Design and fabrication of a micro gas turbine using an automotive turbocharger for distributed power generation

A Thesis Submitted
in Partial Fulfillment of the Requirements
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in

Mechanical Engineering

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JALANDHAR

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CERTIFICATE

We hereby certify that the work being presented in this report titled, **"Design and fabrication of a micro gas turbine using an automotive turbocharger for distributed power generation"** in partial fulfillment of the requirement for the award of Degree of Bachelor of Technology (Mechanical Engineering) submitted to the **Department of Mechanical Engineering** of **Dr B R Ambedkar National Institute of Technology, Jalandhar** is a record of our work carried out during **2014-15** under the supervision of **Dr. Subhash Chander**.

The matter presented in this report has not been submitted to any other university or institute for the award of any degree.

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This is to certify that the above statement made by the candidates is correct to the best of my knowledge.

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ABSTRACT

Microturbines are small generators that burn liquid and gaseous fuels, performing a Brayton cycle and ranged less than 250 kW. This kind of turbo-generators are ideally suitable for distributed generation applications due to their capability in supplying reliable and stable power in a stand alone or parallel operation, they also are used as auxiliary power units on aircrafts.

In relation to the Indian power scenario, microturbines can serve as a viable energy source for places without the grid due to their high power to weight ratio .Utilization of off-the-shelf turbochargers in microturbine development helps in lowering costs and leads to an efficient design.

This thesis encompasses the design and fabrication of a gas turbine engine running on a gaseous fuel making use of an automotive turbocharger. The combustor design was based on the principles of aircraft combustor development and was fabricated entirely using locally available parts and materials. As the turbocharger provided the rotor assembly, the other requirement was a lubrication system which was also made out of locally available parts.

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LIST OF SYMBOLS USED IN THE TEXT

A: Area

C_d: Coefficient of discharge

C_p: Heat capacity at constant pressure

H: Enthalpy

K: Pressure loss coefficient

$\dot{m}$: Mass flow rate

MW: Molecular weight

P: Pressure

q: Dynamic pressure

r_p: Pressure ratio

R: Gas constant

T: Temperature

U: Velocity

V: Volume

Greek Symbols

Δ: Change

Φ: Equivalence ratio

η: Efficiency

μ: Bleed ratio

γ: Ratio of heat capacities

ρ: Densities

Subscripts

0: Initial value/stagnation value

1: Compressor inlet value

2: Impeller exit value

3: Compressor exit value

4: Turbine inlet value

5: Turbine exit value

ref: Reference value

h: Hole

ft: Flame tube

A: annulus

LIST OF FIGURES IN THE TEXT

CHAPTER 1

The motivation behind the project

This chapter briefly examines:

- The Indian power production scenario and the problems associated with it.
- Distributed power generation and its applicability as a solution to the problems India is facing.
- The current project as a source of distributed power generation
- The current project's viability and applicability in the Indian scenario.

1.1 The Indian power scenario

Indian power sector has made considerable progress in the last decade but there is a whole gamut of challenging areas in the power sector that India needs to address on priority in order to meet its growth targets.

The importance of electricity as a prime driver of growth is very well acknowledged and in order to boost the development of power system, the Indian government has participated in a big way through creation of various corporations such as, State Electricity Boards (SEB), NTPC Limited, NHPC Limited and Power Grid Corporation Limited (PGCL), etc. However, even after this the country is facing power shortage.

India has come a long way as far as the power production is concerned with an installed capacity of 271 GW as compared to 42 GW in 1985 (refer fig 1.1 &

1.2). The gross electricity generated by utilities stands at 1106 TW-h as compared to 183 TW-h in 1985.

Despite of what India's power sector has achieved, there are many problems faced by the power sector and these need to be addressed. **One of the issues plaguing the power sector in a big way is shortage of equipment**. This has been a significant reason for India missing its capacity addition targets. While the shortage has been primarily in the core components of Boilers, Turbines and Generators. The current power infrastructure in India is not capable of providing sufficient and reliable power supply. **Nearly 350 to 400 million people have zero access to electricity since the grid does not reach their areas** (refer to fig 1.3). Another problem is **unstable power supply.** There are frequency fluctuations caused by load generation imbalances in the system and this keeps happening because consumer load keeps changing. Frequency should ideally be close to the rated frequency all the time, it has been a serious problem in India**.** **Poor power quality control** has knock-on effects on equipment operation, including large-scale generation capacity.

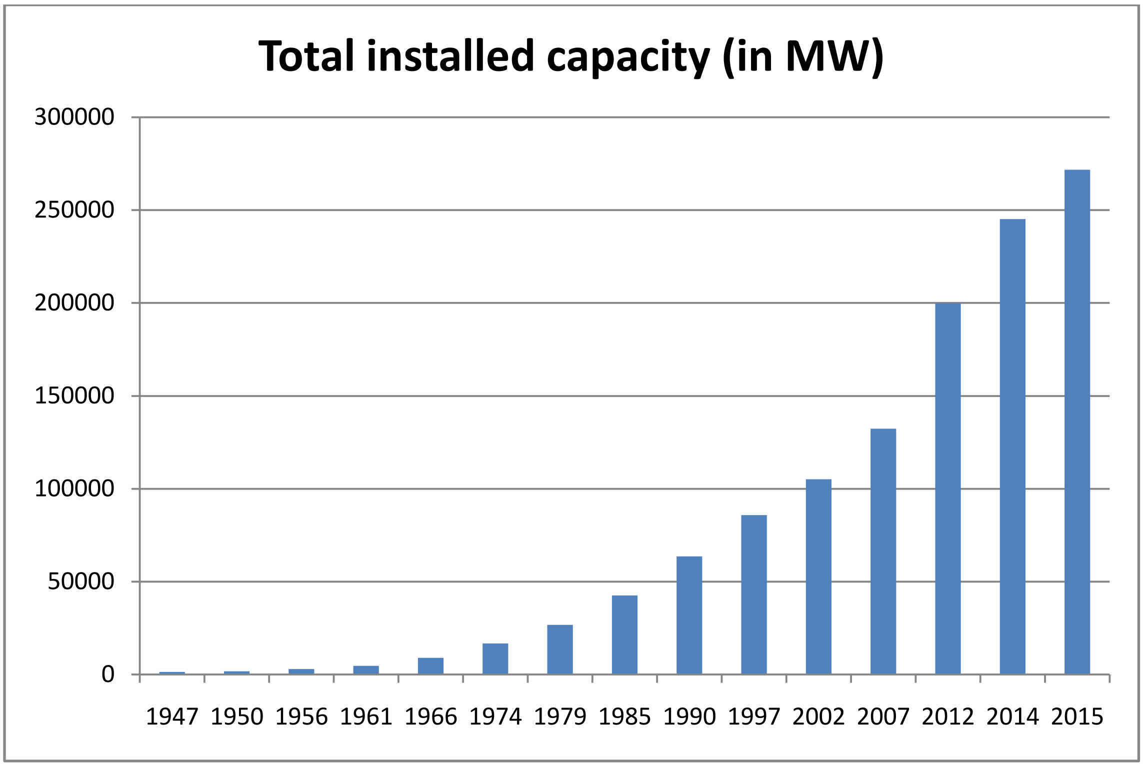
Total installed capacity (in MW)
300000
250000
200000
150000
100000
50000
0
1947 1950 1956 1961 1966 1974 1979 1985 1990 1997 2002 2007 2012 2014 2015

Fig. 1.1

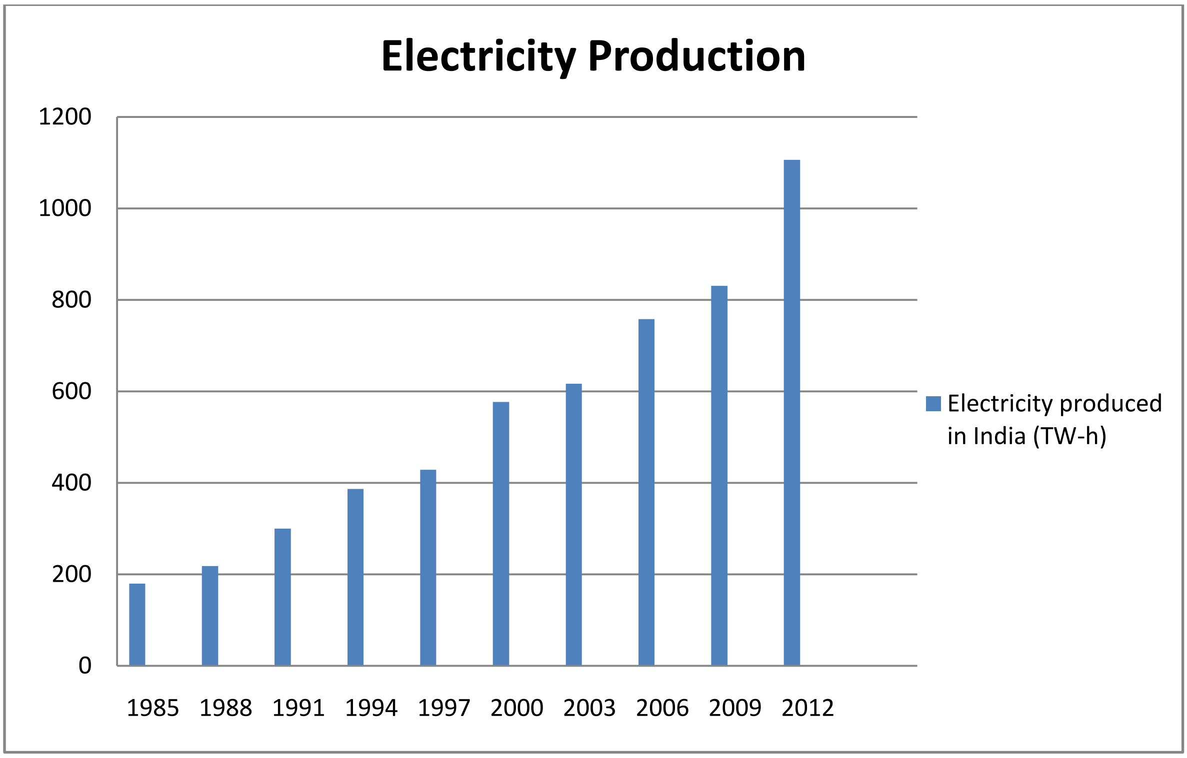
Electricity Production
1200
1000
800
600
400
200
0
1985 1988 1991 1994 1997 2000 2003 2006 2009 2012
Electricity produced in India (TW-h)

Fig. 1.2

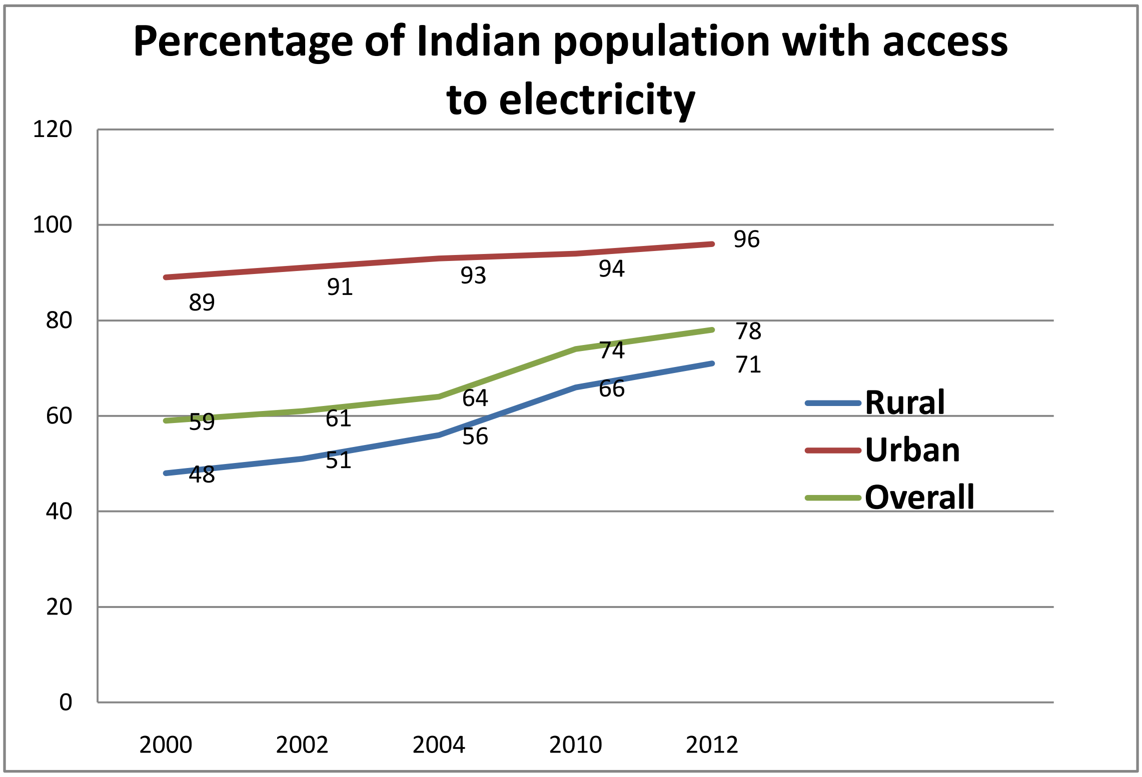

Fig. 1.3

1.2 Power for all: Key considerations

Looking at Figs 1.1, 1.2 and 1.3, there is hardly any doubt that the power sector is growing and it has grown substantially over the years but there are things that this data does not tell us, let's take a look:

Current Profile of Electricity Deficit

Statistics from the National Sample Survey confirm that India's remaining 350 million people without electricity include the country's poorest people living in the most remote areas. About 93 percent of those without electricity reside in rural areas. Out of India's 35 states and union territories, just five states account for more than four-fifths of people without electricity access (Fig.1.4) Bihar has the lowest overall access rate, at 25 percent, followed by

Uttar Pradesh at 43 percent, Orissa at 56 percent, West Bengal at 59 percent, and Rajasthan at 77 percent.

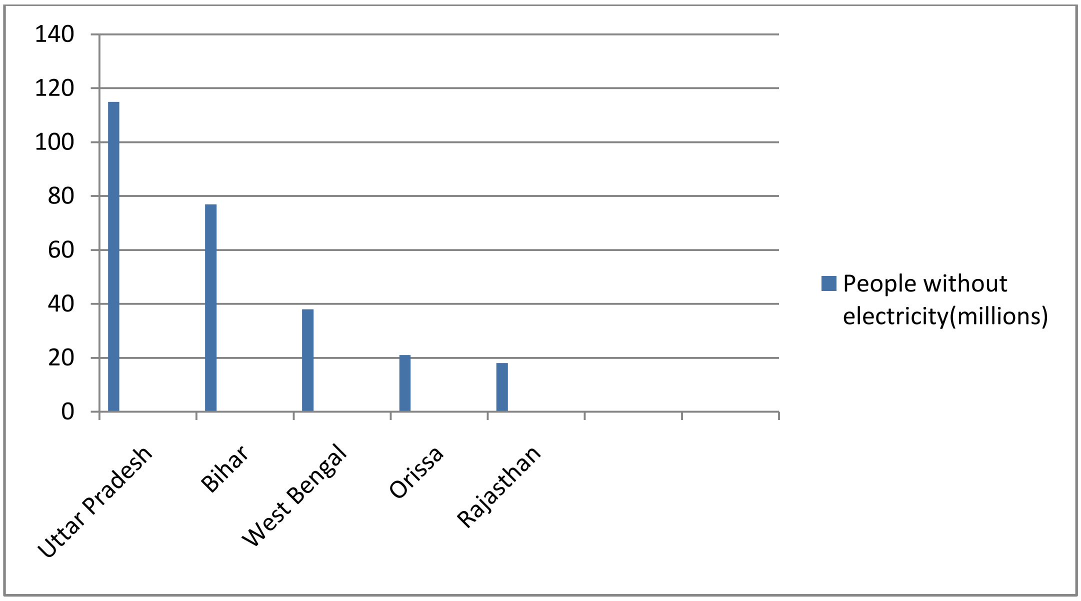

Fig. 1.4 People without electricity in key states

Poor Reliability coupled with low electrification rate

Unreliable power supply undermines India's large investments in rural electrification and its significant benefits to both individual households and the country overall. Without lighting in the evenings, children cannot study and people cannot watch television. In farmers' fields, water cannot be pumped to crops, thus lowering yields.

People without electricity are less likely to adopt service, reasoning that they should not have to pay monthly costs for a service that is unavailable at the times they need to use it. There is no use in investing in lines, transformers, and poles if power does not flow through them.

When poor reliability is coupled with low electrification rate what we witness are regions with no grid connectivity or even if there is grid connectivity, an erratic power supply. Poor reliability also reduces the amount of revenue collected by the electricity companies, which, in turn, provides them little incentive to invest in maintaining service and grid

connectivity, hence it is no surprise that regions with lowest rate of electrification also have heavy electricity outages.

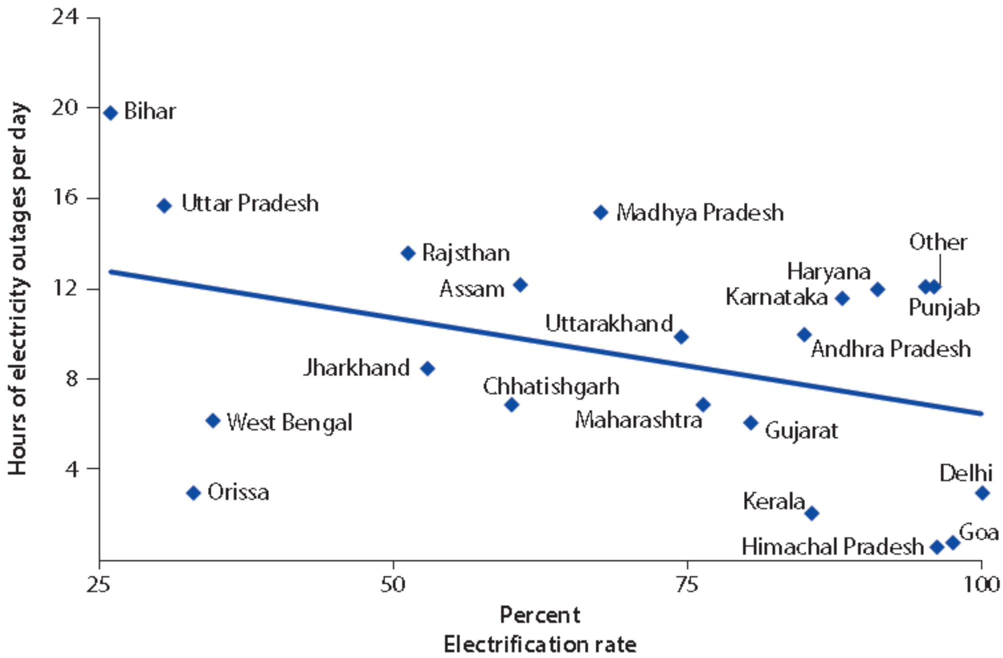

Fig. 1.5 Electrification rate and unreliability plot

Transmission and Distribution losses

In India, average T & D losses have been officially indicated at nearly 18% of the electricity generated in 2012 (Fig 1.6). However in studies carried out by independent agencies these losses have been estimated to be as high as 40% in some states. This huge loss is bleeding the country's resources when compared to the world's average which sits at around 8% and the United States' average at nearly 6% (Fig 1.6). To put these figures into perspective, out of 1106000 million kWh of total electricity generated in 2012, 201000 million kWh of electricity was lost or unaccounted for. With this electricity, nearly 250 million more people could have had access to electricity. This figure is of far greater concern because nearly 30% of rural India is still un-electrified.

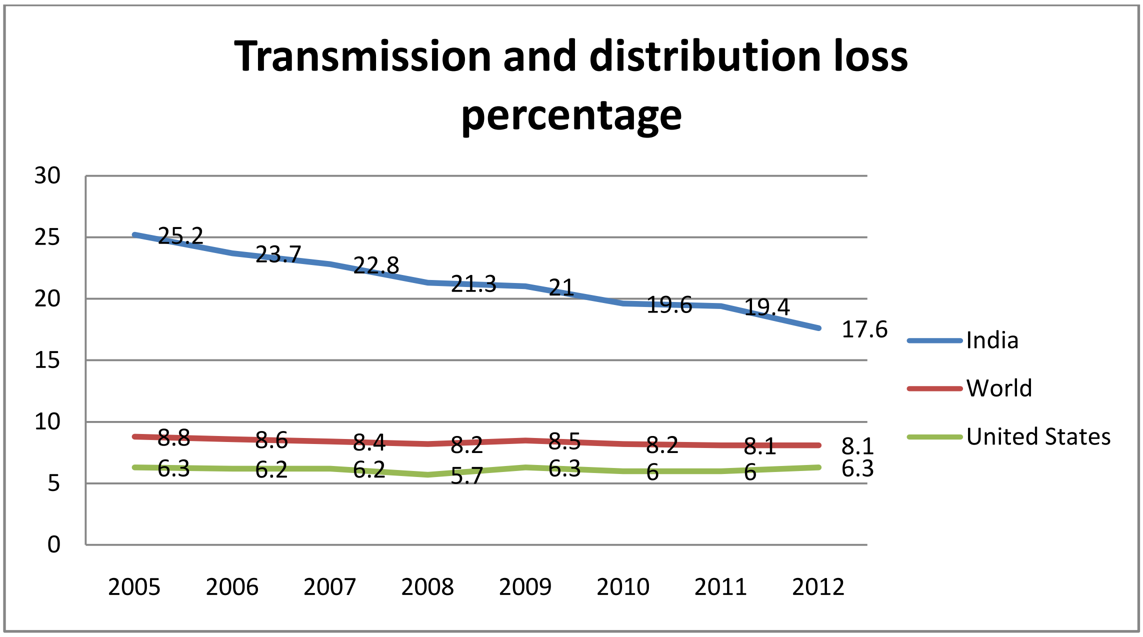

Fig. 1.6

1.3 The Solution: Distributed power generation

Distributed generation is defined as installation and operation of **small modular power generating technologies** that can be combined with energy management and storage systems. **It is used to improve the operations of the electricity delivery systems at or near the end user**. These systems may or may not be connected to the electric grid.

A distributed generation system can employ a range of technological options from renewable to non-renewable and can operate either in a connected grid or off-grid mode. The size of a distributed generation system typically ranges from less than a kilowatt to a few megawatts.

Why do we think that it can be a solution to India's power issues?

- **They can be installed quickly, often in a matter of days or weeks compared to years for central power stations**. Rapid deployment is extremely useful in cases where there is unmet energy demand and supply must ramp up quickly. Quick build time is also useful when restoring power in the wake of natural disasters or within the context of chronically unreliable power systems.

- Distributed power technologies require less money to buy, build and operate. In regions where capital is constrained, **it is increasingly important to provide critical infrastructure such as electricity without having to raise hundreds of millions of dollars** in capital to finance infrastructure projects. **The Indian scenario demands something that is inexpensive and can reach places where the grid is unavailable or is impractical to take.**

- Because of their small size, **distributed power technologies enable energy providers to match the level of demand with the level of supply and to increase supplies incrementally as needed**. Centralized power stations require large capital investment and are available in sizes that are often not appropriate for the required level of supply. **Incremental distributed power development is the appropriate development path for many parts of India.**

- Distributed power technologies are sited at or near demand, **this facilitates a local level of control, operations and maintenance that is not possible with central power stations.** This enables system owners and operators to monitor and customize distributed power solutions to meet their specific needs**. Putting things in the Indian context, this might help lower T&D losses.**

- If coupled with the grid, **it can help get rid of erratic power supply**, that most of the parts of our country face.

These points clearly justify the suitability of Distributed Generation in the Indian scenario.

1.4 The Scope of distributed generation

Globally, distributed power generation investments and installations are on the rise.
By 2020, $206 billion will be invested annually and distributed power applications will account for 42 percent of global capacity additions. (refer to fig 1.7

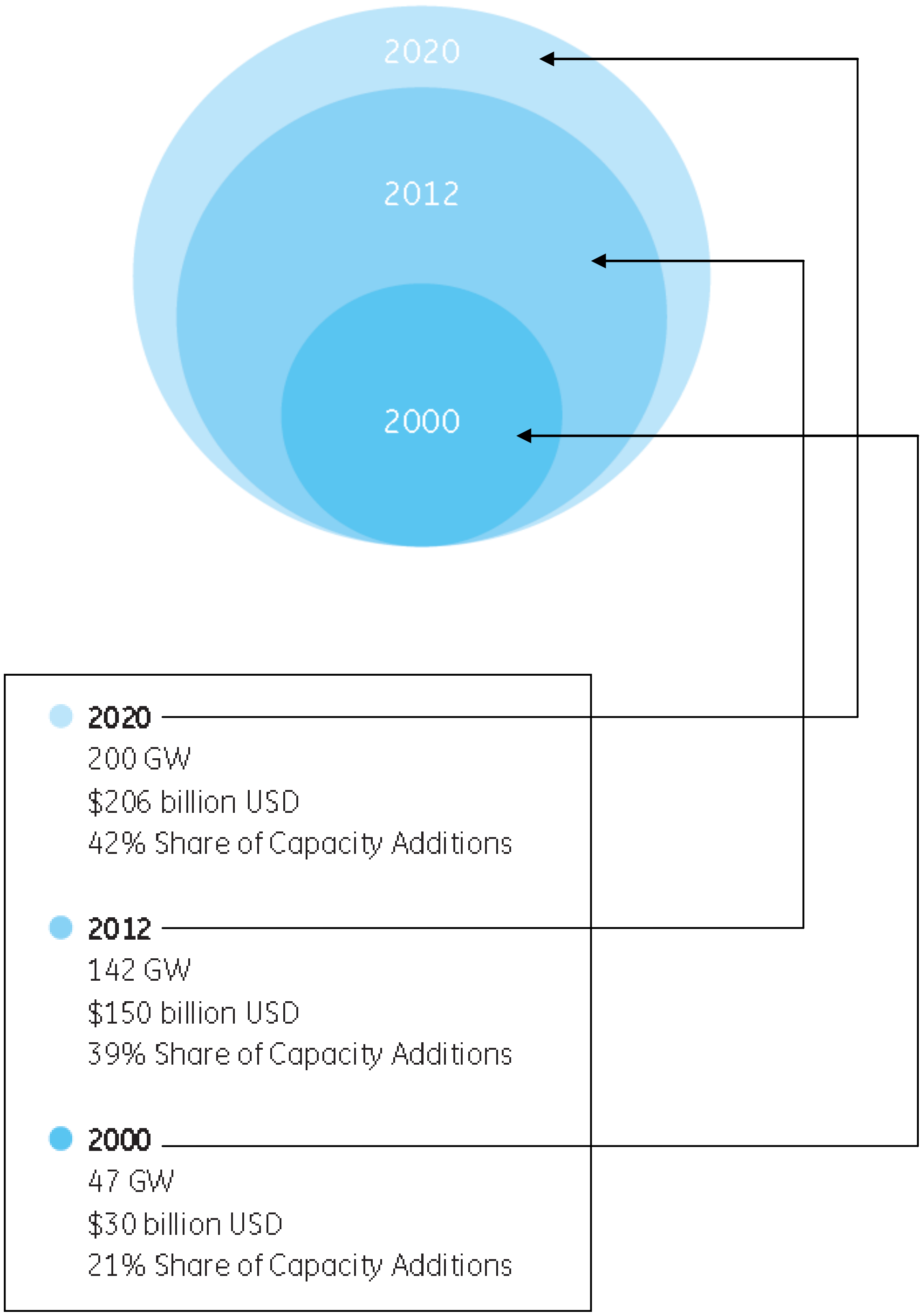

Fig. 1.7: Projected growth of distributed power generation.

1.5 Distributed power generation and the micro gas turbine.

Microturbines are miniature electricity generators that burn gas and liquid fuels in a turbine to create rotation that drives a generator. They range in size from 1 kW to 250 kW. Like their larger cousins, gas turbines, microturbines work by compressing and heating air and then igniting the compressed air, which expands and rotates the turbine blades in order to drive a generator and produce power. **Microturbine compressors and turbines are radial-flow designs, unlike gas turbines, which have axial designs and multiple stages of blades.**

The fact that micro gas turbines are still in the development phase provides us with a terrific opportunity to work on and a niche as far as its commercial status is concerned (refer to Fig 1.8)

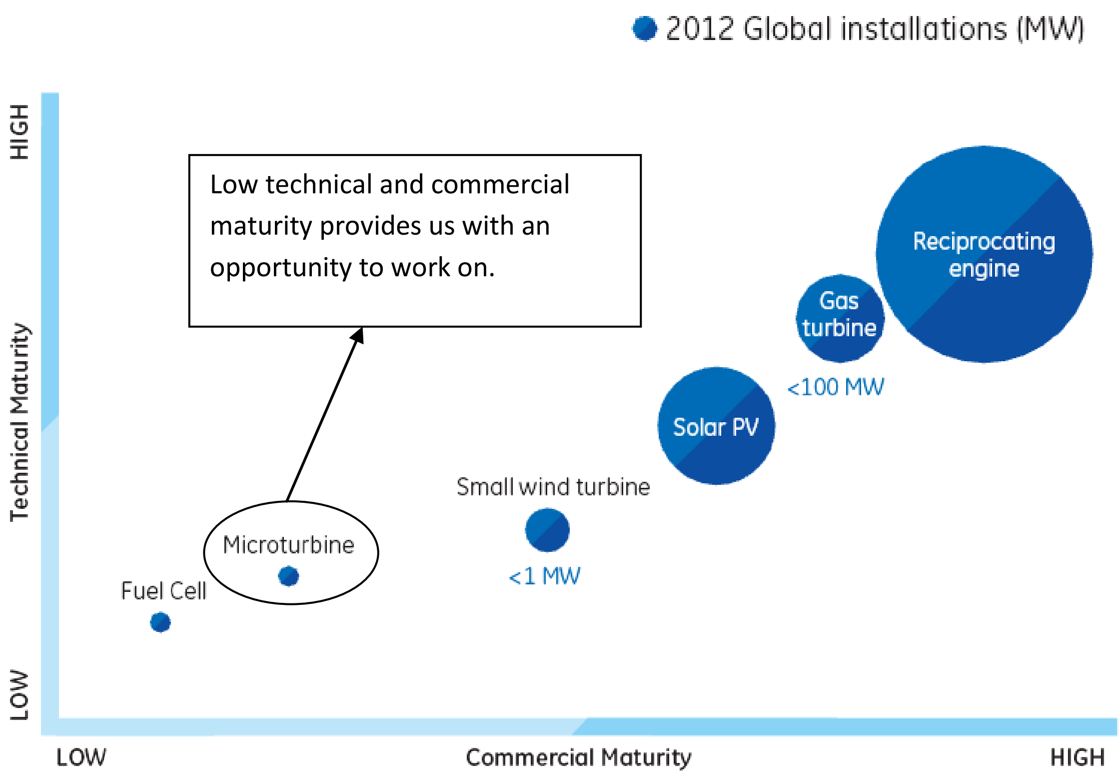

Fig.1.8: Technical and commercial maturity of various distributed power production systems.

1.6 The current project

During the last few decades, **several attempts have been done to develop microturbines with efficiency levels close to those of larger gas turbines. Particularly for microturbines with power rated below 100 kW**.

Many developments have failed to obtain sufficient efficiency, reliability and cost effectiveness due the small-effects: low Reynolds in the turbo machinery passages causing relatively high viscous losses, relatively high tip clearances due the manufacturing tolerances and bearing limitations, large area-to-volume ratios resulting in high heat losses and inadvertent heat transfer to the compressor and high auxiliary system losses. Another factor is that the development of efficient **turbo machinery optimized for a particular cycle is very expensive and can only be justified with very large production volumes.** This factor finds an interesting opportunity in using small automotive turbocharger components. **Off-the-shelf automotive turbochargers already present sufficient efficiency for gas turbine cycles and the price is low due the large production volumes**

This project therefore uses an automotive turbocharger to make a prototype of a micro gas turbine.

These social, technological and commercial implications led us to choose this project.

In our opinion, distributed power generation is the future of power production and we consider our project to be a promising solution to several power problems of our nation.

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CHAPTER 2

The Fundamentals

This chapter is intended to introduce the reader to:

- Gas turbines and the their cycles
- Combustion chamber :Basic design considerations
- Combustion chamber types
- Combustion chamber characteristics
- Combustion chamber aerodynamics
- Combustion chamber chemistry
- The automotive turbocharger and its components
- Gerotor pump and its components

The ideas discussed in this chapter will be used in subsequent chapters and the reader will be repeatedly directed to this chapter.

2.1 Gas Turbines

Of the various means of producing mechanical power the turbine is in many respects the most satisfactory. The absence of reciprocating and rubbing members means that balancing problems are few, that the lubricating oil consumption is exceptionally low, and that reliability can be high. The inherent advantages of the turbine were first realized using water as the working fluid, and hydro-electric power is still a significant contributor to the world's energy resources. Around the of the twentieth century the steam turbine began its career and, quite apart from its wide use as a marine power plant, it has become the most important prime mover for electricity generation. Steam turbine plants producing well over 2000 MW of shaft power with an efficiency of 40 per cent are now being used. In spite of its successful development, the steam turbine does have an inherent disadvantage. It is that the production of high-pressure high-temperature steam involves the installation of bulky and expensive steam generating equipment, whether it is a conventional boiler or a nuclear reactor. The significant feature is that the hot gases produced in the boiler furnace or reactor core never reach the turbine; they are merely used indirectly to produce an intermediate fluid, namely steam. A much more compact power plant results when the water to steam step is eliminated and the hot gases themselves are used to drive the turbine. Serious development of the gas turbine began not long before the Second World War with shaft power in mind, but attention was soon transferred to the turbojet engine for aircraft propulsion. The gas turbine began to compete successfully in other fields only in the mid nineteen fifties, but since then it has made a progressively greater impact in an increasing variety of applications.

In order to produce an expansion through a turbine a pressure ratio must be provided and the first necessary step in the cycle of a gas turbine plant must therefore be compression of the working fluid. If after compression the working fluid was to be expanded directly in the turbine, and there were no losses in either component, the power developed by the turbine would just equal that absorbed by the compressor. Thus if the two were coupled together the combination would do no more than turn itself round. But the power developed by the turbine can be increased by the addition of energy to raise the temperature of the working fluid prior to expansion. When the working fluid is air a very suitable means of doing this is by combustion of fuel in the air which has been compressed. Expansion of the hot working fluid then produces a greater power output from the turbine, so that it is able to provide a useful output in addition to driving the compressor. This represents the gas turbine or internal-combustion turbine in its simplest form.

The original general purpose gas turbine engine design, known as the turbojet, was developed independently by two engineers, Frank Whittle of the United Kingdom, and Hans von Ohain of Germany in the 2930s. The turbojet has since been modified into a number of variants, including turboprop, turbo-shaft, turbofan, and the more recent prop-fan (also known as open rotor or non ducted turbofan). Although each type of turbine engine has very unique purposes and performances, they each operate on the same basic principles, which in the thermodynamics are known collectively as the Brayton Cycle.

The key components of a gas turbine engine i.e. its compressor, turbine and the combustion chamber are shown schematically in Fig.2.1

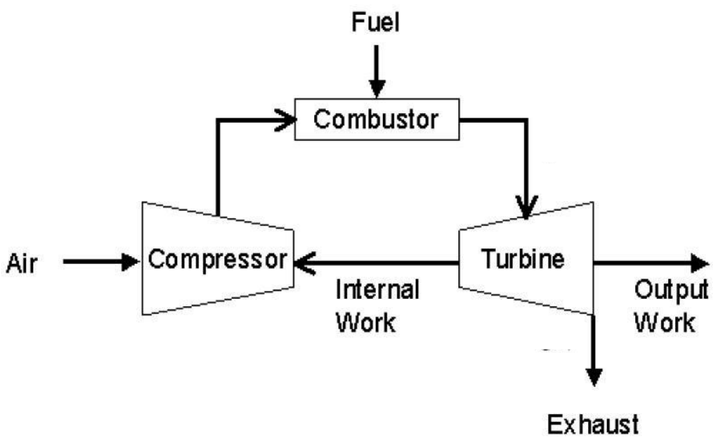

Fig.2.1: Schematic representation of a gas turbine

Ideally, this system is based on the Brayton cycle which assumes isentropic compression and expansion while constant pressure heat addition and removal. A cycle that is closer to reality has non isentropic compression and expansion and a pressure loss in the heat addition and removal processes. Fig2.2 shows ideal and real Brayton cycles, 1-2-3-4 being the ideal cycle while 1-2′-3′-4′ being the real one. The x-axis represents entropy while the y-axis represents the temperature.

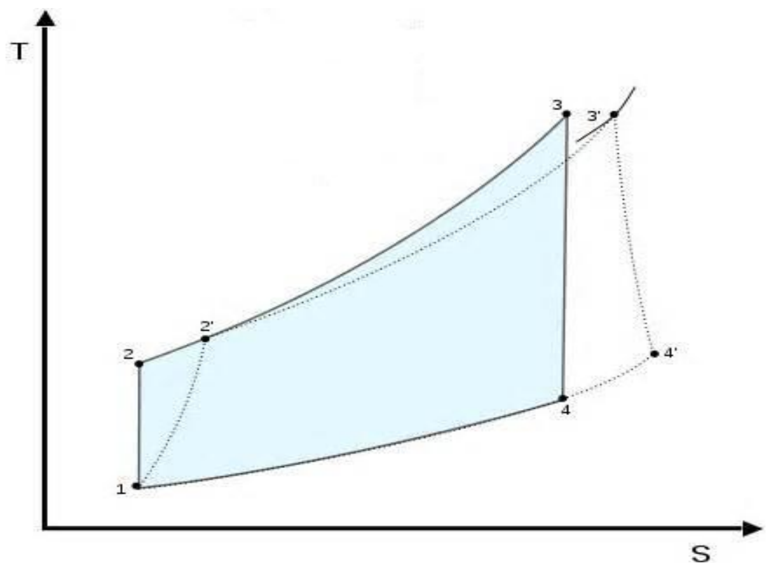

Fig.2.2: Ideal and real Brayton cycle representations.

2.2 The Combustion Chamber

2.2.1 Basic design consideration

At the very heart of the current project lied the development of the combustor or the combustion chamber. Therefore, it is of interest to examine the considerations that dictate the basic geometry of the "conventional" gas turbine combustor. Figure 2.3a shows the simplest possible form of combustor—a straight-walled duct connecting the compressor to the turbine. Unfortunately, this simple arrangement is impractical because the pressure loss incurred would be excessive.
The fundamental pressure loss because of combustion is proportional to the square of the air velocity and, for compressor outlet velocities of the order of 270 m/s, this loss could amount to almost one-third of the pressure rise achieved in the compressor. To reduce this pressure loss to an acceptable level, a diffuser is used to lower the air velocity by a factor of about 5, as shown in Figure 2.8b.

Having fitted a diffuser, a flow reversal must then be created to provide a low-velocity region in which to anchor the flame. Figure 2.3c shows how this may be accomplished with a plain baffle. The only remaining defect in this arrangement is that to produce the desired temperature rise, the overall chamber air/fuel ratio must normally be around 30–40, which is well outside the limits of flammability for hydrocarbon–air mixtures. Therefore, it is required to have a region where air/fuel ratio is low and this is known as the primary combustion zone.

Ideally, the air/fuel ratio in the primary combustion zone should be around 28, the primary zone is designed (will be discussed in a later chapter of this thesis) in a way that it has a lower air/fuel ratio and the burning initiates and the air not required for combustion is admitted downstream of the combustion zone (primary and secondary zones, to be discussed in a later chapter) to mix with the hot burned products, thereby reducing their temperature to a value that is acceptable to the turbine. Figure 2.3 thus illustrates the logical development of the conventional gas turbine combustion chamber in its most widely used form.

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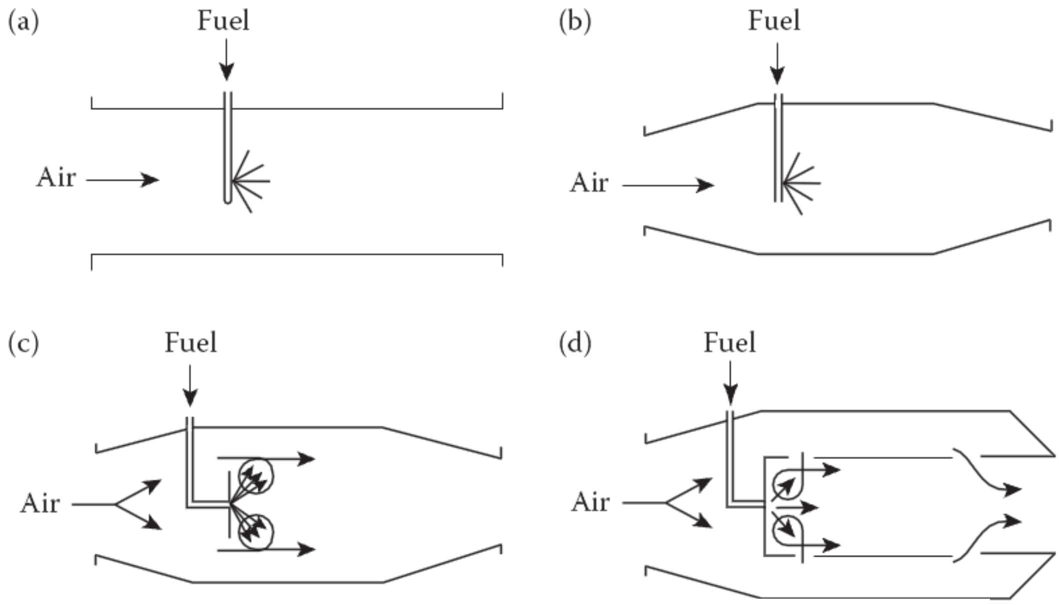


Fig2.3 Derivation of conventional combustor configuration

2.2.2 Combustor Types

A combustor is named after the flame tube it has i.e. the region in which the fuel burns. The choice of a particular combustor type and layout is determined largely by the overall engine design and by the need to use the available space as effectively as possible. There are two basic types of combustor, tubular and annular.

1.) Tubular Combustor

A tubular (or "can") combustor is comprised of a cylindrical liner (flame tube) mounted concentrically inside a cylindrical casing.
We have used a tubular combustor in the current project. Fig 2.4 shows the flame tube of a tubular combustor, the air inlet holes on the flame tube allow the air from the compressor to reach the fuel.

Fig 2.4: The flametube of a tubular combustor

2.) Annular Combustor

In this type, an annular liner is mounted concentrically inside an annular casing. In many ways it is an ideal form of chamber, because it's clean aerodynamic layout results in a compact unit of lower pressure loss than other combustor types. Its main drawback stems from the heavy buckling load on the outer liner. Fig 2.5 shows the outer and inner holes of an annular combustor.

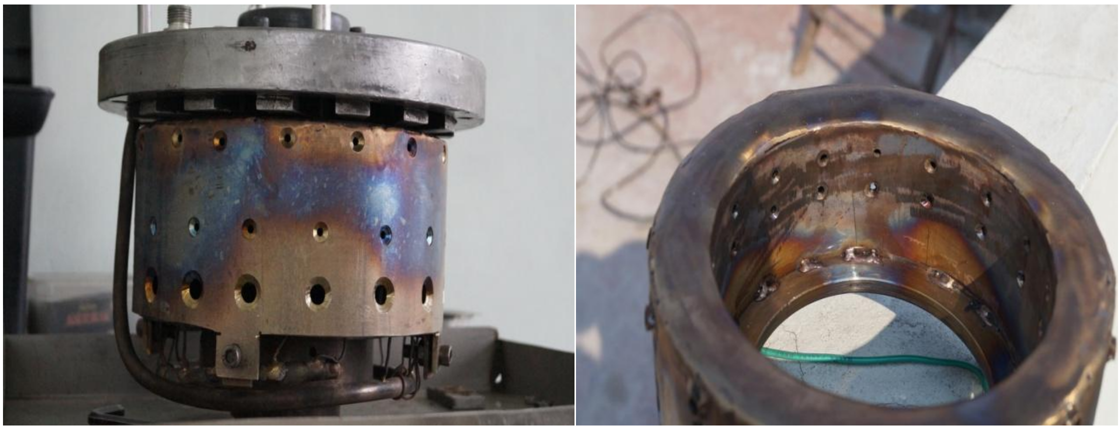


Fig 2.5: The annular flame tube with holes on both sides of the annulus.

Although tubular and annular are the two basic types of combustors, a compromise is made between these two extremes which results in the tuboannular" or "can-annular" combustor, in which a number of equispaced tubular liners are placed within an annular air casing.
The three different combustor types are illustrated in Figure 2.6.

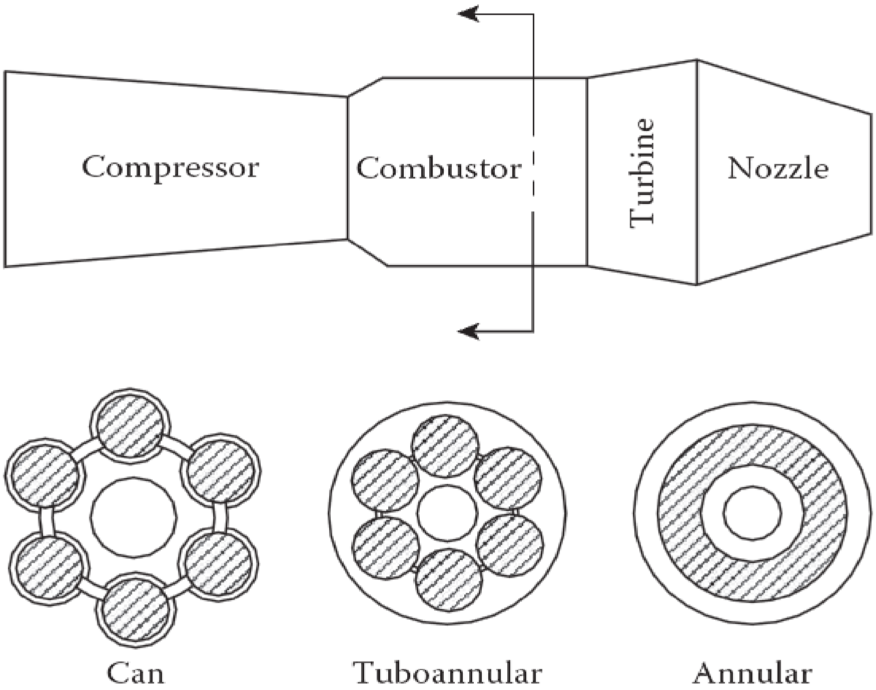


Figure 2.6 Illustration of three main combustor types used in aircrafts

2.2.3 Essential features of a combustion chamber

a.) The Diffuser

As explained in section 2.2.1, it is essential to lower the speed of the air coming from the compressor.

Lower annulus velocities promote:

- Flame stabilization
- flow uniformity,
- jet penetration at steeper angles,
- lower skin-friction losses and
- lower sudden downstream expansion losses.

The goal of diffuser design is to minimize the total pressure loss incurred while recovering as much dynamic velocity head as possible. A good design

- has high static pressure recovery with low pressure losses,
- is insensitive to fluctuations in inlet conditions,
- is short in length.

In aircraft combustors, two types of diffuser are generally used:

1.) Dump type diffusers

It consists of a short conventional diffuser or a prediffuser (can be conical pipe) in which the air velocity is reduced to almost half its inlet value. At exit, flow is dumped into the highly separated step region. Refer to Fig 2.7

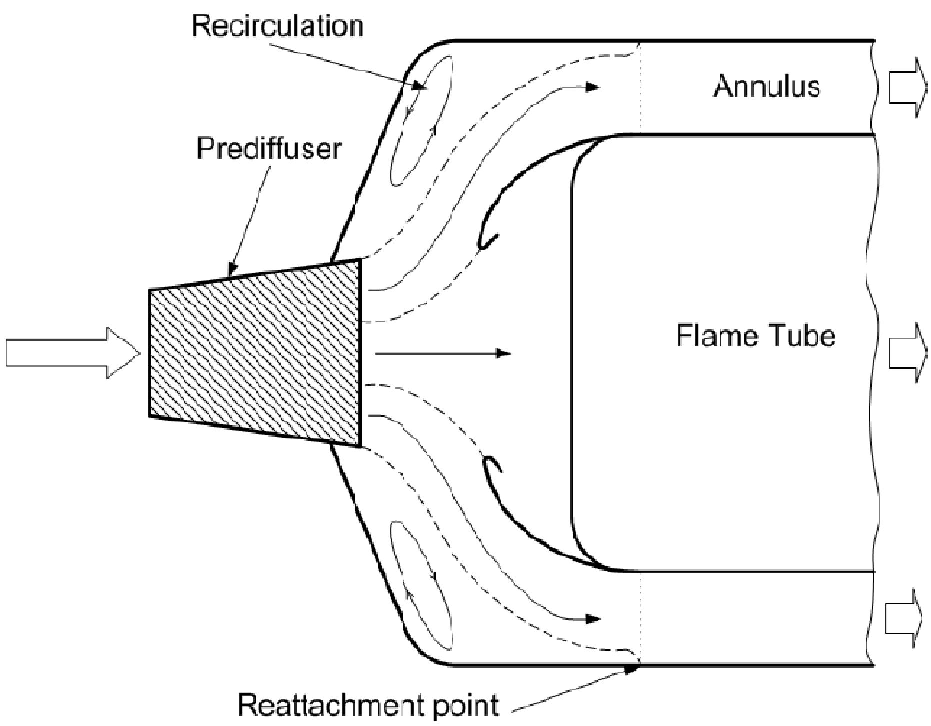


Fig 2.7: A Dump type diffuser*

2.) Faired diffuser

In this type of the diffuser, instead of dumping the flow at a step, two passages are made to direct the flow (Refer to Fig 2.8).

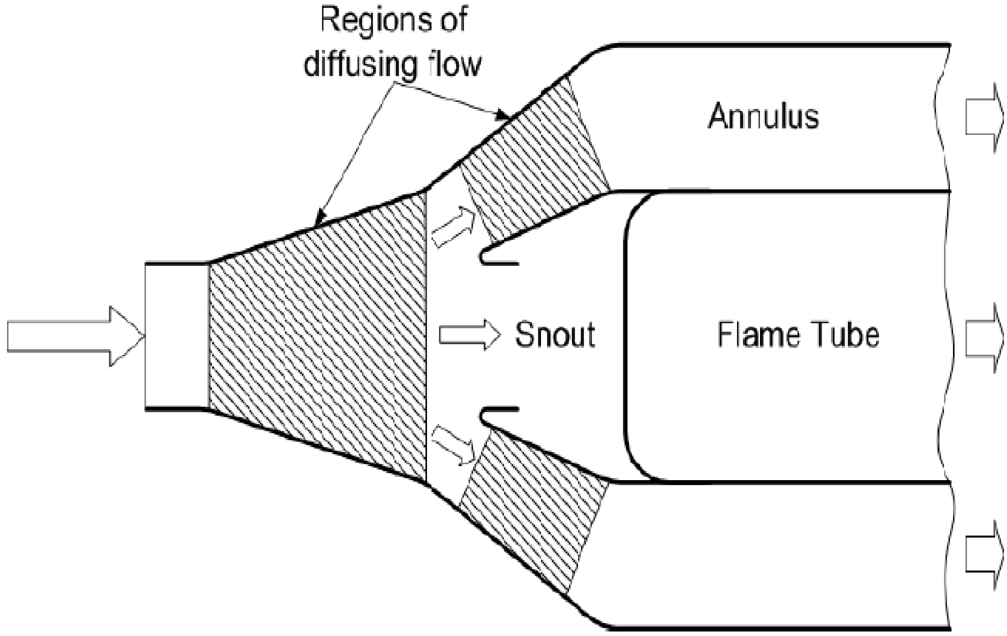


Fig 2.8: A faired type diffuser

**** The central diffuser passage discharges the remaining air into the dome region, which provides air for atomization and dome cooling. This representation is for an aircraft's annular flame tube.***

b.) The zones of combustion in the flame tube.

1.) The primary zone:

The main function of the primary zone is to anchor the flame and provide sufficient time, temperature, and turbulence to achieve essentially complete combustion of the incoming fuel–air mixture. The importance of the primary-zone airflow pattern to the attainment of these goals cannot be overstated. Many different types of flow patterns are employed, but one feature that is common to all is the creation of a toroidal flow reversal that entrains and recirculates a portion of the hot combustion gases to provide continuous ignition to the incoming air and fuel.

Some combustors use air swirlers to create the toroidal flow pattern, whereas others have no swirler and rely solely on air injected through holes drilled in the liner wall (The combustor used in the current project falls in the latter category due to the fact that we are using a gaseous fuel.). Both methods are capable of generating flow recirculation in the primary zone.

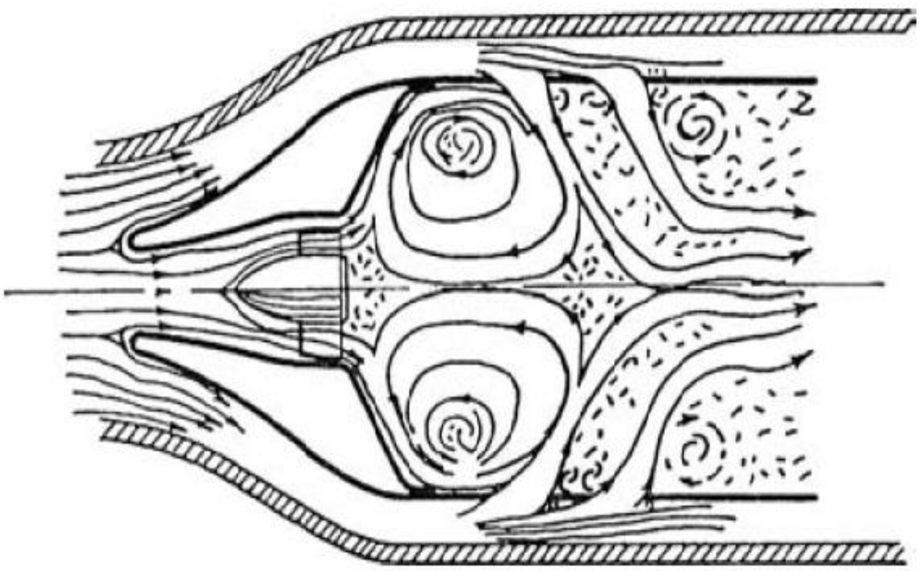


Fig 2.9: The primary recirculation zone

Both swirling air and primary air jets are used to produce the desired flow reversal. Although each mode of air injection is capable of achieving flow recirculation in its own right, but if both are used, and if a proper choice is made of swirl vane angle and the size, number, and axial location of the primary air holes, then the two separate flow recirculations created by the two separate modes of air injection will merge and blend in such a manner that each one complements and strengthens the other. Fig. 2.9 shows the air recirculation patterns as a result of both primary hole jet recirculation and the swirling air.

Since our combustor was going to use a gaseous fuel, it was decided that we would rely only on the primary air jet recirculation.

2.) The secondary/intermediate zone

If the primary-zone temperature is higher than around 2000 K, dissociation reactions will result in the appearance of significant concentrations of carbon monoxide (CO) and hydrogen (H_2) in the efflux gases. Should these gases pass directly to the dilution zone and be rapidly cooled by the addition of massive amounts of air, the gas composition would be "frozen," and CO, which is both a pollutant and a source of combustion inefficiency, would be discharged from the combustor unburned. To get rid of these species, an intermediate combustion zone is provided.
In the intermediate zone, the temperature is dropped to an intermediate level by the addition of small amounts of air which encourages the burnout of soot and allows the combustion of CO and any other unburned hydrocarbons (UHC) to proceed to completion.

3.) The tertiary/dilution zone

The properties of the turbine blade materials put an upper limit to the maximum turbine inlet temperatures, as a result of which hot gases coming from the secondary zone have to be cooled before they enter the turbine.
This is done by providing additional holes after the secondary zone. Tertiary zone holes can allow 30 to 50% of the total mass flow of the air coming from the compressor. The size and shape of these holes are selected to optimize the penetration of the air jets and their subsequent mixing with the main stream.
The length of the tertiary zone is selected in order to promote mixing and give a uniform temperature profile at the turbine inlet. In practice, however, it is found that mixedness initially improves greatly with an increase in mixing length and thereafter at a progressively slower rate. This is why the length/diameter ratios of dilution zones all tend to lie in a narrow range between 1.5 and 1.8.

The locations of the three main zones described above, in relation to the various combustor components and the air admission holes, are shown in Fig. 2.10.

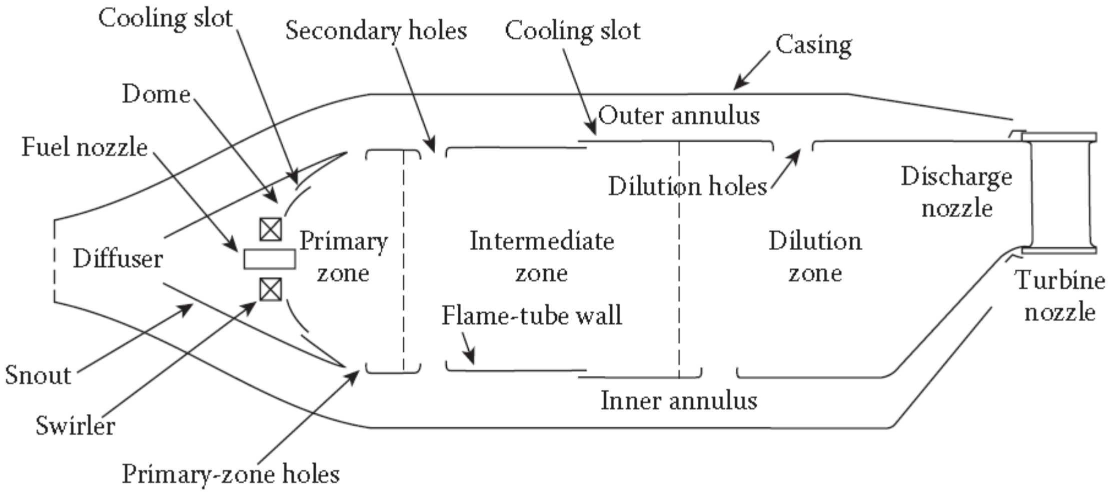


Fig. 2.10: Various components of a conventional combustor

2.2.4 Combustor Aerodynamics

Aerodynamic processes play a vital role in the design and performance of gas turbine combustion systems. In the diffuser and annulus, the main objectives are to reduce the flow velocity and distribute the air in prescribed amounts to all combustor zones, while maintaining uniform flow conditions with no flow recirculation. Within the combustion liner itself, attention is focused on the attainment of large-scale flow recirculation for flame stabilization, effective dilution of the combustion products, and efficient use of cooling air along the liner walls.

Mixing processes are of paramount importance in the combustion and dilution zones. In the primary zone, good mixing is essential for high burning rates and to minimize soot and nitric oxide formation, whereas the attainment of a satisfactory temperature distribution in the exhaust gases is very dependent on the degree of mixing between air and combustion products in the dilution zone.

A primary objective of combustor design is to achieve satisfactory mixing within the liner and a stable flow pattern throughout the entire combustor, with no parasitic losses and with minimal length and pressure loss.

1.) Pressure loss across the combustor

Pressure loss in the combustor is a sum of two losses:
a) Pressure drop in the diffuser and b) Pressure drop across the liner

$$\Delta \boldsymbol{P}_{3-4} = \Delta \boldsymbol{P}_{diffuser} + \Delta \boldsymbol{P}_{liner} \qquad \textbf{(2.1)}$$

It is important to keep ΔP_{diff} to a minimum, since any pressure loss incurred in the diffuser makes no contribution to combustion. In practice, there is little the combustion engineer can do to minimize diffuser pressure loss other than observe the recognized principles of diffuser design.
It is equally important to minimize the liner pressure-loss factor; although in this case there is an important difference in that a **high liner pressure drop is beneficial to the combustion and dilution processes. It gives high injection air velocities, steep penetration angles, and a high level of turbulence, which promotes good mixing and can result in a shorter liner.**

Two dimensionless properties are also important in combustor design; one is the ratio of the total pressure drop across the combustor to the inlet total pressure ($\Delta P_{3-4}/\Delta P_3$), and the other is the ratio of the total pressure drop across the combustor to the reference dynamic pressure ($\Delta P_{3-4}/q_{ref}$), reference values have been defined in the appendix. The two parameters are related by the equation

$$\frac{\Delta \boldsymbol{P}_{3-4}}{\boldsymbol{P}_3} = \frac{\Delta \boldsymbol{P}_{3-4}}{\boldsymbol{q}_{ref}} \cdot \frac{\boldsymbol{R}}{\boldsymbol{2}} \cdot \left(\frac{\dot{\boldsymbol{m}}_3 \boldsymbol{T}_3^{.5}}{\boldsymbol{A}_{ref}\boldsymbol{P}_3}\right)^2 \qquad \textbf{(2.2)}$$

From this equation we can find the reference area of the casing of the combustor, A_{ref} and using the equation

$$\frac{\Delta \boldsymbol{P}_{liner}}{\boldsymbol{q}_{ref}} = \left(\frac{\boldsymbol{A}_{ref}}{\boldsymbol{A}_{h,eff}}\right)^2 \qquad \textbf{(2.3)}$$

We can find the effective area of all the holes of the liner.

Also, the effective area of the holes is related to the actual hole area as

$$A_{h,eff} = \sum_{i=1}^{i=n} C_{D,i} A_{h,i} \qquad (2.4)$$

Where $C_{D,i}$ and $A_{h,i}$ are the discharge coefficient and actual area of the *i*th hole

Table 2.1 gives some pressure loss values for various types of combustors

Type of chamber	$\frac{\Delta P_{3-4}}{P_3}$	$\frac{\Delta P_{3-4}}{q_{ref}}$
Tubular	.07	37
Tuboannular	.06	28
Annular	.06	20

Table 2.1

2.) Flow in the annulus region

Flow conditions in the annulus have a substantial effect on the airflow pattern within the liner and influence the level and distribution of liner wall temperatures.

The mean velocity in the annulus is governed by the combustor reference velocity and the ratio of liner area to casing area. Although a high annulus velocity augments the convective cooling of the liner walls, low velocities are generally preferred because they provide the following benefits:

- Minimum variation in annulus velocity and static pressure, ensuring that all the liner holes in the same row pass the same airflow
- Higher hole discharge coefficients
- Steeper angles of jet penetration
- Lower skin-friction loss
- Lower “sudden-expansion” losses downstream of liner holes

3.) Flow through the liner holes

The flow through a liner hole depends not only on its size and the pressure drop across it, but also on the duct geometry and flow conditions in the vicinity of the hole, which can strongly influence its effective flow area. It has been observed that some deflection of the flow streamlines occurs in the vicinity of the liner holes, to an extent that depends on the geometry of the system, the approach velocity, and the pressure drop across the liner.
Thus, in practice, the coefficient of discharge of liner holes is affected by

- Type (e.g., plain or plunged
- Shape (e.g., circular or rectangular)
- Ratio of hole spacing to annulus height
- Liner pressure drop
- Distribution of static pressure around the hole inside the liner
- Presence of swirl in the upstream flow
- Local annulus air velocity

The curves in Fig. 2.11 and 2.12 depict how the shape and type of holes influence the discharge coefficient at a given pressure drop coefficient (K),which is the ratio of the dynamic pressure of the jet to the dynamic pressure of the flow in the annulus, therefore lower the flow velocity in the annulus, higher the value of K.

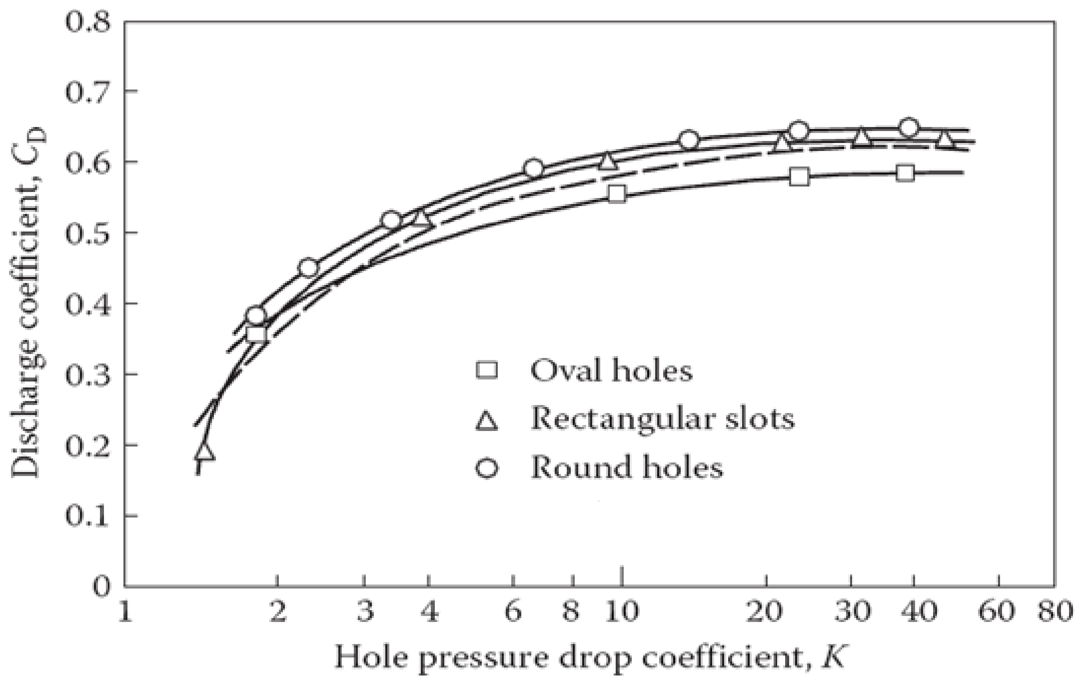


Fig 2.11: Influence of hole shape on the discharge coefficient

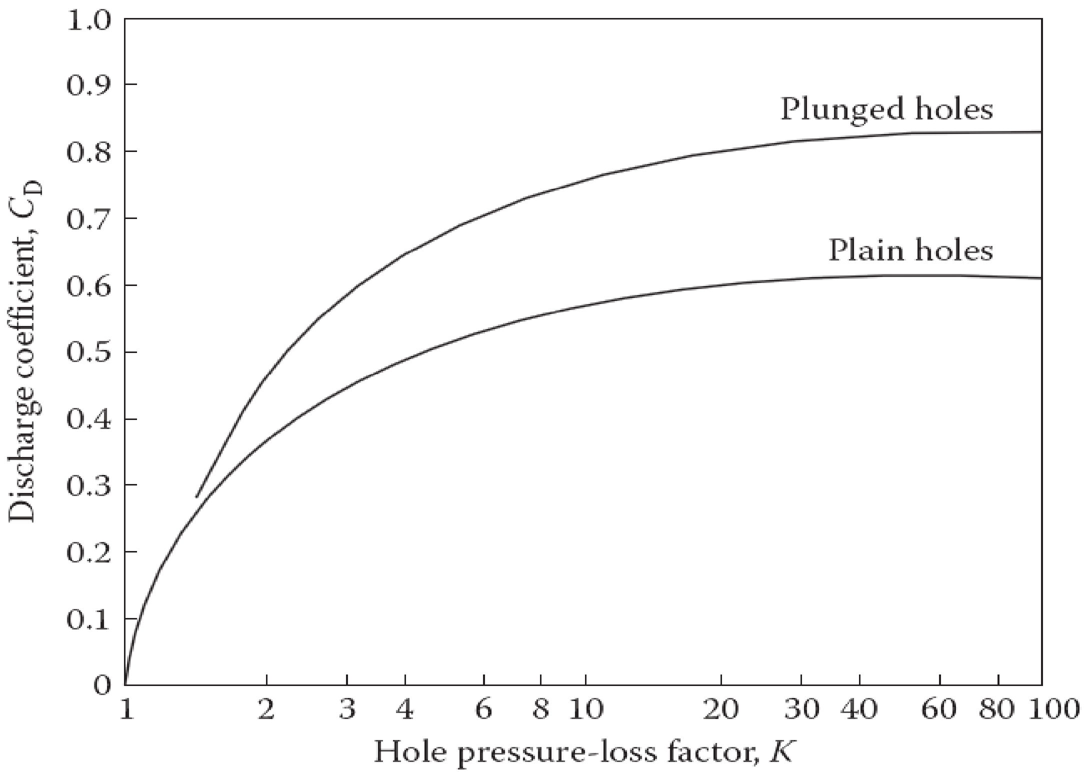


Fig 2.12: Influence of hole type on the discharge coefficient (see appendix for more information on hole types)

The initial jet angle, which is defined as the angle made by the centre of a jet with the axis of the flame tube is also affected by the shape of the hole.
Fig. 2.13 shows the initial jet angle

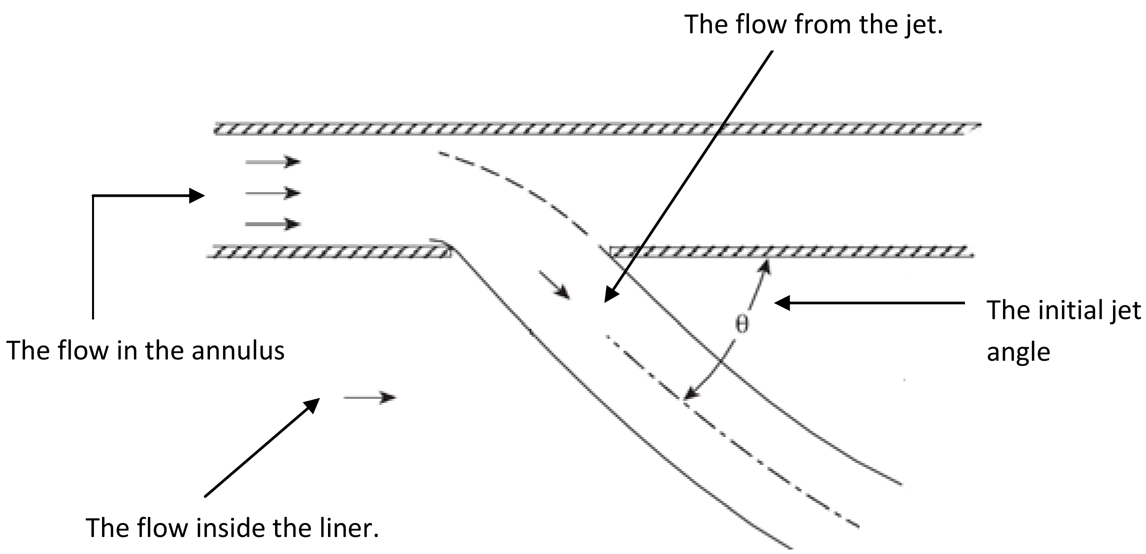


Fig 2.13: The initial jet angle and various flows

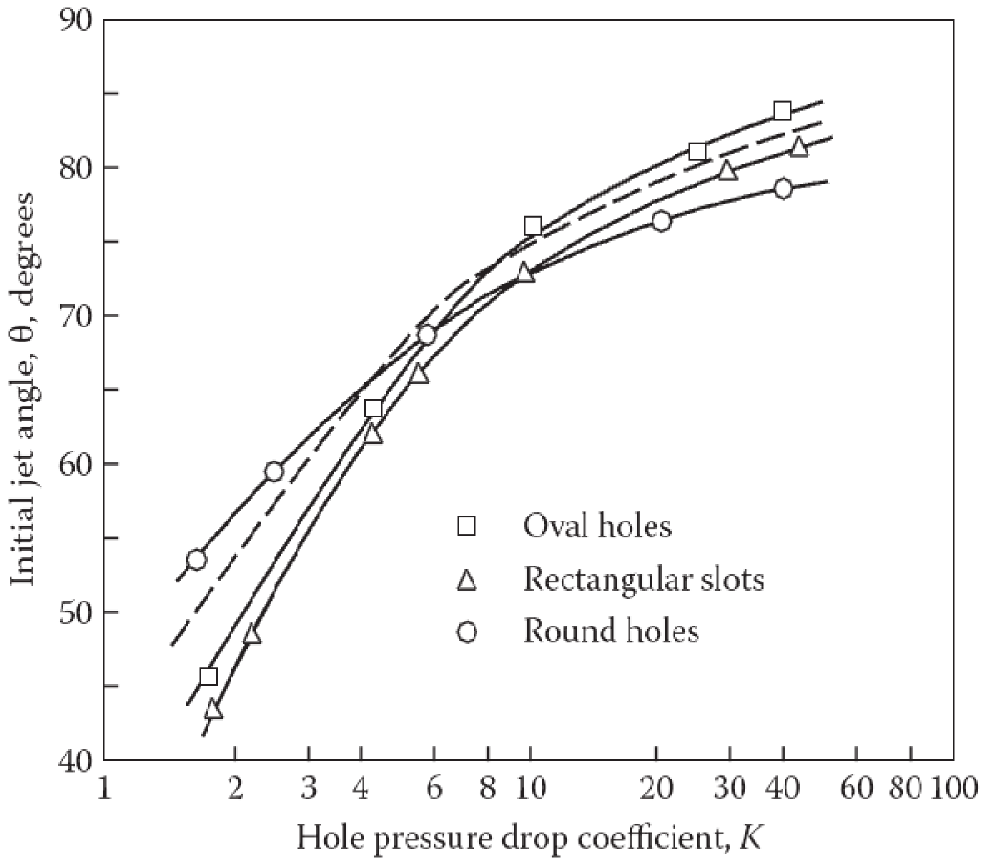


Fig 2.14: The influence of shape on the initial jet angle

It is worth noting from Fig.2.14 that steeper jet angles are formed at higher pressure drop coefficients, suggesting the importance of lower annulus speeds.

4.) Stirring and mixing

In the combustion chamber of a gas turbine, it is essential that the fuel is mixed with the air from the primary and the secondary zones. Apart from this the exhaust going to the turbine inlet should be properly cooled, which requires a higher jet penetration than the primary and secondary zones.

Since the current project is based on a gaseous fuel, the mixing is dependent upon the recirculation zone formed by the primary hole airflow.

Fig. 2.15 shows the motion of air as it moves in a region without any crossed flow i.e. there is no moving air, only static air is present.

In this case as the air moves into the hole a vena contracta is formed and due to the occurrence of shearing, vortices similar to smoke ring structures are formed, the shearing region increases as the fluid moves downstream of the hole.

The vortices tend to engulf the particles of the liner flow and this is termed as "macro-mixing". Another phenomenon that occurs is the "micro-mixing", which occurs considerably downstream of the hole and tends to destroy the interface of the jet and liner fluid by the means of molecular mixing.

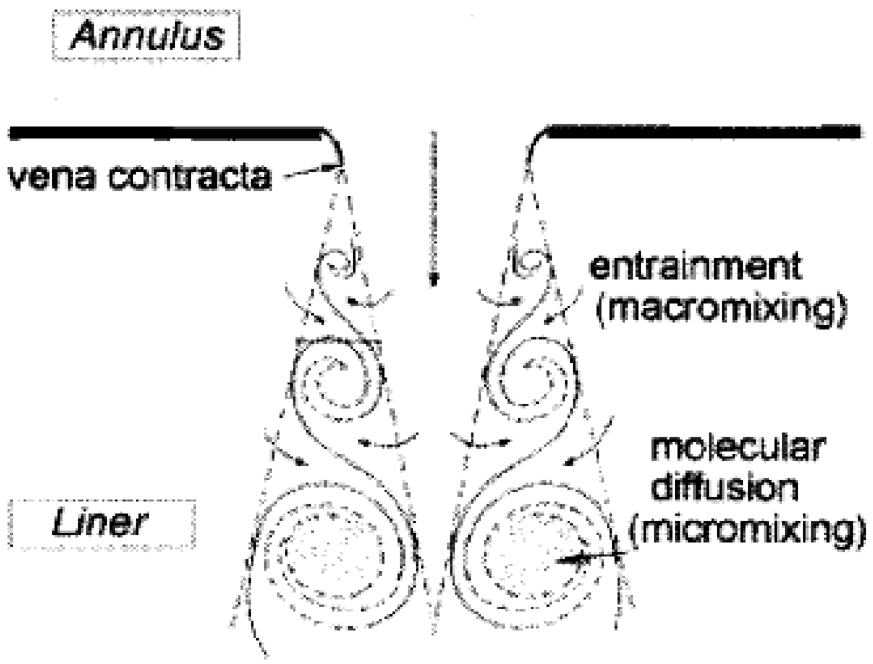


Fig. 2.15: Zones of mixing of a jet in a non crossed flow situation.

Fig. 2.16 shows how the diameter of the jet affects the penetration of the jet in a non crossed flow situation and also justifies the use of larger diameter holes for the tertiary zone.

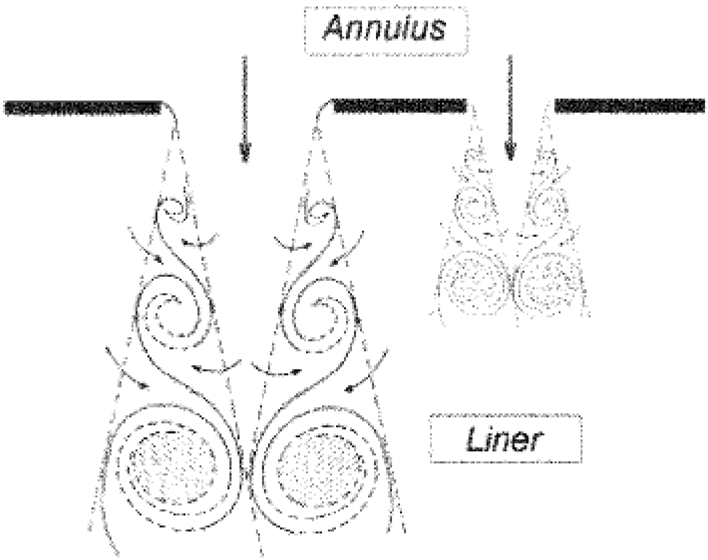


Fig 2.16: Two round jets with the same velocity but diameters differing by a ratio of 2.

In actual practice, the jets from the holes in the liner encounter a crossed flow i.e. a flow along the axis of the liner. The liner flow sees the vertical jet flow as an obstruction and exerts an aerodynamic drag on it, the jet being a compliant structure bends horizontally under the force and is washed downstream.

Fig. 2.17 shows a crossed flow situation. In this case, apart from the ring type vortices, horseshoe vortices with vertical vortex filaments in the wake region are formed.

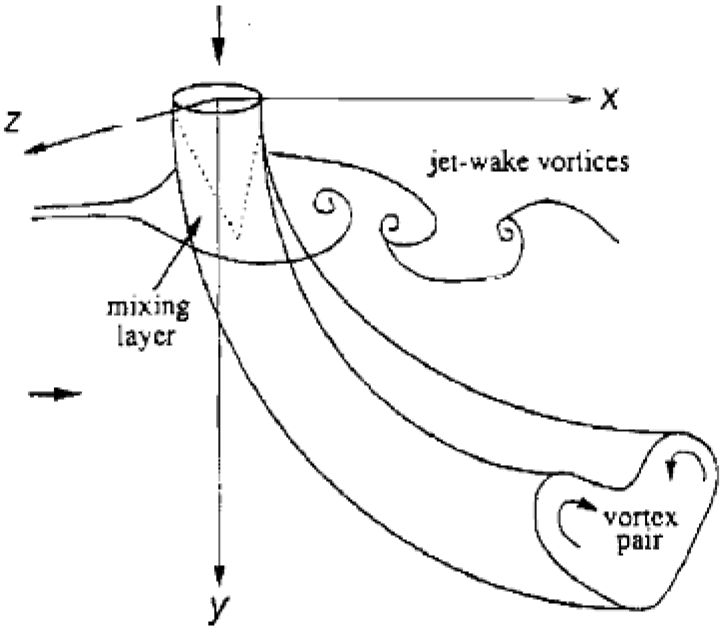


Fig. 2.17: Jet trajectory in case of a crossed stream.

We would like to reiterate that the cases considered here are highly simplified, inside the liner of the combustor several jets are mixing together in a complex fashion.

In general, the rate of mixing is influenced by the following factors:

- The size and the shape of the hole through which the jet issues
- The initial angle of jet penetration
- The momentum ratio of the jet and the liner flow
- The presence of other jets, both adjacent and opposed
- The length of the jet mixing path
- The proximity of walls
- The inlet velocity and temperature profiles of the jet and the hot gases.

Proper mixing is essential to have a uniform temperature profile of the gases going to the turbine.

2.2.5 Combustion chemistry

Combustion is an exothermic reaction in which fuel and oxidizer, air in most cases, react to release heat. Stoichiometric combustion occurs when the exact quantity of oxidizer required to completely burn the fuel is supplied to the combustion reaction. The mixture is said to be lean if more than the stoichiometric quantity of oxidizer is supplied and rich if the amount of oxidizer is less than stoichiometric. The ratio of this stoichiometric quantity of air to fuel mass is determined from an atom balance. The stoichiometric relation for the combustion of a hydrocarbon fuel given by in air is

$$\boldsymbol{C_\alpha H_\beta + a(O_2 + 3.76\ N_2) \longrightarrow \alpha\ CO_2 + \frac{\beta}{2}\ H_2O\ + 3.76\ a\ N_2} \qquad \textbf{(2.5)}$$

Where a = $\alpha + \frac{\beta}{4}$. Thus the stoichiometric air fuel ratio

$$\boldsymbol{\left(\frac{Air}{Fuel}\right)_{stoichionetric} = \frac{4.76a\ MW_{air}}{1\ MW_{fuel}}} \qquad \textbf{(2.6)}$$

Where MW_{air} and MW_{fuel} are the molecular weights of fuel and air respectively. The equivalence ratio Φ is used to quantify whether the mixture is rich, lean, or stoichiometric. It is defined as

$$\Phi = \frac{\boldsymbol{\left(\frac{Air}{Fuel}\right)_{Stoichionetric}}}{\boldsymbol{\left(\frac{Air}{Fuel}\right)_{Actual}}} \qquad \textbf{(2.7)}$$

The equivalence ratio is unity for a stoichiometric mixture. Values greater than unity indicate that the mixture is fuel rich, whereas values less than unity indicate a lean mixture.

Equation 2.5 is an idealization; the products of combustion differ at high temperature where hydrocarbon dissociation products and nitrogen oxides are also present.

Combustion performance

Gas turbine combustors must operate with stability over a wide range of operating conditions (temperature, pressure, velocity, etc.). They must also be capable of igniting with ease and re-initiating combustion during operation in the event of a flameout.

Here we discuss some parameters describing combustion performance.

1.) Combustion efficiency

Combustion efficiency is dependent on the rate at which heat is released by the chemical reaction. It is the ratio of the theoretical mass flow rate of fuel divided by the actual value necessary to provide the same heat release. The combustion efficiency is defined as,

$$\eta_{combustion} = \frac{\dot{m}_{theoretical}}{\dot{m}_{actual}} \quad \textbf{(2.8)}$$

Where $\dot{m}$ is the mass flow rate of fuel.

The efficiency is governed by the time it takes to complete evaporation, mixing and the chemical reaction.

2.) Combustion Stability

Stability of a gas turbine combustor encompasses an operation without blowing out and easy ignition when blowout occurs
Stability is measured by running tests at different operating conditions until flame extinction is observed. The resulting range of stable operation is bounded by converging rich and weak limits, as illustrated in Figure 2.18. The parameter along the horizontal axis, the combustor loading I, reflects the influence of the most important parameters on combustion stability: air flow $\dot{m}_{air}$ combustor volume V and pressure P. It is defined as

$$I = \frac{\dot{m}_{air}}{VP^n} \quad \textbf{(2.9)}$$

Where the exponent n is determined experimentally, is typically chosen to be 2.

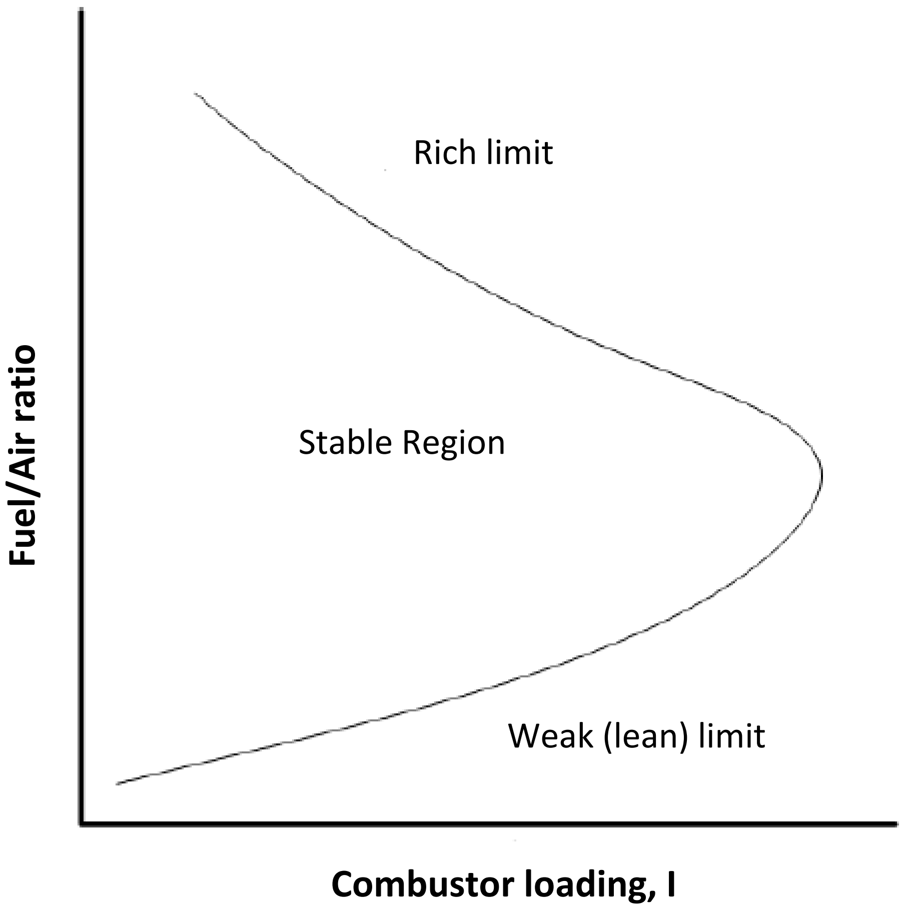


Fig. 2.18: A typical stability loop.

3.) Ignition

Design of a gas turbine combustor requires the selection of an ignition source capable of supplying a sufficient amount of energy to initiate combustion. To substantiate ignition, sufficient energy to create a self-propagating volume or kernel of hot gas is required. The size of the smallest volume which satisfies this criterion is termed the quenching volume. The minimum ignition energy is the amount of energy required to produce a kernel of this critical size, figure 2.19 depicts this concept

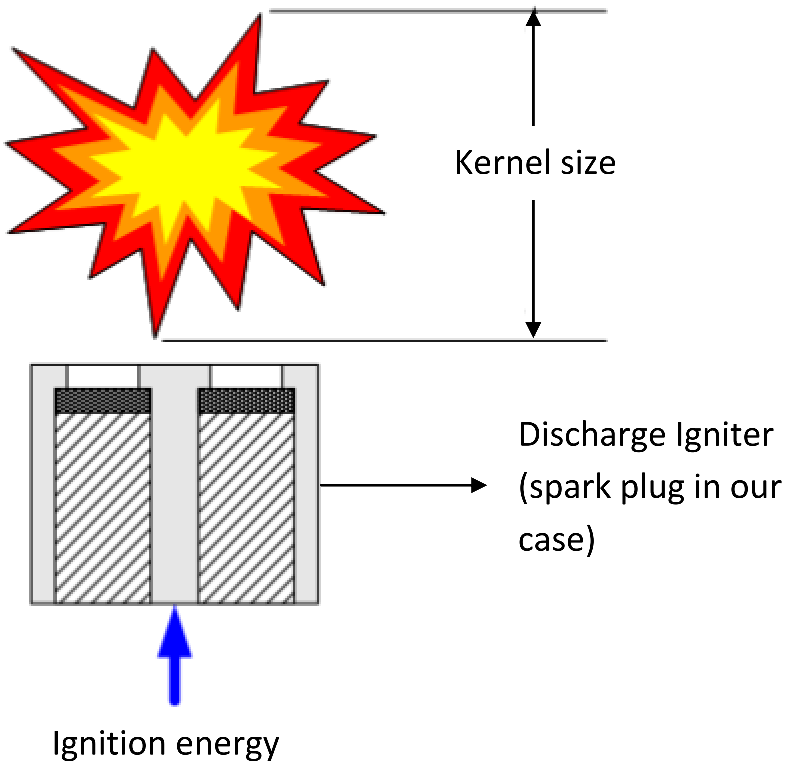


Fig 2.19: An ignition kernel

Our project uses a gaseous fuel; hence the intricacies of using a liquid fuel and the droplet processes have not been included in this review.

Design decisions related to emissions will be discussed in the next chapter.

2.3 The automotive turbocharger and its components

1.) The turbocharger's utility in an automobile and its correlation with an aircraft engine

A turbocharger consists of a compressor and turbine operating on a single common shaft. These are often designed for use on automobile internal combustion engines. When installed, the hot exhaust gases exiting the cylinders pass through the turbine side of the turbocharger, spinning the turbine blades, and thus the shaft. The shaft then transmits power to drive the compressor. The

exhaust gases then continue out to the exhaust manifold and continue as usual. As the compressor spins, it raises the pressure of the incoming air from the air intake. The high pressure air is often directed through a charged air cooler (also known as an intercooler) to further raise the density of the air. The high density air is then ducted into the cylinder and combustion occurs as normal. The larger mass of air, however, allows for more fuel to be burned in the same volume, and thus more power to be extracted by the piston during the power stroke, transmitting more power to the crankshaft and eventually the wheels.
The cycle of an automotive turbocharger follows closely to that of an aircraft turbine engine and thus allows for the correlation between an aircraft engine and that of the automotive based turbine engine to be created.

Fig. 2.20 shows the cycle in which the turbocharger works.

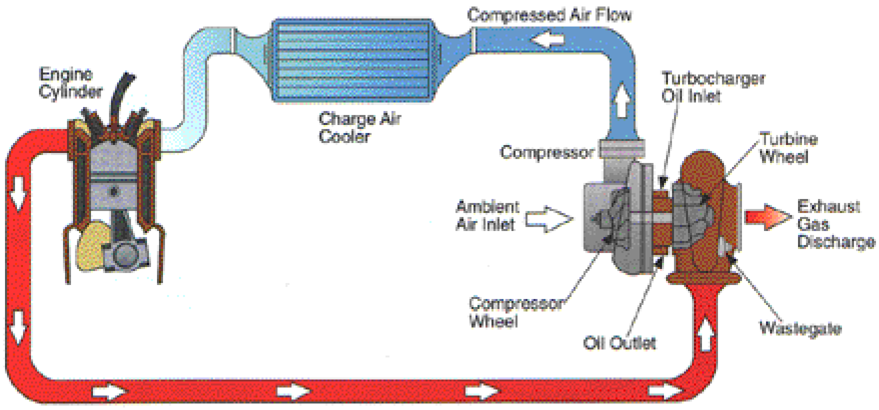


Fig.2.20: Diagram of the function of a turbocharger on a piston-driven internal combustion engine

Components of an automotive turbocharger

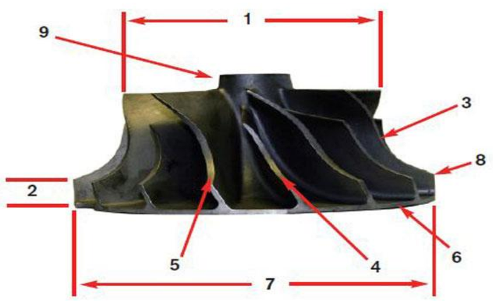
1.) The compressor

The current project uses a radial compressor which uses the circular motion of the compressor wheel to raise the pressure of the air which it intakes.

There are two primary components of the radial compressor: the compressor wheel and the compressor cover. Within these components there are many critical design types and specific features such as the diffuser, a critical feature that is typically designed into the compressor cover.

a) The compressor wheel

Fig. 2.21 shows a compressor wheel and its key components (it is a backward swept blade compressor, similar to the one being used in the current project) and Fig 2.22 shows a compressor wheel cut in half. **It is generally made of out an aluminum alloy.**



1) Inducer diameter

(2) Tip height or tip width

(3) Wheel contour

(4) Splitter blade

(5) Full blade

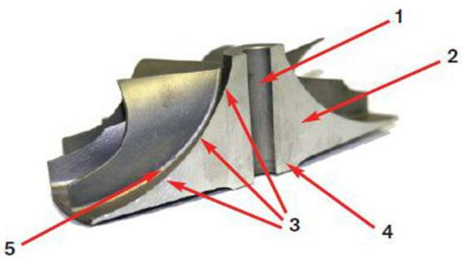
(6) Backwall

(7) Wheel diameter, tip diameter, or exducer diameter,

(8) tip, impeller, or exducer,

(9) Nose.

Fig. 2.21: The compressor wheel and a table of key areas of design consideration.



This compressor wheel has been cut in half.

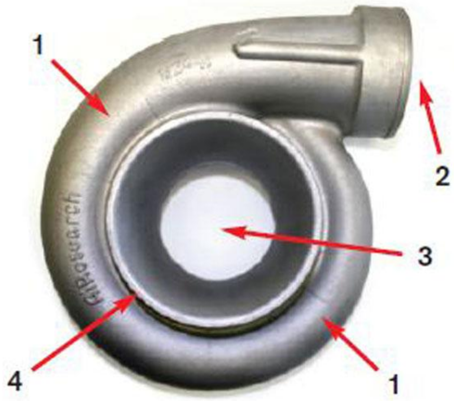
The shaft bore (1) typically runs all the way through the wheel, the compressor wheel hub (2) supports the blades and its shape forms the wheel floor (3) that turns the inlet airflow 90 degrees to make it a radial-type compressor. Some wheels have an extended backwall (4), which strengthens the wheel at its highest point of stress for improved durability. The root of the blades (5) will have a small fillet to support the stresses of compression.

Fig. 2.22: A compressor wheel cut in half and its salient features

b) The compressor cover

The compressor cover, like the compressor wheel, is typically made from an aluminum alloy.

The diffuser portion of the compressor is a critical portion of the overall compressor design. The diffuser is an optimized path for the air as it leaves the compressor wheel on its way to the compressor cover volute. The function of the diffuser is to turn the rapidly compressed and high-speed air leaving the compressor wheel into static pressure as it fills the volute. Most diffusers used in automotive applications are the vaneless type (as in our project) formed by parallel walls between the compressor cover and the bearing housing face. The diffuser has to have a minimum diameter in order to have an effect upon the highly turbulent air exiting the compressor wheel. This is the reason the compressor cover outside dimension is typically so much larger than the outside diameter of the compressor wheel there has to be sufficient diffuser area. Vane-type diffusers are extremely efficient but have flow range restrictions. Refer to fig 2.23.



The compressor cover has four areas that can be easily seen from the outside:

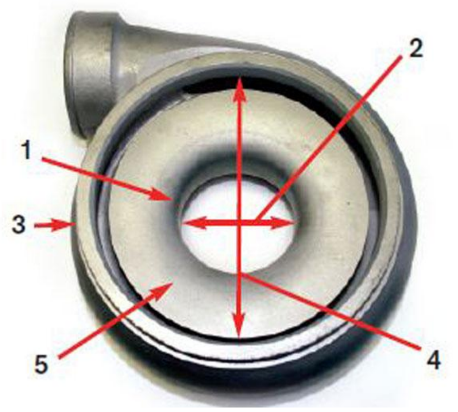
(1) The volute

(2) The discharge

(3) The eye or inducer

(4) The inlet connection diameter.

(a)



From an inside view the following areas can be seen:

(1) The compressor cover contour

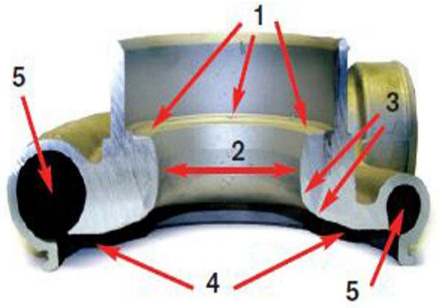
(2) The cover inducer diameter

(3) The bearing housing connection

(4) The bearing housing pilot diameter

(5) The parallel wall diffuser face.

(b)



A compressor cover cut in half reveals a better view of its design features machined into the casting. Air flows into the compressor inlet (1) where a bell-mouth flow nozzle is frequently formed to aid in the smooth air transition into the wheel. The inducer diameter (2) forms the limiting flow of the compressor. The compressor contour (3) cut into the cover matches the contour cut in the compressor wheel, but provides a typical running clearance of anywhere between 0.009–0.012 inch, depending upon model and use, but in some models the contour can be as much as 0.020 inch. The diffuser face (4) forms one side of the parallel wall diffuser. The bearing housing flange, or seal plate, forms the other wall once the turbo is assembled. The volute section (5) gathers the air as it exits the compressor wheel. The cross-sectional area becomes larger as the volute approaches the compressor discharge because it gathers more air and the increased size slows the air helping to convert it from high-speed flow to static pressure.

(c)

A vaned diffuser shown inside the compressor cover.

(d)

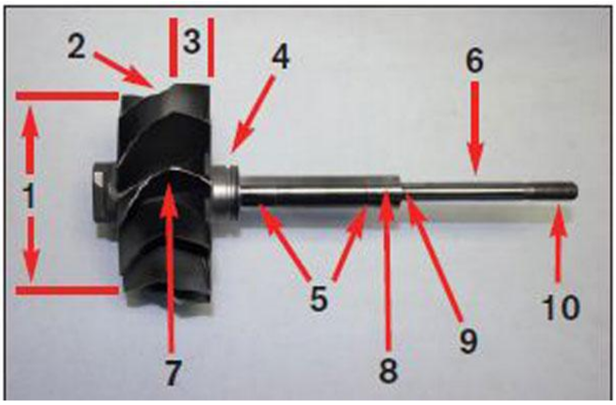
Fig. 2.23: The compressor cover and its features.

(c) The turbine wheel and the shaft assembly

The turbine wheel and shaft assembly is commonly called the turbine wheel. It is the most critical and costly of all turbocharger parts. The turbine wheel is made from INCONEL, an austenitic nickel chromium iron alloy or titanium alloys.

INCONEL made turbines have a maximum allowable turbine inlet temperature of about 800°C.

Fig 2.24 show the vital parts of a turbine assembly.



(1) The exducer diameter,

(2) The turbine wheel contour,

(4) The seal ring, or piston ring groove,

(5) The bearing journal or shaft bearing surface,

(6) The stub shaft,

(7) the turbine wheel,

(8) The turbine shaft,

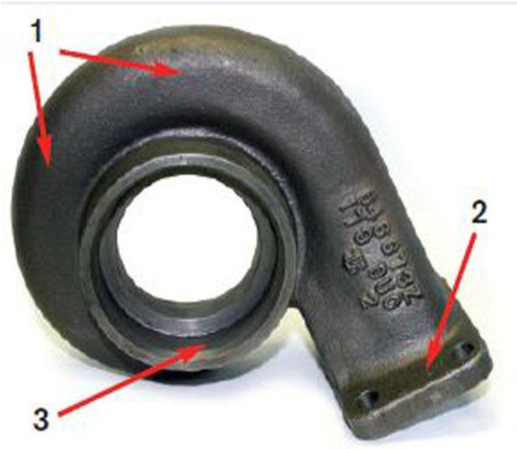
(9) The shoulder, and

(10) The rolled threads used to clamp the compressor wheel onto the shaft.

Fig. 2.24 Key areas of a turbine assembly

(d) The turbine housing

The exhaust from the engine or in our case from the combustion chamber enters the turbine housing and is then directed radially to run the turbine. Fig.2.25 shows the key areas of a turbine housing. **It is typically made of cast iron.**



(1) The turbine volute

(2) The turbine foot

(3) The outlet connection

(a)

(1) The turbine housing contour

(2) Bearing housing diameter

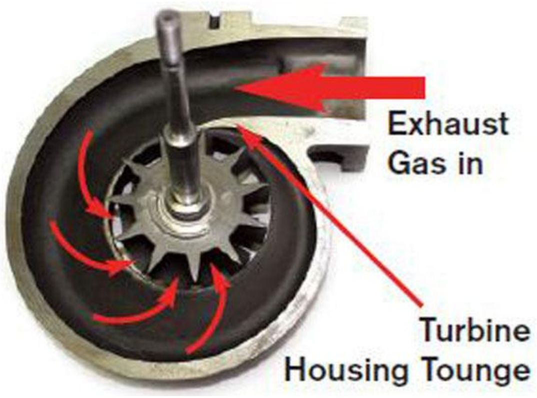
(3) Turbine throat area

(4) Bearing housing connection

(b)

Exhaust inlet with a central division

(c)



Radial entry of hot gases in the turbine

(d)

Fig. 2.25: The turbine housing

(e) The bearing system

The turbine and compressor are machines supported by a common bearing system. The bearing system has many considerations that go into its design. The bearing system must allow for rotational speeds in excess of 100,000 rpm; it must also withstand shaft motion gyrations imparted onto the turbine shaft by the engine's pulses at or near peak torque, handle radial and axial or thrust loads and withstand high temperatures. The most common type of bearing system is the three-piece bronze bearing system that includes two journal bearings and one thrust bearing. The journal bearings are typically full floating in oil and have specific clearances between the turbine shaft and bearing inside diameter, as well as the bearing outside diameter and the bearing housing bore. The two journal bearings typically rotate in relation to the bearing housing while the turbine shaft rotates in relation to the bearings. The third piece of the three-piece bronze bearing system is the thrust bearing. The thrust bearing has two bearing lands, one on each side. The oil enters from the center via a small oil galley and flows outwardly between the bearing surface and the thrust rings made from hardened steel held to very close tolerances.

Fig. 2.26 shows the oil passages and the journal and thrust bearings. The bearing has been indicated by an arrow and the channels over the bearings are the oil passage (not arrowed).

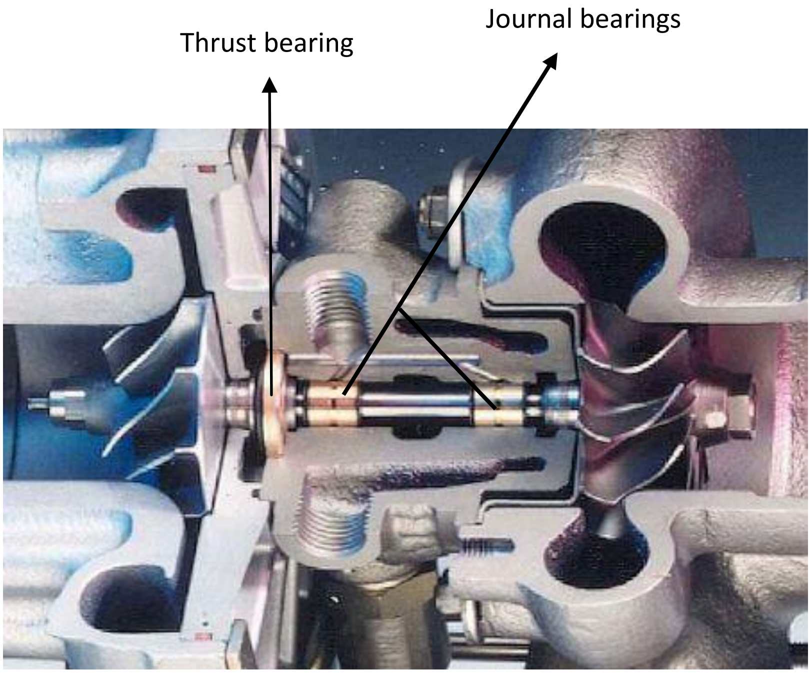


Fig. 2.26: The bearing assembly of a turbocharger

Other components of a turbo include the bearing housing and the seals which prevent the air and the exhaust gases from entering the centre housing.

2.4 The gerotor pump

As far as the present project is concerned, we were looking for a pump that

- **Could work with lubrication oil and produce a pressure of 60-100 psi at a flow rate of 5-10 liters/minute.**
- **Would work without any sort of leakage.**

Most of the locally available pumps were engine lubricant pumps which are supposed to work inside the engine block, submerged in an oil sump and hence presented a leakage problem.

For our utility, we wanted our pump to work well without submerging it into an oil sump (which would have presented us with several practical problems related to the coupling of the motor and the pump.). The gerotor pump provided a valid solution and fulfilled the flow rate and the pressure requirements.

A gerotor pump is a positive displacement pump. A gerotor unit consists of an inner and outer rotor. The inner rotor has N teeth, and the outer rotor has N+1 teeth with N defined as a natural number greater than 2. The inner rotor is located off-center and both rotors rotate. The geometry of the two rotors partitions the volume between them into N different dynamically-changing volumes. During the assembly's rotation cycle, each of these volumes changes continuously, so any given volume first increases, and then decreases. A volume increase creates a vacuum which leads to the suction of the oil. A subsequent volume decreases leads to the pressure rise. Refer to figure 2.27 and 2.28..

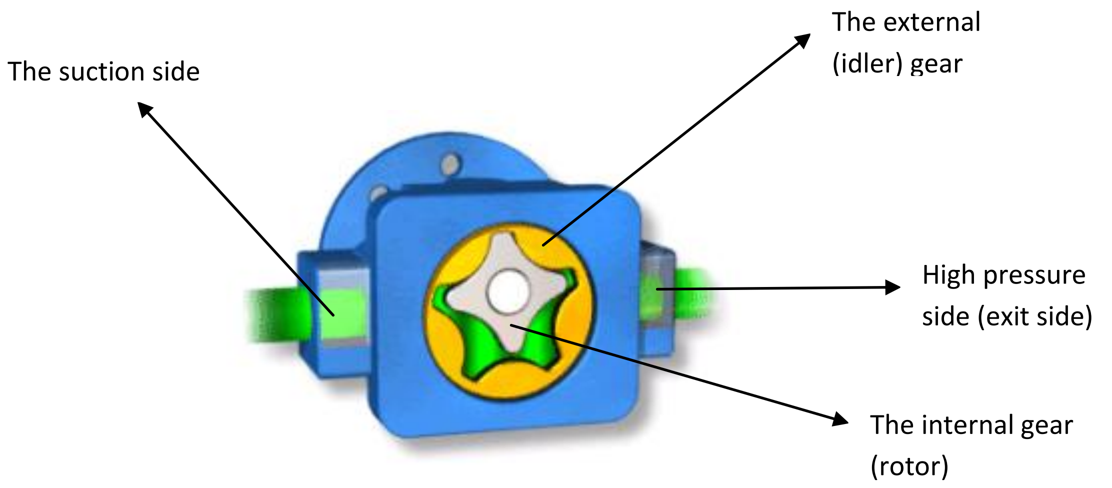


Fig.2.27: A gerotor pump and its components (assumed to be rotating in the counterclockwise direction)

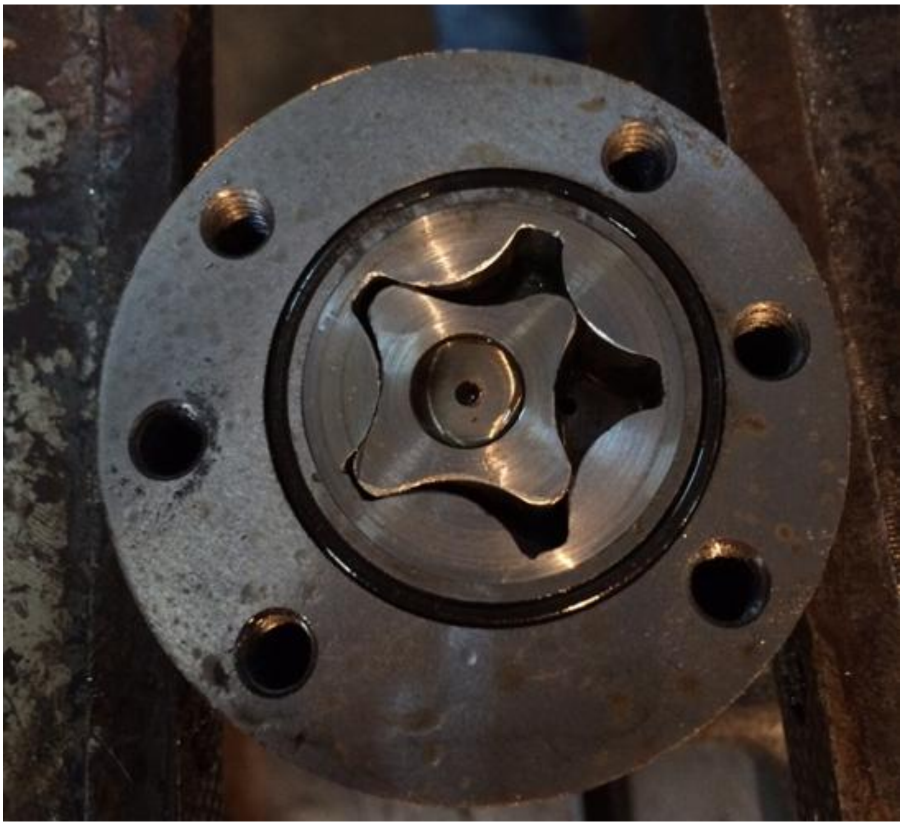


Fig.2.28: The internal gear system of the gerotor used

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CHAPTER 3

Designing process: Putting things in context of the project

This chapter looks at the design methodology we used before fabricating the gas turbine engine, this chapter examines the

- choice of the combustor type
- tentative design of the engine
- the turbocharger model used and the choice of the design point
- Combustion chamber design process; including solid modeling and flow simulations.
- Material selection justification

3.1 Choice of the combustor type

During the last few decades, several attempts have been done to develop microturbines with efficiency levels close to those of larger gas turbines. Particularly for microturbines with power rated below 100 kW, many developments have failed to obtain sufficient efficiency, reliability and cost effectiveness due the small-effects:

- **low Reynolds in the turbo machinery passages causing relatively high viscous losses,**
- **relatively high tip clearances due the manufacturing tolerances and bearing limitations,**
- **Large area-to-volume ratios resulting in high heat losses and inadvertent heat transfer to the compressor and high auxiliary system losses.**

Keeping in mind these problems, it was decided to design a micro gas turbine which made use of an off the shelf turbocharger, which would alleviate these problems as

- **Off-the-shelf automotive turbochargers already present sufficient efficiency for gas turbine cycles.**
- **The price is low due the large production volumes.**
- **The assembling of compressors and turbines of different models may be done in order to reach best efficiency**

The decision of using a turbocharger without any major modification inevitably affected the choice of the combustor type to be used. An annular combustor requires modifications in the diffuser section of the compressor and since the turbine casing cannot be used, nozzle guide vanes have to be designed for the turbine. Apart from this inlets for both compressor and the turbine have to be designed and fabricated.

These series of modifications would have led to a steep rise in the fabrication costs with a host of other problems that would have crept in, the biggest one of which would have been of the attainment of tight clearances.

In view of these issues, it was decided that **a tubular combustor would be used for the present project.**

3.2 The tentative design of the gas turbine

Fig. 3.1 shows the components of the gas turbine assembly as they were imagined in the context of the current project.

Of these components, the main component was the combustion chamber, which was to be designed and fabricated.

Apart from this, the turbocharger bearings were to be provided with lubrication oil at around 50-60 psi, for which a motor pump assembly was to be made (not shown here).

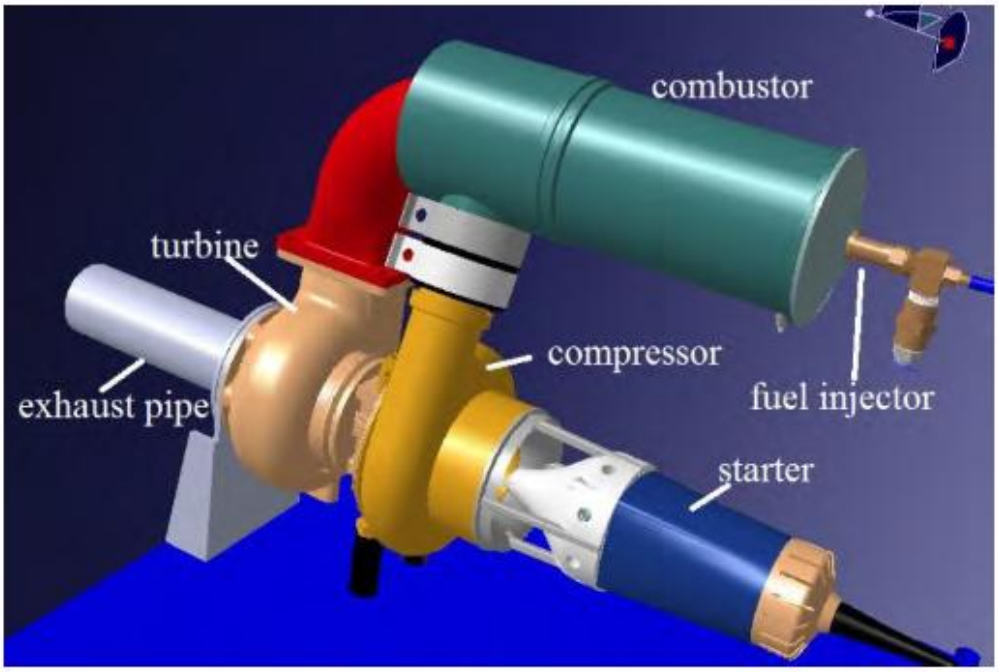


Fig. 3.1: The key components of the gas turbine assembly; the imagined assembly before any design or fabrication was done.

3.3 The turbocharger model used and the choice of the design point

A HOLSET HX-35 (non wastegated) was used and the combustion chamber was designed accordingly.

The choice was based on the ease of availability of the model and a sufficient mass flow.

The compressor map was used to choose the design point; it is shown in fig. 3.2.

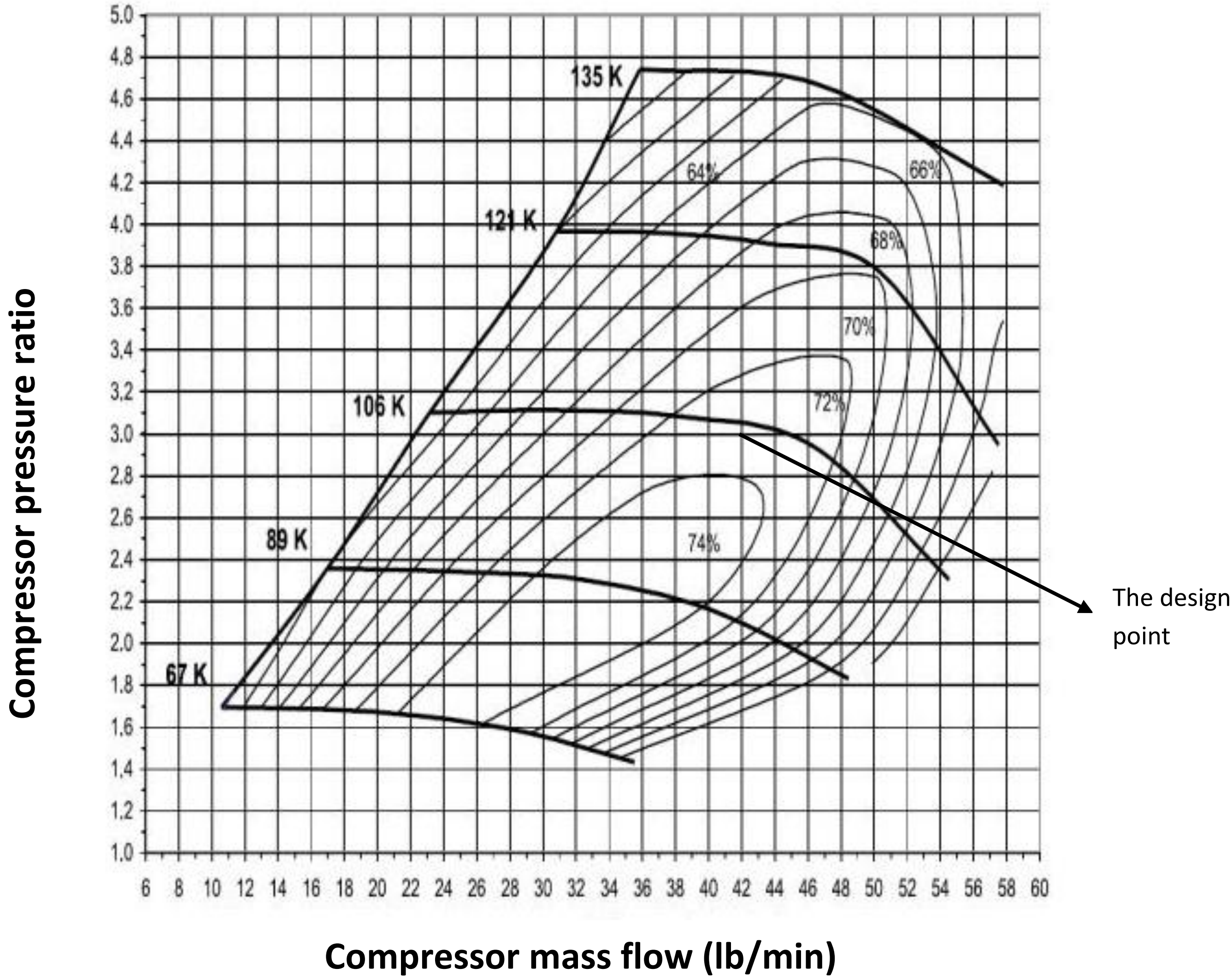


Fig. 3.2: Compressor map for HOLSET HX-35

The values found from the compressor map and by the use of some thermodynamic calculations (please refer to the appendix for the formulas used) are summarized in table 3.1. The maximum turbine inlet temperature is a manufacturer defined value

Parameter	Symbol	Value
Mass flow rate (in kg/s)	$\dot{m}_3$	.3175
Exit pressure (in kPa)	P_3	310
Compressor isentropic efficiency	η	73%
Compressor exit temperature (K)	T_3	452
RPM		103500
Maximum turbine inlet temperature(°C)	T_4	800

Table 3.1

3.4 Combustion chamber design

The combustion chamber design encompassed the determination of the following:

1.) The outer casing diameter

2.) Flame tube/holder diameter

3.) Hole area for the various zones of the flame tube

4.) Length of the various zones of the flame tube

5.) Hole area distribution

3.4.1 Determination of the outer casing diameter

1.) Using Aerodynamic considerations

Using equation 2.2 from section 2.2.4 gives us

$$A_{ref} = \left[\frac{R}{2} \cdot \left(\frac{\dot{m}_3 T_3^{.5}}{P_3}\right)^2 \cdot \frac{\Delta P_{3-4}}{q_{ref}} \left(\frac{\Delta P_{3-4}}{P_3}\right)^{-1}\right]^{.5} \quad \textbf{(3.1)}$$

Using the values from table 2.1 and table 3.1 we calculate the value of A_{ref}.

2.) Using combustion efficiency design chart

Fig 3.3 shows a combustion efficiency design chart for conventional combustors

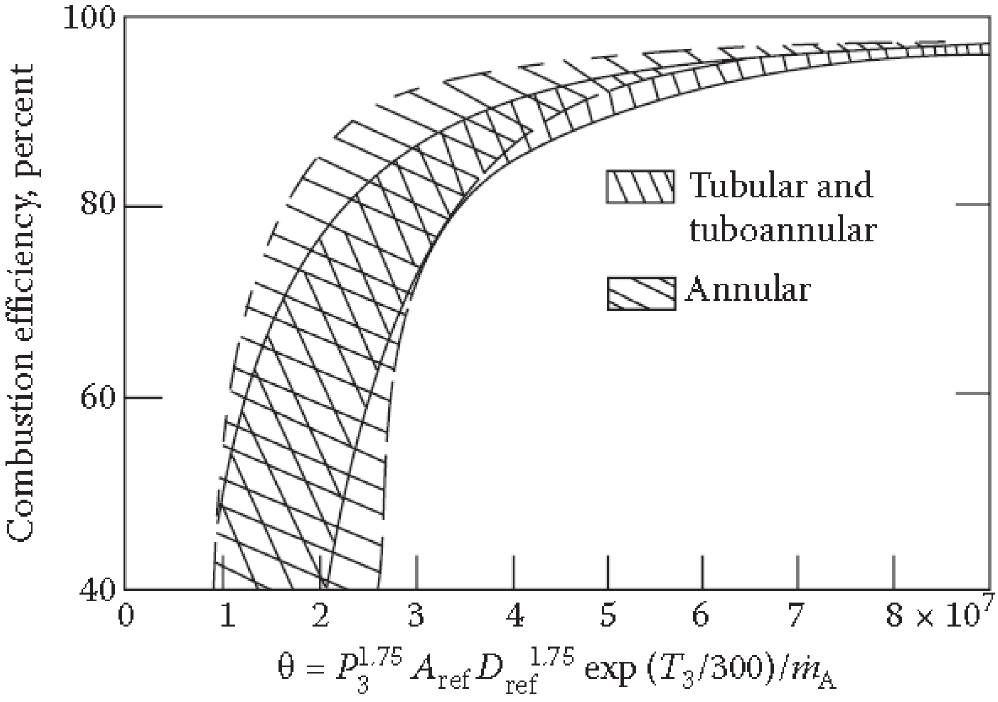


Fig. 3.3: Design chart for conventional combustors

This design chart is used as follows:

1.) A suitable combustion efficiency is chosen (we took a value of 90 to 95%)

2.) The value of ϴ corresponding to that value is read (taken as 8×10^8 here)

3.) Using the values from table 3.1 and using the fact that we have a uniform cylinder as the outer casing (and hence A_{ref} and D_{ref} are related), we find the reference diameter.

The second approach tends to give larger diameters than the first one, but since our project doesn't impose any area/volume constraints we chose the diameter based on the second approach.

Values from both the approaches are summarized in table 3.2

Approach type	**D_{ref} (in mm)**	**D_{ref} (in inches)**
Aerodynamics based	95	3.74
Combustion efficiency based	186	7.32

Table 3.2: The casing diameter values

3.4.2 The diameter of the flame tube

In order to have a low annulus velocity and hence a high static pressure ,the annulus area should be high, at the same time for proper jet mixing, the flame should be sufficiently dimensioned.

Based on a tradeoff between these factors the ratio between the flame tube area and the reference area for tubular combustion chamber should be around 0.7.

$$A_{ft} = .7 \times A_{ref} \qquad (3.2)$$

For the chosen value of the reference diameter i.e. 7.32 inch, the value of the flame tube diameter (D_{ft}) comes out to be 6.12 inch.

Before finding the dimensions of the holes and the lengths of the various zones of the flame tube, a survey of the local market was conducted to check the availability of the pipes matching our requirements and the values of the casing diameter and the flametube diameter were amended according to the available pipes.

Table 3.3 shows the old and the amended values

Component	**Design value (in inches)**	**Available value (inner dia. of the pipe in inches)**
Outer casing	7.32	7.98
Flame tube	6.12	6.06

Table 3.3

Hence, the value of D_{ref} in all the upcoming calculations would be taken as 7.98 inch or 202.7 mm.

3.4.3 Hole area calculation for various zones

This involves the following:

1.) Determination of the preliminary airflow distribution

The design point assumes an overall equivalence ratio of .5 (in actual practice much leaner mixtures are used) with an airflow equal to the amount required for a stoichiometric reaction entering the primary and the secondary zone and the rest entering the tertiary zone for the cooling. Table 3.4 shows the airflow distribution in various zones of the combustor.

	Overall	**Primary zone**	**Secondary zone**	**Tertiary zone**
Equivalence ratio (Φ)	.5	.6	.4	-
Airflow (%)	100	30	20	50
Bleed ratio(μ)	-	.3	.2	.5

Table 3.4

2.) Finding the total hole areas of various zones

The whole area for various zones is found out assuming the following:

1.) Circular plain holes (refer to fig.2.11 and 2.12)

2.) Annulus flow remains constant

3.) The pressure loss across the holes is about 6%

4.) Hole discharge coefficient is .5 for all the holes (refer to fig. 2.11)

Based on these assumptions, calculation is made using the following formula

$$A_{total} = \left(\frac{\frac{R}{2}.(\mu \dot{m}_3)^2.T_3}{P_3^2 \cdot C_d^2 \cdot \frac{\Delta P_h}{P_3}} \right)^{.5} \qquad 3.3$$

Table 3.5 shows the total area allocated to each zone based on equation 3.3

	Primary zone	**Secondary zone**	**Tertiary zone**
Area (mm^2)	740	426	1065

Table 3.5

3.4.4 The length distribution among various zones

The primary zone length can be taken within 2/3 to 3/4 of the flame tube diameter. The latter value is related with better combustion efficiency and has been chosen. A value of .5 times the flame tube diameter is suggested for the secondary zone. A length of 1.5 times the flame tube diameter is recommended for the tertiary zone.

The length of the primary zone is the distance between the starting point of the flame tube and the midpoint of the last row of primary holes and the first row of the secondary holes, the length of the secondary zone is the distance between the endpoint of the primary zone and the midpoint of the last row of secondary holes and the first row of the tertiary holes. Finally, the length of the tertiary zone is the distance from the end of the secondary zone till the turbine inlet.

The turbine inlet (shown in fig.2.25c) is smaller (a rectangular entry of 2.5 inch × 2 inch) as compared to the flame tube and hence a converging passage has to be attached, this region is also the part of the tertiary zone length.

The values are summarized in table 3.6.

	Primary	**Secondary**	**Tertiary**
Length(in mm)	110	82.5	220

Table 3.6

3.4.5 Distribution of the total zonal hole area

The total hole area of various zones found in section 3.4.3 was distributed as given in table 3.7.

The hole diameters were chosen so as to:

1.) Help the flame survive in the primary zone

2.) Lower the primary zone temperature by the use comparatively higher penetrating jets and hence burn the CO and soot coming from the primary zone.

3.) Achieve maximum penetration levels and cool the hot gases before they enter the turbine. Refer fig. 2.15.

A region in each zone was left without any holes, this was done in order to promote homogeneity within the gaseous flow.

	Primary	**Secondary**	**Tertiary**
Total area(mm^2)	740	426	1065
Hole diameter calculated (mm)	5.9	7.8	11.65
Hole diameter used (mm)	6	8	12
Total no. of holes	27	9	10
No. of rows	3	1	1

Table 3.7

The used hole diameters are the diameters closest to the calculated value for which drill bits were easily available.

3.4.6 Combustor model

Based on the data obtained in the previous sections, a model of the combustor assembly was made using SOLIDWORKS.

Fig 3.4 shows a drawing of the assembly, fig. 3.5 shows the solid views and fig. 3.6 highlights the features of its design.

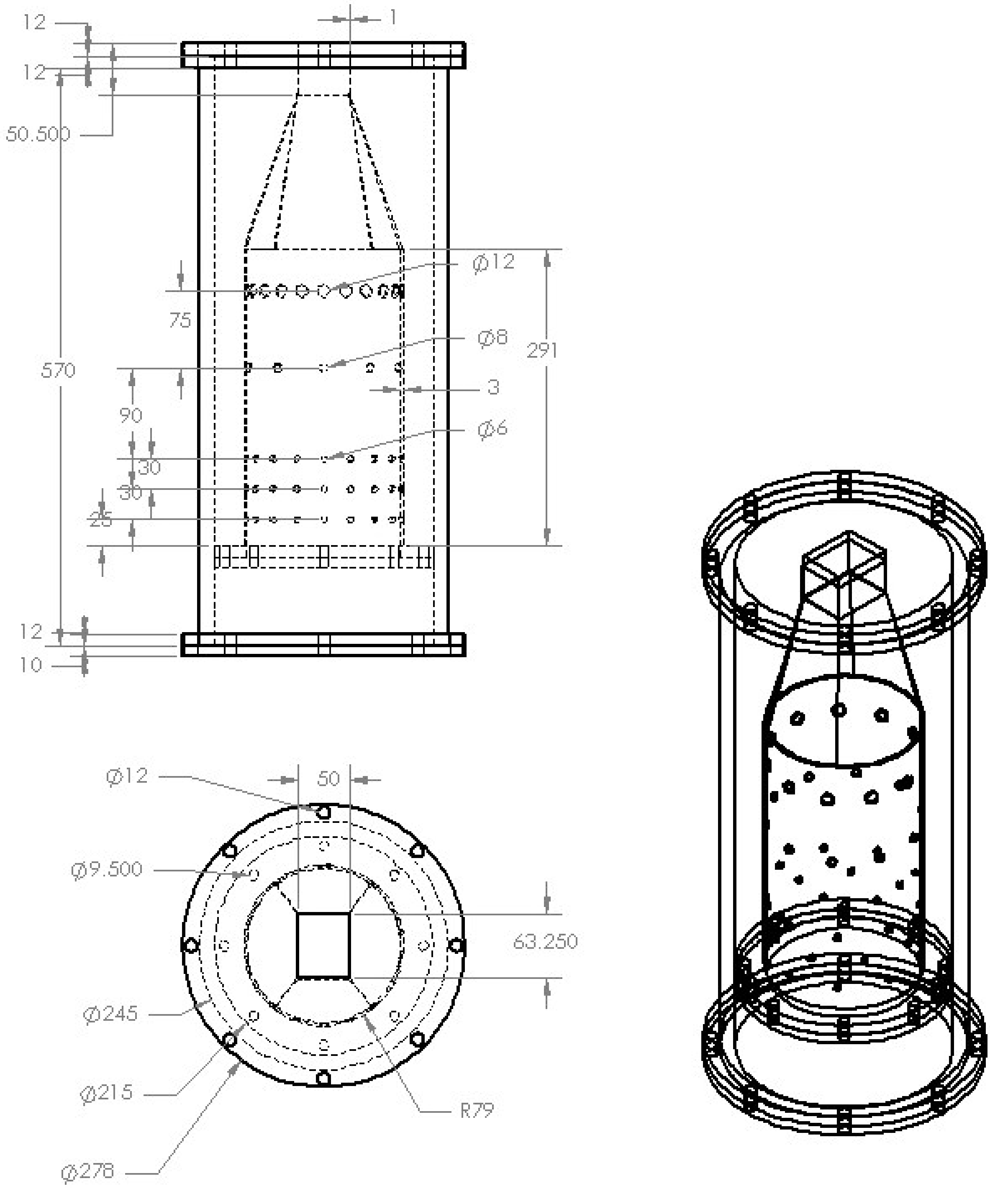
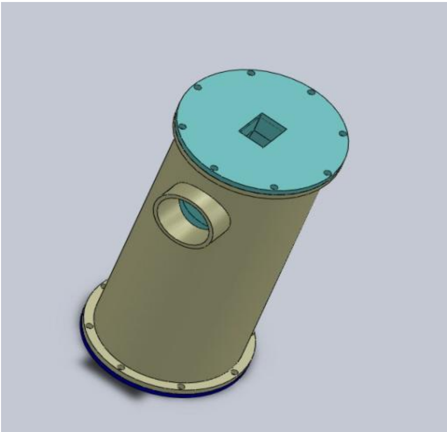


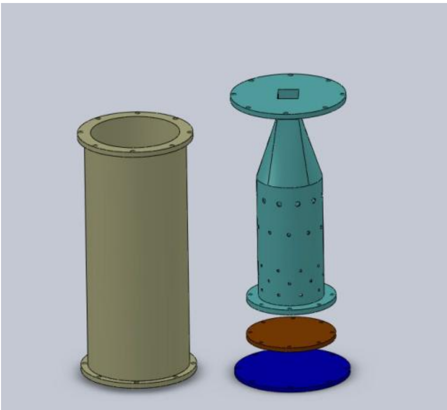
Fig 3.4: Drawing showing the combustor assembly and its dimensions

Fig 3.5 (a), (b), (c) show the assembled, component and the sectional view, respectively.



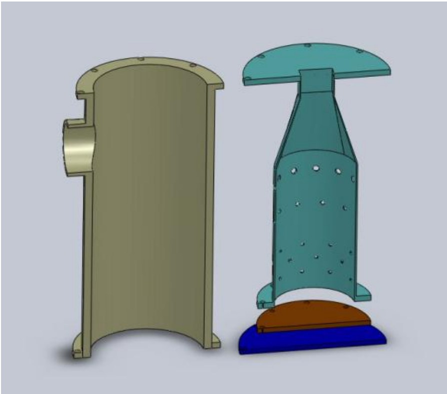
(a)

An assembled view showing the outer casing, compressor inlet (2” dia.) and the turbine outlet (2.5”×2”).The air inlet has no offset. Offset inlets induce tangential flows along the axis of the tube which are detrimental



(b)

An exploded view showing the outer casing, flame tube, flame tube flanges and the covering plates. It is worth noticing that the turbine inlet is considerably smaller than the flame tube and hence the converging pipe.



(c)

A sectional view of the components

Fig 3.5 (a), (b), (c) show the assembled, component and the sectional view, respectively.

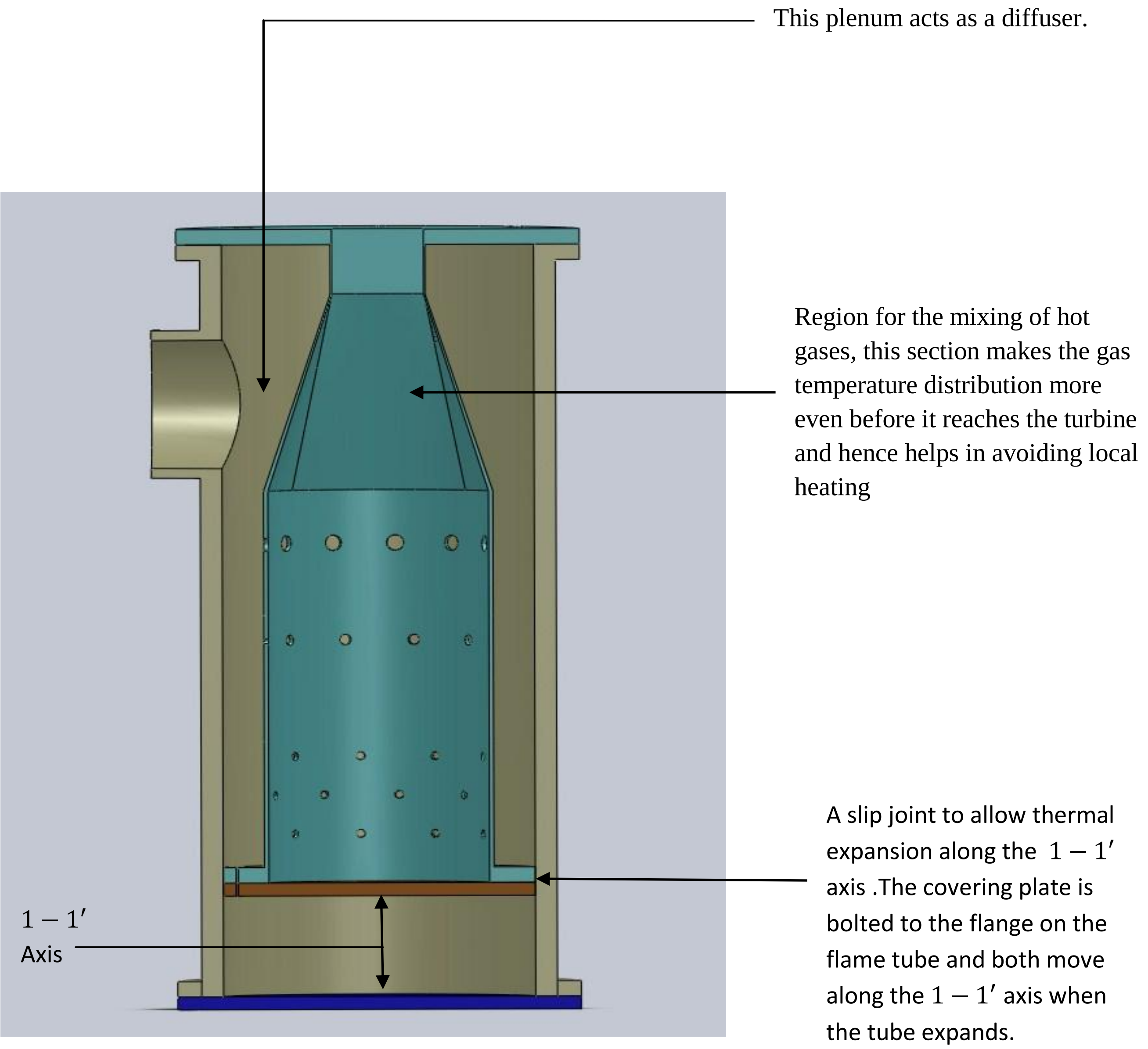


Fig 3.6: Key design features of the combustor assembly.

An effort was made so as to make the combustor assembly as modular as possible, leaving room for modifications and a hassle free replacement of any failed component.

3.4.7 Flow simulations

Based on the data obtained from the previous section, a preliminary model was generated and was put into ANSYS FLUENT to get an initial idea of the validity of the design.
Two simulations were carried out:
1.) Cold flow simulation
2.) Combustion simulation

1.) Cold flow simulation

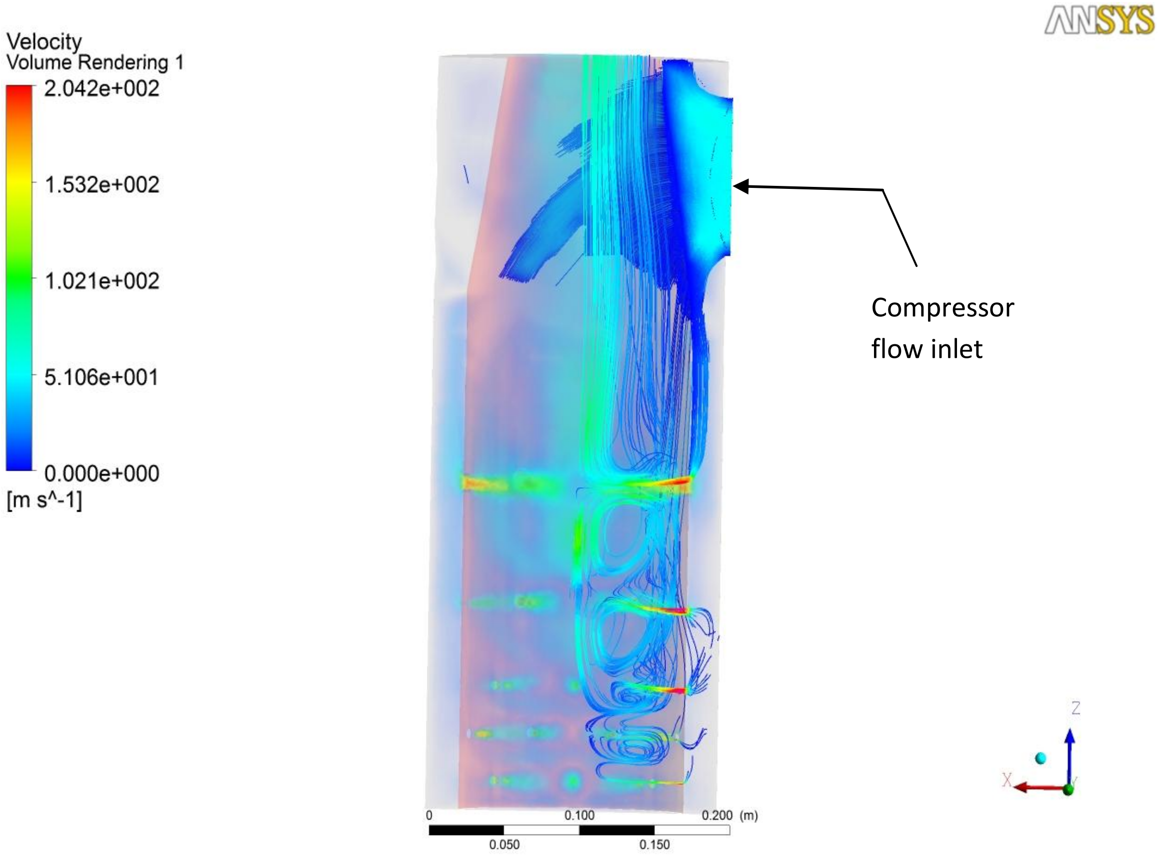


Fig. 3.7: A cold flow simulation depicting air paths and velocities

Inferences drawn from fig 3.7:

1.) This simulation shows recirculation patterns and low penetration in the primary zone with slightly higher velocity and penetration in the secondary zone.

2.) The tertiary jets are penetrating till the core

3.) Low annulus velocities

4.) Steep jet admission angles

The flow patterns obtained from this simulation; though without combustion, were pretty encouraging.

2.) Combustion simulation

For this simulation, a fuel flow rate of propane was taken as 10.242×10^{-3} Kg/s. The zone wise equivalence ratio and airflow percentages are summarized in table 3.4

A radially spewing fuel injector was considered as the fuel inlet.

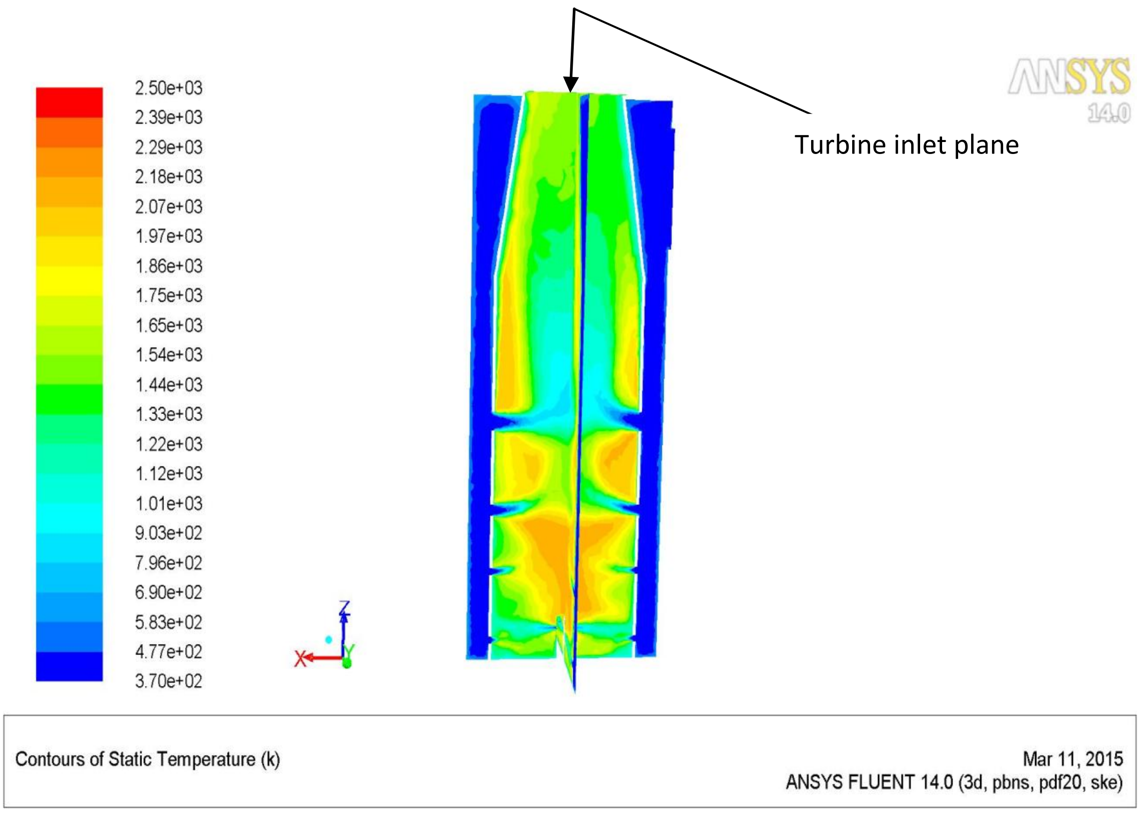


Fig. 3.8: Temperature distribution in the combustor

Inferences drawn from fig. 3.8:

1.) At the chosen design point, the turbine inlet temperature value is higher than the recommended turbine inlet temperature but it should be remembered that in actual practice the engine would run at much leaner mixtures, giving lower temperatures.(leaner, but mixtures leading to temperatures at which the flame can sustain.)
2.) The temperature distribution at the turbine inlet plane seems pretty homogenous, signifying a good amount of mixing.
3.) The air cushion that protects the flame tube from melting can also be clearly seen.

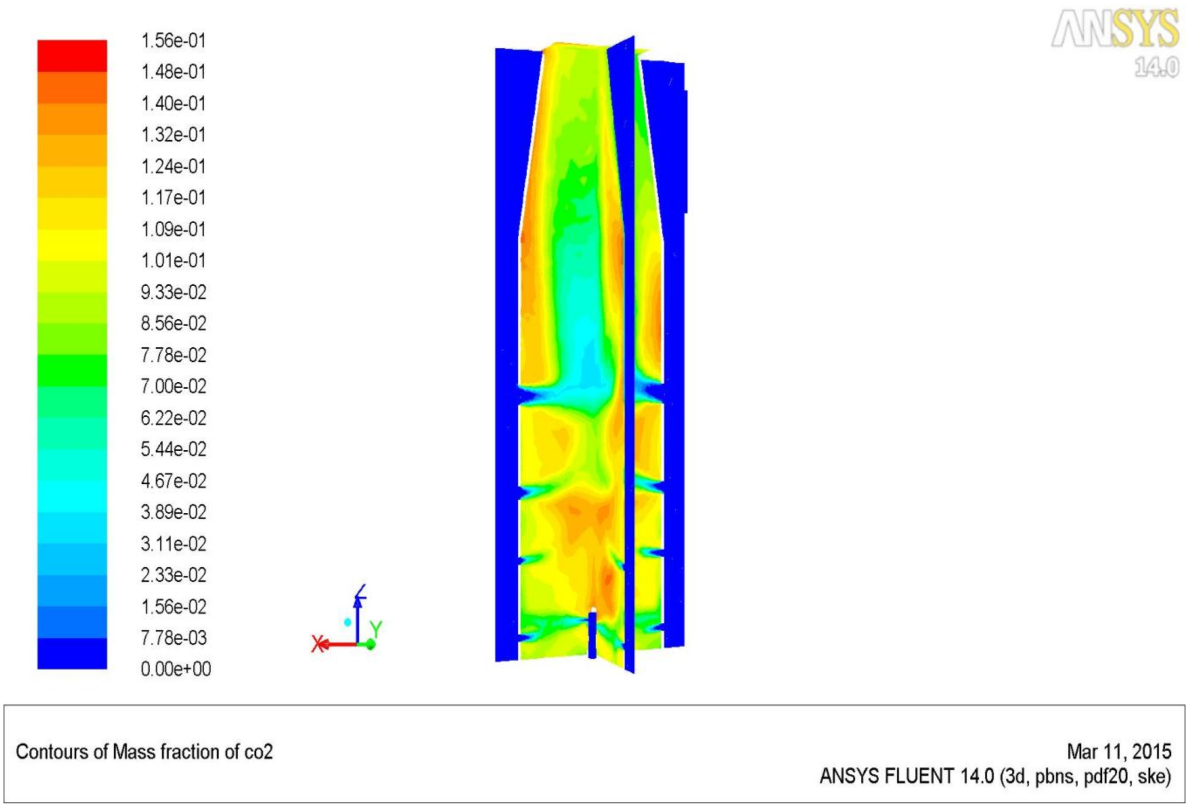


Fig. 3.9: Mass fraction distribution of carbon dioxide in the combustor

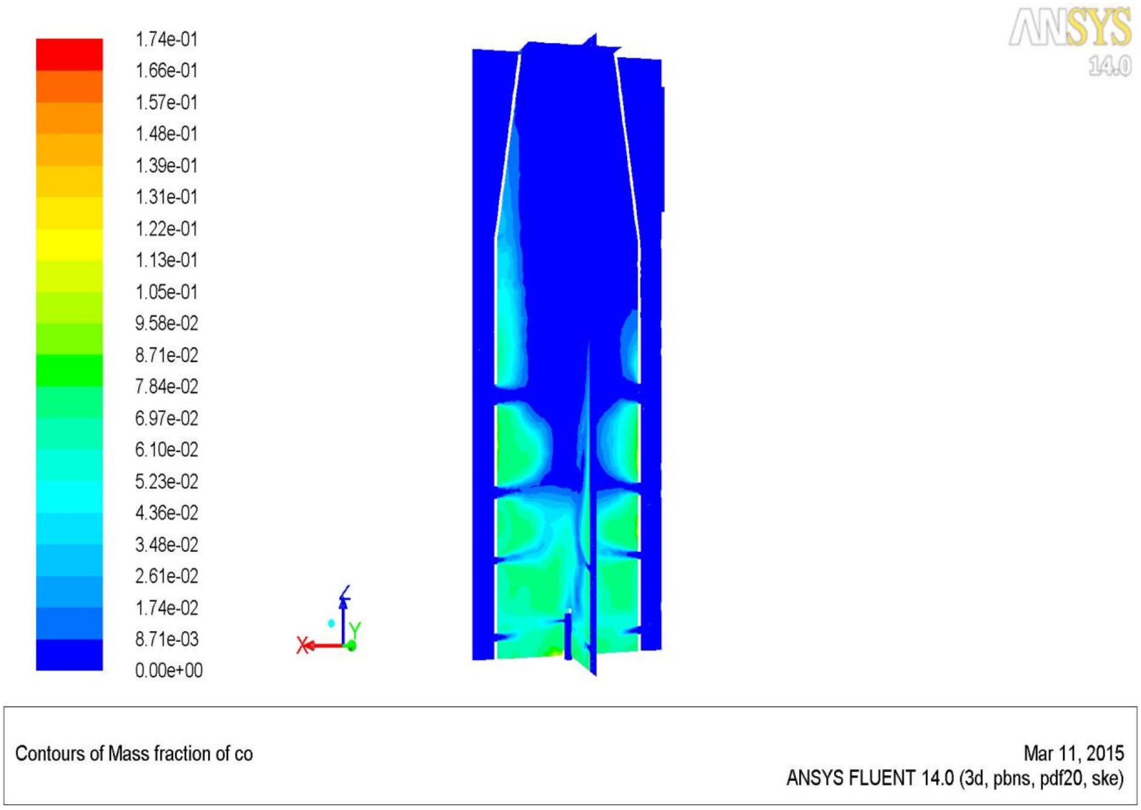


Fig. 3.10: Distribution of carbon monoxide in the combustor

Inferences from fig. 3.9 and 3.10

1.) Both fig. 3.9 and 3.10 indicate clean burning in the combustor

2.) Looking at fig. 3.10, one can see the importance of the secondary zone, a lot of carbon monoxide is burnt in the secondary zone.

Based on the flow simulations, it was decided that the combustor would be fabricated with the same dimensions.

3.5 Material selection justification

We decided to use stainless steel (SS 302) for the combustion chamber based on its successful use in similar applications before.
To confirm its efficacy at elevated temperatures we decided to run a structural simulation in a horizontal position.
Refer to fig. 3.11 and 3.12

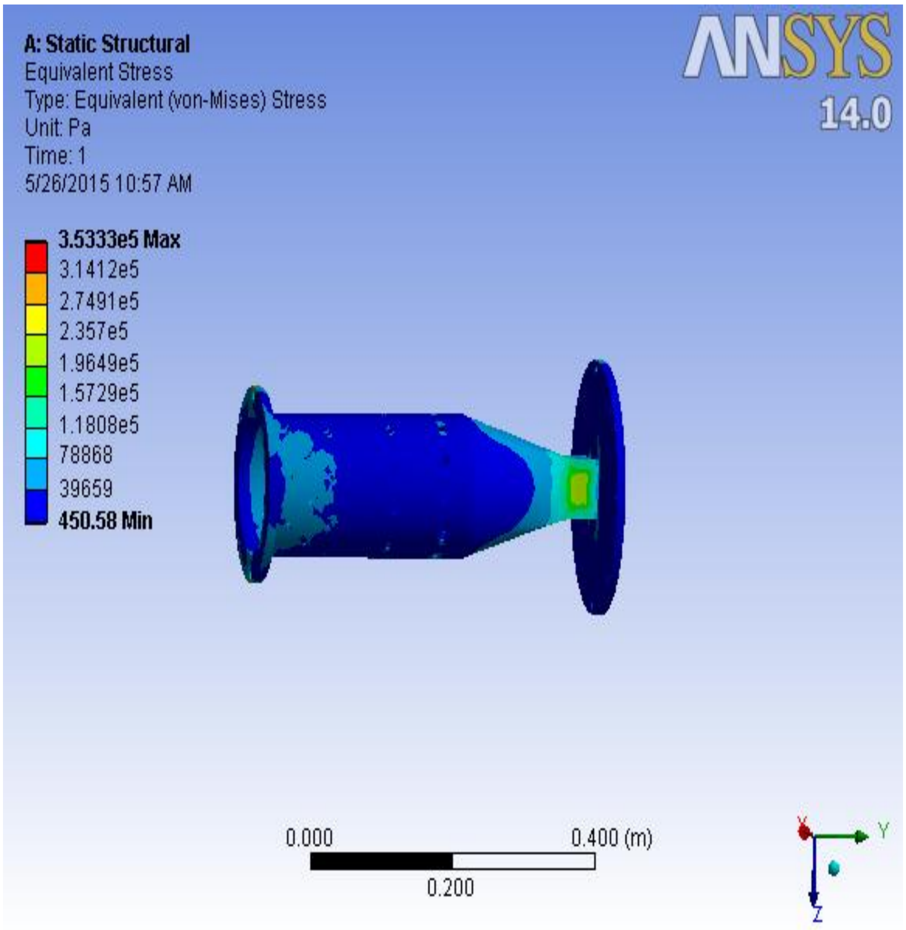


Fig. 3.11: Stress distribution in the flame tube in a horizontal position

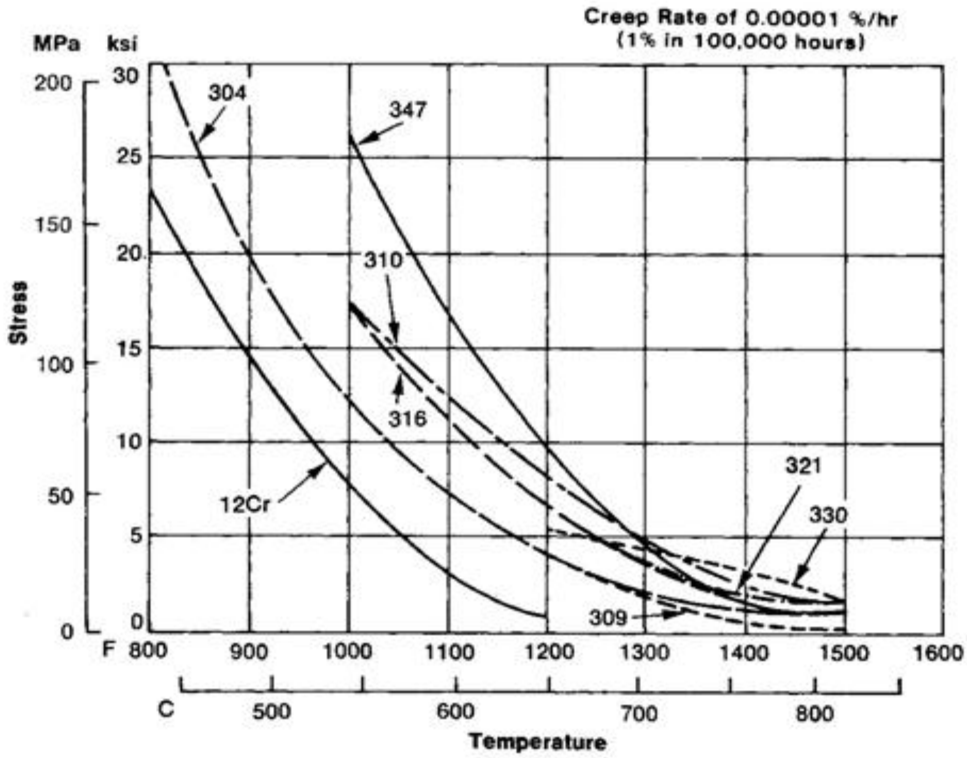


Fig. 3.12: Creep rate curves for several types of steels

Inference from Fig.3.11 and 3.12

- The maximum stress on the structure is about .25MPa, while consulting Fig.3.12 at 900°C and for the given creep rate, the Stress is about 1MPa; therefore at our design stress there will be an even lower creep.

References:

1. Lefebvre, A. H and Ballal D.R. "Gas turbine combustion" 3rd edition.2010. Taylor & Francis.

2. Superturbodiesel.com “compressor map for Holset HX-35”

3. Conrado, Lacava, Filho, Sanches “Basic design principles for a gas turbine combustor” 2004.
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CHAPTER 4

Fabrication: Converting the design into a working entity

The penultimate stage of the project included the fabrication of the combustor and putting the turbocharger, the lubrication and the measurement systems in place.

This chapter is concerned with the:

- Materials used for the combustion chamber fabrication.
- Manufacturing processes used
- The fuel injection system and ignition system
- Lubrication system
- Measurement system
- Supporting frame fabrication
- Some miscellaneous fabrication practices

4.1 Materials used for the combustion chamber fabrication

Section 3.5 looked at the choice of the material of the flame tube, which is most stressed component of the combustor assembly and has to face the highest temperatures.

Table 4.1 shows materials used for various parts of the combustor assembly (refer to fig. 3.5, 4.1 for components and section 2.3 for turbocharger materials)

Component	**Material**
Outer casing	Stainless steel-202
Flame tube	Stainless steel-302
Flame tube flanges and cover	Stainless steel-302
Flanges at the end of the outer casing	Mild steel
Outer covers	Stainless steel-202
Bend from the compressor exit	Aluminum alloy
Inlet pipe	Stainless steel-302

Table 4.1

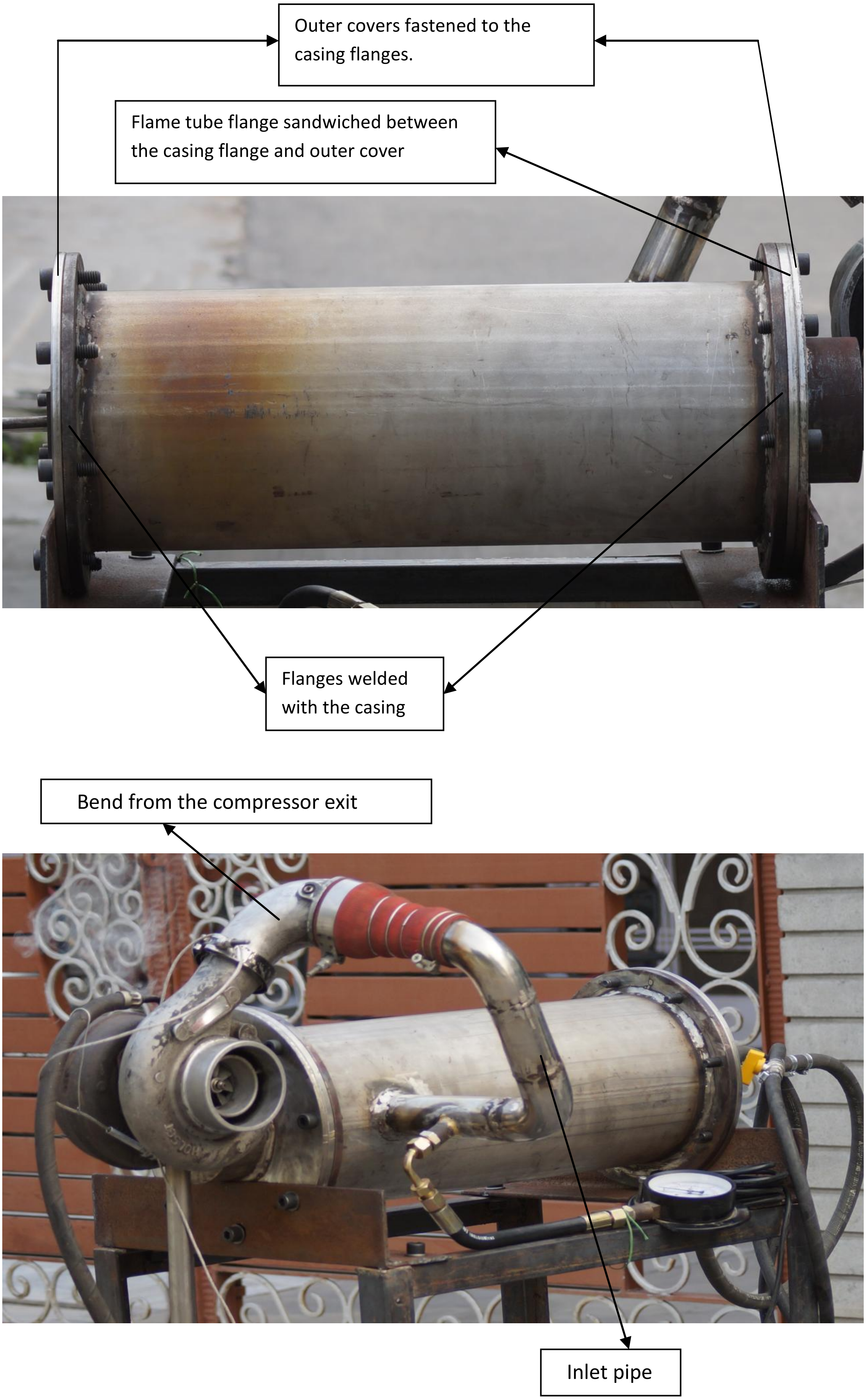


Fig 4.1: Components of the combustor in context of the material table

4.2 Manufacturing practices used in the fabrication

The key manufacturing processes used were:

1. Machining
2. Welding
3. Drilling
4. Fastening
5. Sheet metal

1.) Machining

Machining was used in:

- Facing various pipes and turning plates in order to assign correct dimensions to them.
- Making flanges for the outer casing and the flame tube out of circular plates.

2.) Welding

Welding was used in:

- Attaching flanges to the flame tube and the outer casing
- Connecting the converging section to the flame tube.
- Connecting the compressor feed pipe to the combustor casing.
- Adding pressure gauge duct in the compressor feed pipe.
- The fuel injection system (refer section 4.3)
- Frame fabrication (refer to section 4.6).

3.) Drilling

Drilling was used in:

- Making holes of various sizes on the flame tube.(refer fig. 3.4 for the dimensions)
- Making fastener holes in the flanges and cover plates, the holes in outer casing's flanges were tapped with threads as well. (refer to fig. 3.4 for the dimensions)
- Fuel injector and spark plug slots in the outer casing as well as flame tube cover plates. (refer to fig. 4.2)

- Fuel injection holes in the injector
- Frame fabrication (refer to section 4.6)

4.) Fastening

Fasteners were used to:

- Attach the turbocharger with the combustion chamber
- Attach the flame tube to the outer casing
- Attach the covering plates to the casing and the flame tube flanges.
- Attach the combustion chamber with the frame.
- To seal the fuel injection region on the outer cover using a cork gasket.

5.) Sheet metal

Sheet metal practices were used in:

- Making the converging passage of the flame tube to the turbine inlet (refer to fig 3.4 and 3.5)
- The lubrication system(refer to section 4.4)

4.3 The fuel injection system

As mentioned earlier, the current project makes use of a gaseous fuel; as a result a simplified fuel injection system was required.

A radially spewing injector was proposed in order to assist fuel mixing and burning.

The fabrication of the fuel injection system consisted of:

1. Drilling holes of approximately 1.58mm on all the faces of a hexagonal nut.
2. Covering one of the open faces of the nut.
3. Welding a fuel pipe made of SS and having an inner diameter of approximately 10mm to the other face of the nut.

The hexagonal nut with drilled faces which was used for the fuel injector.

(a)

The fuel injector fitted in the flame tube covering plate along with the spark plug (post testing photograph)

(b)

The fuel pipe coming out from the casing plate and connected to the fuel supply (This is the same pipe that's shown connected to the nut in the above photograph)

(c)

Fig. 4.2 (a), (b), (c): The fuel injection system

4.4 The Ignition system

The ignition system comprised of the following:

1. A spark plug
2. A 4V to 400kV step up module

The spark plug was inserted in its slot in the flame tube cover plate beside the fuel injector [refer to fig.4.2 (b)] and was connected to the step up module (fig. 4.3).

Sparks were easily created by grounding one end of the step module and connecting the other to the spark plug terminal coming out of the covering plate of the casing (this terminal had to be insulated for the spark to be created).

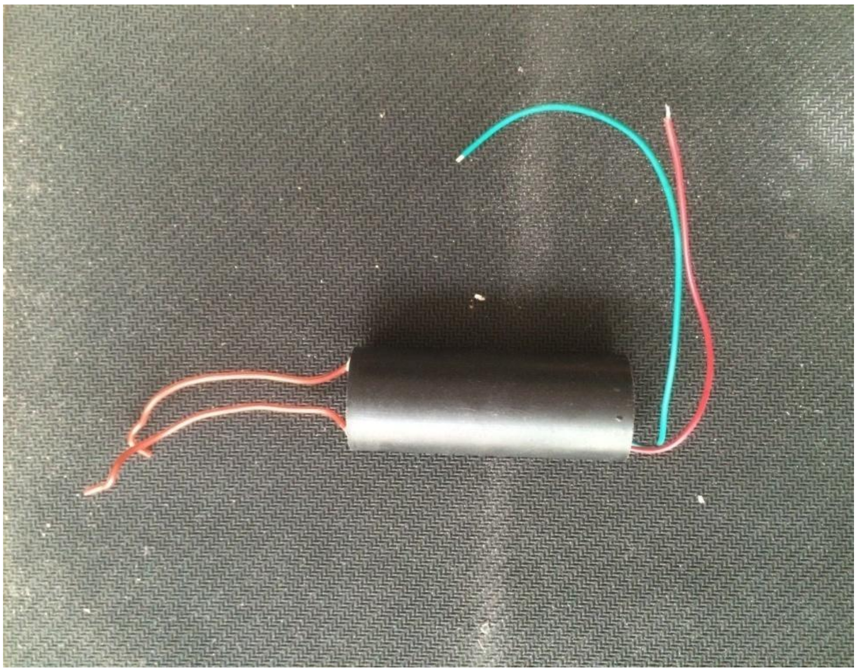


Fig.4.3: The voltage step up module

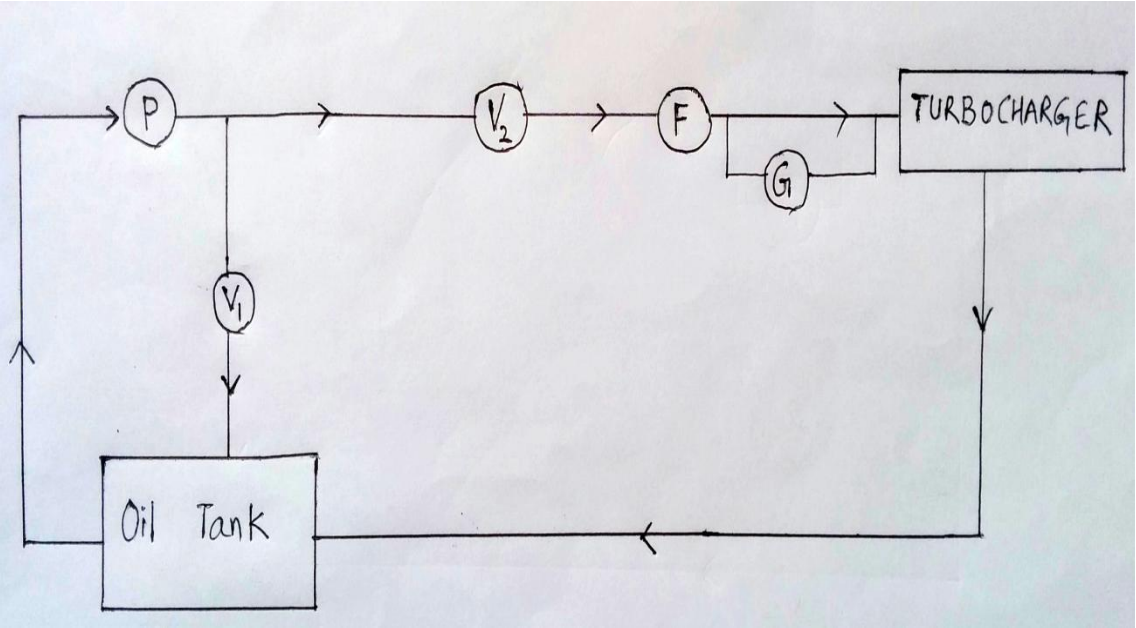
4.4 The lubrication system

For the turbocharger to work without any problems, it needs clean lubrication oil to be supplied at around 50-100 psi**.**

In order to achieve this, a .5 hp, single phase motor was coupled with a gerotor pump (section 2.4) and an oil supply circuit was established.

4.4.1 The oil supply circuit and its components

Fig. 4.4 shows the oil supply circuit used in this project



Key:

P= Gerotor pump

V_I= Pressure relief valve

V_2= Hydraulic needle valve

F= Oil filter

G= Pressure gauge

Fig. 4.4: The oil supply circuit diagram

The components of the lubrication system:

1.) The gerotor pump coupled with the .5 hp, single phase motor.

The pump, coupled to the motor (shown in fig. 4.5) served as the driving element

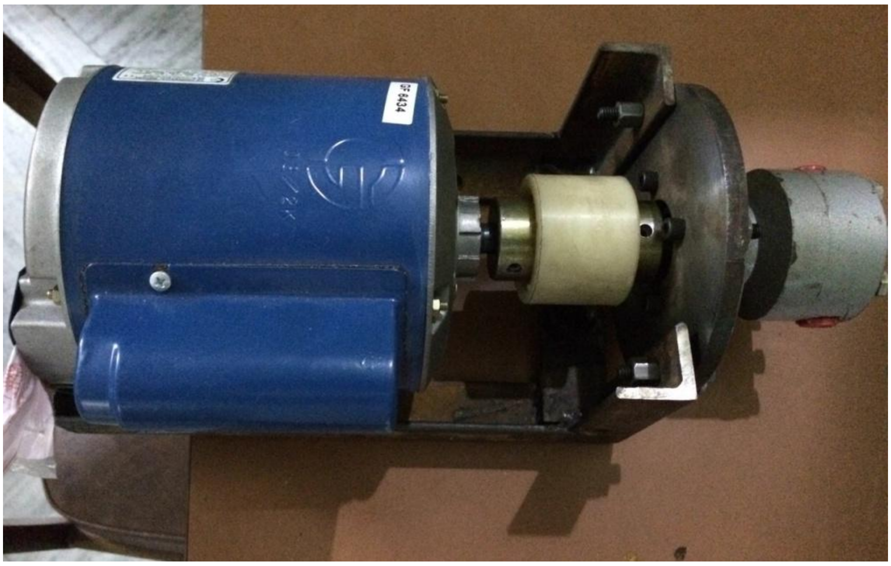


Fig. 4.5: The pump coupled with the motor

2.) Hydraulic needle valve

During the spool-up of the engine, the oil pressure has to be restricted in order to have a relatively free motion of the rotor. As the rotor rpm increases, it requires higher lubricant flow.

In order to achieve this, a hydraulic needle valve (V_2 in fig.4.4) was used.

Fig. 4.6 shows a hydraulic needle valve.

Fig. 4.6: A hydraulic needle valve

3.) Pressure relief valve

The pump used in this project was able to generate pressure up to the level of 200 psi, in order to restrict the pressure reaching the turbocharger; a pressure relief valve (shown as V_1 in fig.4.4) was incorporated in the circuit.

The relief valve can be easily calibrated by the use of an Allen key, in our case the valve was calibrated in such a way that it allowed a pressure of 70psi to the turbo when the needle valve was fully open.

The most important reason for using the relief valve was the uneven response of the needle valve (for e.g. the pressure gauge would rise from 0 to 20 psi in two rotations of its knob while reaching 150 from 20 in just one), which posed a problem in controlling the lubricant flow and most of the times a higher pressure was sent to the turbocharger. This problem was eliminated using the relief valve.

Fig. 4.7 shows the relief valve used.

Fig 4.7: A generic pressure relief valve for a Sonalika tractor used in the current project

4.) Oil cleaning equipments

To clean the oil reaching the turbocharger two oil filters were used, one in the oil tank to get rid of coarse particles and one in the oil supply circuit (an automotive oil filter). Apart from these, a magnet was dropped in the oil tank to collect magnetic particles (due to the fabrication process and could not be removed.)

Fig. 4.8 show these components

The automotive oil filter used in the engine.

(a)

The oil filter inside the oil tank

(b)

Fig. 4.8: Oil cleaning components of lubrication system

5.) Engine oil

20 liters of 10W-40 engine oil were used in the lubrication system.

The choice was based on its ability to maintain viscosity at high temperatures.

4.5 The measurement system

In order to judge the engine's performance, the following components were installed:

1.) A 30 psi pressure gauge to measure the compressor pressure

2.) A K-type thermocouple [shown in fig.4.9 (b)] along with a digital temperature gauge to measure the turbine outlet temperature.

3.) A 150 psi pressure gauge for the lubrication system

4.) A laser tachometer for RPM measurement. (One of the compressor blades was marked with a white paint.)

Most of these components were put on a dashboard to ease the data recording process [see fig. 4.9 (a)].

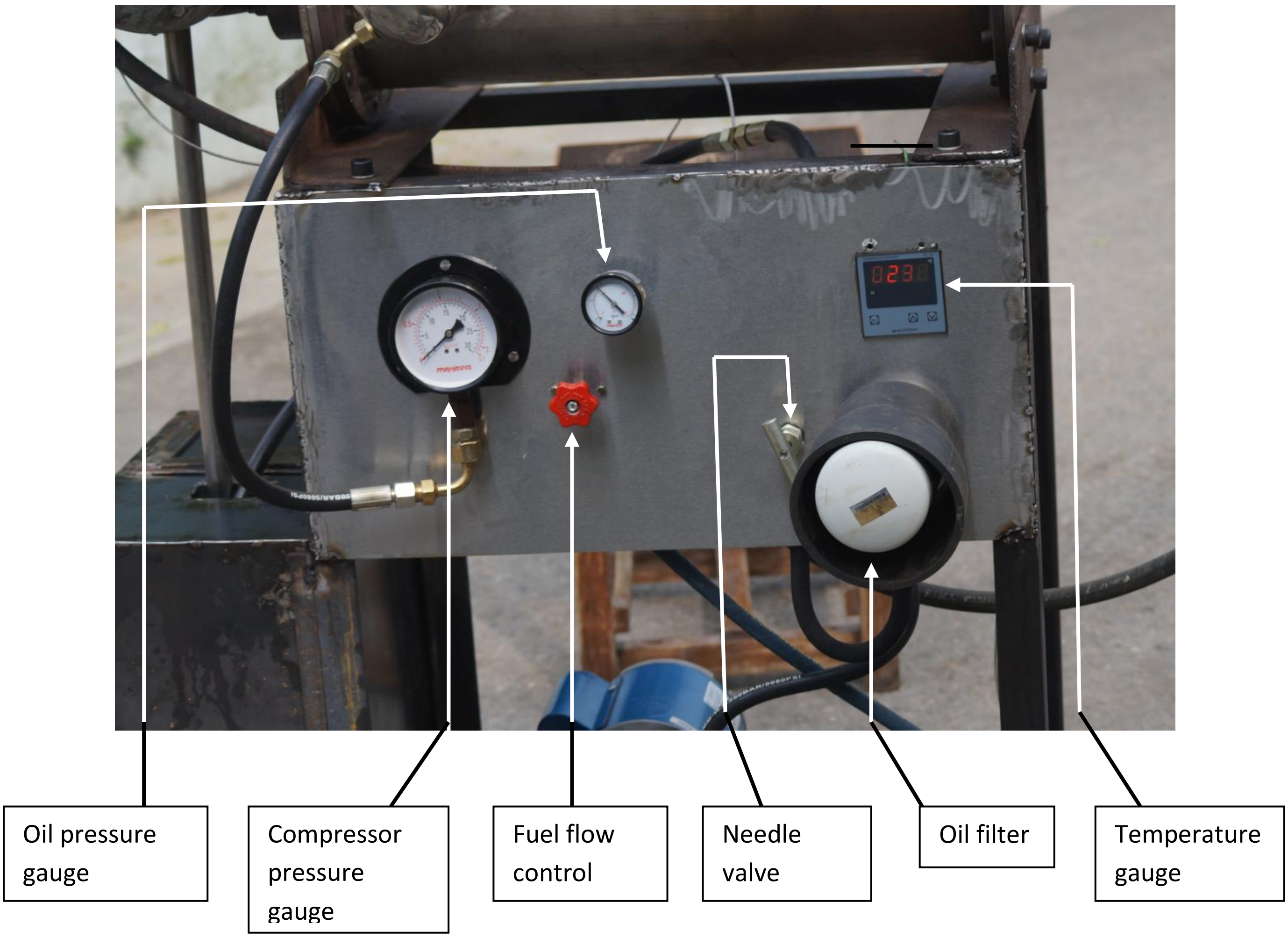


Fig.4.9 (a): The dashboard and the installed components

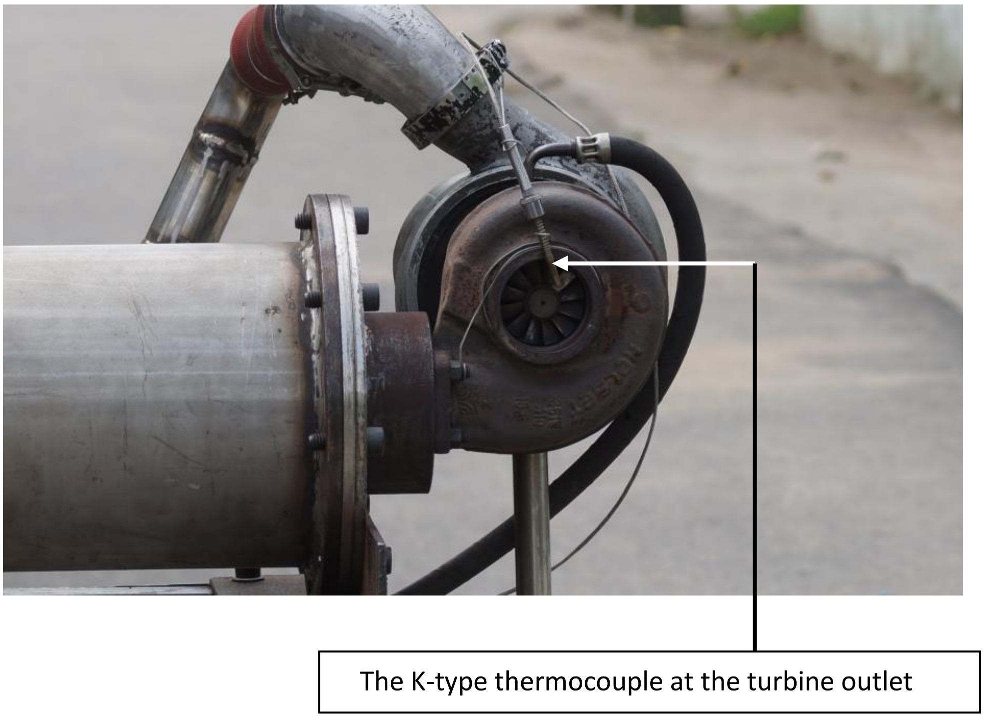


Fig.4.9 (b): The installed thermocouple

4.6 The supporting frame

A frame to put all the engine components together was made by welding mild steel angles. Wheels were provided to ease the engine's movement (fig. 4.10)

Fig. 4.10: The engine frame

4.7 Some miscellaneous practices undertaken during the fabrication process

1.) Countersinking of flame tube holes

The flame tube holes were countersunk in order to increase the value of the discharge coefficient. Fig.4.11 shows countersunk holes in the primary zone (holes in all the zones were countersunk)

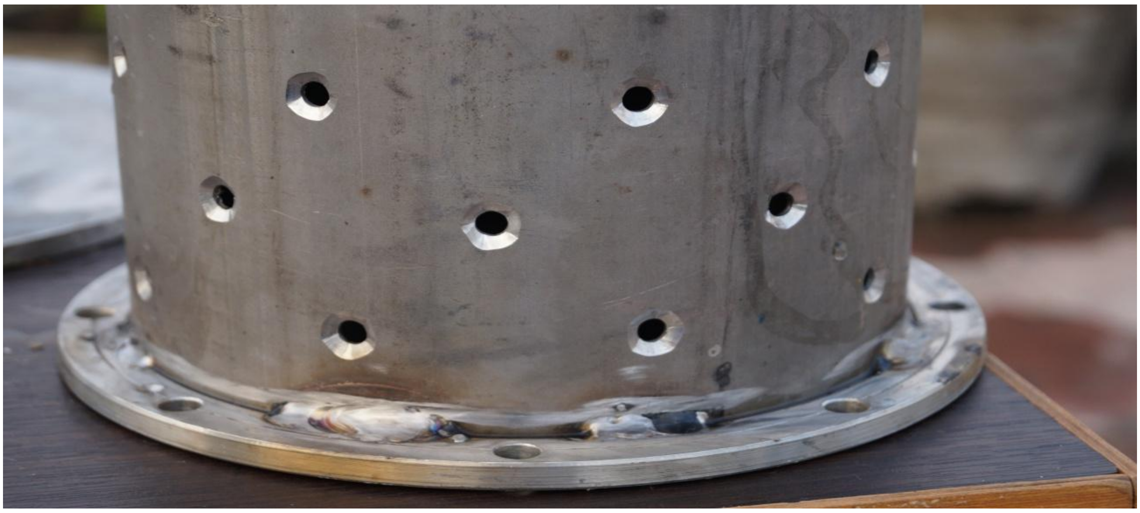


Fig. 4.11: Countersunk holes

2.) Rotor balancing

Since the rotor reaches RPM values topping 100000, shaft balancing is of the utmost importance.

Shafts were checked and balanced before running the engine.

The shafts were balanced at the local turbocharger workshop.

The ease with which the balancing can be done also speaks to the viability of using a turbocharger.

Fig. 4.12 shows a shaft balancing machine.

Fig.4.12: A shaft balancing machine in action

3.) Modular design

The engine design used fasteners instead of welding as much of possible.

As mentioned earlier, a modular design opens the roads for any future modifications and helps in easy the rectification of problems.

4.) Sealing

Cork gaskets were used to prevent leakage from the combustion chamber's outer cover plate.

CHAPTER 5

PERFORMANCE DETERMINATION

Contents of this chapter include:

- Qualitative testing
- Quantitative testing

5.1 Qualitative testing

Our methodology for qualitative testing included:

1. **Examination of the color of the turbine blades while the engine is in operation.**
2. **Examination of the flame tube post operation.**

Fig.5.1 shows photographs of the turbine outlet while the engine was running at the maximum fuel flow rate during two separate runs.

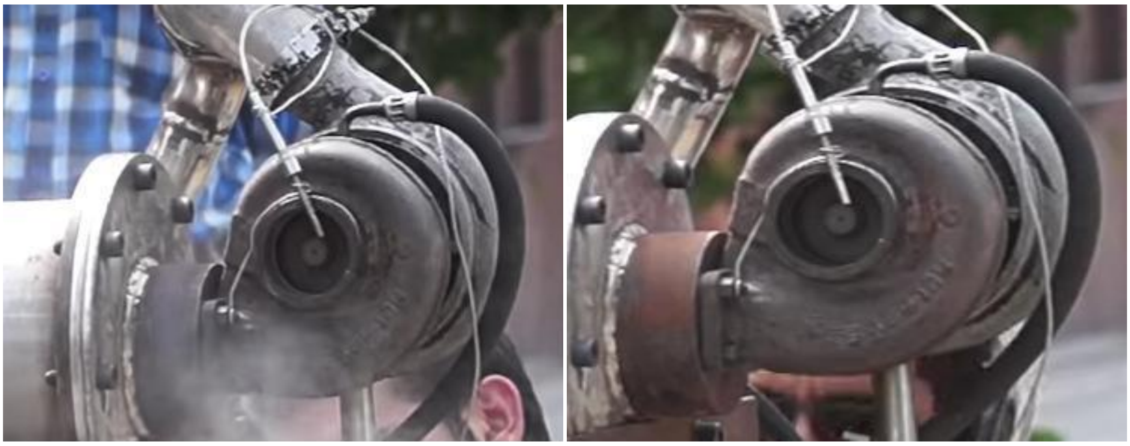


Fig.5.1: Turbine blades' condition

These photographs show no blade glow. With such a visual inspection one can qualitatively conclude that the turbine inlet temperatures were within the safe limits.

2.) Examination of the flame tube after a run

An idea of the temperatures to which the flame tube was subjected to **(not the same as the core temperature of the flame tube)** can be gotten from a comparison of the colors formed on the surface of the flame tube with a heat tint color chart (refer to the appendix for more info. on heat colors.). Table 5.1 shows a heat tint color chart

Fig. 5.2 shows the colors on the flame tube post operation.

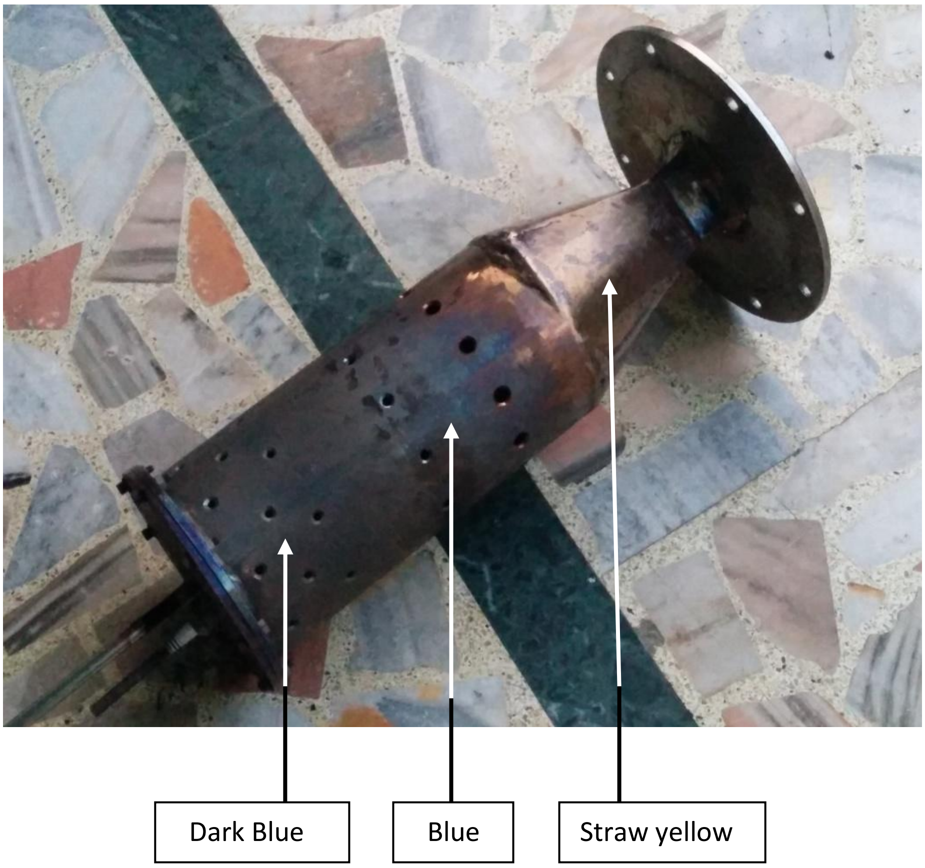


Fig. 5.2: The heat colors of the flame tube

Color Formed	Approx Temperature(° C)
pale yellow	290
straw yellow	340
dark yellow	370
brown	390
purple brown	420
dark purple	450
blue	540
dark blue	600

Table 5.1: The heat tint color chart for steel

Comparing the flame tube with the color chart, we can deduce the temperatures to which the flame tube was subjected as give in table 5.2.

Zone	**Primary**	**Secondary**	**Tertiary**
Color	Dark blue	Blue	Straw yellow
Approx. temp. (°C)	600	540	340

Table 5.2

Although no exact conclusions can be drawn from this data but it can be safely concluded that the temperature of the gases after the tertiary holes dropped significantly, which is as per the design requirement.

This also speaks to the fact that although core temperatures reach values over the melting point but the air cushion provided by the air coming through the holes saves the flame tube.

5.2 Quantitative testing

The idea here is to answer the following two questions:

1. **Is the compressor performing efficiently?**
2. **Is the turbine inlet temperature within the prescribed limits?**

In order to answer these questions, the following methodology was used

1. Finding the compressor pressure ratio using the compressor pressure gauge (**the pressure ratio is a ratio of absolute pressures and hence the atmospheric pressure must be added to the gauge pressure**).
2. Finding the compressor RPM using the laser tachometer.
3. Finding the efficiency of the compressor using this data and the compressor map (fig. 3.2). This is done by finding the point of intersection of the pressure ratio and the RPM reading and then reading the efficiency at that point.
4. Finding the turbine inlet temperature using the following formula (details of which are in the appendix)

$$\boldsymbol{T_4 = T_5 + 271.13\left(r_p^{.284} - 1\right)} \qquad \textbf{(5.1)}$$

Test Results:

Three test runs were conducted and the results are given in tables 5.3, 5.4 and 5.5.

The first run was done with an old turbocharger, the second run with the same turbocharger but new bearings and the third one with a new turbocharger.

Pressure gauge reading(psi)	15	20	25	30
Pressure ratio(r_p)	2.006	2.347	2.687	3.027
RPM of the compressor	74700	82800	-	-
Turbine outlet temperature(°C)	560	588	-	-
Compressor Efficiency % (η)	69	70	-	-
Turbine inlet temperature (°C)	619.27	662.33	-	-

Table 5.3: Results from the first run

Pressure gauge reading(psi)	15	20	25	30
Pressure ratio(r_p)	2.006	2.347	2.687	3.027
RPM of the compressor	76600	84800	101200	-
Turbine outlet temperature(°C)	538	543	563	-
Compressor Efficiency % (η)	71	71	70	-
Turbine inlet temperature (°C)	597.27	617.33	650.87	-

Table 5.4: Results from the second run

Pressure gauge reading(psi)	15	20	25	30
Pressure ratio(r_p)	2.006	2.347	2.687	3.027
RPM of the compressor	76900	85900	103700	108100
Turbine outlet temperature(°C)	523	532	548	565
Compressor Efficiency % (η)	71	71.5	72	72
Turbine inlet temperature (°C)	582.27	606.33	635.87	665.22

Table 5.5 Results from the final run

Inferences from the data:

1.) The engine performance improved by changing the bearing and by the change of the turbocharger, both compressor efficiency and turbine inlet temperature showed improvements.

2.) The efficiency at the design pressure ratio was at 72% vs. 74% of the design value.

3.) The compressor worked in the non-surging, non--choking region i.e. the operational points were neither too right nor too left in the compressor map.

4.) The maximum turbine inlet temperature reached was about 670°C, well below the 800-850°C limit (but we have to remember that there was no load on the engine during these tests.).

We concluded from the tests that the engine was working as per our expectations and according to the design.

CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

Motivated by the current power scenario in the country, a micro gas turbine engine was designed and fabricated for distributed power generation and to cater area. The turbocharger helped us in eliminating several design problems and was cost effective as well.

The design was verified using numerical analysis tools and the results from the testing of the fabricated engine showed good agreement. Test results showed stable working of the engine as the compressor efficiency was in the 70-72% range and the turbine outlet temperatures were under 600°C. Apart from that, the liner tempering colors suggested a temperature distribution along the flame tube which met our design expectations.

6.2 Recommendations

The recommendations for future work include:

- Making the engine ready for liquid fuel.
- Use of swirlers for mixing of liquid fuel in the primary zone.
- Reducing size of the combustor.
- Using different compressor and turbine wheels to reach a better efficiency.
- Life prediction based on environmental effects and cycle fatigue.
- Investigation of combustor off-design performance
- More rigorous numerical analysis
- Heat transfer analysis of the combustor.
- Design, fabrication and testing of other flame tube geometries.

APPENDIX

1.) Reference Quantities

A number of flow parameters have been defined to facilitate the analysis of combustor flow characteristics and to allow comparison of the aerodynamic performance of different combustor designs.
These parameters include the reference velocity, U_{ref} and reference dynamic pressure, q_{ref} .

U_{ref} is the mean velocity across the plane of maximum cross-sectional area of the casing.

q_{ref} is the dynamic pressure across the plane of maximum cross-sectional area of the casing.

A_{ref} is the area of the plane of maximum cross-sectional area of the casing.

$$U_{ref} = \frac{m_3}{\rho_3 . A_{ref}} \quad \textbf{(A1)}$$

$$q_{ref} = \frac{\rho_3 . U_{ref}^2}{2} \quad \textbf{(A2)}$$

2.) Hole Types used in combustors

Orifices have been successfully designed using different sizes and shapes, each configuration with their own advantages and disadvantages. Of the different types of holes (Fig. A1), the round punched hole is the simplest to manufacture, however, it performs poorly when the dynamic pressure in the annulus is large compared to that of the jet. The resulting shallow penetration angle does not promote mixing near the combustor centerline where the temperature profile peaks. However, round holes operate well when the annulus dynamic pressure is low. A thimble or a scoop may be used to help guide the flow and overcome the shortcomings of round punched holes. Hole shape can also be important when

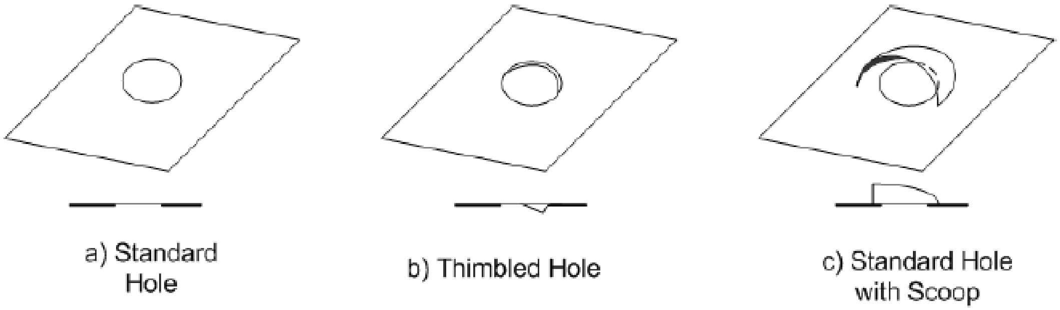


Fig. A1: Types of holes commonly used

the available space is limited. In this case, elliptical or rectangular holes with the major axis aligned along the circumferential direction can be used. Rectangular holes are seldom used as they introduce large stress concentrations near the hole corners. The use of these non-circular holes has proven to have negligible effect on turbulent mixing and jet trajectories.

3.) Thermodynamic relations used

Compressor efficiency:

$$\frac{p_{03}}{p_{01}} = \left[1 + \frac{\eta_{c.}(T_{03} - T_{01})}{T_{01}}\right]^{\frac{\gamma}{\gamma-1}} \qquad \textbf{(A3)}$$

This formula was used in section 3.3 to the compressor exit temperature, T_3.
Although this formula makes use of stagnation values, one can safely use total values due to low mach numbers.

Turbine inlet temperature:

Since the compressor and the turbine are mounted on the same shaft, mechanical efficiencies can tend to 100%.
Assuming it to be 99% and using conservation of energy, we get:

$$\Delta H_c = .99\Delta H_t$$

This can be written as:

$$\left(C_p.\Delta T\right)_c = .99\left(C_p.\Delta T\right)_t \qquad \textbf{(A4)}$$

C_p values are different at the turbine and the compressor end.

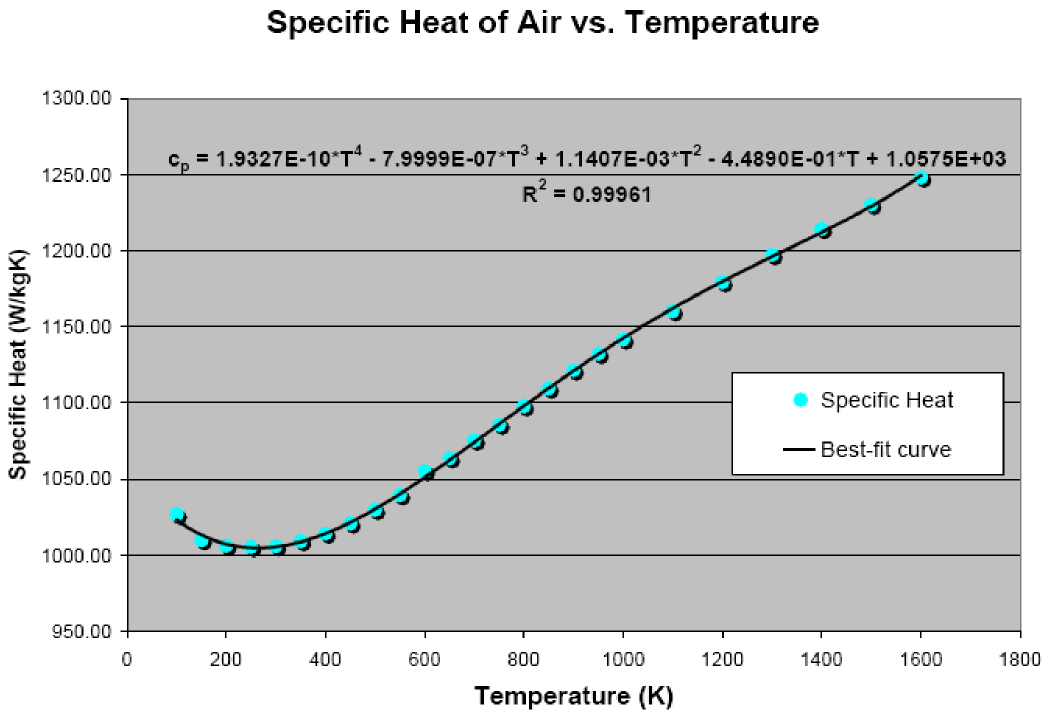


Fig. A2

Using fig.A2, C_p at 373K ≈ 1020 W/kgK and at 1000K ≈ 1140 W/kgK and γ at 373K was taken as 1.397; these were taken to be the representative temperature values for the compressor and the turbine respectively.
Also,

$$\Delta T_c = T_1.\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right) \qquad \textbf{(A5)}$$

Since T_1 is the inlet temperature, the ambient temperature, we can reach equation 5.1 using A4 and A5.

4.) Heat colors/ temper colors on stainless steel heated in air

The color formed when stainless steel is heated, either in a furnace application or in the heat affected zone of welds, is dependent on several factors that are related to the oxidation resistance of the steel. The heat tint or temper color formed is caused by the progressive thickening of the surface oxide layer and so, as temperature is increased, the colors change.

There are several factors that affect the degree of color change and so there is no a single table of color and temperature that represents all cases. The colors formed can only be used as an indication of the temperature to which the steel has been heated.

Factors affecting the color formed:

1.) Steel composition
The chromium content is the most important single factor affecting oxidation resistance. The higher the chromium, the more heat resistant the steel and so the development of the heat tint colors is delayed.

2.) Atmosphere
The level of oxygen available for the oxidation process also affects the colors formed. Normally heating in air (i.e. approx. 20% oxygen) is assumed. In welding, the effectiveness of the shielding gas or electrode coating and other weld parameters such as welding speed can affect the degree of heat tint color formed around the weld bead.

3.) Time
Laboratory tests done to establish the published heat tint color charts have usually been based on heating for one hour. As exposure time is increased, the temper colors can be expected to deepen i.e. make it appear that a higher exposure temperature may have been used.

4.) Surface finish
The original surface finish on the steel can affect the rate of oxidation and the appearance of the color formed. Rougher surfaces may oxidize at a higher rate and so could appear as deeper colors for any given set of conditions. As the colors formed are by light interference, then the smoothness of the surface can also affect the appearance of the colors formed. There is no specific data published that compares the effect of surface finish, but it is worth noting that surface finish can influence the conclusion on heating temperature, from the colors seen.

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2. Cohen, Rogers, Saravanamuttoo “Gas turbine theory” 4th edition.1996 Longman Group.

3. British stainless steels association “heat colors of stainless steels”.