENGES

If this turbo-compressor is to be incorporated into a gas turbine, it is going to run at a speed of 150000-200000 rpm, therefore the pressure ratio of the turbo-compressor increases automatically. The following are the challenges that might be posed due to the increase in the speed and the pressure ratio:

- The relative velocity between the blades and inlet air is very high, therefore shock maybe induced.
- Due to the high speeds, turbulence is created, which in-turn creates vibrations, which might lead to failures.
- Since turbo-compressors work at an optimum rpm the output air flow might be less due to pressure build up, hence surging might occur.
- Also if the air flow is too high then chocking might occur.

4.1 SURGE: To understand surge one needs to know the working principle of the compressor, which is imparting kinetic energy to the fluid at the impeller and then sacrificing this kinetic energy at the diffuser to increase the static potential of the fluid and decreasing the kinetic energy of the fluid.[4] If maximum head capacity is reached, then pressure in diffuser will be greater than pressure at impeller outlet. This will prevent fluid from moving further at impeller outlet and causes the fluid in diffuser to flow back, i.e. flow reversal takes place. This can be deteriorating as it has potential to damage the bearings and other rotating parts, and also cause high vibrations.

4.2 SHOCK: When the relative velocity between the working fluid and blades reach the speed of sound, there are characteristic waves produced causing damage to the blades.

4.3 CHOKE: It is operating point at minimum flow capacity condition where high velocity air flow causes severe damage on the compressor creating instability [5].

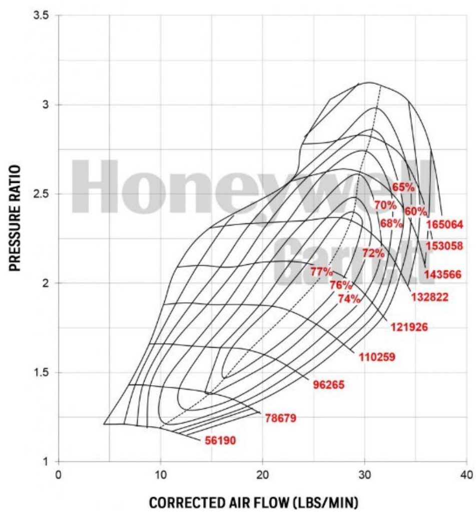


FIG 9: Performance map of Centrifugal Compressor

CHAPTER-5

EXPERIMENTAL PROCEDURE

5.1 Introduction:

The design of pre-whirl is based on research of blade angles where they are different blade angles at pre-whirl blades are designed. At first we designed a straight blade were the flow is un-uniform and decided to study on blade design and came up with an initial half design was straight to ensure smooth transition of blade angle and swirl component.

Then we designed another blade with a half parabolic and other half straight where still there was no uniform flow of air, so we came up with another blade design were the design of the blade is such that the blade angle is increasing from the root tip till the edge of the blade casing to compensate for the speed increase along the radius of the blade.

Compressor we have designed similar to that of GARRETT GT-28 turbo charger which is famous because it was used in Nissan skyline GT-R and its characteristics are as follows, the flow capacity of a turbo is about 5-35$\frac{lb}{min}$ and works well for engine of capacity from 1litre-3.1litre engine. The trim of the compressor is 62. The Inducer is 44.60mm, Exducer is 60.1mm and A/R ratio is about 0.42.

The simulations were done using ANSYS Fluent. The flow field analysis was made to find out the effect of Inlet Guide Vane on Compressor performance investigation was done at 150000 RPM. As predicted the Inlet Guide Vanes create noticeable change in the Compressor characteristics. Positive angled vanes change the characteristics curves and reduce the mass flow rate which improves the efficiency at very high rotational speeds. With increase in the positive angle it becomes easier to control surge at low rotational speeds but the pressure ratio will also decrease with the angles.

Negative vane angles have the opposite effect on Compressor performances. The efficiency of the Compressor is reduced with increase in the negative angle also there is no appreciable change in the pressure ratio. The probability of surge is very high at low rotational speeds but choke is prevented at higher rotational speeds.

The following steps were taken to simulate the above procedures.

5.2 Design

- **Design of Inlet Guide Vanes in Solid Edge**
  1. Diameter of Hub = 13 mm
  2. Diameter of shroud = 48.9 mm
  3. The Straight Blades were designed by using data from NACA 0012 aerofoil.
  4. The Blade angle at Hub (for 30° blade) = 15°.
  5. Angle of Blade at shroud = 45°.
  6. The length of the entire assembly = 21 mm

The Inlet Guide Vanes were designed and assembled in solid edge and separated at a distance of 100mm.

5.3 Procedure

- **Procedure for Simulation in ANSYS Fluent.**

5.3.1 Geometry – in this step the Compressor Guide Vane assembly is imported and appropriate material is selected for the simulation.

- **Creation of enclosure** – an enclosure is created to obtain the flow field required for flow analysis. Enclosure details =
- **Boolean** – since we want to simulate only the flow we have to remove the solid parts hence Boolean is created for subtraction of solid parts. The Compressor has to be rotated for this reason we have to create another flow field which contains the area which encloses the spinning Compressor.

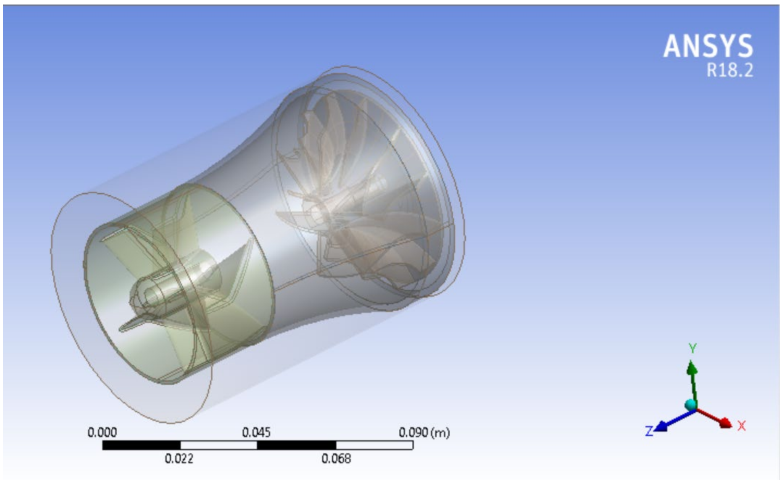


FIG 10: Assembly of Inlet Guide Vanes and Turbo-Charger Compressor GT28

5.3.2 Meshing – in this step the entire flow field is divide into number of elements so that when different parameters are applied they are distributed evenly along the entire field. For this particular application we used tetrahedral elements. Mesh relevance is fine, the inlet, outlet and wall are named. Mesh is generated.

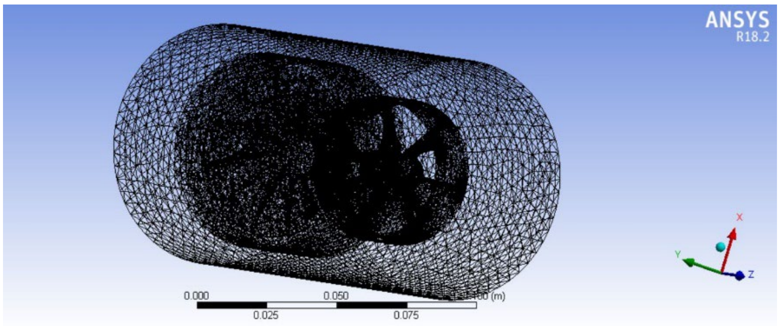


FIG 11: Meshed model

5.3.3 Solver - in this step all the parameters and boundary conditions are defined and the simulation is solved by using mathematical equations.

Steps during the process

- General
  - Solver
    - i. Type - Pressure based
    - ii. Velocity Formulation – Absolute
    - iii. Time – Transient

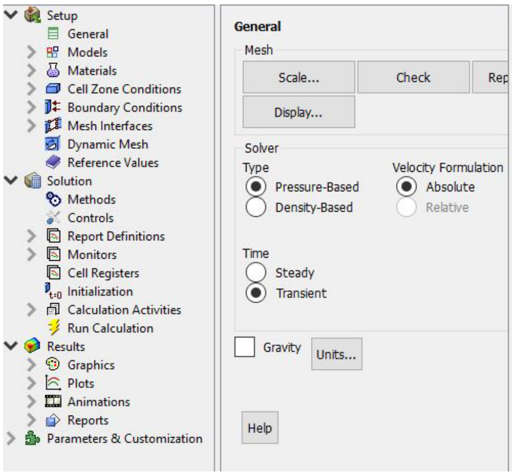


FIG 12: Selection of computational solution

We select pressure based equations because it contains turbulent motion with swirl components and pressure gradiance along the length. We choose transient time because the properties and parameters change with respect to time.

- Models
    - Viscous – Realizable k-e, Scalable Wall Function.
        1. K-Epsilon (2 equation)
            - I. K-Epsilon model – Realizable.
        2. Near Wall Treatment – Scalable Wall Function

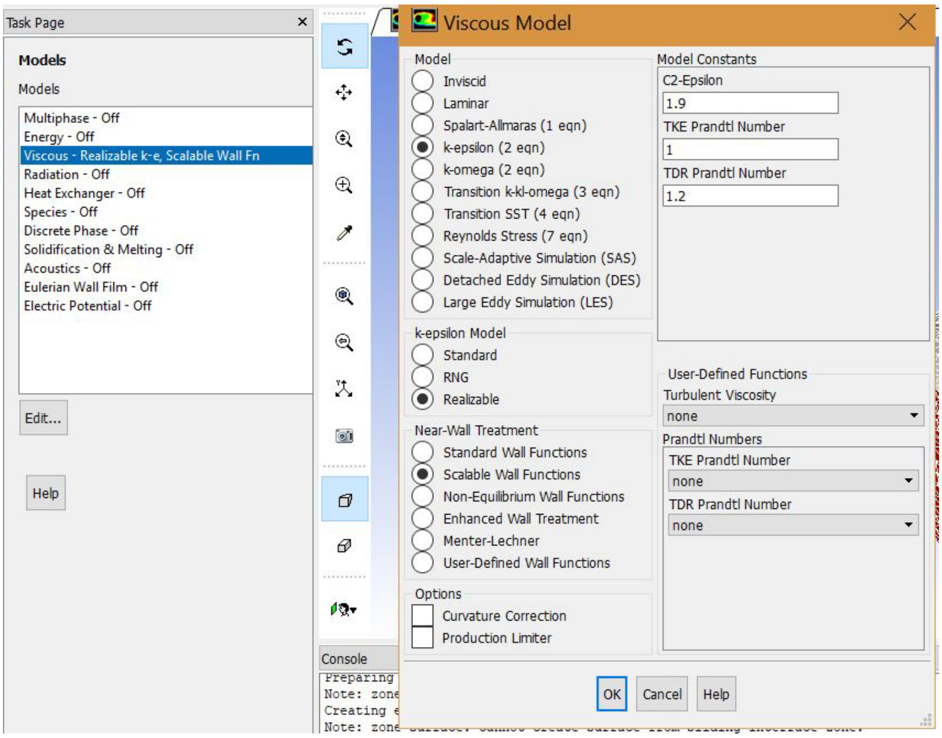


FIG 13: Flow models

K-Epsilon turbulence model is a linear eddy viscosity model which was created by Harlow and Nakayama in 1968 is the most popular two equation eddy-viscosity–using turbulence model. This model lays emphasis on the mechanisms and process that affect the turbulent kinetic energy. This model assumes that turbulent viscosity is isotropic. The ratio between Reynolds stress and mean rate of deformation is same in all directions. It requires near wall treatment.

Realizable K-Epsilon is suitable for complex shear flows involving rapid strain, moderate swirl, vortices and locally transitional flows which may contain boundary layer separation, massive separation, vortex creation behind bodies, stall in diffusers etc. this is used for our modelling because it is accurate and easy to converge

The turbulence kinetic energy equation is given by:

$$\frac{\partial}{\partial t}(\rho\, k) \;=\; \nabla\cdot(\rho\, D_{\kappa}\, \nabla\kappa) \;+\; \rho\, G \;-\; \frac{2}{3}\rho\,(\nabla\cdot u)\,\kappa \;-\; \rho\,\epsilon \;+\; S_{\kappa}$$

The dissipation rate is given by:

$$\frac{\partial}{\partial t}(\rho\,\epsilon) = \nabla\cdot(\rho\, D_{\epsilon}\,\nabla\epsilon) \;+\; C_1\,\rho\,|S|\,\epsilon \;-\; C_2\,\rho\,\frac{\epsilon^2}{\kappa + (\nu\,.\epsilon)^{0.5}} \;+\; S_{\epsilon}$$

The turbulence viscosity is calculated by using:

$$\nu_t \;=\; C_{\mu} \,*\, \frac{\kappa^2}{\epsilon}$$, where C_{μ} is given by

$$C_{\mu} = \frac{1}{A_0 + A_S U * \frac{\kappa}{\epsilon}}$$

- Materials
  - ➢ Flow Medium – Air
    - Density – 1.225 $\frac{kg}{m^3}$
    - Viscosity - 1.7894e-05 $\frac{kg}{m-s}$
  - ➢ Compressor – Aluminum7075
    - Density – 2719 $\frac{kg}{m^3}$

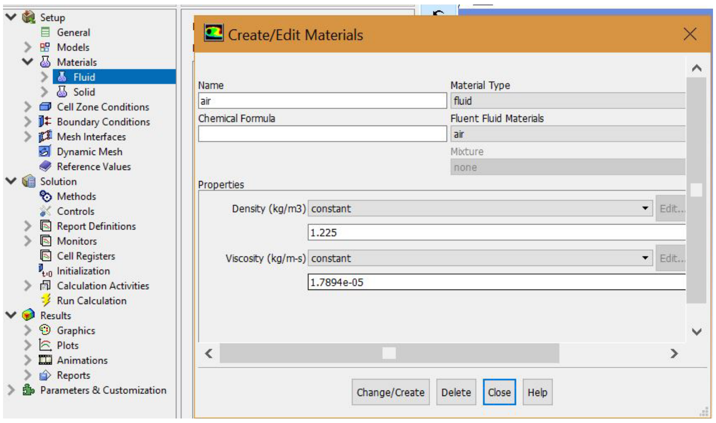


FIG 14.1: Fluid material properties

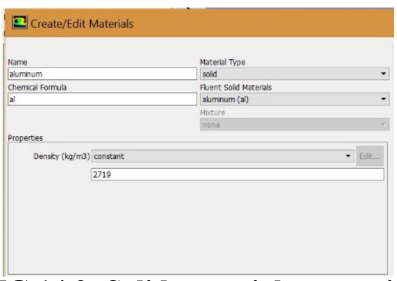


FIG 14.2: Solid material properties

- Cell zone conditions
    - ➢ Compressor Enclosure
        - Material name – air
        - Mesh motion – rotational axis

Origin- (0, 0, 0)

Direction – (0, 0, -1)

Rotational velocity – 150000 RPM

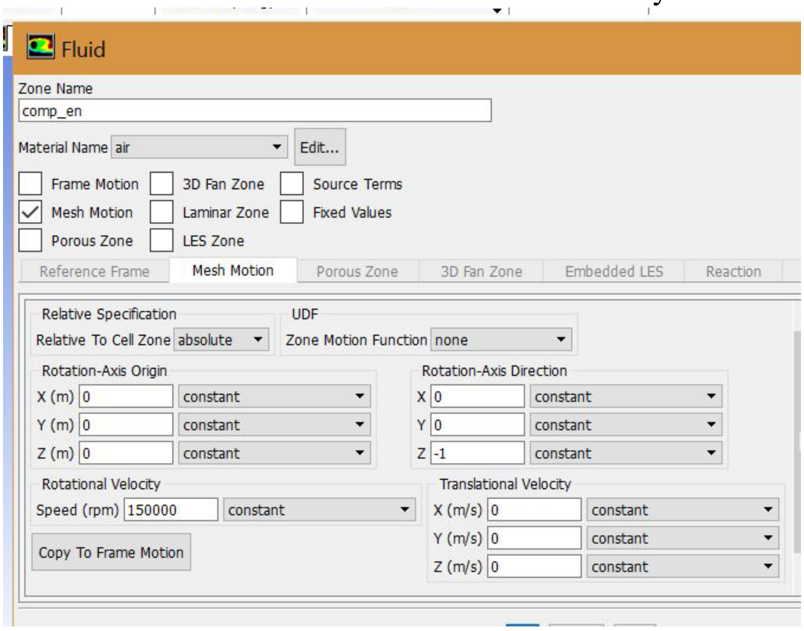


FIG 15: Cell boundary condition for Compressor

Mesh motion is one of the methods in which radial velocity can be imparted to the compressor in a transient flow field. The rotation axis origin here is (0, 0, 0) and the direction of axis is (0, 0, -1). Since ANSYS programmes follow Ampère's right-hand grip rule, in which the rotating component spins along the flow similar to the wrapping of fingers on a rod and the thumb depicting the direction of the axis. Here we need to rotate the compressor in the counter-clockwise direction, hence the direction of axis is given as negative in accordance with the right hand rule.

- Boundary conditions
- ➢ Inlet – pressure inlet
- ➢ Outlet – pressure outlet
- ➢ Wall – Non shear wall

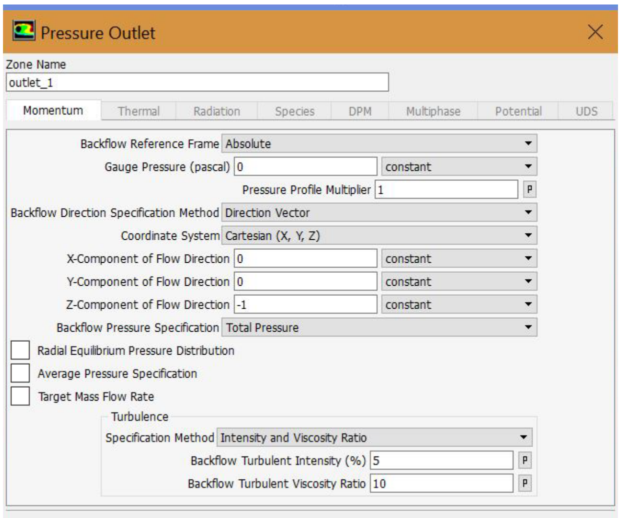


FIG 16: Inlet boundary conditions

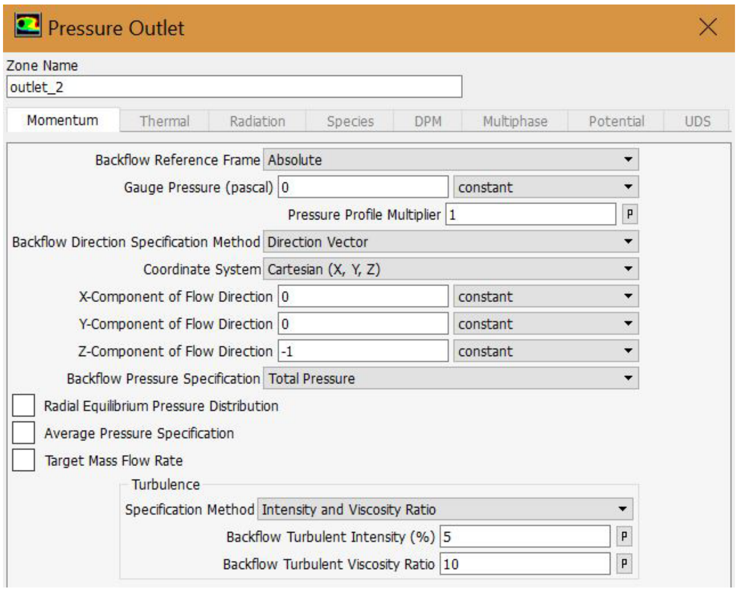


FIG 17: Outlet boundary conditions

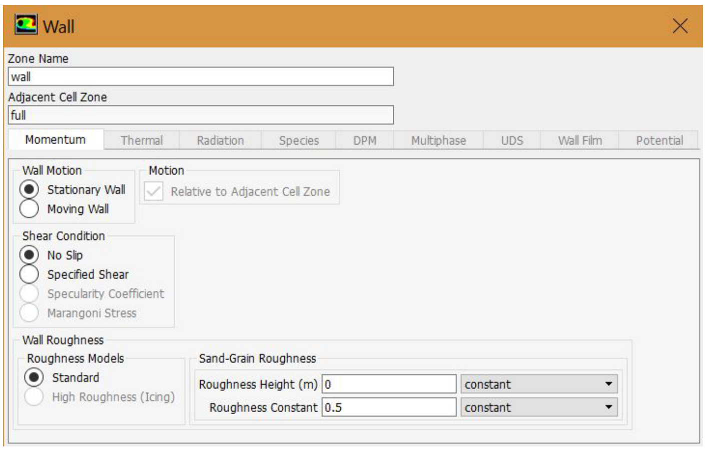


FIG 18: Wall boundary conditions

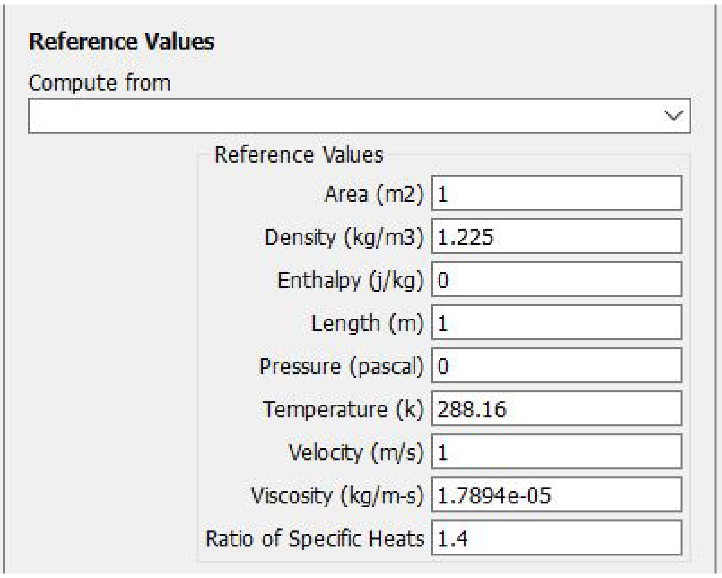


FIG 19: Initial flow condition

The inlet and outlet of the flow field are free flow boundary conditions and atmospheric conditions are taken at those areas. Wall is stationary and non-slip. The reference values for atmospheric parameters are taken at 15^{o}c.

- Solution

➢ Methods

- Pressure-Velocity Coupling
    i. Scheme – simple
- Spatial Discretization
    i. Gradient – least squares call based
    ii. Pressure – second order
    iii. Momentum – second upwind
    iv. Turbulent kinetic energy – first order upwind
    v. Turbulent dissipation rate – first order upwind
- Transient formulation
    i. First order implicit

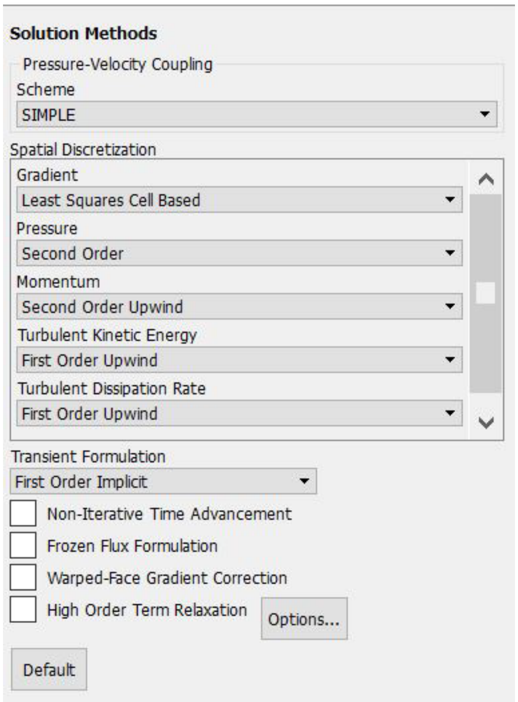


FIG 20: Solution methods

Upwind scheme is a simple and stable discretization scheme. It is also dissipative according to the flow that is simulated. First order upwind scheme contains only 2 data points whereas second order upwind contains 3 data points. First Order Upwind is easier to converge and less accurate but Second Order Upwind scheme is more difficult to converge and more accurate.

➢ Controls

- Under relaxation factors

    i. Pressure – 0.3
    ii. Density -1
    iii. Body forces – 1
    iv. Momentum -0.4
    v. Turbulent kinetic energy -0.8
    vi. Turbulent dissipation rate -0.8
    vii. Turbulent viscosity -1

Factors which have lesser values are given more importance. Factors whose value is 1 are neglected.

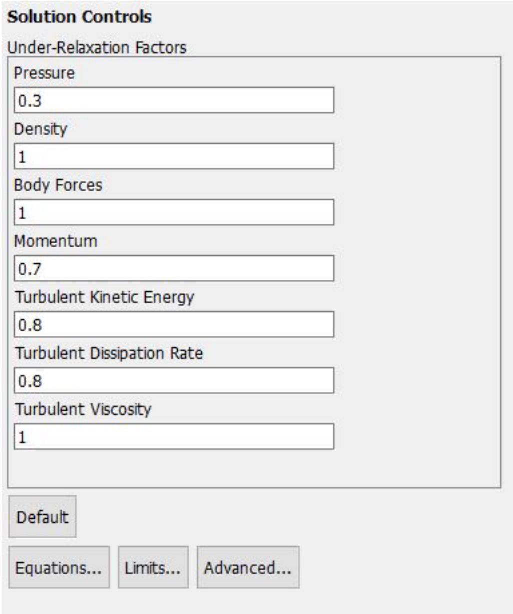
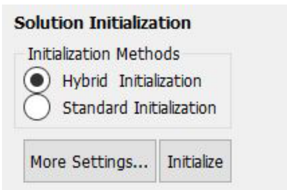


FIG 21: Solution controls

- Initialization
  - Hybrid initialization
- Calculation activities
  - Automatic export
    - i. File type- CFD-post compatible
    - ii. Cell zones
      - I. Compressor enclosure
      - II. Entire volume
    - iii. Quantities
      - I. Static pressure, pressure coefficient, dynamic pressure, absolute pressure, total pressure, relative total pressure, density, density all, velocity magnitude, x- velocity, y- velocity, z- velocity, axial velocity, radial velocity, tangential velocity, relative velocity magnitude, relative (x, y, z) velocity, relative tangential velocity, mesh (x, y, z) velocity, velocity angle, relative velocity angle, vorticity magnitude, helicity, (x, y, z) vorticity, cell Reynolds number, cell convective courant number, turbulent kinetic energy, turbulent intensity, turbulent dissipation rate (epsilon), production of k, (turbulent, effective) viscosity, turbulent viscosity ratio, wall Ystar, wall Yplus, molecular viscosity, wall shear stress, (x, y, z)wall shear stress, skin friction coefficient,

active cell partition, stored cell partition, cell ID, cell element type, cell zone type, cell zone index, partition neighbours, cell weight, (x, y, z)coordinate, axial coordinate, angular coordinate, Abs.angular coordinate, radial coordinate, face area magnitude, (x, y, z) face area, cell volume, orthogonal quality, cell equiangle skew, cell equivolume skew, face handedness, mark poor elements, interface overlap fraction, cell wall distance, adaption function, adaption curvature, adaption space gradient, adaption iso-value, boundary cell distance, boundary normal distance, boundary volume distance, cell surface area, cell warpage, cell children, cell refine level, mass imbalance, strain rate.

- Run calculation
    - Time step size -0.03 seconds
    - Number of time steps – 30
    - Max iterations per time step – 15
    - Calculate.
- Results are obtained from Fluent. Animations and pressure graphs are obtained.



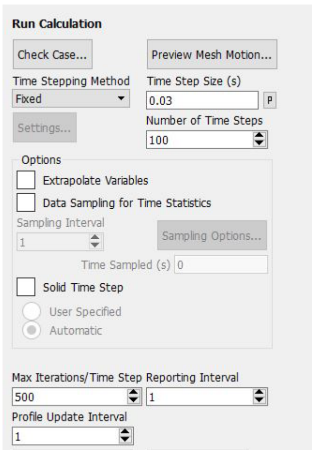


FIG 22: Solution initialization and calculation steps

In unsteady flow, conditions vary with time. So time interval is the period of time elapsing while the phenomenon is happening (from beginning to end) and time step is the incremental change in time for solving the governing equations. Iterations is the number of times governing equations are solved.

CHAPTER-6

RESULTS & CONCLUSION

The project team has designed a turbo-compressor similar to the geometry of the turbocharger mentioned in the paper (GARRETT GT-28). We have designed different guide vanes to impart different whirl angles to the inlet air. The configuration of blades can be in the following ways

i. Straight blades.
ii. Parabolic blades.
iii. Straight and Parabolic blades.
iv. Reverse blades. .

The straight and parabolic guide vanes were placed 100mm away from the compressor and were enclosed in a common casing which connects the guide vanes and the compressor. We estimated distance to be 100mm when we simulated the effects of the guide vanes in ANSYS Fluent and most of the swirling action happened around the 100mm length area and also by referring different journals.

The simulation was done on ANSYS Fluent, the following procedures were followed

1. Importing of the assembly model of Compressor, Guide vane and casing.
2. Enclosing it in an enclosure of the cylindrical size (0.001, 0.001, and 0.001).
3. Naming all boundary parts i.e., inlet, outlet, wall, compressor assembly.
4. Generating a mesh of the above assembly and enclosure.
5. Importing the boundary conditions which are as follows (for air at 25°C)
   a. Inlet - pressure = 1atm.
   b. No slip solid wall.
   c. Compressor = 150000RPM.
   d. Assembly made up of Aluminium.

The flow was set up to be solved in the CFX Solver, the results were inconclusive because of discrepancy in the software used. All the parameters taken and generated are present in the ANSYS REPORT, which is situated in the following pages.

We also authored a paper which we presented at cogNIEscience-2018(A national level technical paper presentation contest) under the guidance of Prof, Mr Ronald Reagon.

RESULTS:

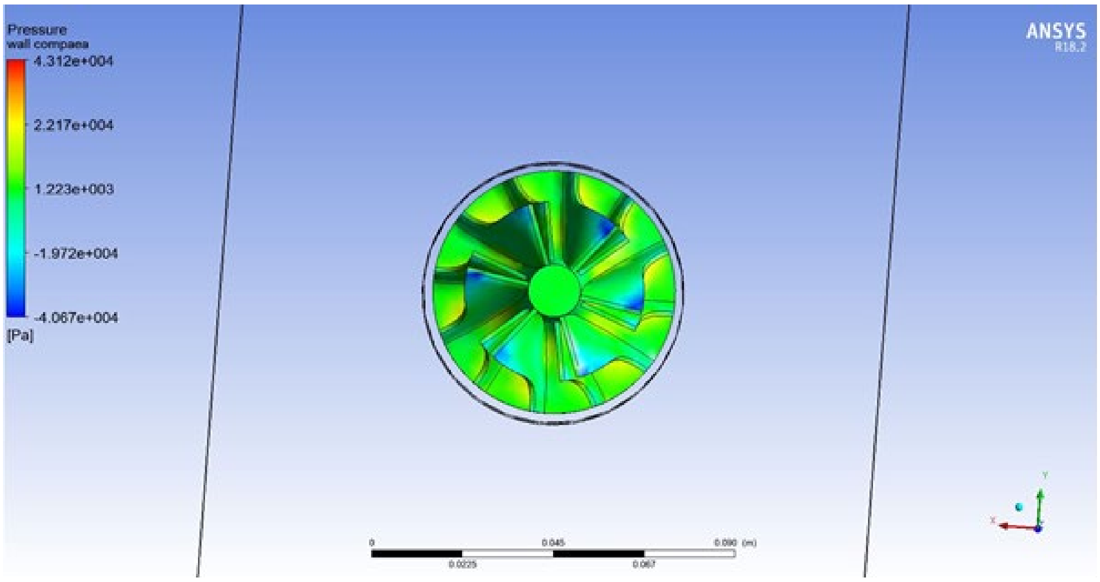


FIG 23: Pressure distribution along Compressor

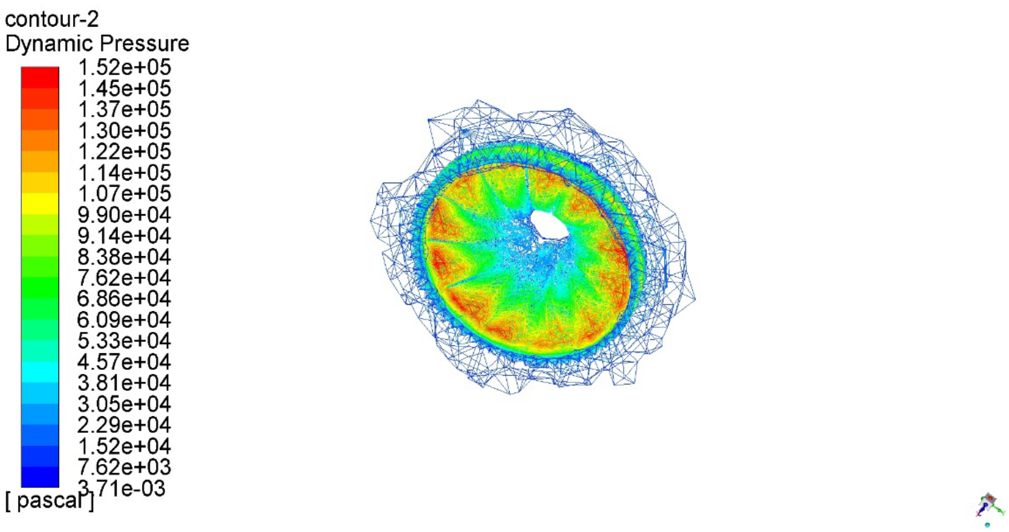


FIG 24: Dynamic pressure along Compressor

As seen from the above diagrams, the pressure is the lowest at the front face end of the inducer blades. This is due to the design of the compressor blades which keep pushing the air away from the forward face of the inducer. The pressure is maximum at the exducer blade wall with closed surface area.

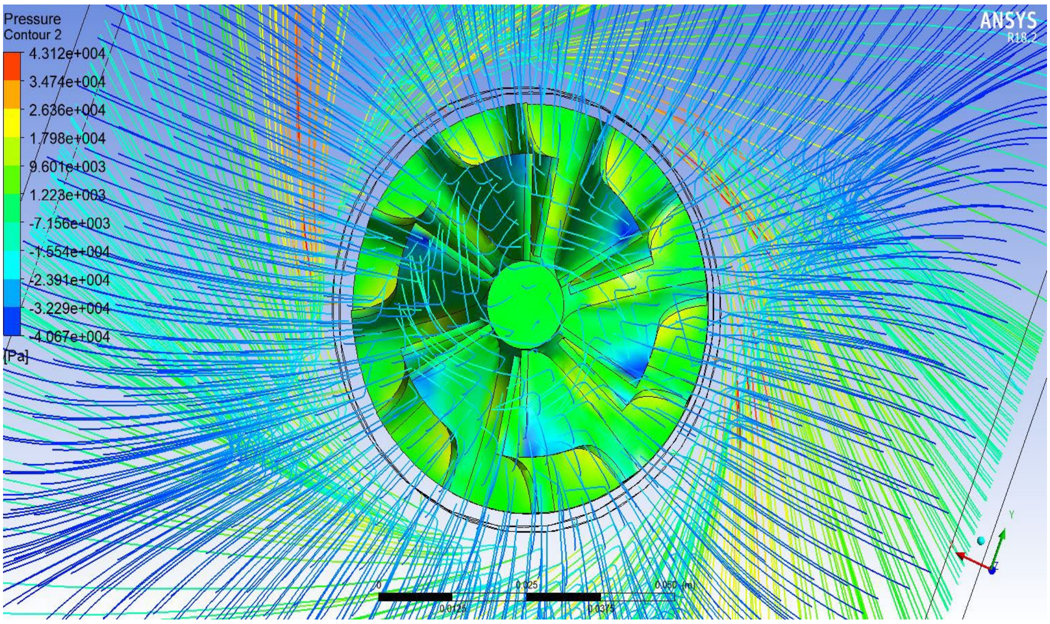


FIG 25: Top view-velocity streamlines on Compressor

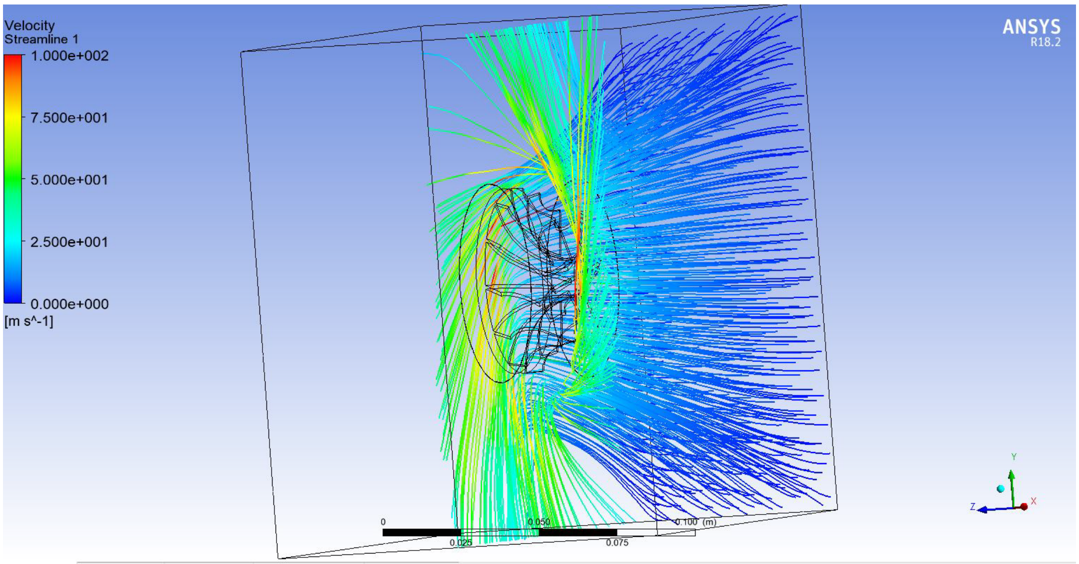


FIG 26: Side view- velocity streamlines on Compressor

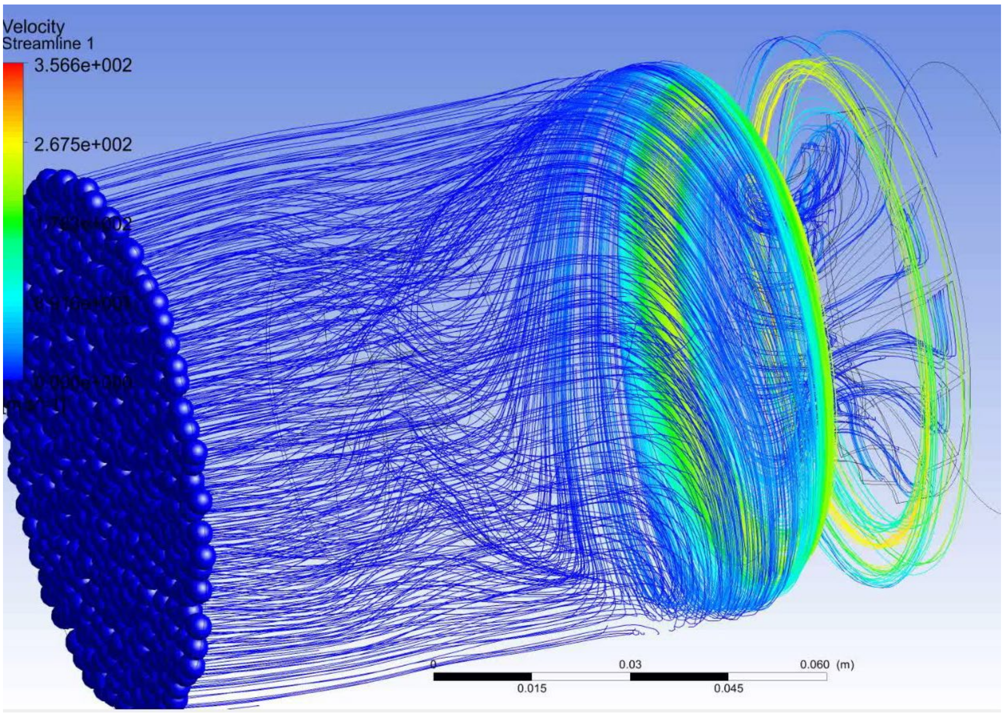


FIG 27: Reverse mixed blade assembly velocity streamlines

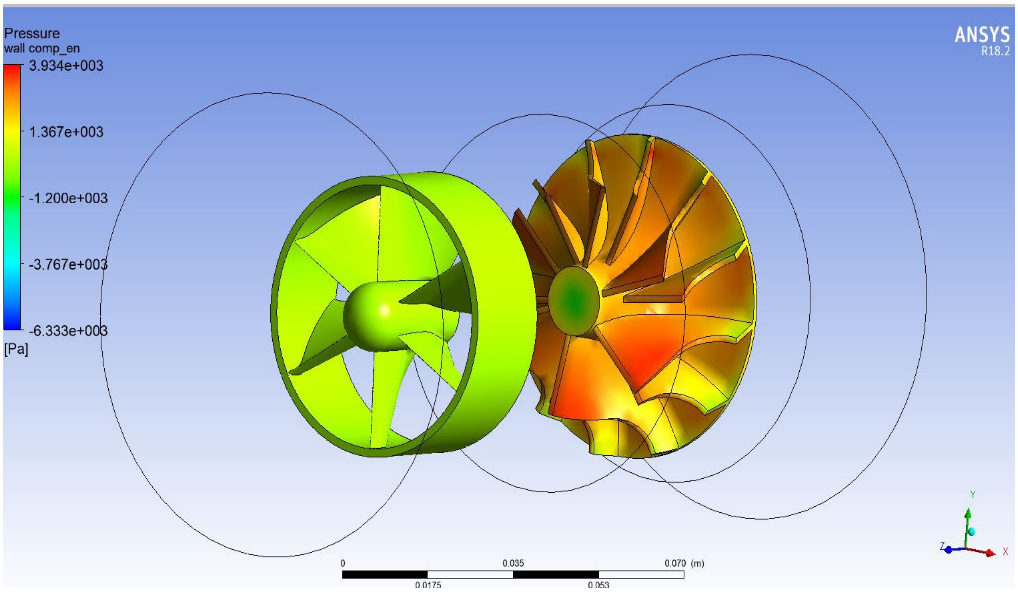


FIG 28: Pressure along IGV and Compressor-reverse mixed blades

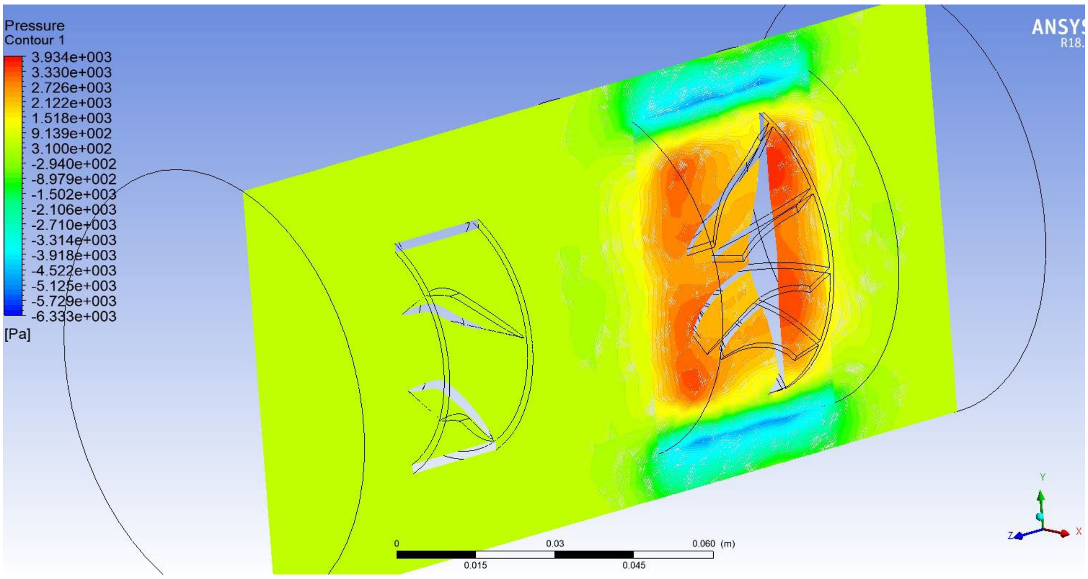


FIG 29: Pressure distribution along ZY plane reverse mixed blades

As seen in the above pictures, the reverse blades increase the angle of attack in the flow field and hence mass flow rate decreases. This increases overall pressure inside the compressor. It is also seen that the pressure is highest at the front face end of the inducer blades. Whereas it was lowest in the compressor. The pressure remains high even at the exducer blade surface

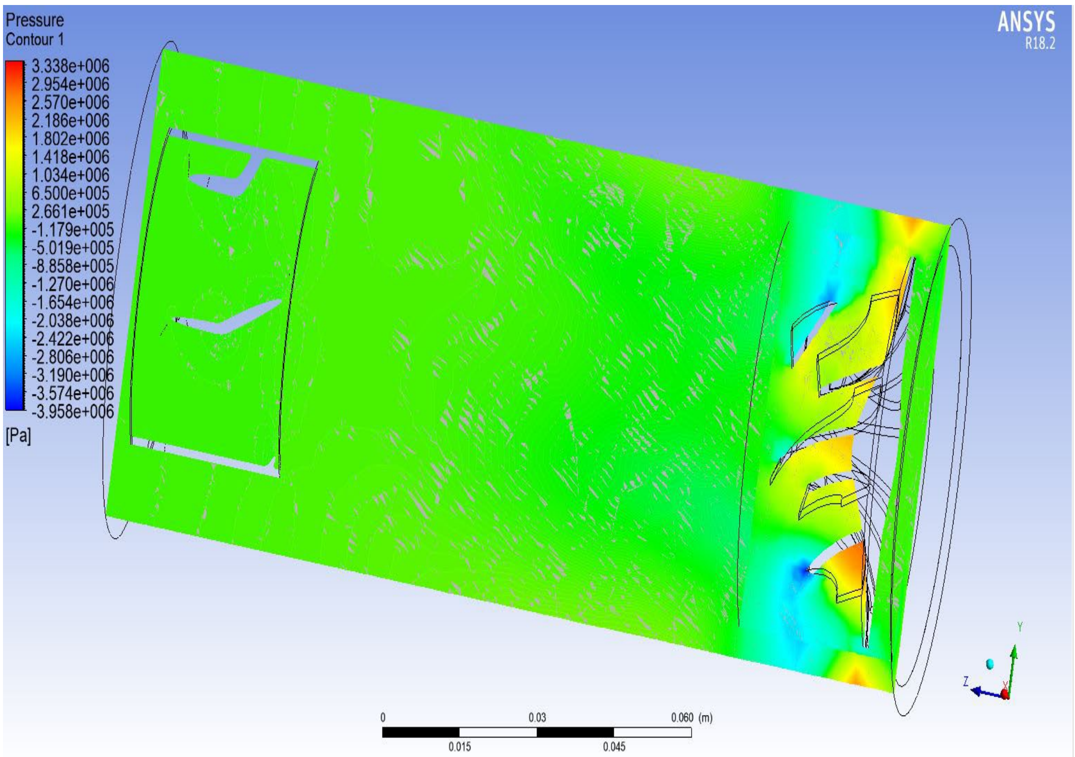


FIG 30: Pressure distribution along ZY plane forward mixed blades

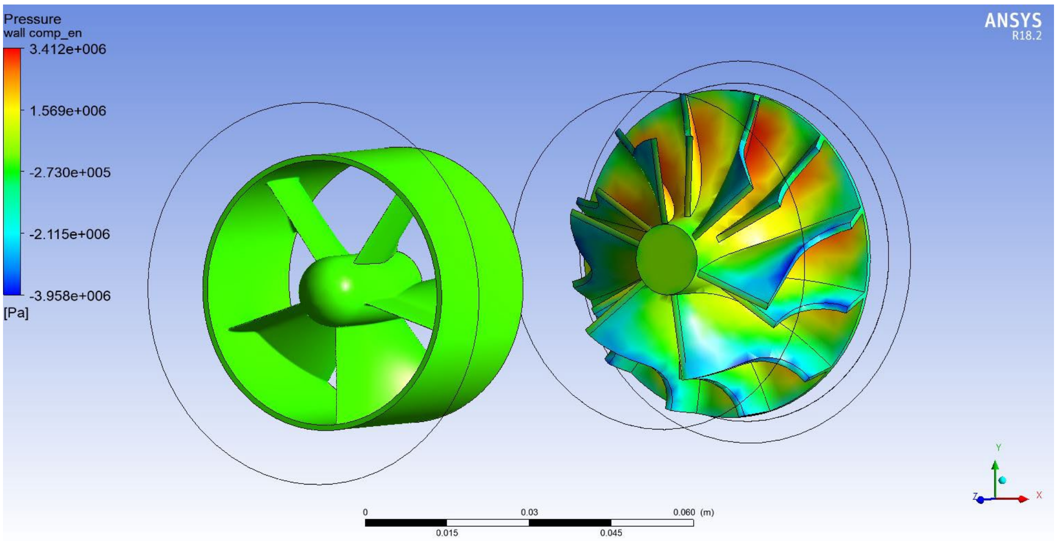


FIG 31: Pressure along IGV and Compressor-forward mixed blades

From the blade assembly diagrams given above, we can see that a large area of the compressor blade in its outer edge, the presence of a low pressure field. This is due to the decrease in the angle of attack induced by the guide vanes. There are still areas of high pressure at the inner surfaces of the blades the mass flow rate is relatively high.

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