

Design of Impulse Turbine for Energy Regeneration from Kinetic Energy of Exhaust Gas

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ABSTRACT: As we know efficiency of any Diesel engine is only 40% remaining 60% of energy got waste. This waste energy carried away by cooling water or coolant use in engine to maintain temperature of I C engine, unaccounted source like friction in moving components in the system, leakages of air fuel mixture etc. And through exhaust gases which carry 400°C to 500°C amount of heat energy up to 7 bar of pressure energy.

Waste energy carried away by cooling water and unaccounted source we cannot control because cooling of engine cylinder is necessary to avoid deformation and engine seizing also we can't control waste energy from unaccounted source, mechanical joints, linkages always create friction and high pressure in I.C. engine always have leakages because piston in cylinder is moving part. So here our concerns are with exhaust which carry away 30-35% of energy which contain up to 4-7 bar pressure at 450-550°C. This pressure energy can utilize by turbocharger but in turbocharger reaction turbine is used which work on mostly pressure energy of fluid and turbocharger only have 70-75% of efficiency so remaining 25-30% of energy still gets waste which is in the form of kinetic energy at nearly 400°C temperature. This waste energy also controllable by saving this kinetic energy using impulse turbine.

KEYWORDS: Waste Exhaust Energy, Exhaust Energy Regeneration, Waste Energy After Turbocharging, Axial Impulse Turbine, EN24, Impulse Turbine Design, CFD on Impulse Turbine, Static Structural Analysis of Turbine.

I. INTRODUCTION

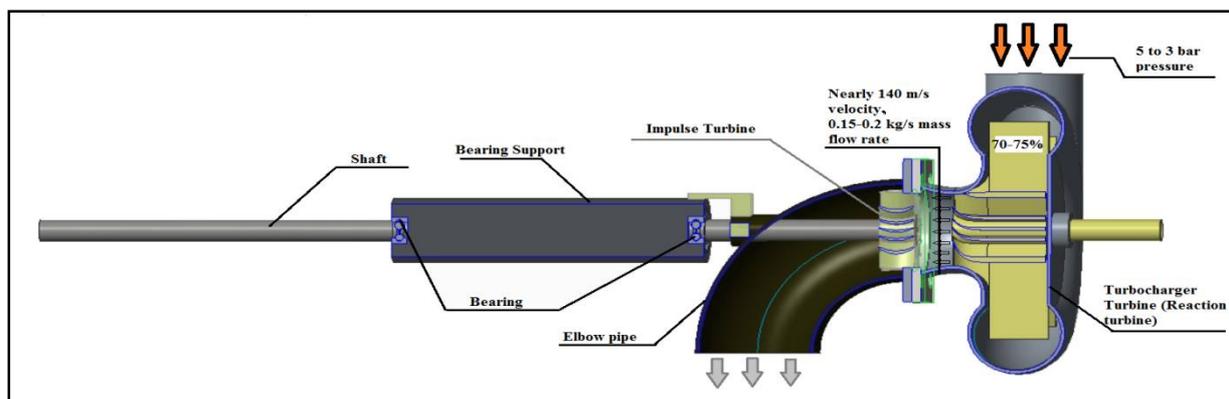


Figure 1 Design of Impulse Turbine for Energy Regeneration from K.E. Of Exhaust Gas

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After the exhaust stroke burnt gases contain up to 4-7 bar pressure at 450 to 550°C temperatures as per Vincenzo De Bellis et.al. [1] represents 1D simulation and experimental analysis of a turbocharger turbine for automotive engine under steady and unsteady flow conditions. So, considering all losses in valve, pipes bent losses, at the inlet of the turbocharger it has 3-5 bar pressure. This assumption taken using M. S Chiong et.al. [3] paper presents non-adiabatic pressure loss boundary condition for modelling turbo charger turbine pulsating flow and M.H. Padzillah et.al. [2] which presents influence of speed and frequency towards the automotive turbocharger turbine performance under pulsating flow condition. A simplified methodology of pulse flow turbine modelling. Turbocharger utilize that pressure by using reaction turbine to run the compressor which use to compresses inlet air charge in C.I engine. At outlet of turbo charger pressure is fall up to nearly 1bar because of reaction turbine which mostly work on pressure energy of fluid, but burnt gases still contain kinetic energy nearly 140 m/s of velocity and nearly 0.1-0.2 kg/s of mass flow rate. So, this kinetic energy can utilize by implementing Impulse turbine. The term impulse means sudden force act to bring desire change. It denotes as product of force and time interval for which it acts. Impulse turbine works on the same principle, i.e. it works on the kinetic energy of fluid [7] [10]. This impulse turbine rotates the shaft attached to it and rotational energy can be converted into electrical energy using alternator. Electrical energy can use to charge the battery of automobile. This is the research-based project so we built up the conceptual model and aim is to find out how much power it can produce. Figure-1 shows the labelled diagram for our model.

Diagram shows the inlet to the turbocharger at 5 to 3 bar pressures. It is use to run turbocharger turbine which is reaction turbine of efficiency 70-75%. At outlet the pressure reduces to the 2 to 1 bar. Inlet to our model to save kinetic energy using impulse turbine. To see the exact half section view of our project figure 2 shows the wireframe diagram for our project.

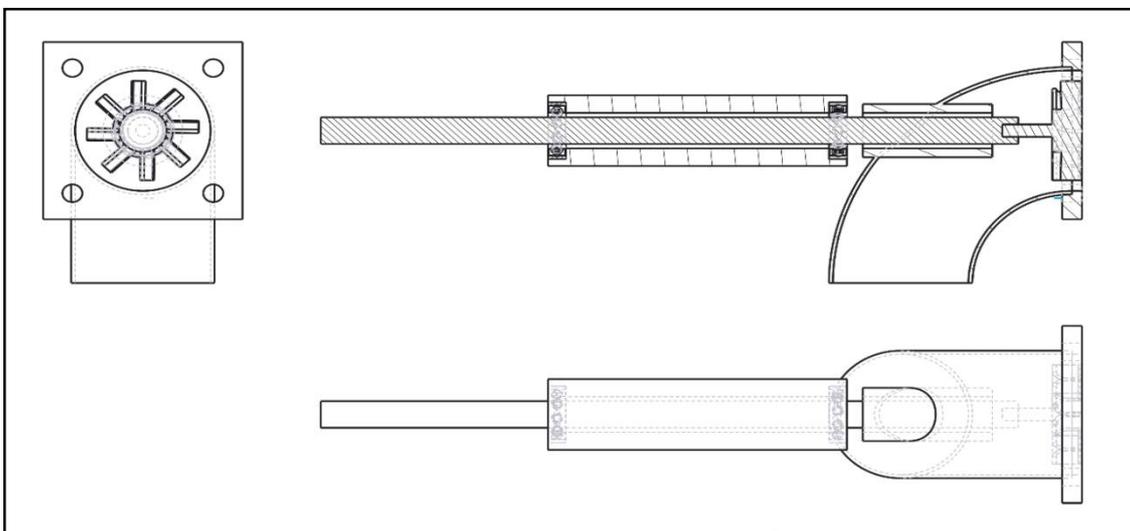


Figure 2 Wireframe diagram and half sectional view of project model

➤ Objectives

- To prevent wastage of energy from exhaust gases.
- Find how much power this conceptual model can produce.
- To use energy from exhaust gases to charge battery in automobiles so there are no friction losses.
- Maximize use of engine fuel power.
- Compare conceptual model working with actual model calculation.
- Find out it can eliminate alternator attached to the engine shaft or not.

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II. DESIGN CALCULATION

For design calculation we selected the truck model TATA Tipper which is according to availability and capacity of truck. Following are the design calculation

1 Calculation for waste Kinetic Energy

Model **TATA 1613 Tipper** (engine 697 tcic)

Capacity – 5883 cc

So, for mass flowrate,

Let's assume engine running at 1500 rpm

Also $1500 \text{ rpm} = \frac{1500}{60} \text{ rps} = \mathbf{25 \text{ rps}}$

There are two rotations required to complete 1 cycle in 4 stroke IC engine

i.e.

- First rotation – suction and compression stroke
- Second rotation – expansion and exhaust stroke

So, for total no. of strokes per second

$$= 25 \times 2$$

$$= 50 \text{ strokes per second}$$

But for mass of air at inlet we only consider suction strokes

$$\text{i.e. } \frac{50}{4} = 12.5 \text{ suction strokes per second}$$

So, capacity of engine 5.883 liters

$$\text{So } 12.5 \times 5.883 = 73.5625 \text{ liters/second}$$

$$\text{i.e. } 73.5625 \text{ liters/second} \leftrightarrow 0.073565 \text{ m}^3/\text{s}$$

$$\text{we know that, } d = \frac{m}{v} \quad \dots [8]$$

i.e. density = mass/volume

$$\text{density of air} = 1 \text{ kg/m}^3$$

property of diesel considers negligible because very small amount of diesel is added in air fuel mixer

$$\text{therefore } 1 = \frac{m}{0.073565}$$

$$\cdot m = 0.073565 \text{ kg/sec}$$

With application of turbocharger fresh air charge compressed from 1 bar to 2-3 bar at the inlet of engine cylinder

So, the volumetric efficiency of engine cylinder will be twice

$$\cdot m = 0.073565 \times 2$$

$$\cdot \mathbf{m = 0.14713 \text{ kg/sec}}$$

We know that,

$$P v = m R T$$

... [6]

where,

$$p = \text{pressure in N/m}^2$$

$$v = \text{volumetric flow rate in m}^3/\text{sec}$$

$$m = \text{mass flow rate kg/sec}$$

$$R = \text{gas constant } (0.287 \text{ kpa.m}^3/\text{kg.k})$$

$$T = \text{temperature in } ^\circ\text{K } (400^\circ\text{C} + 273 = 673 \text{ } ^\circ\text{K})$$

$$1 \times 10^5 \times v = 0.14713 \times 0.287 \times 673$$

$$\cdot \mathbf{V = 0.2841830663 \text{ m}^3/\text{sec}}$$

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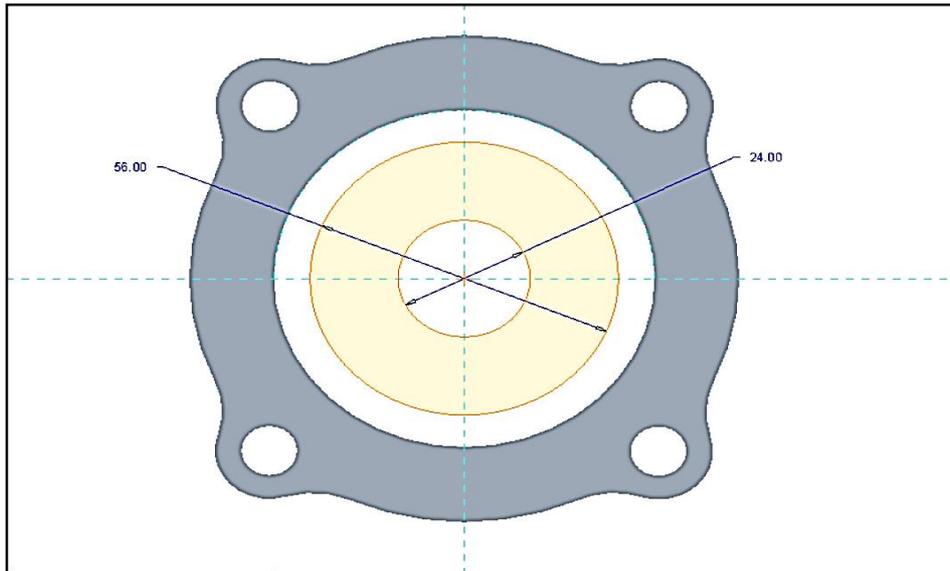


Figure 3 Representation of Area of Flow Rate Using Turbocharger Gasket

Now we know the volumetric flow rate of the exhaust to calculate the velocity of gas we need the area at the outlet of the turbine, by using the gasket dimension we can have the outlet dimension of turbocharger. So, we can easily design turbine shroud diameter and hub diameter which is equal to shroud diameter of turbocharger turbine and hub of turbocharger turbine. Figure 3 show the diagram for area of flow using turbocharger outlet gasket.

Flow area(A) = (Blade tip diameter - Hub diameter) - (thickness of turbocharger turbine blade × height of turbocharger turbine blade × number of turbocharger turbine blade)

$$A = \left(\frac{\pi}{4} \times 0.056^2 - \frac{\pi}{4} \times 0.024^2 \right) - (0.00025 \times 0.032 \times 12)$$

$$A = 1.914619298 \times 10^{-3} m^2$$

For velocity,

$$v = A \times V_e \quad \dots [5]$$

- $0.2841830663 = 1.914619298 \times 10^{-3} \times V_e$
- $V_e = 156.2630874 \text{ m/sec}$

For kinetic energy,

$$K.E. = \frac{1}{2} \times m V_e^2 \quad \dots [6]$$

- $\frac{1}{2} \times 0.14713 \times 156.2630874^2$
- $K.E. = 1796.321337 \text{ watt}$

1.2 Calculation for turbine design

For max. efficiency (η_{p-max}) = $\cos^2 \alpha$

Consider $\alpha = 10^\circ$

Inlet vel. = $V_e = V_1 = 156.2630874 \text{ m/sec}$

$$\text{Speed ratio (s)} = \frac{V_b}{V_1} = \cos \alpha / 2 \quad \dots [5]$$

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- $V_b = \left(\frac{\cos\alpha}{2}\right) V_1$
- $V_b = \left(\frac{\cos\alpha}{2}\right) \times 156.2630874$
- $V_b = 76.9445499 \text{ m/s}$

1.2.1 Calculation for inlet velocity triangle

Velocity of flow at inlet is 90° so the inlet angle is taken 90° as no guide blade are provided in this conceptual model.

i.e. $\alpha = \theta = 90^\circ$

$$V_{w1} = V_1 \cos \alpha = 156.2630874 \times \cos 90^\circ$$

$$V_{w1} = 0$$

$$V_{f1} = V_1 \sin \alpha = 156.2630874 \times \sin 90^\circ$$

$$V_{f1} = 156.2630874 \text{ m/s}$$

$$BD = V_{w1} - V_b = 156.2630874 \text{ m/s}$$

$$V_{r1} = \sqrt{Vf^2} = \sqrt{156.2630874^2}$$

$$V_{r1} = 156.2630874 \text{ m/s}$$

1.2.2 Calculation for outlet velocity triangle

For maximum possible degree of outlet angle 10° outlet angle is selected.

Also, $V_{r1} = V_{r2} = 156.2630874 \text{ m/s}$

$$V_{f2} = V_1 \sin \phi = 156.2630874 \times \sin 10^\circ$$

$$V_{f2} = 27.13480036 \text{ m/s}$$

$$AB = V_{r2} \cos \phi = 156.2630874 \times \cos 10^\circ$$

$$AB = 153.8891 \text{ m/s}$$

$$V_{w2} = AB - V_b = 153.8891 - 76.9445499$$

$$V_{w2} = 76.9445499 \text{ m/s}$$

$$\beta = \tan^{-1} \left(\frac{V_{f2}}{V_{w2}} \right) = \tan^{-1} \left(\frac{27.13480036}{76.9445499} \right)$$

$$\beta = 19.4254^\circ$$

$$V_2 = \frac{V_{w2}}{\cos \beta} = \frac{76.9445499}{\cos 19.425}$$

$$V_2 = 81.5887759 \text{ m/s}$$

1.3 For power output of turbine

$$P_{\text{output}} = m \times (V_{w1} + V_{w2}) V_b$$

$$P_{\text{output}} = 0.14713 \times (0 + 76.9445499) \times 76.9445499$$

$$P_{\text{output}} = 871.0778329 \text{ watt}$$

1.4 For turbine efficiency

$$\eta_{\text{impturb}} = \frac{P_{\text{output}}}{\text{K.E. supplied}}$$

$$\eta_{\text{impturb}} = \frac{871.0778329}{1796.321337}$$

$$\eta_{\text{impturb}} = 0.4849 = 48.49\%$$

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By using all calculation of inlet and outlet triangle following is the representation of velocity diagram for turbine blade.

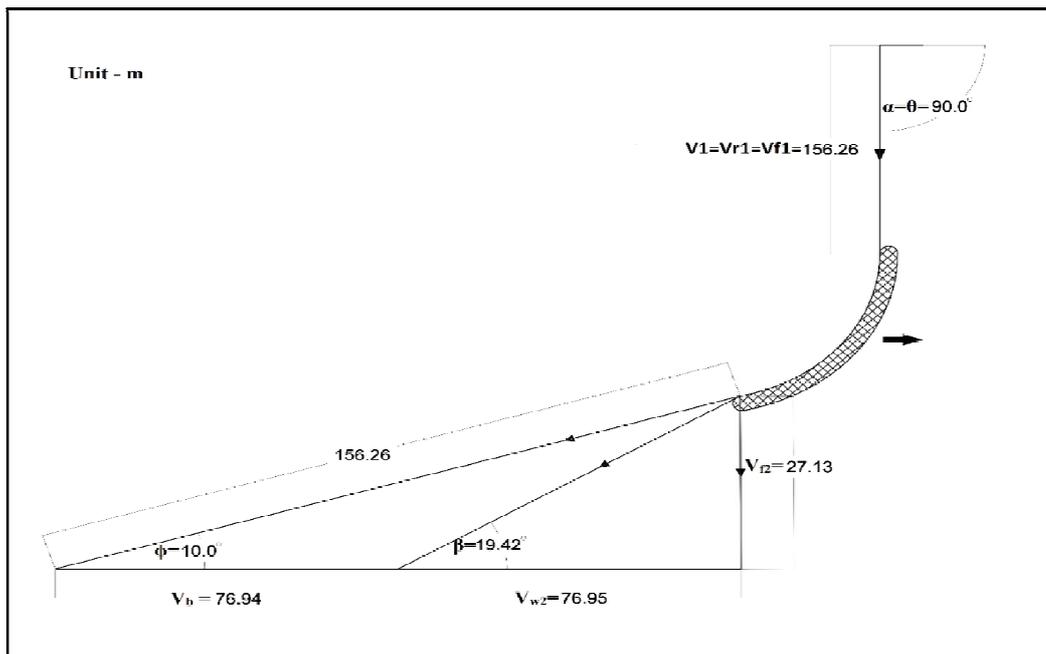


Figure 4 Inlet and Outlet Blade Velocity Triangle of turbine blade

above calculation give the theoretical result for turbine which is turbine of our conceptual model can produce power up to 871.0778329 watt. which have theoretical efficiency of 48.49%. Now, next we find out how it works experimentally. To build up actual model we start from proper material selection according to type of loading acting on various components and surrounding working environment of our model.

III. MATERIAL SELECTION

At turbine power plant for temperature between 400 °C to 600 °C Stainless steel '304/304L' is use. This project also works under same temperature ranges. Alternative to Stainless steel '304/304L' 'EN24-alloy steel' also can use in this temperature range. For selection of material Phillip Downson et.al. [4] represents selection of material and material related process for centrifugal compressor and steam turbine in the oil and petrochemical Industry. In our project high tensile force act on the turbine blade due to centrifugal force. Crushing force is act on the bolted area of plate so high strength properties are required. Alloy steel EN24 steel is a high tensile alloy steel renowned for its wear resistance properties and also where high strength properties are required. EN24 alloy is nearly half of the price of Stainless steel '304/304L' which is Rs 80-100 /- kg. and easily available. Heat treatment on alloy steel: -

- **In hardening process of EN24** heat uniformly to 823/850°C until heated through. Quench in oil.
- **Then for tempering steel** heat uniformly and thoroughly at the selected tempering temperature, up to 660°C and hold at heat for two hours per inch of total thickness. Tempering between 250-375°C is not recommended as this can seriously reduce the steel's impact value.
- **For Stress Relieving of material** Heat slowly to 650-670°C, soak well. Cool the EN24 tool in a furnace or in air.

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Properties of Outline Row 3: ALLOY STEEL (EN24)				
	A	B	C	D E
1	Property	Value	Unit	
2	Density	7840	kg m ⁻³	
3	Isotropic Secant Coefficient of Thermal Expansion			
4	Coefficient of Thermal Expansion	1.2E-05	C ⁻¹	
5	Reference Temperature	22	C	
6	Isotropic Elasticity			
7	Derive from	Young's Mod...		
8	Young's Modulus	2.07E+11	Pa	
9	Poisson's Ratio	0.3		
10	Bulk Modulus	1.725E+11	Pa	
11	Shear Modulus	7.9615E+10	Pa	
12	Field Variables			
13	Temperature	Yes		
14	Shear Angle	No		
15	Degradation Factor	No		
16	Tensile Yield Strength	680	MPa	
17	Compressive Yield Strength	680	Pa	
18	Tensile Ultimate Strength	680	Pa	
19	Compressive Ultimate Strength	0	Pa	

Figure 5 Properties edited in Ansys for analysis of various component of project

IV. COMPONENTS AND DIMENSIONS

According to model TATA 1613 turbocharger exhaust we have to design our project so following are the reasons for selecting appropriate size of each components.

1 Plate

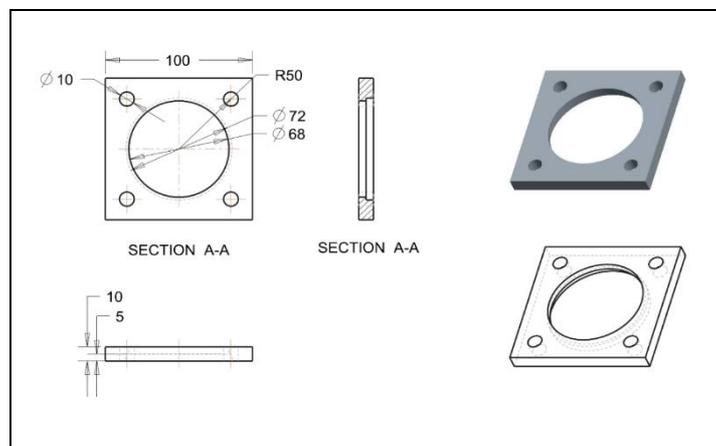


Figure 6 Plate dimensions

We find out dimension of the plate with the reference of gasket between connection of turbocharger outlet and exhaust pipe towards silencer. Material selection for plate is EN24 alloy steel. It is 100×100mm square plate with hole

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in centre of diameter 68 mm. It also bore with 72 mm diameter up to 5 mm Height concentric to 68 mm diameter hole. 1mm×4 holes are provided for bolting it with turbocharger outlet. Thickness is considered of 1mm which is very strong to carry overall weight of the model project. For safety plate analysis also done by using Ansys software which shows design is safe for the plate Figure 6 show the dimension of plate.

2 Elbow Pipe

Elbow shape pipe selected because it separates the alternator and bearings from exhaust gases, which can damage it. Material selection for pipe is EN24 which is good to use 450 to 550°C. Its inner and outer diameter is 68mm and 72mm respectively. Pipe is welded to the plate for strong structural joint. Pipe also has a hole of diameter 30mm concentric to the inlet pipe hole. One cylindrical pipe is welded in that hole of length 65mm with inner diameter 20mm which can prevent critical speed of turbine shaft, obviously it is for safety precaution. Figure 7 shows the detention of elbow pipe.

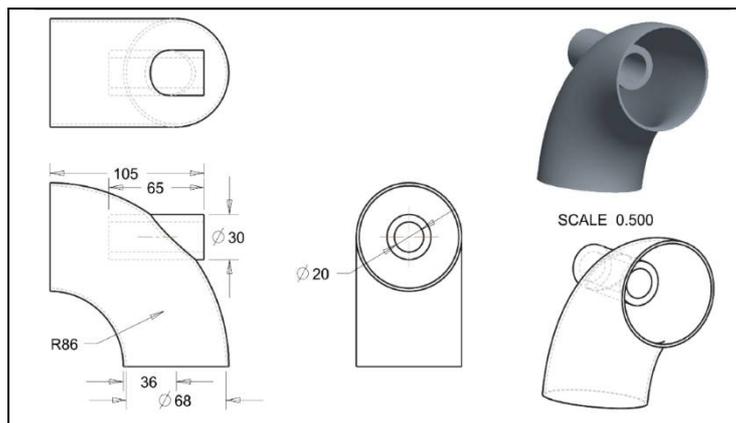


Figure 7 Elbow pipe dimensions

3 Bearing Support

Bearing support is supports the bearing which constrain the turbine shaft and only provide one translation and rotation along x-axis.

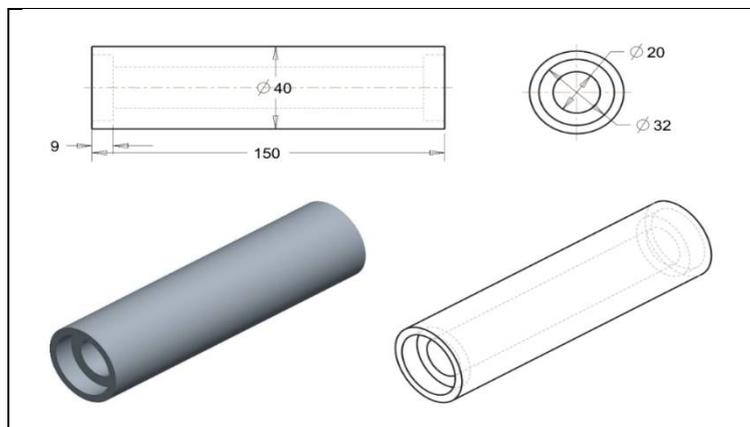


Figure 8 Bearing supports dimensions

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Bearing supports design according to bearing dimensions. Figure 8 shows the detention of bearing support. Its Outer diameter is 40mm and inner diameter is 20mm. to support the bearing at both end it has 32mm diameter which is the outer diameter of bearing.

4 Bearings

Two deep groove ball bearing is used for supports made up of stemless steel. It can be used for up to 32000 rpm with outer and inner diameter 15mm and 32mm respectively, thickness 9mm. Figure 9 shows the detention of bearings.

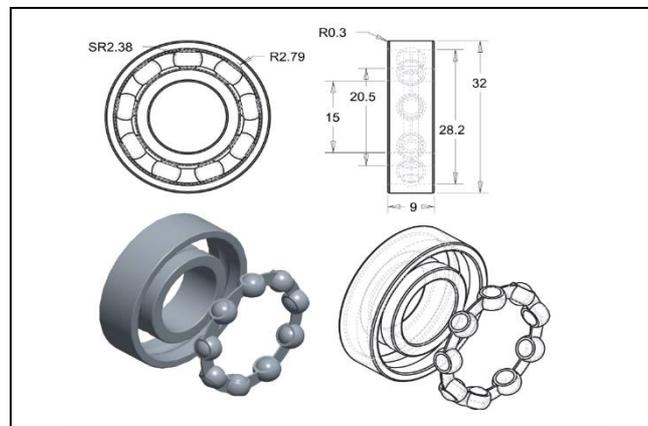


Figure 9 Bearing dimensions

5 Impulse Turbine

Turbine dimension and Number of blades

Outer and hub dimension of turbine shroud is decided by considering outlet diameter of turbocharger turbine blades.

i.e. Outer diameter of impulse turbine = Outer diameter of turbocharger turbine

$$D = 0.056 \text{ m} = 56 \text{ mm}$$

Also, Hub diameter of impulse turbine = Hub diameter of turbocharger turbine

$$d = 0.024 \text{ m} = 24 \text{ mm}$$

- Blade height = $D - d = 0.056 - 0.024$
- **Blade height = 0.032 m = 32 mm**

By taking the blade thickness 2 mm

Total number of blades: -

Number of blades of turboturbine = Number of blades of regenerator turbine

Now, Flow area = $\frac{\pi}{4} \times D^2$ – area occupied by blade

And, Area occupied by blade = Thickness of blade \times Height of blade \times Number of blade

$$1.818619298 \times 10^{-3} = \frac{\pi}{4} \times 0.056^2 - (0.002 \times 0.032 \times \text{Number of blade})$$

- Number of blade = $8.56 \approx$ **8 number of blades**

By using inlet and outlet angle of turbine blade and above dimensions we design the turbine model using PTC Cero parametric software.

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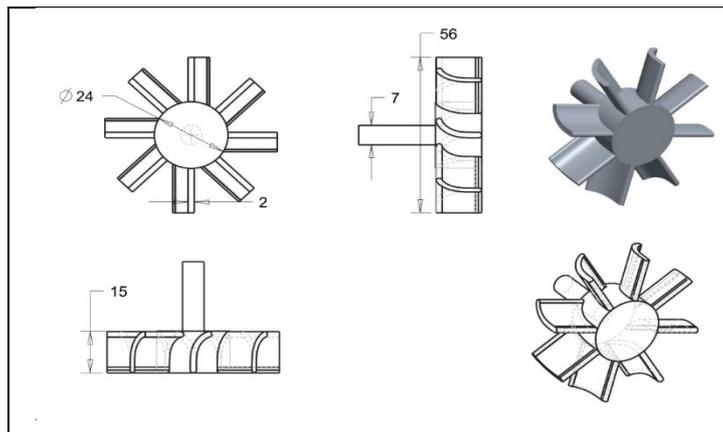


Figure 10 Impulse turbine dimensions

Above CAD Figure 10 shows the dimensions of impulse turbine. By using CAD diagram, the actual concept model turbine figure 11 shows the actual we manufacture manufactured part of turbine.



Figure 11 Actual picture of turbine

6 Alternator

As we know the according to rpm we design the alternator and this is concept model so we use the DC motor as alternator which motor to alternator conversion factor or power reduction factor is 15. We use the RS-775WC-8514 model which is brush type DC motor. Figure 12 shows performance table of DC motor,

Performance Table													
MODEL	VOLTAGE			NO LOAD		AT MAXIMUM EFFICIENCY				STALL			
	OPERATING RANGE	NOMINAL V		SPEED r/min	CURRENT A	SPEED r/min	CURRENT A	TORQUE mNm	TORQUE gcm	OUTPUT W	TORQUE mNm	TORQUE gcm	CURRENT A
RS-775WC	8514	6 - 20	18	19500	2.7	17040	18.7	153	1561	273	1216	12396	130

※ CCW shifted commutation (CCW+)
※ Generator conversion factor 15 (Approximate)

Figure 12 Performance table of RS-775WC-8514

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For calculating actual power of turbine, we refer the following graph we know the current and torque of the generator by using following relationship chart we find the rpm of the generator figure 13 shows Efficiency, current, speed relationship with torque of motor.

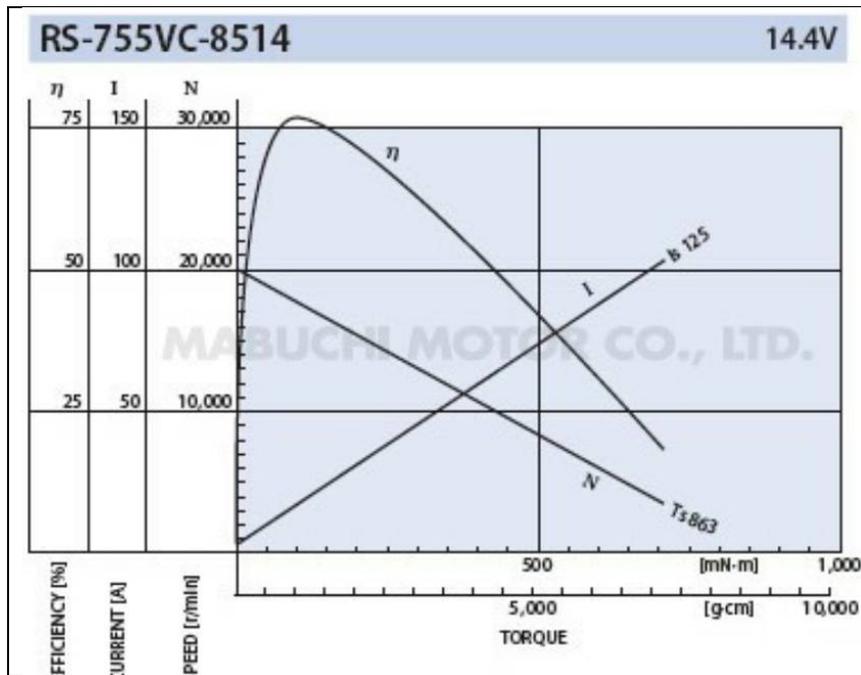


Figure 13 Efficiency, current, speed relationship with torque of RS-755VC-8514

For the assembly between rod and alternator we manufacture connector between shaft and alternator. Fix this connector it with the alternator as per shown in figure 14 which is actual picture of alternator.



Figure 14 Actual picture of RS-755VC-8514 Motor

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7 Multimeter

Multimeter is use for measuring the current, voltage and resistance of circuit. Figure 15 show the actual picture of multimeter.



Figure 15Multimeter

8 Digital Thermometer

We use digital thermometer to measure the exhaust gas temperature. Figure 16 shows the actual picture of digital thermometer.



Figure 16Digital thermometer

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By using digital thermometer, we calculate the temperature every 5 minutes for total 20 minutes so the following observation for the temperature we get. Table1 shows the observation of temperature of exhaust gas.

Sr no.	Time(0-5min)	Temperature
1	Initial level	89.7 °C
2	After 5 min	136.4°C
3	After 10 min	206.7°C
4	After 15 min	248.5°C
5	After 20 min	253.2°C

Table1 Observation table of temperature of exhaust gas

9 Circuit

Here electric circuit require to generate load on the alternator. So, we provided two bulbs of 35watt connect it with the motor and two probes of multimeter. Measured resistance of circuit is 32.5 Ω. Figure 17 shows the circuit diagram.

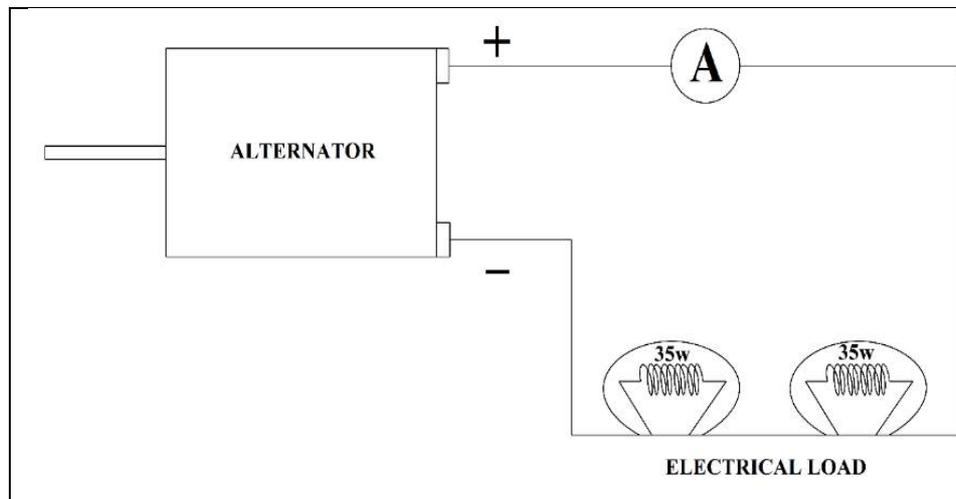


Figure 17 Electric circuit

V. ANALYSIS OF PROJECT

After deciding the design dimensions, it is important to check that design is safe for working environment and working loads. So, following are the CAE and CFD analysis of the component under load.

1 Turbine analysis

For the analysis of turbine, we first perform the CFD (Computational fluid dynamics) on the turbine, from which we get the total pressure and forces on the turbine for static analysis of turbine.

1.1 CFD on turbine

for the CFD of turbine we give the velocity input of 156 m/s of air. We keep the energy equation on to input temperature of 400 °C. Turbulence modelling is a complicated area [9]. The choice of model depends on the application. Here, the Realizable k-epsilon model is used. This is an improvement on the well-established Standard. In CFD analysis following pressure and velocity contour we observe.

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following fig shows the pressure contours over turbine and turbine blade in which the maximum pressure reach on the turbine is up to 35650 Pa.

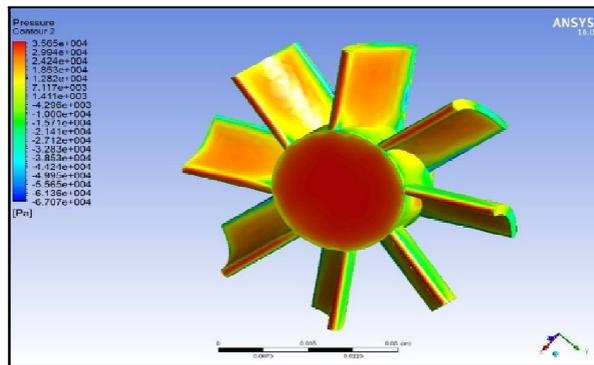


Figure 18 Pressure contours on the turbine

How pressure acts through the overall flow over the blade shown in the following figure 19 which shows pressure contour around the blade of turbine.

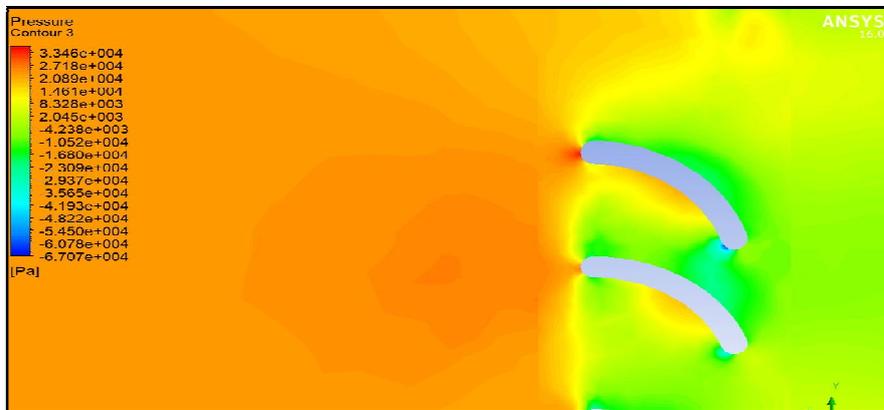


Figure 19 Pressure contours around the turbine blade

For the velocity over the turbine figure 20 shows the velocity contour over the turbine.

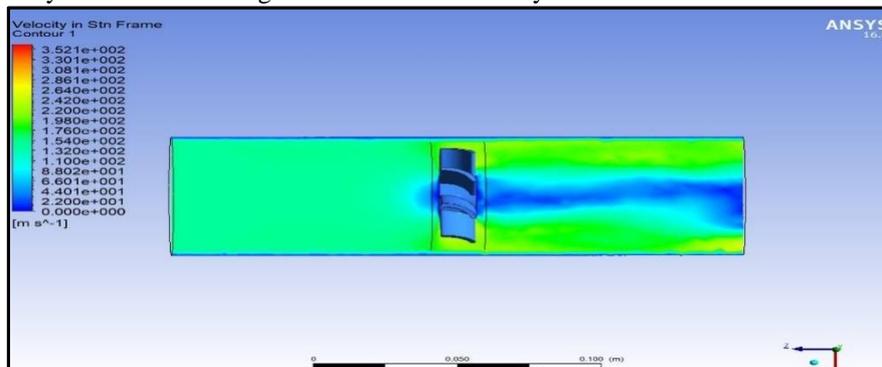


Figure 20 Velocity contours over the turbine

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To see the velocity of flow over the turbine blade figure 21 shows velocity contour over the turbine blade.

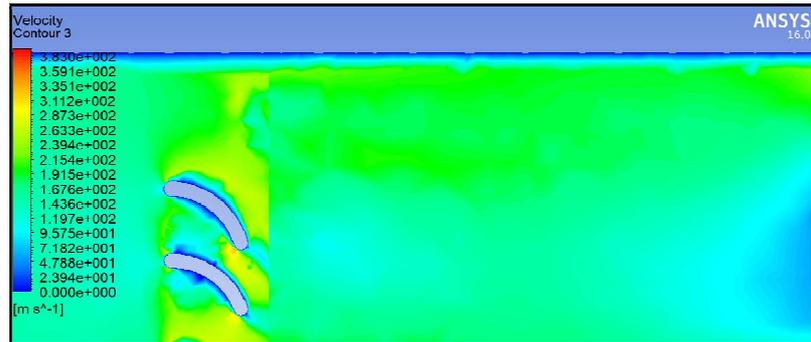


Figure 21 Velocity contours over the turbine blade

After computing we get the following results.

Force on turbine blade	= 51.0152 N
Max pressure on the turbine blade	= 36600.2 Pa
Torque on the turbine	= 0.852723 Nm
Power output of turbine at 5000 rpm	= 446.484 J
Power output of turbine at 7000 rpm	= 625.078 J

Above result we can compare with our result calculated by analytical method and experimental method to validate our work as the concept design work properly. Above parameter like pressure and force on the turbine also use for structural analysis of the turbine. To find out design of turbine is safe or unsafe under the working condition and environment.

1.2 Static structural analysis of turbine

We use the maximum force on the turbine calculated by the CFD analysis for the static structural analysis of turbine.

Following figure 22 shows the boundary conditions we applied on the turbine for the structural analysis of turbine.

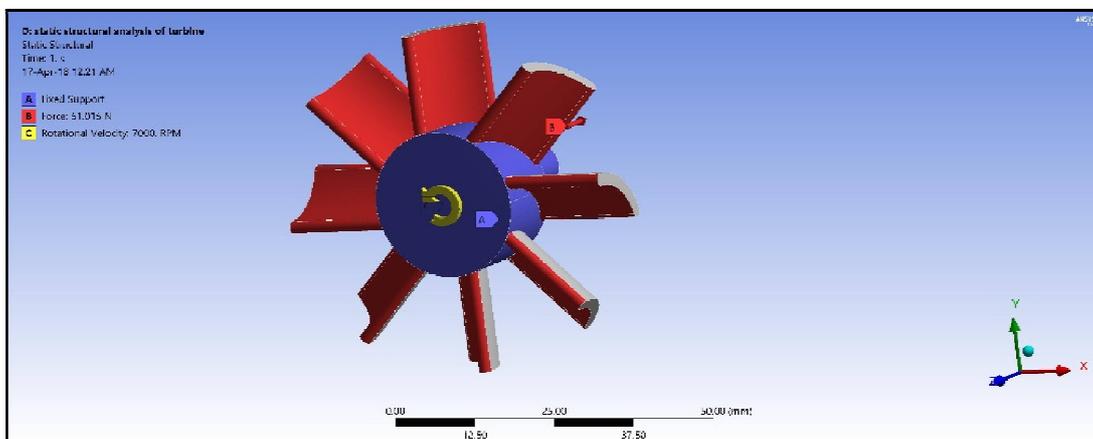


Figure 22 Boundary condition applied over the turbine

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Working environmental temperature, we give as 400 °C. fig 6.6 shows the diagram for given temperature environment for the turbine.

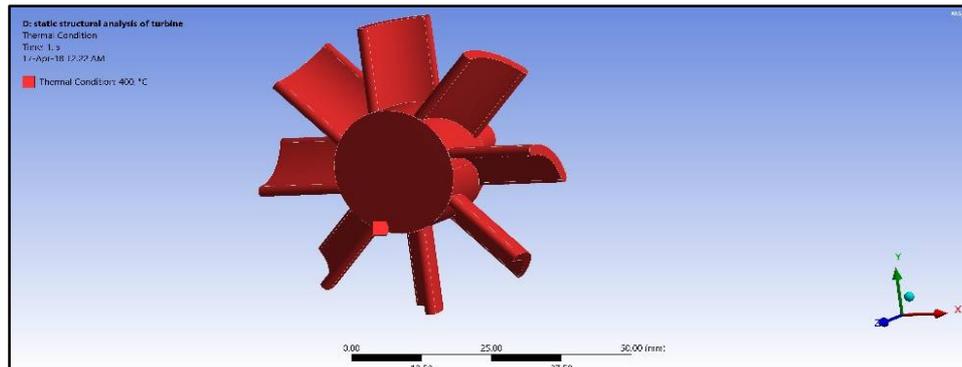


Figure 23 Boundary condition applied over the turbine

After applying all the boundary conditions and solving above analysis we get the total deformation of turbine. Figure 24 shows the total deformation for the turbine.

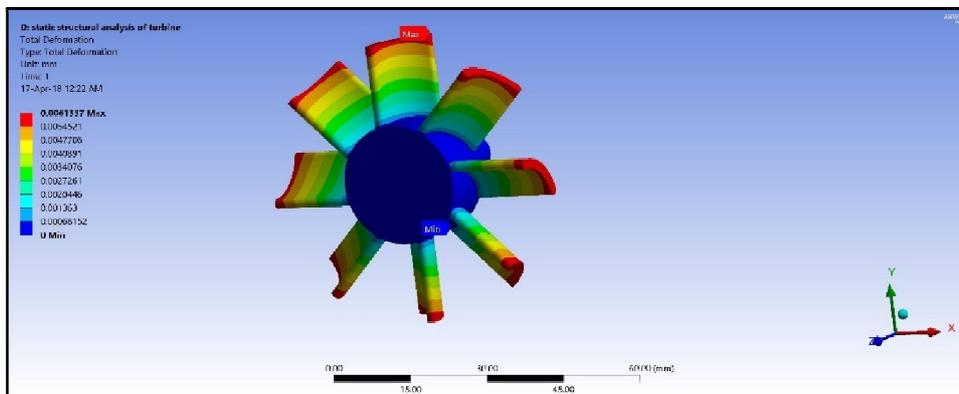


Figure 24 Total deformation of turbine blade

Following figure 25 shows the diagram for the stresses on the turbine blade

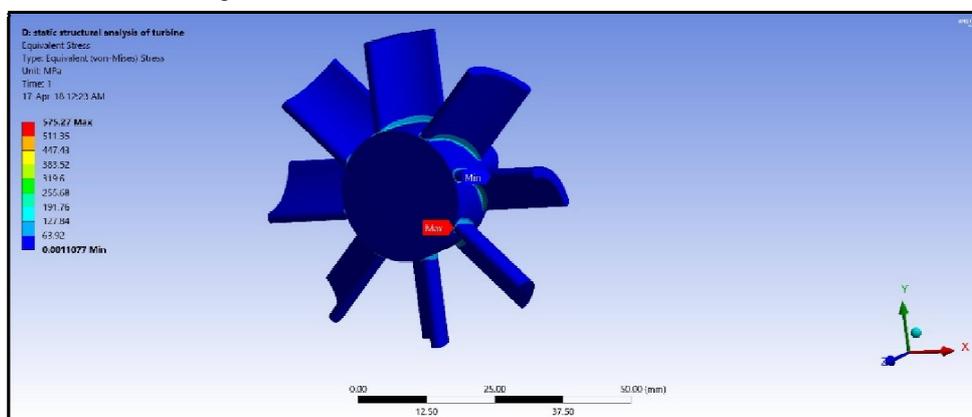


Figure 25 Stresses on the turbine blade

Figure 26 shows the safety factor for turbine blade under the static structural load.

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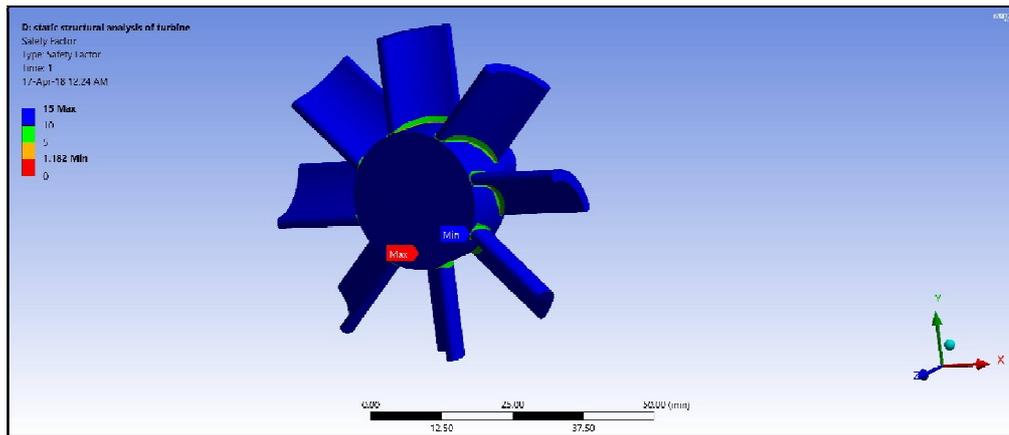


Figure 26 Stresses on the turbine blade

1.3 Centrifugal force on the blade of the turbine

Centrifugal force generated to the blades of turbine during rotation of turbine at high rpm. Which creates tensile force on the blade.

$$\text{Force} = \rho A \omega^2 \left(\frac{0.056^2 - 0.024^2}{2} \right) \quad \dots [6]$$

Where, ρ = Density of material kg/m^3

A = Blade cross-sectional area $\text{m}^2 \times$ number of blade ($3.0039 \times 10^{-5} \text{ m}^2$ calculated from Ansys)

ω = Rotational velocity = $\frac{2\pi \times N}{60}$ rad/s

N = Speed rpm

Maximum rpm can be calculated by using blade speed

$$V_b = \frac{\pi \times D \times N}{60}$$

$$76.9445499 \times 0.4849231 = \frac{\pi \times 0.056 \times N}{60} \quad \dots (48.49\% \text{ efficiency})$$

$$N = 12725.18448 \text{ rpm}$$

$$F = 7840 \times 3.0039 \times 10^{-5} \times 8 \times \left(\frac{2\pi \times 12725.18448}{60} \right)^2 \times \left(\frac{0.056^2 - 0.024^2}{2} \right)$$

$$F = 4282.396746 \text{ N}$$

From force we can find the tensile stress

$$\text{Stress} = \frac{F}{A} = \frac{4282.396746}{3.0039 \times 10^{-5}}$$

$$\text{Stress} = 142561228.6 \text{ N/m}^2 = 142.5612286 \text{ N/mm}^2$$

Factor of safety

$$F_s = \frac{\text{Yield stress}}{\text{working stress}}$$

Where yield stress of Alloy Steel EN24 is 290 N/mm^2 at 400° temperature

$$F_s = \frac{680}{142.5612286}$$

$$F_s = 4.76988032$$

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1.4 Tensile stress in blade on Ansys

Turbine is the most important part of this project and for our design. Which design should be the safest at operating temperature. So, to cross verify above calculation we also analyze the centrifugal force on turbine blade by using Ansys software at static structural type.

Following figure 27 shows the total deformation of the turbine blade under centrifugal force.

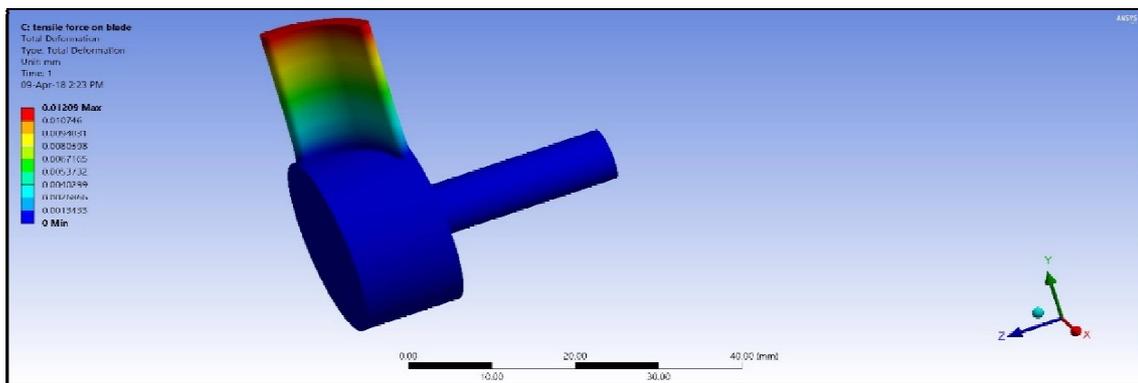


Figure 27 Total Deformation of blade under tensile load

Following figure 28 shows the Equivalent stress over turbine blade under centrifugal force.

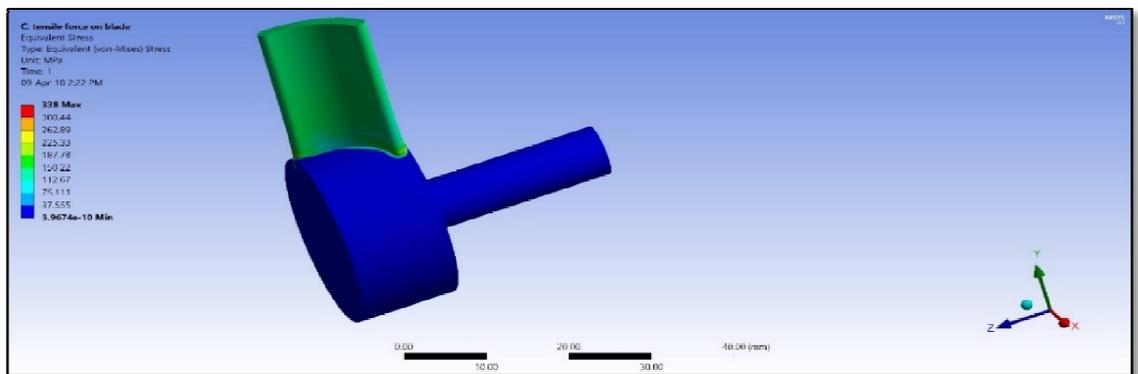


Figure 28 Stresses on blade under tensile load

Following figure 29 shows the Factor of safety of turbine blade under centrifugal force.

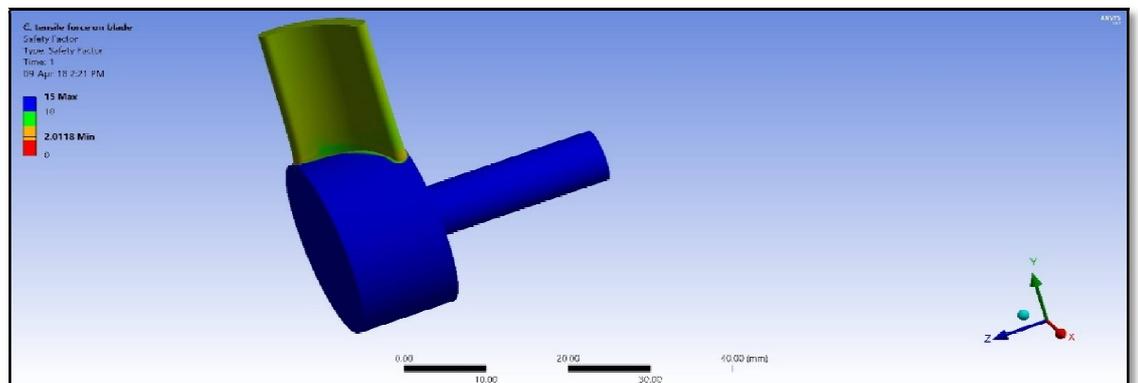


Figure 29 Factor of safety of turbine blade under tensile load

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2 Stresses and deformation on plate

Maximum stresses are generated near bolted area because of the overall weight of the project carried by plate. And plate bolted to the outlet of turbocharger. So, this is important to analyse. This checked by using Ansys software where the structural analysis type is used. Following figure 30 shows the equivalent stresses on the plate.

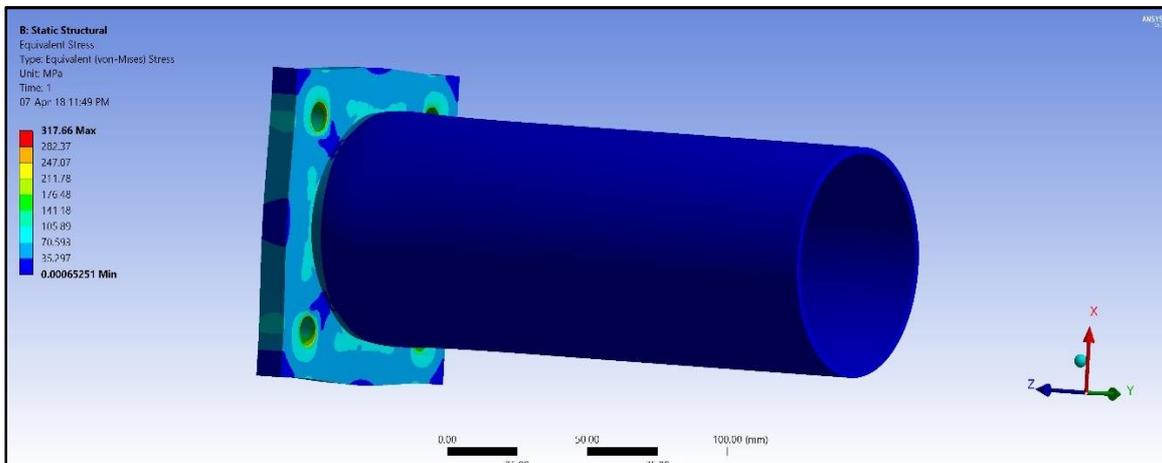


Figure 30 Stresses on the plate

Figure 31 shows static structural analysis in which factor of safety of the plate is 2.1406 which make the design is safe.

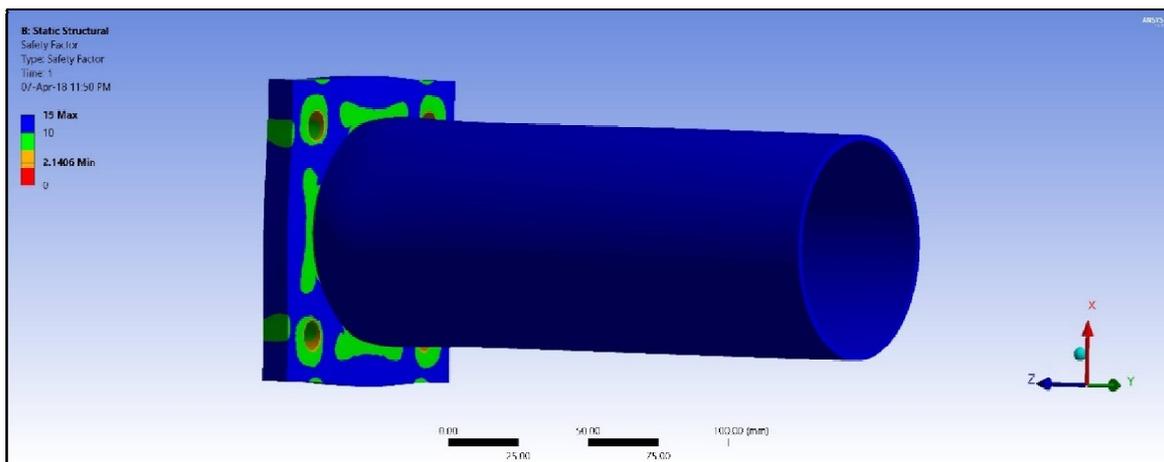


Figure 31 Factor of safety for plate

3 Critical speed of shaft

While high rotating speed of turbine shaft, shaft can be reach to the critical speed. Which can cause failure and accident so the checking of this parameter is important. This checked by using Ansys software, where transient type of analysis is used. Following figures shows the deformation, stress and factor of safety of rod. Figure 32 shows the Total deformation of rod under critical speed of shaft.

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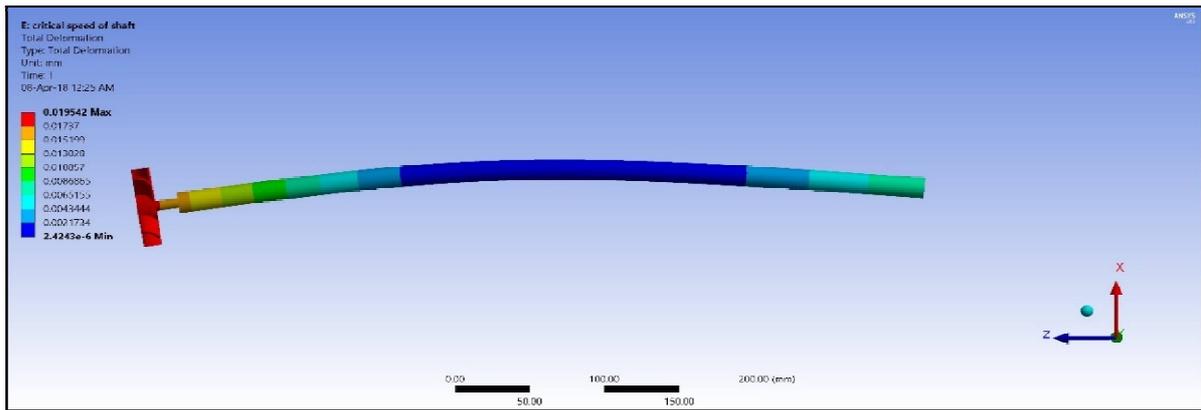


Figure 32 Total deformation of rod under critical speed of shaft

Figure 33 shows transient analysis of shaft in which the Stresses on rod under critical speed of shaft.

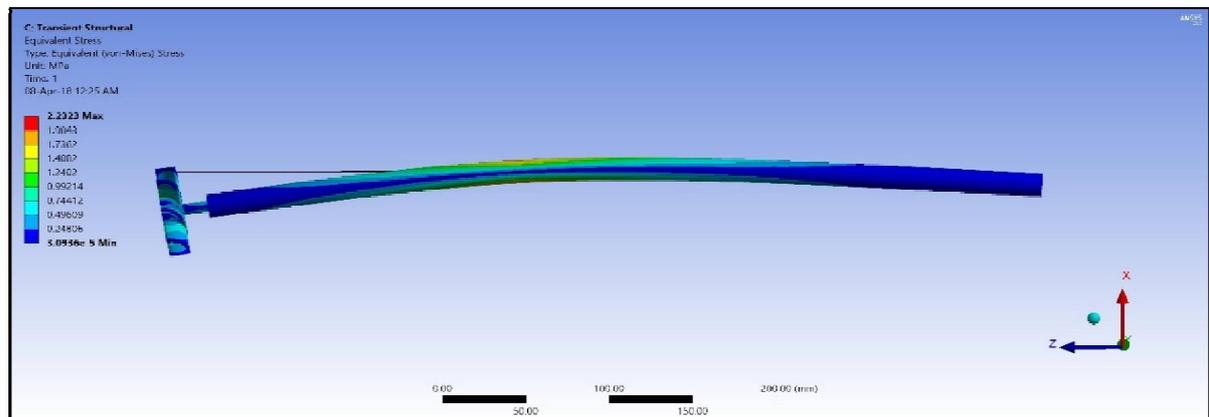


Figure 33 Stresses on rod under critical speed of shaft

Figure 34 shows transient analysis of shaft in which the safety factor of rod under critical speed of shaft.

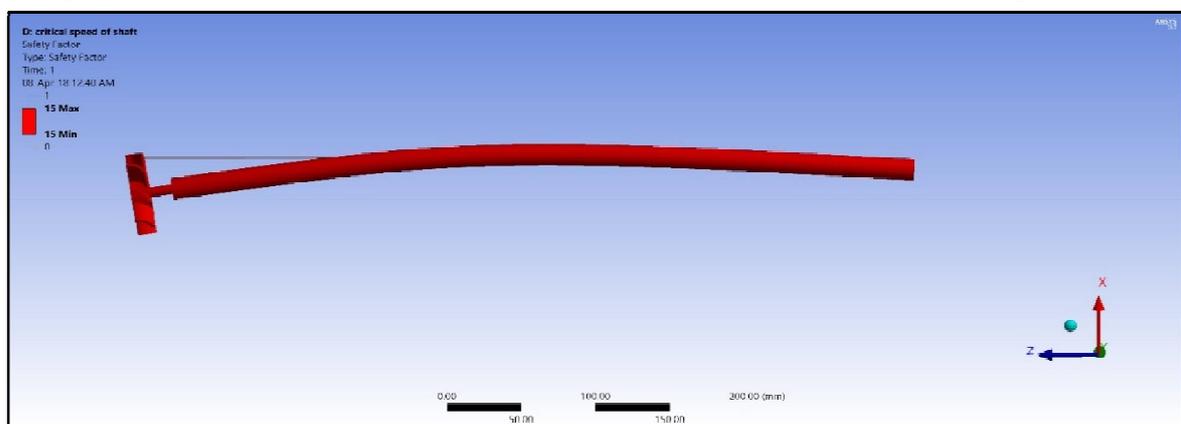


Figure 34 safety factor of rod under critical speed of shaft

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VI. EXPERIMENTAL SETUP

In experimental setup we connect our project setup to the outlet at the turbocharger of TATA 1613 model. As alternator we use DC motor RS-775WC-8514 brush motor from Mabuchi Motors use it as a generator, which motor to generator power conversion factor is 15. This motor connects to the impulse turbine shaft. Between turbine and motor distance maintain as exhaust gases should not affect the generator. To calculate the power output, we measure current from multimeter. For this measurement we design a small circuit which measured resistance is 32.5Ω which is constant with respect to time. The experiment is measured on the engine during its ideal condition means no load on the truck engine. Figure 35 shows the actual setup of the project.



Figure 35 Experimental setup of project

Figure 36 shows the actual picture of our project model which we had attached to the turbocharger outlet.



Figure 36 Actual picture of project

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1 Experimental readings

Following are the readings of current generated by generator in Ampere. It varies time to time because of pulsating flow of exhaust gas and according to acceleration to the engine.

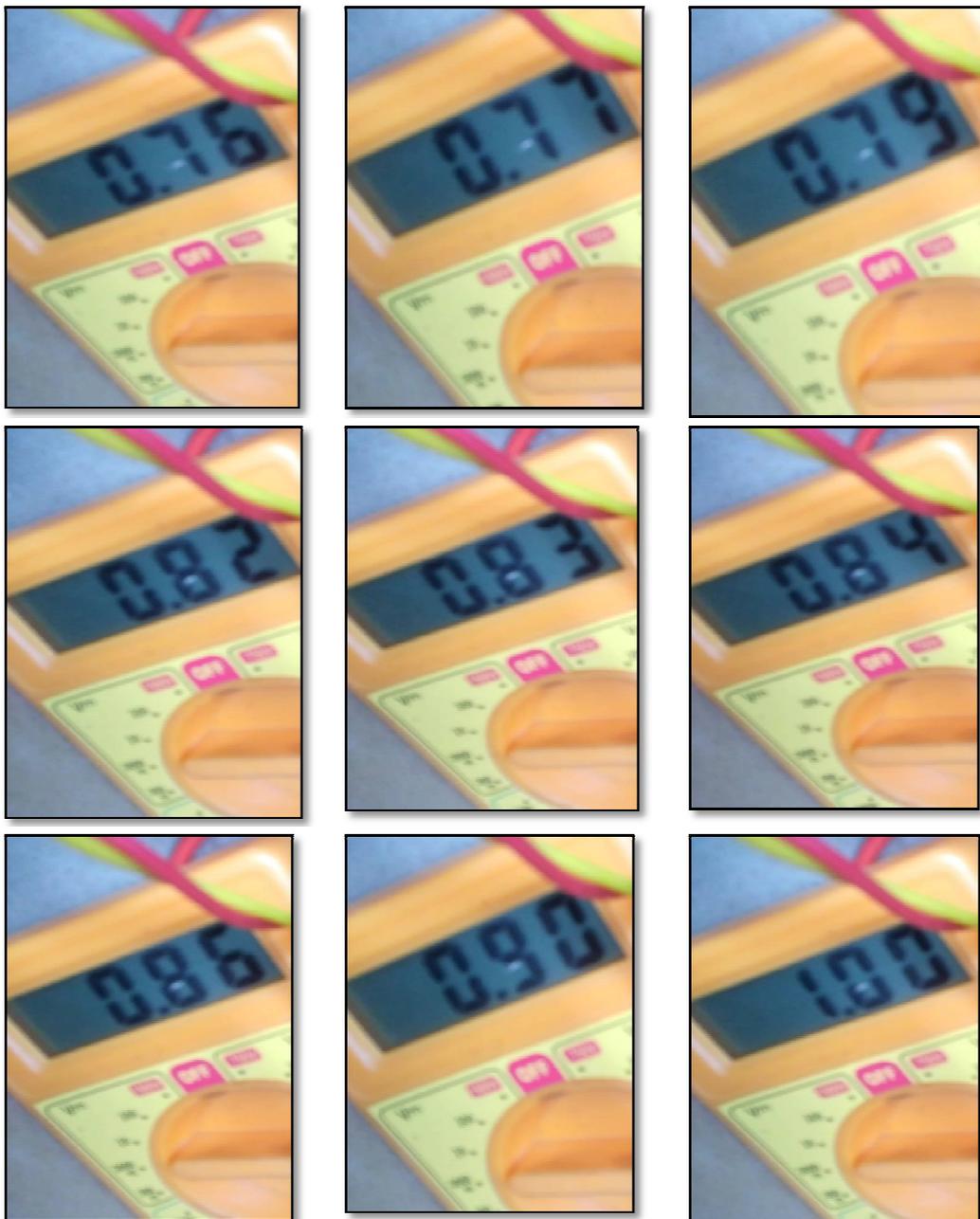


Figure 37 Experimental Readings of current generated in Ampere

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Following figure 38 shows the reading of resistance in circuit. Resistance is given to the circuit to generate torque to the alternator.

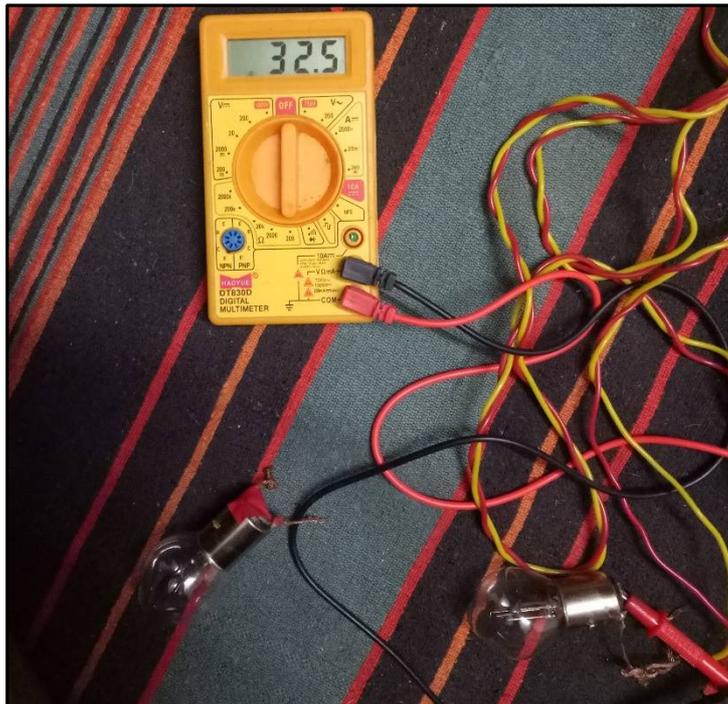


Figure 38 Readings of resistance in circuit

2 Calculation of power output using experimental values

Observation: -

Resistance (R) = 32.5 Ω

Max Current (I) = 1 A

Avg. Current (I) = 0.85 A

Motor to Generator Conversion factor = 15

For power output

$$P_{\max} = I^2 R = 1^2 \times 32.5 = 32.5 \text{ watt}$$

$$P_{\text{avg}} = I^2 R = 0.85^2 \times 32.5 = 23.48125 \text{ watt}$$

It will give two values, Motor to Generator Conversion factor is 15

Max power of motor = **487.5 watt**

Avg power of motor = **352.21875 watt**

So, by using torque of motor we can find rpm. Following relation can be used to find rpm

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

$$352.21875 = \frac{2\pi \times N \times 0.6559}{60} \text{ and } 487.5 = \frac{2\pi \times N \times 0.6559}{60}$$

$$N = \mathbf{5127.74 \text{ rpm}} \text{ to } \mathbf{7097.22391 \text{ rpm}}$$

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Therefore, while experiment turbine rotates in range of 5127.74 rpm to 7097.22391 rpm. And maximum and average power output is 487.5 watt and 352.2185 watt respectively.

VII. CALCULATION FOR ACTUAL TURBINE EFFICIENCY

For examine the results of this project we use concept model. Which saved lot of money and give fine results. But actual model is different which consist of slightly different turbine design. For actual model we will provide stator blades between turbocharger and impulse turbine. Which will give velocity inlet angle at 30°. So, we will keep blade inlet angle also at 30° angle. Following are the calculation for actual turbine.

1 Calculation for turbine design

$$\text{For max. efficiency } (\eta_{b-\max}) = \cos^2 \alpha \quad \dots [6]$$

Consider $\alpha = 10^\circ$

Inlet vel. = $V_e = V_1 = 156.2630874$ m/sec

$$\text{Speed ratio } (s) = \frac{V_b}{V_1} = \cos \alpha / 2$$

$$\bullet V_b = \left(\frac{\cos \alpha}{2}\right) V_1$$

$$\bullet V_b = \left(\frac{\cos 10^\circ}{2}\right) \times 156.2630874$$

$$\bullet V_b = 76.9445499 \text{ m/s}$$

1.1 Calculation for inlet velocity triangle

Velocity of flow at inlet is 30° so the inlet angle is taken 30° according to guide blade will be provided in actual model.

i.e. $\alpha = \theta = 30^\circ$

$$V_{w1} = V_1 \cos \alpha = 156.2630874 \times \cos 30^\circ$$

$$V_{w1} = 135.3278034 \text{ m/s}$$

$$V_{f1} = V_1 \sin \alpha = 156.2630874 \times \sin 30^\circ$$

$$V_{f1} = 78.1315437 \text{ m/s}$$

$$CD = V_{w1} - V_b = 135.3278034 - 76.9445499$$

$$CD = 58.3832531 \text{ m/s}$$

$$V_{r1} = \sqrt{V_{f1}^2 + CD^2} = \sqrt{78.1315437^2 + 58.3832531^2}$$

$$V_{r1} = 97.53533905 \text{ m/s}$$

$$\theta = \tan^{-1} \left(\frac{V_{f1}}{CD} \right)$$

$$\theta = \tan^{-1} \left(\frac{78.1315437}{58.3832531} \right)$$

$$\theta = 53.23133159^\circ$$

1.2 Calculation for outlet velocity triangle

For maximum possible degree of outlet angle 30° is selected.

Also, $V_{r1} = V_{r2} = 97.53533905$ m/s

$$AB = V_{r2} \cos \phi = 97.53533905 \times \cos 30^\circ$$

$$AB = 84.46808138 \text{ m/s}$$

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$$V_{w2} = AB - V_b = 84.46808138 - 76.9445499$$

$$V_{w2} = 7.523531484 \text{ m/s}$$

$$V_{f2} = V_{r2} \sin \phi = 97.53533905 \times \sin 30^\circ$$

$$V_{f2} = 48.76766953 \text{ m/s}$$

$$\beta = \tan^{-1} \left(\frac{V_{f2}}{V_{w2}} \right) = \tan^{-1} \left(\frac{48.76766953}{7.523531484} \right)$$

$$\beta = 84.4154^\circ$$

$$V_2 = \frac{V_{w2}}{\cos \beta} = \frac{7.523531484}{\cos 84.4154}$$

$$V_2 = 77.31080193 \text{ m/s}$$

By using all calculation of inlet and outlet triangle following figure 39 representation of velocity diagram for turbine blade

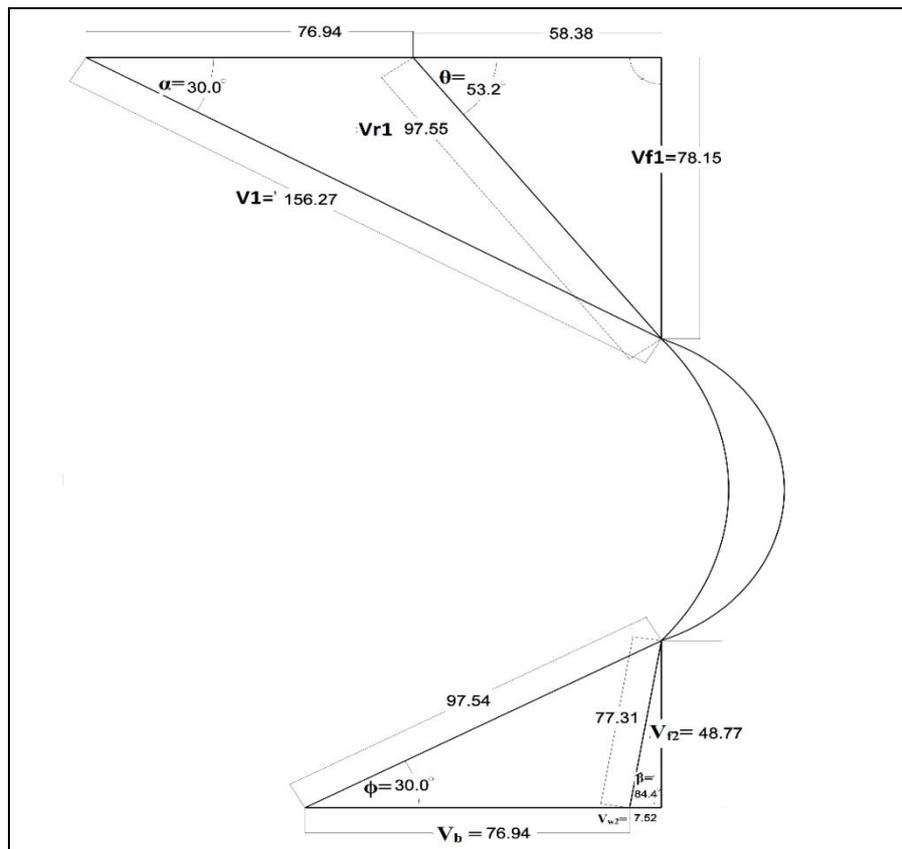


Figure 39 Inlet and Outlet Blade Velocity Triangle For actual turbine

2 For power output of turbine

$$P_{\text{output}} = m \times (V_{w1} - V_{w2}) V_b$$

$$P_{\text{output}} = 0.14713 \times (135.3278034 - 7.523531484) \times 76.9445499$$

$$P_{\text{output}} = 1446.853195 \text{ watt}$$

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3 For turbine efficiency

$$\eta_{\text{turb}} = \frac{P_{\text{output}}}{\text{K.E. supplied}} = \frac{1446.853195}{1796.321337}$$

$$\eta_{\text{turb}} = 0.8054 = 80.54\%$$

VIII. RESULTS

- Maximum power output from concept model at ideal condition = **487.5 watt**
- Average power output from concept model at ideal condition = **352.21875 watt**
- Maximum rpm of the turbine of conceptual model at ideal condition = **7097.22391 rpm**
- Average rpm of the turbine of conceptual model at ideal condition = **5127.74 rpm**
- Percent of efficiency of actual turbine more than conceptual model is
= 80.54 - 48.49 = 32.05 %
- Maximum power output actual turbine can produce at ideal condition
 $487.5 \times 0.3205 = 156.24375$ watt
 - Max power = $156.24375 + 487.5$
 - **Max power = 643.74375 watt**
- Average power output actual turbine can produce at ideal condition
 $352.21875 \times 0.3205 = 112.88610938$ watt
 - Avg power = $112.88610938 + 352.21875$
 - **Avg power = 465.10485938 watt**
- CFD results for the turbine power output
Power of turbine at 5000rpm = **446.484 watt**
Power of turbine at 7000rpm = **625.078 watt**

IX. CONCLUSION

After completion of this project we concluded that our project can produce maximum power output of 487.5 watt at ideal condition which turbine efficiency is 48.49%. after compare its calculation with actual turbine we found that maximum power output at ideal condition increase up to 643.74375watt as it 80 % efficiently work.

At theoretical calculation concept model and actual model power output are 871.0778329 watt and 1446.853195 watt which is double than the experimental results this is because power losses in 6 mm gap between turbine tip and casing which we keep for safety purpose to avoid critical speed of shaft. And second reason is that experiment is only conducted in ideal condition i.e. when there is no load on engine. When there is load on the engine is more, more amount of fuel is added for combustion which increase the pick pressure inside the engine cylinder results into more amount of kinetic energy after turbocharging. So, it will increase the power output. And to by using actual designed turbine it will easily generate the required amount of electrical energy to charge the battery of automobile vehicles.

Apart from all this CFD results also give the nearly same results as experimental results by considering alternator efficiency as 80% we get the electric power output of 357.1872 watt at 5000 rpm and 500.0624 watt at 7000 rpm. Which is very close results to our experiment results which validate our design and results and proof of our concept model work exactly as it designed.

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By considering above all parameter and results we can save the sufficient amount of energy from exhaust gases which can easily replace the alternator which is directly attached to the engine shaft which can improve the efficiency of engine and Save the exhaust gas waste kinetic energy.

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