MARMARA UNIVERSITY

FACULTY OF ENGINEERING

MODELING OF SIX CYLINDERS GASOLINE ENGINE USING 0D ANALYSIS PROGRAMS

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GRADUATION PROJECT REPORT

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ISTANBUL, 2022

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by

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Jun 29, 2022, Istanbul

SUBMITTED TO THE DEPARTMENT OF MECHANICAL ENGINEERING IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE

OF

BACHELOR OF SCIENCE

AT

MARMARA UNIVERSITY

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ACKNOWLEDGEMENT

First of all, I would like to thank my supervisor Assoc. Dr. Mustafa Yılmaz, for the valuable guidance and advice on preparing this thesis and giving me moral and material support.

June 2022 Gökhan Vanlı

Halil Kaan İlyasoğlu

Contents

ABSTRACT

In this study, literature studies on modeling studies and cooling systems used in engines were examined. In today's world, diesel engines are widely used in automotive industry, marine industry, and defense industry. Although recent emissions standards in the automotive industry appear to have reduced the use of diesel engines, efficient fuel consumption still makes diesel engines a good choice for other industries. All these developments have increased the importance of modeling diesel engines and their subcomponents for more precise control. As first part of this study, modeling in cylinders is considered. Furthermore, Cooling systems in internal combustion engines are considered. The purpose, features and classification of the cooling system are mentioned.

A program for computational simulation of internal combustion engines is presented. One- dimensional gas dynamics equations are used for model the flow through pipes and manifolds, and the remaining components in the engine (cylinders, valves, etc.) are modeled by using 0D models. The 0D analysis program is able to simulate spark ignition and compression-ignition, two-stroke and four- stroke, multi-cylinder, and multi-valve engines, naturally aspirated or turbo-charged. 0D program can run the program according to the desired conditions. The purpose of this study is to reach the performance curves and BSFC egg curve of the desired engine by using the program.

SYMBOLS

ε	: Thermal efficiency
R_{max}	: Maximum pressure rise rate
°C	: Celsius degrees
NO_x	: Nitrogen oxides
CO_2	: Carbon dioxide
X_b	: Mass fraction burned
$\theta(i)$	: Instantaneous crank angle
θ_0	: Spark-advance
θ_b	: Burn duration
P	: Pressure of an ideal gas
V	: Volume of the gas
m	: Mass of the gas
R	: Universal gas constant
T	: Mean gas temperature
B	: Cylinder bore
l_r	: Connecting rod length
a	: Crank radius
s	: Instantaneous distance between the crank axis and the piston pin axis
V_c	: Clearance volume
V_d	: Displaced cylinder volume
C_r	: Compression ratio
Q	: Total energy transferred into the system
W	: Work transferred out of the system
ΔU	: Change in internal energy within the system

c_v	: Specific heat of the combustion chamber gas
γ	: Specific heat ratio
η_c	: Combustion efficiency
$\frac{dQ_w}{d\theta}$	: Instantaneous change in heat loss to the cylinder walls
AF_{ac}	: Actual air-fuel ratio
h_c	: Convective heat transfer coefficient
h_r	: Radiative heat transfer coefficient
k_{gas}	: Gas thermal conductivity
Nu	: Nusselt number
ρ_{gas}	: Instantaneous cylinder gas density
$\overline{S_P}$	: Mean piston velocity
μ_{gas}	: Instantaneous gas viscosity
R_{air}	: Fluid-specific gas constant

ABBREVIATIONS

LPG	: Liquefied petroleum gas
CFD	: Computation fluid dynamics
A/F	: Air/Fuel
EGR	: Exhaust gas recirculation
SCR	: Selective catalytic reduction
PM	: Particulate matter
DEF	: Diesel exhaust fluid
LHV	: Lower heating value of the fuel
TDC	: Top dead center
ATDC	: After top dead center
BTDC	: Before top dead center
BDC	: Bottom dead center
ABDC	: After bottom dead center
BBDC	: Before bottom dead center
LES	: Lotus engine simulation
HT	: Heat transfer
RPM	: Revolutions per minute
BSFC	: Brake specific fuel consumption

LIST OF FIGURES

LIST OF TABLES

1. INTRODUCTION

1.1 Historical Development of Diesel Engine

Engines with the diesel cycle were mostly simple internal combustion engines with mechanical control. However, with the development of technology over time, the electronic control on the diesel engine has become more complex with the increase of sensors and actuators and has become a mechatronic system. Thus, mechanical systems have been replaced by electromechanical systems, and mechanical control units have been replaced by electronic control units. As seen in the figure, diesel engine systems have become more complex with new technologies added to diesel engines and new standards introduced over the years. Sensors and actuators, which are included in the diesel engine and the number of them is increasing day by day, together with the problem of the difficulty of controlling the engine. has brought. The key role in coping with the difficult control problem is the modeling of the engine. A system that is not modeled correctly cannot be controlled with high accuracy, and the whole system must be modeled with maximum accuracy for successful and precise control of diesel engines. Appropriate parameters and operating limits should then be determined for the modeled diesel engine. The importance of engine modeling clearly emerges when all components of the diesel engine system work in harmony.

1.2. Working Principle of Diesel Engine

The combustion cycle starts with the compressor connected to the same shaft as the turbine part of the turbo. The compressor uses the rotation of the shaft and creates a vacuum effect from the external environment and compresses the air inside. Compressed air is lowered in temperature to obtain a better combustion efficiency and increase the oxygen density of the air per unit volume. It passes through the intercooler. The cooled air then enters the cylinders and the intake manifold, which is the last point before combustion. When the intake valves of the cylinders are opened, the valves are closed due to the pressure difference between the air inside the cylinders and the intake manifold. flows until it is closed again. This step is the first step of the timed combustion process. In the second stage, the upward cylinder motion compresses the air. The compression ratio can differ from engine to engine and plays a key role in calculating the cylinder exit temperature. In the third stage, fuel is injected into the cylinder and combustion occurs. In some cases, the fuel is heated for a while before injection to increase combustion efficiency. Another important point of combustion efficiency is the air/fuel ratio. The most suitable air/fuel ratio in diesel engines is considered to be 14.7:1. At low rates, unburned fuel may remain in the cylinder. In the fourth stage of the combustion cycle, the exhaust gas It is ejected from the cylinders to the exhaust manifold by the upward movement of the cylinder. In the fourth stage, since there is a dead volume (top dead center) at the top of the cylinder, a small amount of exhaust gas cannot escape and remains in the cylinder and mixes with the air coming from the intake manifold in the next combustion cycle. high temperature exhaust gas increases the pressure inside the exhaust manifold. Thanks to the difference between this high pressure and the atmospheric pressure, the gas flow generated enables the turbine of the turbo to rotate. The higher the pressure inside the exhaust manifold, the faster the turbine spins. Speeds of the compressor connected to the same shaft as the turbine speed becomes equal. To control airflow through the cylinders, the exhaust manifold pressure is controlled using the wastegate. Because the exhaust manifold pressure affects the turbine speed, it affects the airflow out of the compressor. The increase in intake manifold pressure due to the amount of flow is the main factor affecting the air flow into the cylinders. factor. In simple words, the waste gate; It is a valve that reduces the exhaust manifold pressure by throwing the exhaust gas directly into the atmosphere.

1.3 Engine Modeling

One of the crucial conditioning in this engine development process is 1D engine simulation, which for many decades has been an established system in the development of internal combustion engines, starting from fairly simple methods like filling and emptying calculations to more sophisticated style to give more important computation and results.

Generally speaking, the engine model is constructed using a 1D tool which takes account of the wave dynamic engine manifold multifarious via the robust solution of the Navier-Stokes equations. The model comprises several sub models such as combustion, heat transfer, and friction losses and it must be validated with more than 100 measurements. Figure 1 shows an illustration of a 1D engine model with some in-cylinder results.

Once the model had been validated working with traditional fuels the model is ready to perform simulations and estimations with the engine working with the alternative fuel studied, such as natural gas, LPG etc. These estimations will help the engineers' decision during the development circles. Eventually, it is important to mention that the 1D engine simulation can be done alone or can be an input provider to other software similar as CFD to perform more complex calculations.

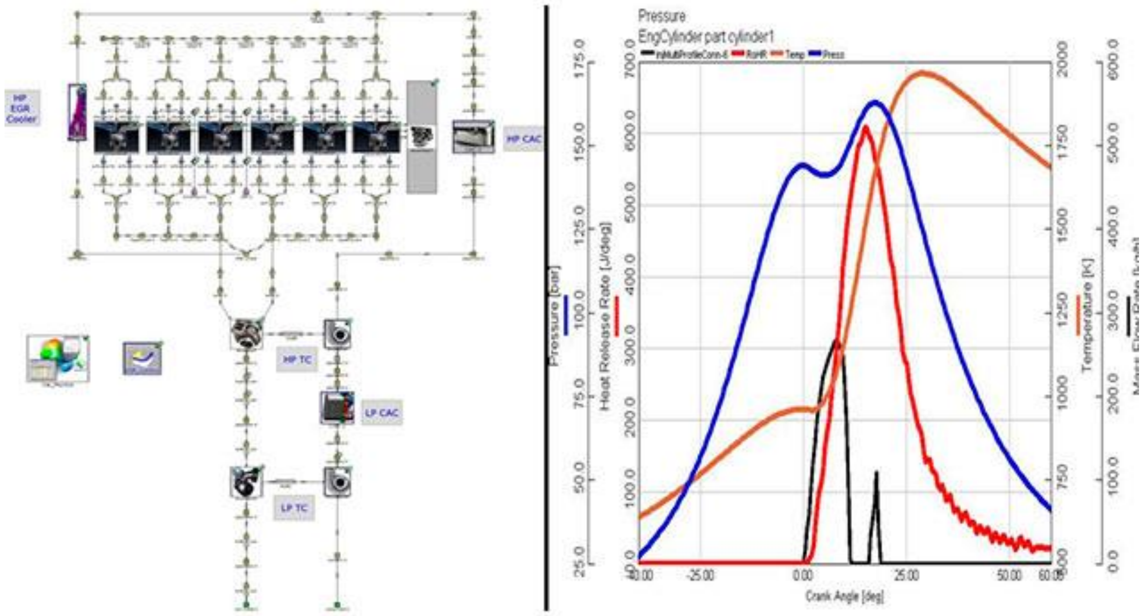


Figure 1: 1D engine model (left), In-cylinder results (right) (Barroso, 2018)

Diesel engines are the most efficient internal combustion engines and diesel engines are designed to maximize engine performance at the commercial speed. Large low-speed 2-stroke Diesel engines achieve over 50% efficiency and can be fueled with low quality liquid hydrocarbons, including bunker-oil (i.e., the remaining crude oil fraction after raffination of gasoline and diesel). Smaller 4-stroke direct injection turbocharged Diesels can reach efficiencies of approximately 40%. These engines are now one of the most likely prime movers for the efficient cars of the future. (Guzzella & Amstutz, 1998)

There are two reasons for this:

The thermal efficiency of all internal combustion engines increases with increasing compression ratio ε. Unavoidably, by compressing any working fluid its temperature is increased, such that there is a natural limit for ignitable fluids (for air-gasoline-mixtures typically around ε = 12). The Diesel engine avoids all self-ignition problems (“knock” or “detonation”) by compressing air only in its cylinders. The fuel is injected at the desired crank-angle into the cylinder and ignites spontaneously and (almost) immediately. The study Meininger and his team showed that lower cetane number fuels showed more knock tendency than high cetane number fuels for the tested aviation diesel engine. In this study, maximum pressure rise rate, or R_{max} was used as a parameter to define knock criteria for aviation diesel engine. (Meininger, et al., 2016)

- The torque of Diesel engines is controlled by changing the A/F-ratio of the mixture to be burned (its “quality”). Due to the extremely hot air in the cylinder (700 K and more), very lean mixtures are possible (relative A/F ratios typically between 1.4 and 7) such that throttling of the intake air can be completely avoided.

The intake and exhaust system of a diesel engine is related to the performance of the engine. In a four-stroke engine, the gas exchange process releases combustion gases at the end of the power stroke, bringing further air into the next intake stroke. The volumetric efficiency of an engine appears in a form similar to torque and is affected by the mass flow rate. It may be affected by various variables similar as engine speed, air–fuel ratio, compression ratio, intake and exhaust valve geometries, and intake and exhaust pipe length.

The engine was operating outside of the commercial speed, the volumetric efficiency decreases, and a lot of environmental pollutants are discharged. Greenhouse gases, sulfur oxides, and nitrogen oxides are regulated, and more regulation will be tightened in the future.

- Scrubber,
- Exhaust gas recirculation (EGR)
- Selective catalytic reduction (SCR) (after treatment)

Scrubber, exhaust gas recirculation (EGR), selective catalytic reduction (SCR), are used as devices to reduce the emission of environmental pollutants. (Kyong-Hyon & Kyeong-Ju, 2020)

Wet scrubbers are used to clean diesel exhaust emissions, by bubbling them through a liquid (usually water) to reduce their temperature and remove some soluble components and particles. Then, these emissions pass through a filter to remove further diesel particulate matter. The PM-capturing mechanism, heat transfer mechanism, and fluid mechanism of a wet scrubber are reviewed. Several parameters have a major influence on wet scrubber performance, such as inlet gas velocity. (Abdulwahid, Situ, & Brown, 2018)

Exhaust Gas Recirculation (EGR) is a system that allows the exhaust gases to be recirculated back into the intake manifold. This process leads to a significant reduction in nitrogen oxides (NO_x) emissions because it reduces the two elements underlying its production: oxygen in excess and combustion temperature. There are two types of EGR. These are iEGR, and EGR. EGR (external) is the main technology used by vehicle manufacturers for the reduction of NO_x emissions on diesel engines. It is more effective than iEGR substantially because the exhaust gases can be cooled before reentering the cylinders, the amount of exhaust gases recirculated is advanced and the flow is better controlled. (Exhaust Gas Recirculation (EGR) complete guide, n.d.)

Selective Catalytic Reduction (SCR) is an advanced active emissions control technology system that injects a liquid-reductant agent through a special catalyst into the exhaust stream of a diesel engine. The reductant source is usually automotive-grade urea, otherwise known as Diesel Exhaust Fluid (DEF). The DEF sets off a chemical reaction that converts nitrogen oxides into nitrogen, water, and tiny amounts of carbon dioxide (CO_2), natural components of the air we breathe, which is then expelled through the vehicle tailpipe.

SCR technology is designed to permit nitrogen oxide (NO_x) reduction reactions to take place in an oxidizing atmosphere. It is called "selective" because it reduces levels of NO_xusing ammonia as a reductant within a catalyst system. The chemical reaction is known as "reduction" where the DEF is the reducing agent that reacts with NO_x to convert the pollutants into nitrogen, water, and tiny amounts of CO_2. The DEF can be rapidly broken down to produce the oxidizing ammonia in the exhaust stream.

1.4 Cooling System in Internal Combustion Engines

Our aim in using cooling system in engines is to keep the material temperature in the engines below the melting point and to remove the heat quickly in order to increase the durability of the engine. Cooling channels not only remove heat from the engine, but also help us reach optimum operating temperature as soon as possible. (Adsul, Kotebavi, Bedekar, & Mishra, 2018)

The first cooling system application in engines was tested in V-type 8-cylinder engines by Cadillac Motor Company of America in 1914.As a result of many system designs and experimental studies carried out since this date, the cooling systems in engines have taken their place today.

While the air-fuel mixture in the cylinder is burning, the temperature of the burned gases rises to 2000-2500 ℃. Some of this heat is absorbed by the cylinder walls, cylinder head and pistons. Therefore, engine parts need to be cooled so that they do not reach the temperature where they can be damaged. Two types of cooling systems are used in today's vehicles, namely air cooling and water cooling.

Engines convert the heat energy generated by the combustion of the fuel in the cylinder into mechanical energy. But the heat energy of the fuel burning in the cylinders cannot be completely converted into mechanical energy, about 26% of this energy is effectively rendered useful. 41% is lost through exhaust gases and friction. The remaining 33% is damped by the engine itself.

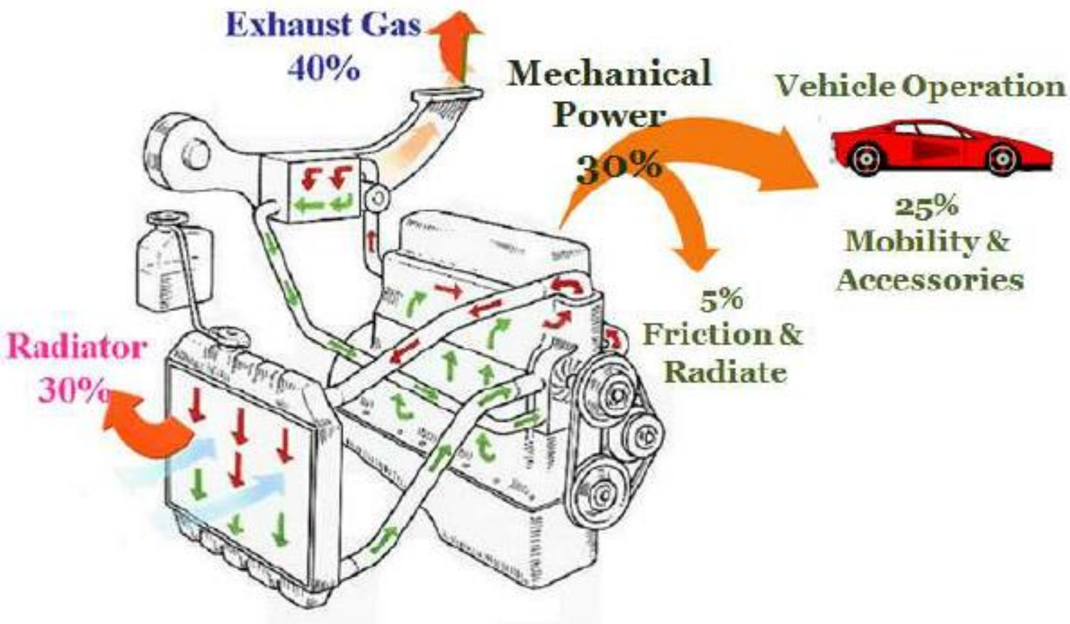


Figure 2: Typical energy split in internal combustion engines (Baatar & Kim, 2011)

1.4.1 Purpose of The Cooling System

The purpose of the cooling system in engines is to keep the engine at the temperature at which it can operate in the most efficient way under all operating conditions and at all engine speeds.

While the air-fuel mixture in the cylinder is burning, the temperature of the burned gases rises to 2000-2500 °C. Some of this heat is absorbed by the cylinder walls, cylinder head and pistons. therefore, engine parts need to be cooled so that they do not reach temperatures where they can be damaged. In addition, the temperature of the cylinder walls should not exceed 250°C. If this temperature limit value is exceeded, the lubricating oil film layer on the cylinder walls may be damaged and it may lose its lubricating property. However, it is difficult to match the characteristics of the oil with this temperature limit value. If the cylinder bores and cylinder head are overcooled, the thermal efficiency of the engine will decrease. Another purpose of the cooling system is to cool 30% to 35% of the heat released in the combustion cycle and to throw it into the air with the help of heat transfer methods.

When we look at the conversion of heat energy to mechanical energy in general, it is desirable to have low cylinder wall and cylinder head temperatures at the beginning of the compression stroke in order to get the highest efficiency from the engine.

High operating temperatures cause oil films to break. In engines that continue to operate in this way, adhesions occur between metal parts. The event that causes the metal to deform over time, caused by working at high temperatures, is friction. An improved cooling system is needed to avoid friction or low efficiency. In this way, the temperature value at which the motors can operate at the desired temperature, that is, with the highest efficiency, can be achieved. As mentioned above, this temperature value is the temperature of the cylinder walls at the limit of 250 °C. Thus, 10 times more efficiency can be obtained from cylinder gases during ignition.

The two main heat transfer fluids in typical cooling system are coolant and air. Under hood air flow when approaching radiator is highly non- uniform, thus 3D computational fluid dynamics is required to estimate air mass flow rates.

Automotive designers should design a robust engine cooling system which is working well in both normal and severe driving conditions. When vehicles are keyed- off suddenly after some distance of driving, the coolant temperature tend to increase dramatically. This is because soaking heat at engine could not be transferred away timely, as water pump and cooling fan have stop working after keyed- off.

Engine cooling system is very crucial for a vehicle as it ensures engine always running at its optimum operating temperature. In water cooling system, coolant flows in water jacket and transfers heat away from cylinder head and body. At heat exchanger/ radiator, hot coolant will transfer the heat to cooling air. A complete numerical study of engine cooling system consists of 2 portions, air side and coolant side. (Pang, Masjuki, Hazrat, & Kalam, 2011)

1.4.2 Features of The Cooling System

In internal combustion engines, the temperature at the time of combustion rises up to 2000-2500 ℃ in a short time. Before the parts covering the combustion chamber have time to feel this temperature, they come into contact with the fresh cold mixture entering the intake valve or, in diesels, with the air. Since the walls of the combustion chamber are always in contact with hot and cold gases, they feel an average temperature.

This average temperature varies according to the type, load, and constructive structure of the engine. This temperature is usually between 500-1000 ℃. This temperature is called the average gas temperature.

Despite the advantage of periodic operation of internal combustion engines, the average gas temperature that occurs is much higher than the temperatures at which the material starts to lose its mechanical strength and the viscosity of the lubricating oils. To eliminate this temperature and for the following reasons, internal combustion engines need to be cooled.

- It is ensured that the delivery efficiency of the fresh mixture entering the cylinder is at its maximum value.
- It eliminates the premature ignition and knocking of the fresh mixture entering the cylinder.
- For a good lubrication supply, the temperature that keeps the oil viscosity constant is maintained.

- Expansion of moving parts such as pistons, rings and cranks under excessive heat is eliminated
- Balanced operation of engine parts in harmony is ensured.

1.5 Classification of Engine Cooling System

Cooling systems in internal combustion engines appear as air-cooled and water-cooled systems. Water is generally preferred to air-cooled systems because of its high specific heat capacity, density, and thermal conductivity. Compared to less volumetric flow and reduced temperature difference, water appears to transmit heat over longer distances. (Adsul, Kotebavi, Bedekar, & Mishra, 2018)

1.5.1 Air Cooled Engine

Most modern cars use water-cooled engines with radiators, water pumps and hoses that circulate a water and coolant mixture throughout the engine. The heat from the engine is transferred to the coolant, and then the coolant is cooled in the radiator and sent back around again.

Air-cooled engines want none of this. But all engines are technically air-cooled because even water-cooled engines use air to cool the fluid in the radiator. In air-cooled engines, fins or extended surfaces are provided on the cylinder walls, cylinder head, etc. Heat generated due to combustion in the engine cylinder will be conducted to the fins and when the air flows over the fins, heat will be dissipated to air.

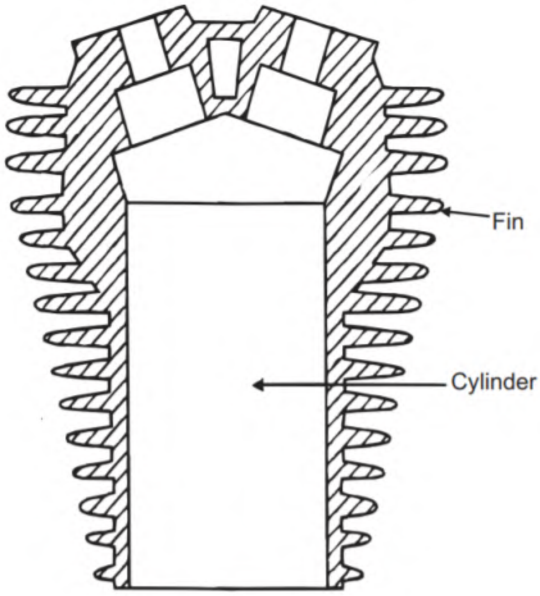


Figure 3: Cooling Fins of an Engine Cylinder (KRISHNAVENI, SOWMYA, & SUDHAKAR, 2014)

In air-cooled engines, there is advantages and disadvantages. To mention the advantages, air-cooled engines don't have coolant leakage problems. Typically, they're lighter than liquid-cooled engines, too, because they have fewer parts. They also warm up a lot faster than liquid-cooled engines and don't have any risk of the coolant freezing. On the other hand, if we look at the disadvantages', air-cooled engines are more likely to overheat. They can also be more expensive to build, and the large fans used to cool the engine can take away a lot of power.

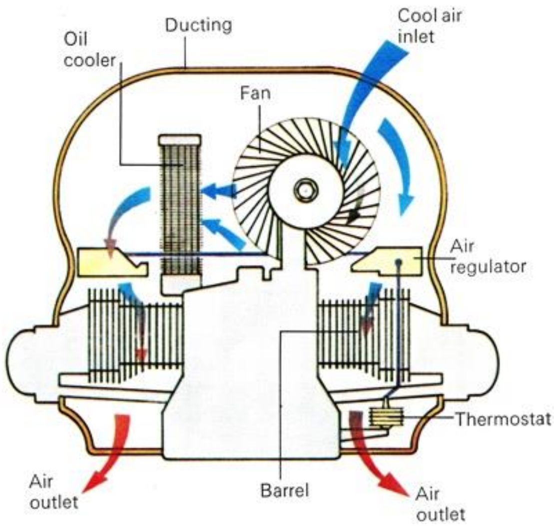


Figure 4: The cooling system used on the most known air-cooled engine, the VW Beetle. (uniquecarsandpart, 2022)

1.5.2 Water Cooled Engine

In water-cooling system, cooling water jackets are provided around the cylinder, cylinder head, valve seats etc. The water when circulated through the jackets, it absorbs heat of combustion. This hot water will then be cooling in the radiator partially by a fan and partially by the flow developed by the forward motion of the vehicle. The cooled water is again recirculated through the water jackets.

There are two types of water-cooling system:

Thermo Siphon System

In this system the circulation of water is due to difference in temperature (i.e., difference in densities) of water. So, in this system pump is not required but water is circulated because of density difference only.

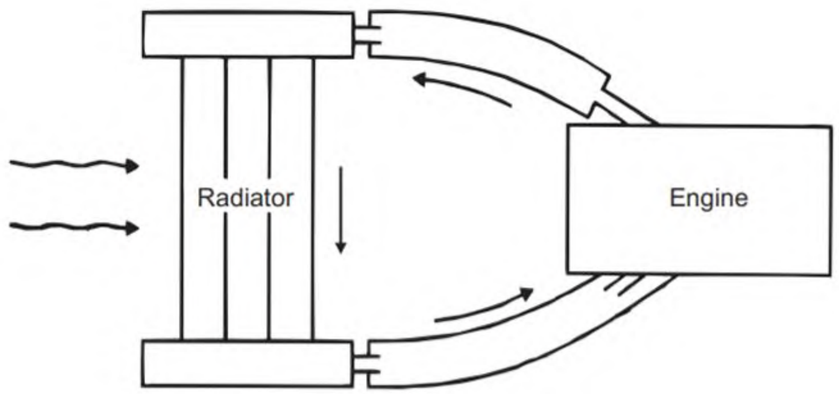


Figure 5: Thermo Siphon System of Cooling. (Prudhvi, Vinay, & Babu, 2013)

Pump Circulation System

In this system circulation of water is obtained by a pump. This pump is driven by means of engine output shaft through V-belts.

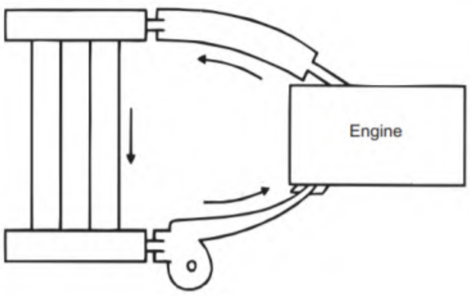


Figure 6: Pump Circulation System. (Prudhvi, Vinay, & Babu, 2013)

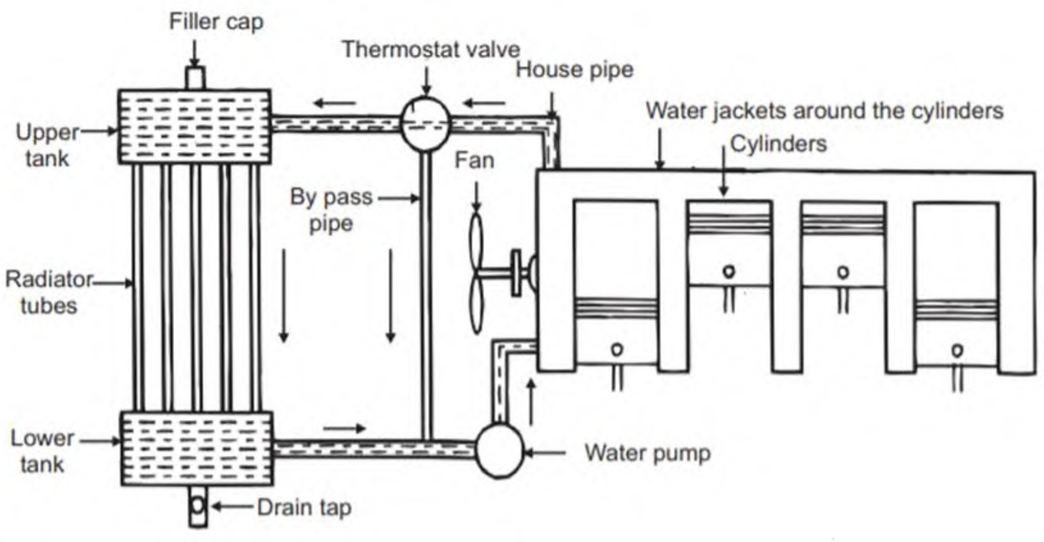


Figure 7: Water Cooling System of a 4-cylinder Engine. (ZAID, 2014)

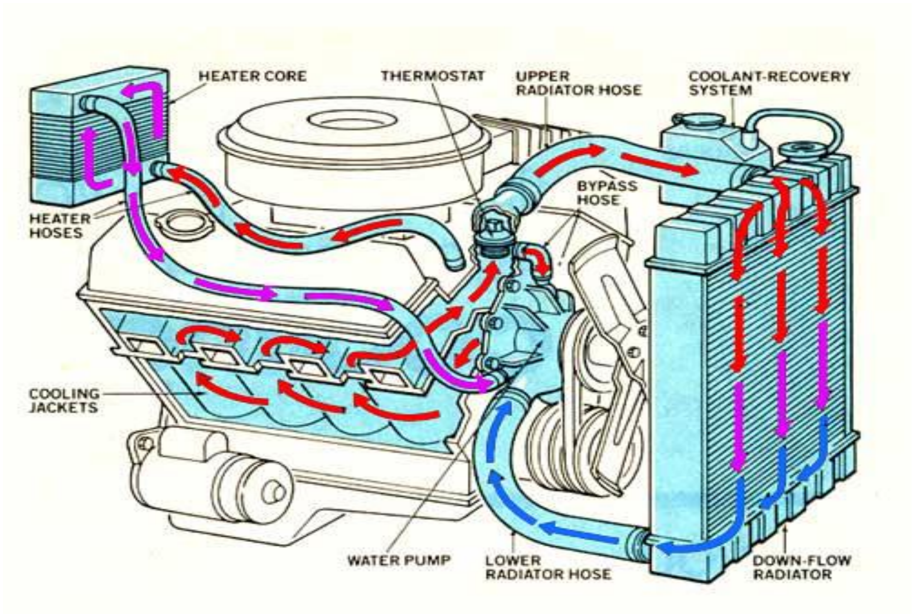


Figure 8: How water flows in an Internal Combustion Engine (Cruz, 2017)

Water cooling system mainly consists of:

- **Radiator**: It mainly consists of an upper tank and lower tank and between them is a core. The upper tank is connected to the water outlets from the engine's jackets by a hose pipe and the lover tank is connect to the jacket inlet through water pump by means of hose pipes.
- **Thermostat valve**: It is a valve which prevents flow of water from the engine to radiator, so that engine readily reaches to its maximum efficient operating

temperature. After attaining maximum efficient operating temperature, it automatically begins functioning. Generally, it prevents the water below 70°C.

- **Water pump**: It is used to pump the circulating water. The pump is driven by means of engine output shaft only through belts. When it is driven water will be pumped.
- **Fan**: It is driven by the engine output shaft through same belt that drives the pump. It is provided behind the radiator, and it blows air over the radiator for cooling purpose.
- **Water Jackets**: Cooling water jackets are provided around the cylinder, cylinder head, valve seats and any hot parts which are to be cooled. Heat generated in the engine cylinder, conducted through the cylinder walls to the jackets. The water flowing through the jackets absorbs this heat and gets hot. This hot water will then be cooled in the radiator.
- **Antifreeze mixtures**: If the water used in the radiator freezes because of cold climates, then ice formed has more volume and produces cracks in the cylinder blocks, pipes, and radiator. So, to prevent freezing antifreeze mixtures or solutions are added in the cooling water.

NO	Air-Cooling System	Water-Cooling System
1	The design of this system is simple and less costly.	The design of this system is complicated and more costly.
2	The weight of the cooling system is very less.	The weight of the cooling system in much more.
3	The fuel consumption is more.	The fuel consumption is less.
4	There is no danger of leakage or freezing of the coolant.	There is a danger of leakage or freezing of the coolant.
5	Its installation and maintenance are very easy and less costly.	Its installation and maintenance in difficult and more costly.

6	It works smoothly and continuously. Moreover, it does not depend on any coolant.	If the system fails, it may cause serious damage to the engine within a short time.
7	Air cooling system is not suitable for multi cylinder engines.	This system can be employed in multi cylinder engines satisfactorily.

Table 1 :Differences between Air Cooling System and Water-Cooling System.

2. Material and Method

2.1 Internal Combustion Engine Modeling

2.1.1 Single-Zone Engine Model

After making decisions regarding the modeling techniques used in this research, a single-site model was developed. The single-zone model formed the basis from which the two-zone model was derived and was used to predict the power characteristics of the engine as well as the bulk cylinder gas temperature and pressure.

The simplest approach to engine modeling is to treat the cylinder contents as a single fluid or region. The single-zone model considers the burnt and unburned gases, residual gases, and unburned HC in the cylinder as an ideal gas at uniform pressure. Single-zone models typically use the Wiebe function to represent chemical energy release as a function of crank angle. (Klein, 2004) The Weibe function has a characteristic "S-shape" and is defined as follows:

$$X_{b(\theta)} = 1 - exp\left[-a\left(\frac{\theta(i)-\theta_0}{\theta_b}\right)^{k+1}\right] \quad (2.1)$$

where **a** and **k** are adjustable constants, $\boldsymbol{\theta(i)}$ is the instantaneous crank angle, $\boldsymbol{\theta_0}$ is the spark-advance and $\boldsymbol{\theta_b}$ is the burn duration. The burn profile is engine-specific, and the constants and k can be adjusted to tune a profile to a specific application.

The ideal gas law forms the basis for the single-zone engine model, and is defined as:

$$PV = mRT \quad (2.2)$$

Where **P** is the pressure of an ideal gas, **V** is the volume of the gas, **m** is the mass of the gas, **R** is the universal gas constant, and **T** is the mean gas temperature. The cylinder gas volume, V, in a reciprocating, IC engine can be related to the engine geometry as a function of crank angle :

$$V(\theta) = V_c + \frac{\pi B^2}{4}(l_r + a - s) \tag{2.3}$$

where **B** is the cylinder bore, $\boldsymbol{l_r}$ is the connecting rod length, **a** is the crank radius, **s** is the instantaneous distance between t1he crank axis and the piston pin axis, and $\boldsymbol{V_c}$ is the clearance volume. The clearance volume is expressed as:

$$V_c = \left(\frac{V_d}{C_r - 1}\right) \tag{2.4}$$

Where $\boldsymbol{V_d}$ is the displaced cylinder volume and $\boldsymbol{C_r}$ is the compression ratio. Figure 9 shows a diagram of these figures and their relationship to engine geometry.

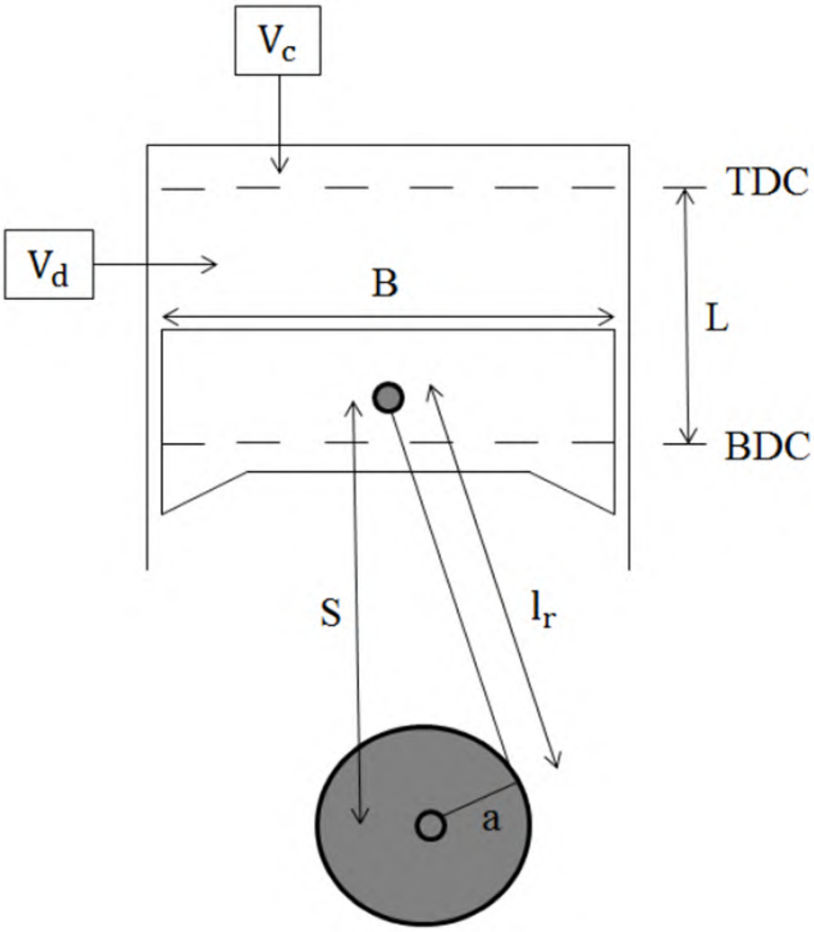


Figure 9: . Engine geometry and variables used in modeling.

In differentiating equation 2.2 with respect to , equation 2.5 is obtained:

$$\frac{dP}{d\theta}V + P\frac{dV}{d\theta} = mR\frac{dT}{d\theta'} \tag{2.5}$$

and upon rearranging equation 2.5 and solving for **dP**, expression 2.6 is obtained:

$$\frac{dP}{d\theta} = \left(-\frac{P}{V}\right)\frac{dV}{d\theta} + \left(\frac{P}{T}\right)\frac{dT}{d\theta} \tag{2.6}$$

where **P**, **V**, and **T** are instantaneous values modeled relative to the engine's crank angle. The same process can be applied to the first law of thermodynamics, which is expressed as:

$$\Delta U = Q - W \tag{2.7}$$

where **Q** is the total energy transferred into the system, **W** is the work transferred out of the system, and $\Delta \boldsymbol{U}$ is the change in internal energy within the system. In differentiating equation 2.7 with respect to $\boldsymbol{d\theta}$, expression 2.8 can be obtained:

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} - \frac{dW}{d\theta} = mc_v\left(\frac{dT}{d\theta}\right) \tag{2.8}$$

where $\boldsymbol{c_v}$ is the specific heat of the combustion chamber gas. Upon dividing the specific heat by the universal gas constant, equation 3.9 can be obtained:

$$\frac{c_v}{R} = \frac{c_v}{c_p - c_v} = \frac{1}{\gamma - 1} \tag{2.9}$$

where $\boldsymbol{\gamma}$ is the specific heat ratio.

Equations 2.10 and 2.11 can be used to describe the formation of work and the net heat input:

$$\frac{dW}{d\theta} = P\left(\frac{dV}{d\theta}\right) \tag{2.10}$$

$$\frac{dQ}{d\theta} = \eta_c LHV\left(\frac{dX_b}{d\theta}\right) - \frac{dQ_w}{d\theta} \tag{2.11}$$

where $\boldsymbol{\eta_c}$ is the combustion efficiency, **LHV** is the lower heating value of the fuel, $\left(\frac{\boldsymbol{dX_b}}{\boldsymbol{d\theta}}\right)$ is the instantaneous change in mass-fraction burned and $\frac{\boldsymbol{dQ_w}}{\boldsymbol{d\theta}}$ is the instantaneous change in heat loss to the cylinder walls. Upon substituting equations 2.9 and 2.10 into equation 2.8, the instantaneous change in temperature is defined as

$$\frac{dT}{d\theta} = T(\gamma - 1)\left[\left(\frac{1}{PV}\right)\left(\frac{dQ}{d\theta}\right) - \left(\frac{1}{V}\right)\left(\frac{dV}{d\theta}\right)\right] \tag{2.12}$$

where the gross change in input heat $\frac{dQ}{d\theta}$ can be found in equation 2.11. With the change in temperature as a function of crank angle designated, the heat input from the fuel can be used to find the change in pressure as a function of crank angle:

$$Q_{in} = \eta_c LHV \left(\frac{1}{AF_{ac}}\right)\left(\frac{P}{RT}\right) V_d \qquad (2.13)$$

where $\boldsymbol{AF_{ac}}$ is the actual air-fuel ratio. Lastly, the change in pressure is defined as:

$$\frac{dP}{d\theta} = \left(\frac{-\gamma P}{V}\right)\left(\frac{dV}{d\theta}\right) + \left(\frac{\gamma-1}{V}\right) Q_{in} \frac{dX_b}{d\theta} + (\gamma - 1)\left(\frac{1}{V}\right)\left(\frac{dQ_w}{d\theta}\right) \qquad (2.14)$$

2.1.2 Wiebe Function

The Wiebe function curve has a characteristic S-shaped curve and is commonly used to characterize the combustion process. The mass fraction burned profile grows from zero, where zero mass fraction burn indicates the start of combustion, and then tends exponentially to one indicating the end of combustion. The difference between those two ends is known as the duration of combustion. Although the Wiebe function simple and robust in specifying the combustion process, there are inherent issues.

The main advantage of the Wiebe functions is that they are normalised by the combustion duration. Thus, the user may quickly change the total combustion duration and be confident of achieving a realistic heat release rate.

The single part Wiebe function is used by default for all combustion systems. The model coefficients are set as a function of the fuel type. The default coefficients are:

Fuel	a	m
Gasoline	10	2
Diesel	6.9	0.5
Methane	5	2.2
Methanol	10	2

Table 2:Lotus’s default values for Wiebe function.

Combustion Duration

The definition of the combustion duration is a function of the type of fuel being used. It is notoriously difficult to reliably measure both the start and end of combustion in spark ignited gasoline and methanol fuelled engines. An approach has therefore been adopted by which the combustion duration of these engines is defined as the number of crank degrees between 10% and 90% mass fraction burnt. For diesel (and some gas) engines however the start and end of combustion are more easily obtained. Thus, for all other engines the combustion duration is defined as the number of crank degrees between 0 and 100% mass fraction burned.

Default combustion durations are available for several fuel / combustion system combinations. These are mainly intended to allow the user to quickly develop a new model and should not be relied upon for accurate modelling of each combustion system / fuel type combination. The available defaults are;

Fuel	Carburettor	Port Injected	Direct Injection	Indirect Injection
Gasoline	Eq 2.15	Eq 2.15	Eq 2.15	Eq 2.15
Diesel	-	-	Eq 2.16	Eq 2.17
Methane	-	-	-	-
Methanol	Eq 2.15	Eq 2.15	Eq 2.15	Eq 2.15

Table 3:Lotus's default values depends on injection types.

$$10-90\%(deg) = 20\left(\frac{Bore}{Stroke}\right) + 0.6\left(\frac{Speed(rpm)*Stroke(m)}{30} - 11\right) \qquad (2.15)$$

$$0-100\%(deg) = 30 + 0\left(\frac{50}{Trapped\ AFR*0.06691*0.7}\right) \qquad (2.16)$$

$$0-100\%(deg) = 30 + 0\left(\frac{50}{Trapped\ AFR*0.06691*0.85}\right) \qquad (2.17)$$

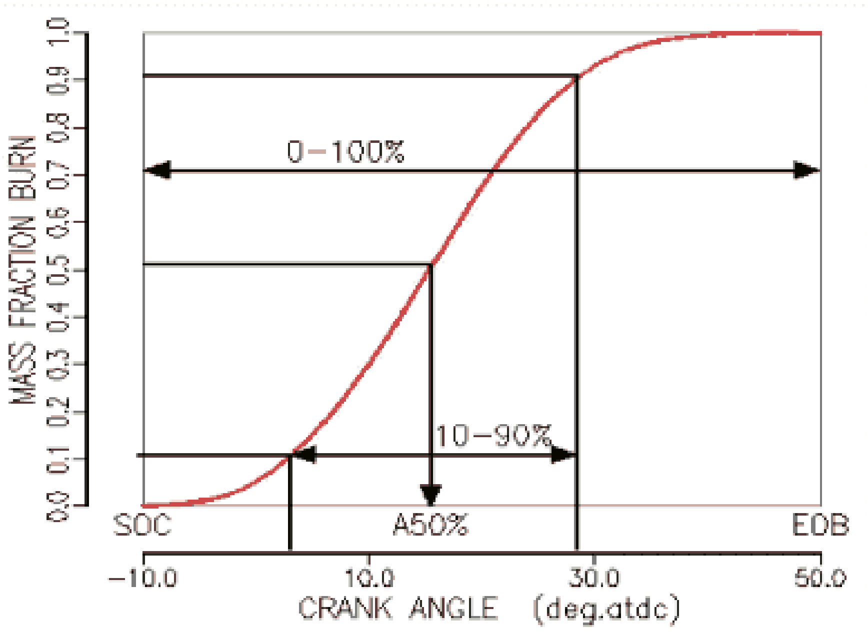
Combustion Phasing

The definition of the combustion phasing is a function of the type of fuel being used. It is notoriously difficult to reliably measure both the start of combustion in spark ignited gasoline and methanol fuelled engines. An approach has therefore been adopted by which the combustion phasing of these engines is defined as the number of crank degrees after TDC firing at which 50% of the fuel has been burnt. (Note a negative combustion phasing value for these engines implies an angle of 50% burn before TDC). For diesel (and some gas) engines however the start and end of combustion are more easily obtained. Thus, for all other engines the combustion phasing is defined as the number of crank degrees before TDC at which combustion starts. (Note a negative combustion phasing value for these engines implies a start of combustion timing after TDC).

Default combustion phasings are available for several fuel / combustion system combinations. These are mainly intended to allow the user to quickly develop a new model and should not be relied upon for accurate prediction of performance or maximum cylinder pressure. The available combustion phasing defaults are;

Fuel	Carburettor	Port Injected	Direct Injection	Indirect Injection
Gasoline	A50%. atdc	A50%. atdc	A50%. atdc	A50%. atdc
Diesel	-	-	SOC-5. btdc	SOC-0. btdc
Methane	-	-	-	-
Methanol	A50%-10. atdc	A50%-10. atdc	A50%-10. atdc	A50%-10. atdc

Table 4:Lotus default combustion phasings.



Graph 1: Crank Angle vs Mass Fraction Burn

2.1.3 Cylinder Heat Transfer

In Lotus Engine Simulation, Heat transfer to and from the cylinder gases are calculated at every crank angle increment. These calculations require a knowledge of the wall temperatures and surface heat transfer coefficient.

Cylinder Wall Temperatures

The cylinder wall temperatures are either specified explicitly by the user or calculated via a simple one-dimensional heat transfer calculation for the cylinder head and liner walls. The cylinder walls are assumed to have a wall thickness that is a directly proportional to the bore diameter;

- Head flame face thickness = 0.13 x Bore
- Liner thickness = 0.07 x Bore

The thermal conductivity of the walls is specified by the material index. The assigned wall material properties are;

Material	Conductivity$\left(\frac{W}{m*k}\right)$
Cast Iron	45
Aluminium	150
Steel	48

Table 5: Conductivities of Materials

2.1.4 Annand's Heat Transfer Model

Annand's and Woschni's heat transfer models differed in the fact that Annand's approach separated the convective and radiative terms. Annand's method solved for the heat transfer coefficient by assuming pipe-like fluid dynamics, and using the in-cylinder density, and Reynolds and Nusselt numbers as functions of time. (Blair, 2009)

Using Annand's method, Newton's law of cooling can be broken into convective and radiative terms as follows:

$$\delta Q_w = (h_c + h_r)A_w(T - T_w)dt \tag{2.18}$$

where $\boldsymbol{h_c}$ is the convective heat transfer coefficient and $\boldsymbol{h_r}$ is the radiative heat transfer coefficient. The convective heat transfer coefficient can be extracted from the relationship between the Nusselt number and fluid properties:

$$h_c = \frac{k_{gas}Nu}{B} \tag{2.19}$$

where $\boldsymbol{k_{gas}}$ is the gas thermal conductivity, **Nu** is the Nusselt number, and **B** is the cylinder bore.

With an iterative solver, the thermal conductivity of the cylinder gas can be modeled using a polynomial curve-fitting of experimental data. Blair suggests using the curve fitted equation:

$$k_{gas}\left(\frac{W}{m*k}\right) = 6.1944 * 10^{-3} + 7.3814 * 10^{-5}T(K) - 1.2491 * 10^{-8}T(K)^2 \tag{2.17}$$

where **T** is the instantaneous cylinder temperature **[K]** , and the thermal conductivity is output in units of $\left[\frac{W}{m*k}\right]$.

The Nusselt number can be described relative to the Reynolds number and the type of engine:

$$Nu = aRe^{0.7} \tag{2.20}$$

where **a** is a constant having a value of 0.26 for a two-stroke engine and 0.49 for a four-stroke engine, and **Re** is the instantaneous Reynolds number. The Reynolds number is expressed as:

$$Re = \frac{\rho_{gas}\overline{S_p}B}{\mu_{gas}} \tag{2.21}$$

where $\boldsymbol{\rho_{gas}}$ is the instantaneous cylinder gas density, $\overline{\boldsymbol{S_p}}$ is the mean piston velocity and $\boldsymbol{\mu_{gas}}$ is the instantaneous gas viscosity. Since the model assumes ideal gas behavior, the cylinder gas density can be found by rearranging the ideal gas law:

$$\rho_{gas} = \frac{P}{R_{air}T} \tag{2.22}$$

where $\boldsymbol{R_{air}}$ is the fluid-specific gas constant, and an assumed value of 287$\left[\frac{J}{kg*K}\right]$ was used for this variable.

As with the thermal conductivity, the cylinder gas viscosity was modeled using empirical equations. According to Blair, the cylinder gas viscosity can be expressed as:

$$\mu_{gas}\left(\frac{kg}{m*s}\right) = 7.457*10^{-6} + 4.1547*10^{-8}T(K) - 7.4793*10^{-12}T(K)^2 \tag{2.23}$$

where the instantaneous cylinder temperature must be provided in units of [K], and the gas viscosity is output in units of $\left[\frac{kg}{m*s}\right]$.

Although the radiative heat transfer coefficient is small, it was decided that radiation should be included in considering overall heat losses in the model. The radiative heat transfer coefficient is defined as:

$$h_r\left(\frac{W}{m^2*K}\right) = 4.25*10^{-9}\left(\frac{T^4 - {T_w}^4}{T - T_w}\right) \tag{2.24}$$

where the instantaneous cylinder temperature and wall temperature must be provided in units of [K]. With known pressure and temperature traces from the single-zone calculations, Annand's method could then be used to calculate heat losses.

2.2 Lotus Engine Simulation (LES)

2.2.1 Introduction

The software selected for the simulation portion of the research is the Lotus Engine Simulation program. The software predicts the gas flow regimes and uses thermodynamics modeling to extract the overall performance of the engine in question. The program can simulate two-stroke and four-stroke engines, spark-ignition, or diesel combustion systems, with a variety of fuel delivery options, including direct, indirect, or port injection.

2.2.2 Data Module

A simulation file begins with a blank interface that is user-populated with several elements – there are six types, listed in Table 6. Flow devices are then used to connect basic elements to each other or to additional flow devices. These devices are listed in Table 7.

Element	Description	Additional Features
Cylinder	Zero-dimensional Element	Combustion and HT model
Plenum	Zero-dimensional Element	Heat transfer model
Pipe	One-dimensional Element	HT and friction model
Inlet	Infinite source of inlet gas	Input pressure and temperature
Outlet	Exhaust boundary	Output pressure
Close End	Special geometric element	n/a

Table 6: Major element components in the LES program

Flow Device	Description
Valve	Can be either camshaft-or self-actuated
Throttle	Defines flow area and discharge coefficient
Compressor	Turbo or supercharger; compressor map fully defined
Turbine	Turbocharger turbine – map fully defined
Charge-Cooler	Heat exchanger with pressure loss and heat transfer

Table 7:Flow components in the LES program

An additional composite element, called a Super Element, is available for flow through complex devices such as silencers on intake/exhaust paths, or catalytic converters. Since no Super Elements were used in the simulations that follow, they will not be defined. Rotating elements in the model may be linked to loads or other rotating elements using mechanical links. Control Elements are comprised of sensors and actuators.

2.2.3 Data Module Population

The LES software allows the user to build a complete engine model in the virtual environment by populating an input data file saved in the working folder with a. sim extension. Each engine component is expressed by icons that have specific input fields. This section will be an overview of available user inputs or possible default values where data are not available.

2.2.4 Result Module

Simulation results are available in a variety of ways to the user. Among them are results available as text and graphs (either cycle averaged, or crank angle based). They are saved with LES proprietary extensions (. prs,. mrs, etc.) and can be opened through the program menus once simulations conclude. If the program ended unexpectedly during a single simulation, the resulting data would be incomplete or corrupted and required restarting the simulation process. This turned out to be a frustrating exercise with temporal-heavy parametric runs of 24-48 hours, as the program encountered various faults, or the computer would be restarted because of independent operating system updates. As such, the program would benefit from being run on a laboratory-type server, where computing power is more readily available, and any system restarts would generally be scheduled, and the user could better forecast them.

2.2.5 Text Files

At the conclusion of each single-parameter simulation, a text file is generated that contains the input data, and the results data for each operating condition. A tabulated data that is spreadsheet exportable is also generated, and it is this summary of results which provides the bulk of the result information used in this report. With parametric results, the program sets up multiple text files, one for each set of unique input prompts. For example, in the parametric results discussed in Chapter 7, a 6x7 parametric simulation run will generate 42 sets of text files, numbered consecutively. The data from each one of these files is then separately extracted and disseminated. It was this area of results interpretation which proved to be the most time-consuming process of using LES. Part of the future recommendations for the use of LES is encouraging users to develop Excel Virtual Basic or Python-based methods of automatically extracting specific sets of data from multiple large text files, a task that proved beyond the technical skill of the author. All datasets used here were manually extracted and populated into Excel spreadsheets for analysis. From the LES software, the data sections included the following information:

- Gas flow data is given and includes such data as air mass flows per cycle, scavenge ratios and efficiencies.
- Fueling data includes mass per cycle and equivalence ratio.
- Trapped conditions data gives in-cylinder pressures, temperatures residuals and phase angles.
- Performance data includes mean effective pressures, efficiencies, and power.
- Consumption Data gives Specific fuel consumption and thermal efficiency values.
- Heat Transfer Data includes heat loss rate and fraction of fuel energy (both per cylinder).

One additional feature of the text files is that they contain a copy of the input data, and this can be used to generate a simulation model.

2.2.6 Lotus Simulation

The engine to be simulated using the LES program was chosen as the 2JZ-GTE engine. Engine specifications are given in the figure below.

ENGINE – 2JZ-GTE ENGINE 77

■ENGINE SPECIFICATIONS AND PERFORMANCE CURVE

Item \ Engine			2JZ-GTE	7M-GTE
No. of Cyls. & Arrangement			6-Cylinder, In-Line	←
Valve Mechanism			24-Valve DOHC, Belt Drive	←
Combustion Chamber			Pentroof Type	←
Manifolds			Cross-Flow	←
Fuel System			SFI*[1] [EFI]	MFI*[2] [EFI]
Displacement		cm³ (cu.in.)	2997 (182.9)	2954 (180.3)
Bore X Stroke		mm (in.)	86.0 X 86.0 (3.39 X 3.39)	83.0 X 91.0 (3.27 X 3.58)
Compression Ratio			8.5 : 1	8.4 : 1
Firing Order			1-5-3-6-2-4	←
Max. Output		[SAE-NET]	239 kW @ 5600 rpm (320 HP @ 5600 rpm)	173 kW @ 5600 rpm (232 HP @ 5600 rpm)
Max. Torque		[SAE-NET]	427 N·m @ 4000 rpm (315 ft·lbf @ 4000 rpm)	344 N·m @ 3200 rpm (254 ft·lbf @ 3200 rpm)
Valve Timing	Intake	Open	3° BTDC	6° BTDC
		Close	50° ABDC	←
	Exhaust	Open	52° BBDC	53° BBDC
		Close	4° ATDC	3° ATDC
Fuel Octane Number		RON	96	91
Oil Grade			API SG, EC-II	API SG

*[1]: SFI (Sequential Multiport Fuel Injection)
*[2]: MFI (Multiport Fuel Injection)

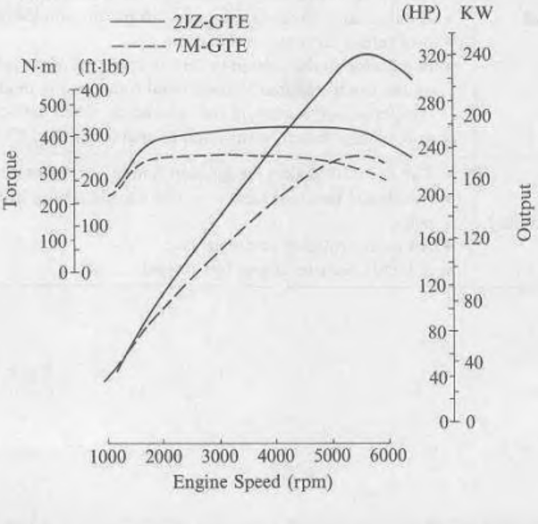


Figure 10:2JZ-GTE Engine Specifications

The simulation diagram was created as shown following figures with the specified engine characteristics.

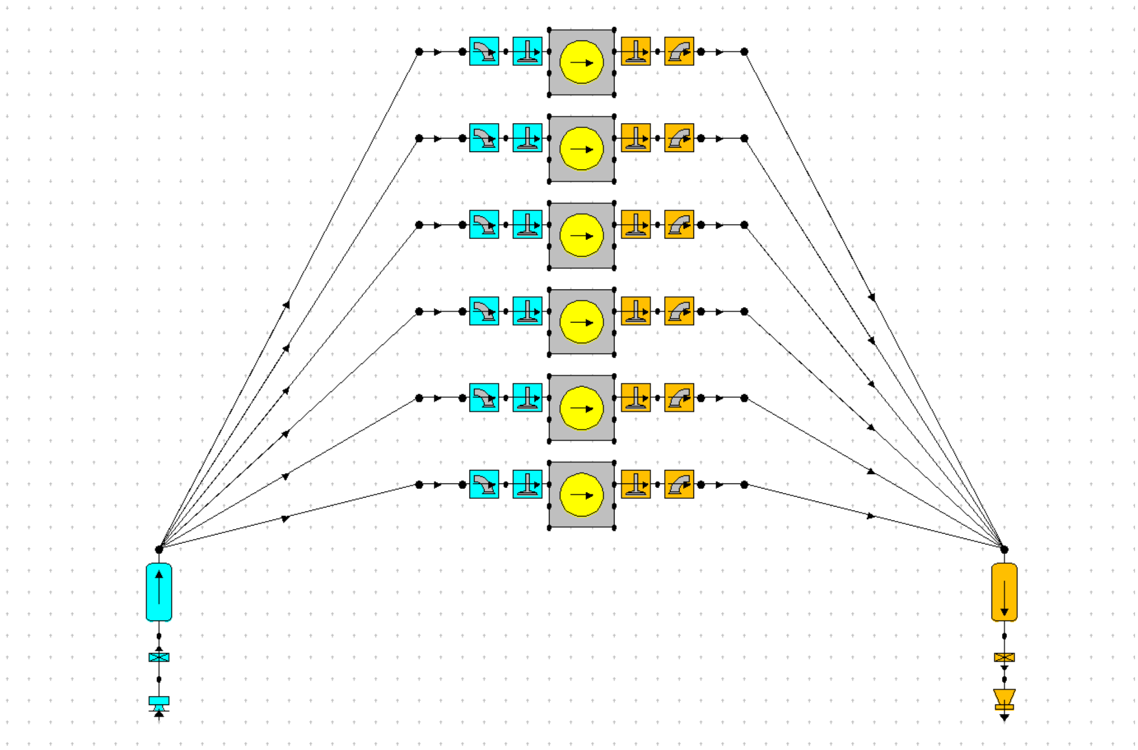


Figure 11: 2JZ-GTE diagram

Label	default cylinder
Bore (mm)	86,0000
Stroke (mm)	86,0000
Cyl Swept Volume (l)	0,49956
Total Swept Volume (l)	2,99734
Con-rod Length (mm)	142,00
Pin Off-Set (mm)	0,00
Compression Ratio	8,50
Clearance Volume (l)	0,066608
Phase (ATDC)	0,00
Combustion Model	
Open Cycle HT	
Closed Cycle HT	
Surface Areas	
Surface Temperatures	
Scavenge-Cylinder	

Figure 12:Cylinder Data Sheet

Bore, stroke, con-rod Length lengths, compression ratio, and phase angle are entered here

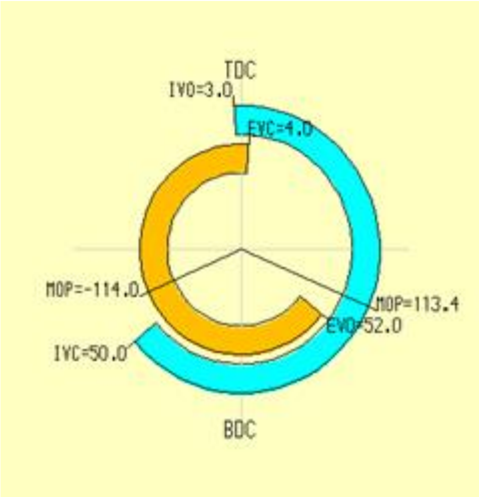


Figure 13:Valve timings

Valve timings are entered here.

Test Point	Speed (rpm)	Combustion Option	Combustion Efficiency (0-1)	Maldistribution Factor	Fueling Option	Cylinder Data	Cylinder No.	Trapped Air Fuel Ratio	Equivalence Ratio	Fueling Rate (mm³/inj)
1	1000	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,100	
2	1500	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,100	
3	2000	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	
4	2500	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	
5	3000	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	
6	3500	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	
7	4000	Efficiency + Maldistribution	0,9500	0,0100	Equivalence Ratio	Common	All		1,000	
8	4500	Efficiency + Maldistribution	0,9500	0,0100	Equivalence Ratio	Common	All		1,000	
9	5000	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	
10	5500	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	
11	6000	Efficiency + Maldistribution	0,9000	0,0100	Equivalence Ratio	Common	All		1,000	

Figure 14:LES Fueling Options

Combustion efficiency, Maldistribution, Equivalence ratio can also be changed in the desired test points. 2 The settings in figure 11 are used for 2JZ-GTE simulation. Here, the Equivalence ratio has been taken as 1.1 to give more air into the cylinder at low RPMs and 1 at other RPMs.

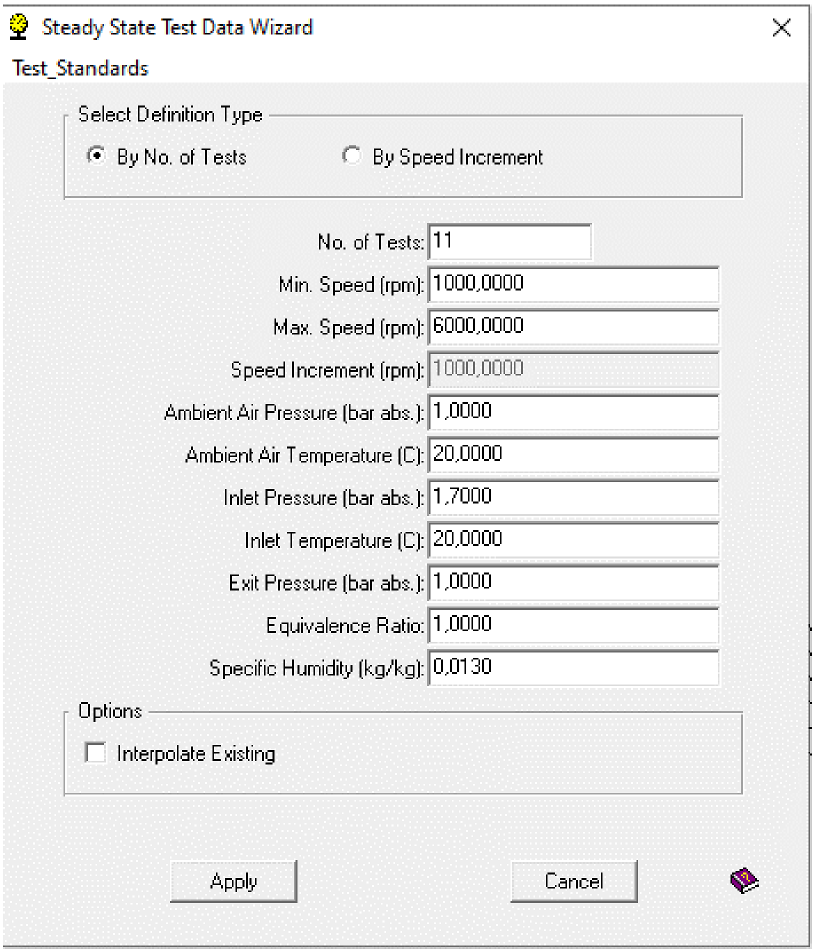


Figure 15: Steady State Test Data Wizard

Here we set the test conditions. We specify the min and max RPMs. We specify how many test points will occur in this interval. We can adjust the settings of Ambient air Temperature, Ambient air pressure, Inlet pressure, Inlet temperature, Exit Pressure, equivalence ratio, Specific humidity. The values used for this simulation are given in Figure 12.

Test Point	Speed (rpm)	Humidity Option	Specific Humidity (kg/kg)	Relative Humidity (0-1)	Ambient Air Pressure (bar-abs)	Ambient Air Temperature (C)	Inlet No.	Inlet Boundary Pressure (bar-abs)	Inlet Boundary Temperature (C)	Exit No.	Exit Boundary Pressure (bar-abs)	Exit Temp Initialisation	Exit Boundary Temperature (C)
1	1000	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
2	1500	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
3	2000	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
4	2500	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
5	3000	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
6	3500	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
7	4000	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
8	4500	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
9	5000	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
10	5500	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	
11	6000	Specific Humidity (kg/kg)	0,0130000	0,8745018	1,00000	20,00	1	1,70000	20,00	1	1,00000	Use Cyl 1 at EVO	

Figure 16:Boundary Conditions

Here we can change the boundary conditions specific to the test points. The boundary conditions of our simulation are as in figure 13.

2.3 ICE Modeling with MATLAB

2.3.1 About MATLAB

MATLAB is a type of computer program often used for engineering and positive science calculations. MATLAB is a 4th generation and multi-paradigm programming language developed by the US-based company MathWorks. MATLAB, which emerged by combining the words "Matrix Laboratory" in English, is a programming language with a matrix-based working system, as the name suggests. MATLAB helps to perform many mathematical calculations such as linear algebra, optimization, statistics, numerical analysis, fourier analysis, optimization quickly and effectively. At the same time, MATLAB is frequently used for drawing 2D and 3D graphics. (mathworks, 2022)

2.3.2 Components of the MATLAB Model

The bulk of the MATLAB model was set up through the use of a script. The model required many unrelated equations, so functions were only used under certain circumstances. The MATLAB code was broken into the following sub-sections:

- Engine geometry and atmospheric inputs.
- Pre-allocation of arrays and matrices.
- Fuel inputs and combustions efficiencies.
- Instantaneous engine and fluid properties.
- Valve simulation.
- Plot statements.

The sections mentioned above are located in our main code, BSFCAnnand.m.

2.3.3 Running MATLAB code

We have 4 files with .m extension. These are correction.m, LOAD2.m, BSFCAnnand.m and BSFC_Mapping_Function.m. The first three of them are function files, while our last file is execution codes. In the next section these codes and how they work will be discussed.

2.3.4 Obtaining the BSFC chart

```
Load = 1.1;                %Start High, Decreased By .1 Each Iteration
N_load = 7;                %Seven Loads

%_______________________________________________________________________
%N_RPM=8;
Torque(1:15,1:7)=zeros;
bsfc(1:15,1:7)=zeros;
for j = 1:N_load
    Load = Load-.1;            %Decreases Load With Each Iteration
    [BSFC,T_ac,N_RPM]=LOAD2(Load);%Feeds "Load" Into Load Function
    BSFC=BSFC';                %Transposes Vector
    bsfc(1:N_RPM,j)=BSFC;      %Creates Matrix Out of BSFC Vectors
    T_ac=T_ac';                %Transposes Torque Vector
    Torque(1:N_RPM,j)=T_ac'; %Creates Matrix Out of Torque Vectors
end
```

Figure 17: MATLAB Codes in BSFC_Mapping_Function.m

To get the BSFC graph, we run our engine at different load values from 0.4 to 1 increasing by 0.1. In deriving the BSFC graph, we are concerned with 3 parameters, they are BSFC, Torque and RPM. What we're doing in this function is actually using the LOAD2 function to get all 3 values and the results for different values, we create a matrix and throw it into it.

```
function [BSFC,T_ac,N_RPM]=LOAD2(Load)
RPM = 1000;                     %Sets Starting Pt. For Loop
N_RPM = 15;                     %15 RPM Data Sets
BSFC(1:N_RPM)=zeros;            %Preallocate Array
T_ac(1:N_RPM)=zeros;            %Preallocate Array
for i = 1:N_RPM
    RPM = RPM+500;
    [BSFC(i),T_ac(i)]=BSFCAnnand(RPM,Load);
end
end
```

Figure 18:MATLAB Codes in LOAD2.m

In this section, we send different load values to our main function BFSCAnnand.m. In this way, we get the results for 105 different values realized in our main function and with the return values of the functions, we will have obtained the BSFC Map, which we want at the last stage.

When we come to BSFCAnnand.m scripting, we see that there are Engine geometry and atmospheric inputs, Pre-allocation of arrays and matrices, Fuel inputs and combustions efficiencies, Instantaneous engine and fluid properties stages. As a result of using the information entered with the formulas mentioned in the section 2.1 internal combustion engine modeling, we have obtained the Brake Torque and Brake Specific Fuel Consumption values.

```
%Plot Statements

figure(1)
grid
contourf(RPM,Torque,bsfc,170)
xlabel('RPM')
ylabel('Torque [N*m]')
title('Toyota Supra 2JZ-GTE BSFC [g/kW-h] MAP')
colorbar
colormap jet
caxis([250 700])
```

Figure 19:Codes of Plot Statements

By using the values we obtained at different rpm and load values, we obtain the BSFC Map graphic with the help of the contourf function.

3. Results and Discussion

3.1 Performance Curve

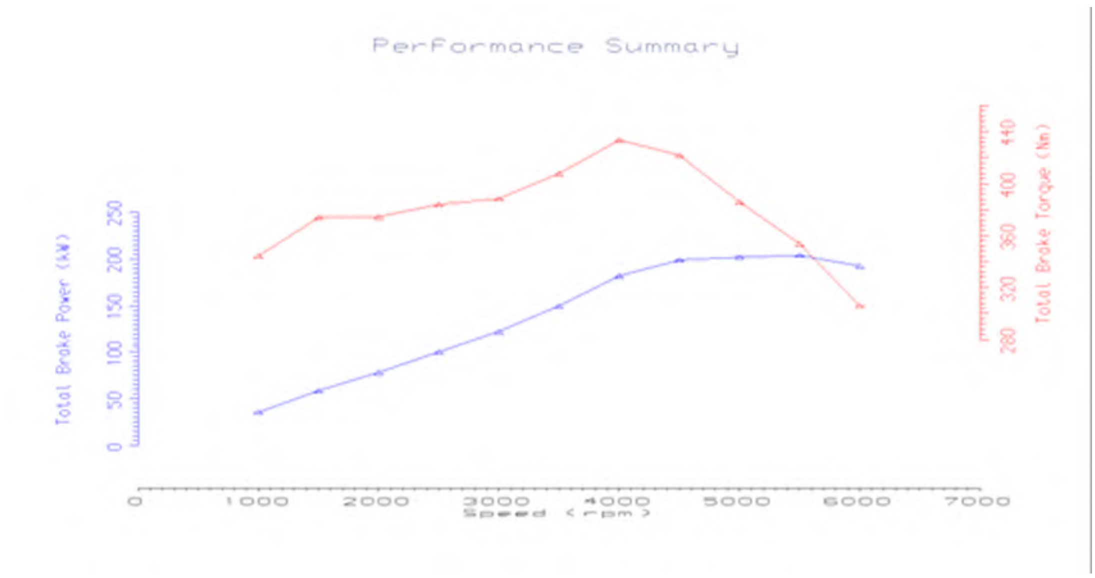


Figure 20:Performance Curves

After entering the above-mentioned engine characteristics and entering the above-mentioned conditions, the torque and power curves related to RPM were obtained as mentioned above. Looking at the performance curves provided by the manufacturer, it is seen that the curve characteristics are in harmony. Since it is not known under which conditions the manufacturer tests the engine, there is a margin of error in terms of value.

3.2 Engine BSFC

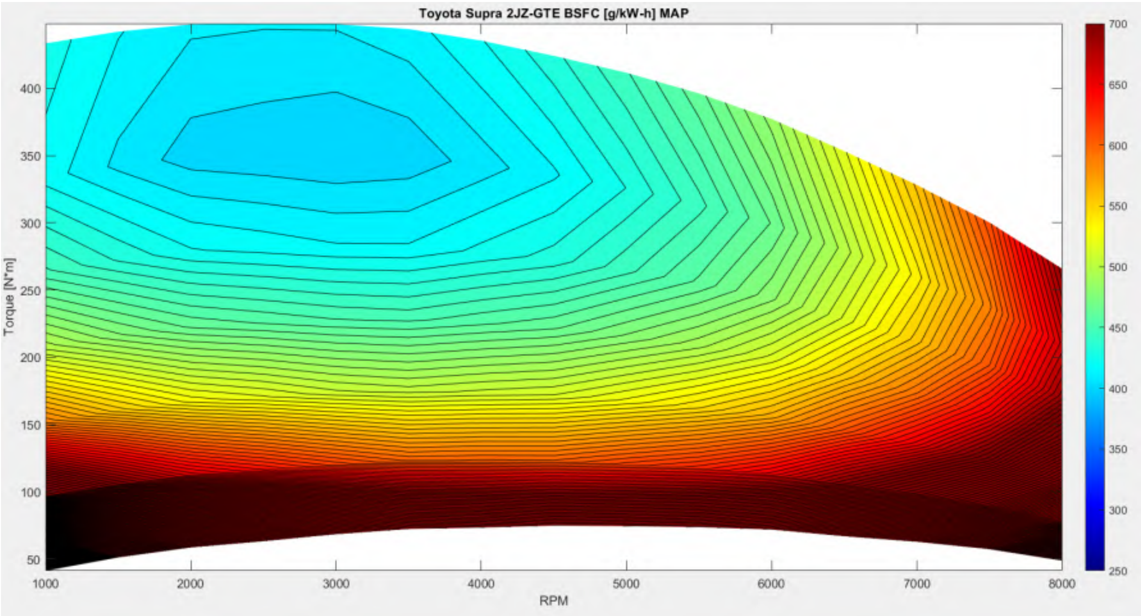


Figure 21: BSFC Mapping of 2JZ-GTE

To obtain the BSFC graph, codes were generated on MATLAB. Thanks to these codes, we have obtained torque and BSFC values at different RPM and load values. After this process, with the contourf command, we plotted the graph with the x-axis RPM, the y-axis Torque, and the 3rd axis BSFC.

As seen in the graph, BSFC is high at low load values. This is because the engine needs more fuel to run. Our optimum working range is between 2000-3500 rpm. However, it was observed that the fuel consumption increased with the increase in RPM values.

3.3 FEASIBILITY AND COST ANALYSIS

RPM	2000	3000	4000
Power (kW)	85.88	138.25	193.95
BSFC (g/kWh)	561.55	488.76	448.72
Fuel Consumption (l/h)	67.924	95.171	122.576
The cost of Gasoline (TL/l)	27.38 (Shell, 2022)		
The cost of consumption (TL/h)	1859.76	2605.78	3356.13

Table 8: Fuel Consumption according to RPM

We used buildingclub.info to convert g/kWh to l/h.

$$Q = \frac{N \times q}{R} \tag{3-1}$$

Where,

Q [l/h] :Maximum theoretical fuel consumption in grams per 1 hour of engine operation at maximum power;

q [g/kWh] : Specific fuel consumption each RPM

N[kW] : Engine power;

R[kg/m^3] : Fuel density.

Gasoline (petrol) fuel density: 710 – 760 kg/m^3

For 2000 RPM ;

$$Q = \frac{N \times q}{R} = \frac{85.88 \times 561.55}{710} = 67.924\ l/h$$

For 3000 RPM;

$$\frac{138.25 \times 488.76}{710} = 95.171\ l/h$$

For 4000 RPM;

$$Q = \frac{N \times q}{R} = \frac{193.95 \times 448.72}{710} = 122.576\ l/h$$

As shown in calculations above, at 2000 RPM the fuel consumption is 67.924 liter per hour and cost of fuel of gasoline is 27.26 TL per liter. So, the cost of consumption for an hour is 1859.76 TL.

4. Conclusions

Within the scope of this project, we worked on the 0D modeling of the internal combustion engine. As a result of these studies, we used the formulas of engine geometries according to different crank angles. Based on these formulas, we got the power and torque values. While doing these steps, we used the Wiebe combustion model. We used Annand model for heat release.

Since we could not reach all the parameters related to the engine in our project, we tried to choose the engine parameters in accordance with the graphic shared by the manufacturer.

We did the single zone modeling in MATLAB, the second program of the project. We adapted the models and formulas used by LES in the background to MATLAB. As a result of these processes, we obtained the required torque and bsfc values. When we look at our BSFC graph obtained with these values, we have accessed extensive information about the fuel consumption of the engine.

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6. Appendices

6.1 BSFC_Mapping_Function.m

```
Load = 1.1;                 %Start High, Decreased By .1 Each Iteration
N_load = 7;                 %Seven Loads

%______________________________________________________________________
%N_RPM=8;
Torque(1:15,1:7)=zeros;
bsfc(1:15,1:7)=zeros;
for j = 1:N_load
    Load = Load-.1;             %Decreases Load With Each Iteration
    [BSFC,T_ac,N_RPM]=LOAD2(Load);%Feeds "Load" Into Load Function
    BSFC=BSFC';                 %Transposes Vector
    bsfc(1:N_RPM,j)=BSFC;       %Creates Matrix Out of BSFC Vectors
    T_ac=T_ac';                 %Transposes Torque Vector
    Torque(1:N_RPM,j)=T_ac';    %Creates Matrix Out of Torque Vectors
end

%_______________________________________________________________________

%Create RPM Matrix
RPM(1:N_RPM,1:N_load)=zeros;
RPM(1,1)=8000;
for i = 2:N_RPM
    RPM(i,1) = RPM(i-1,1)-500;
end
for j = 2:N_load
    RPM(1:N_RPM,j)=RPM(1:N_RPM,j-1);
end
```

```

%______________________________________________________________________
%Plot Statements

figure(1)
grid
contourf(RPM,Torque,bsfc,200)
xlabel('RPM')
ylabel('Torque [N*m]')
title('Toyota Supra 2JZ-GTE BSFC [g/kW-h] MAP')
colorbar
colormap jet
caxis([250 700])
```

6.2 LOAD2.m

```
function [BSFC,T_ac,N_RPM]=LOAD2(Load)
RPM = 1000;                     %Sets Starting Pt. For Loop
N_RPM = 15;                     %15 RPM Data Sets
BSFC(1:N_RPM)=zeros;            %Preallocate Array
T_ac(1:N_RPM)=zeros;            %Preallocate Array
for i = 1:N_RPM
    RPM = RPM+500;
    [BSFC(i),T_ac(i)]=BSFCAnnand(RPM,Load);
end
end
```

6.3 BSFCAnnand.m

```
function[BSFC,T_ac]=BSFCAnnand(RPM,Load)
%Marmara University Engine Simulation Uses "Single Zone" Combustion Analysis

%________________________________________________________________________
%Engine Inputs
%Load = 1;                      %Engine Load (Affects Inlet Pressure)
%RPM = 7600;                    %Revolutions Per Minute [1/min]
L = (86/1000);                  %Stroke of Engine [m]
B = (86/1000);                  %Bore of Engine [m]
l = .142;                       %Length of Engine Connecting Rod [m]
N_cyl = 6;                      %Number of Cylinders [unitless]
C_r = 9;                        %Compression Ratio [unitless]
N_r = 2;                        %Number of Revolutions Per Power Stroke
if Load==1
    theta_0=ceil(-.0013*RPM+154.82);
    theta_b=ceil(.0038*RPM+40);
end
if Load<1
    theta_0=ceil(-.0013*RPM+154.82)-(10-Load*10);
    theta_b=ceil(.0038*RPM+40)-(10-Load*10);
end

theta_f = theta_0+theta_b;      %Final Comb. Angle [degrees]
IVC = 0;                        %Time [degrees] when Intake Valve Closes
EVO = 335;                      %Time [degrees] when Exhaust Valve Opens
```

```
%______________________________________________________________________

%Engine Calculations Based On Previous Inputs
%Assumes Average Surface Area In Which Heat Transfer Occurs
A_p = (pi/4)*B^2;              %Cross Sectional Piston Area [m^2]
A_ch = 2*A_p;                  %Cylinder Head Surface Area (in chamber)
V_d = N_cyl*A_p*L;             %Displaced Volume Of Engine [m^3]
N = RPM/60;                    %Converts RPM to RPS [1/s]
S_bar_p = 2*L*N;               %Calculates Mean Piston Speed [m/s]
a = L/2;                       %Calculates Crank Radius (1/2 stroke)[m]
V_TDC = (V_d/(C_r-1))/N_cyl;   %Calculates Clearance Volume [m^3]
V_BDC = (V_d/N_cyl)+V_TDC;     %Cyl. Volume At BDC [m^3]
%______________________________________________________________________

%Calculating Losses Due To Friction
fmep=((2E-06)*(RPM^2)) + (0.045*RPM) + 121.68;

%Volumetric Efficiency Correction Factor
CF = correction( Load,RPM );

%Initial Preallocation Of Matrices (Second Preallocation In Loops Needs To
%Be Included (Do Not Delete)
V(1:360)=zeros;DV(1:360)=zeros;rho(1:360)=zeros;mu(1:360)=zeros;
C_k(1:360)=zeros;C_R(1:360)=zeros;X(1:360)=zeros;M_F(1:360)=zeros;
DX(1:360)=zeros;Re(1:360)=zeros;Nus(1:360)=zeros;h_g(1:360)=zeros;
DQ_w(1:360)=zeros;DQ(1:360)=zeros;Q(1:360)=zeros;DT(1:360)=zeros;
DP(1:360)=zeros;P(1:360)=zeros;T(1:360)=zeros;W_dot(1:360)=zeros;
W(1:360)=zeros;T_indicated(1:2)=zeros;Q_dot(1:360)=zeros;u(1:360)=zeros;
du(1:360)=zeros;cv(1:360)=zeros;m_b(1:360)=zeros;m_u(1:360)=zeros;
V_u(1:360)=zeros;V_b(1:360)=zeros;T_u(1:360)=zeros;T_b(1:360)=zeros;
A_u(1:360)=zeros;A_b(1:360)=zeros;DT_u(1:360)=zeros;gamma_u(1:360)=zeros;
u_u(1:360)=zeros;du_u(1:360)=zeros;cv_u(1:360)=zeros;DQ2(1:360)=zeros;
DQ_w2(1:360)=zeros;Q2(1:360)=zeros;

%______________________________________________________________________
%Fuel Inputs/Efficiencies
AF_ratio_stoich = 15.09;    %Theoretical Air Fuel Ratio (gravimetric)
if Load ==1
    lambda = .85;
end
if Load==.9
    lambda=.925;
end
if Load==.8
    lambda=.95;
end
if Load<.8
    lambda=.95;
end
AF_ratio_ac = lambda*AF_ratio_stoich; %Actual Air Fuel Ratio
AF_ratio_mol_sotich=14.7; %Molar Air_Fuel Ratio (Stoich)
AF_ratio_mol=lambda*AF_ratio_mol_sotich;
LHV = 44.6e6; %Lower Heating Value Of Fuel Mixture [J/kg]
eta_combmax = .95; %Assumed MAX Comb. Efficiency
%Predicts Combustion Efficiency (Reference To Blair)
eta_comb=eta_combmax*(-1.6082+4.6509*lambda-2.0764*lambda^2);
```

```
%Atmospheric Inputs
P_atm = (101325)*1.7;
T_atm = 278;
P_BDC = Load*P_atm; %Inlet Pressure[Pa]
R_air = 287; %Gas Constant For Air [J/kg-K]
gamma(1:360) = 1.4; %Preallocate Gamma Array (sets initial value)
T_w =350; %Assumed Wall Temperature (Reference Stone)

%________________________________________________________________________

%Polynomials Used To Calculate Gamma As A Function Of RPM
a_1 = .692; a_2 = 39.17e-06; a_3 = 52.9e-09; a_4 = -228.62e-13;
a_5 = 277.58e-17;b_0 = 3049.33; b_1 = -5.7e-02; b_2 = -9.5e-05;
b_3 = 21.53e-09;b_4 = -200.26e-14;c_u = 2.32584; c_r = 4.186e-03;
d_0 = 10.41066; d_1 = 7.85125; d_3 = -3.71257;e_0 = -15.001e03;
e_1 = -15.838e03; e_3 = 9.613e03;f_0 = -.10329; f_1 = -.38656;
f_3 = .154226; f_4 = -14.763; f_5 = 118.27; f_6 = 14.503;
r_0 = -.2977; r_1 = 11.98; r_2 = -25442; r_3 = -.4354;

%________________________________________________________________________
R=R_air/1000;
T_BDC = T_atm; %Assumed Inlet Temperature [K]
%Calculate Mass of Air In Cylinder/ Mass Of Fuel Based On AFR
rho_a = P_BDC/(R_air*T_BDC); %Air Density kg/m^3
m_a = rho_a*V_d; %Mass of Air In Cylinder [kg]
m_f = m_a/AF_ratio_ac; %Mass Of Fuel In Cylinder [kg]
m_c = m_a + m_f; %Mass In Cylinder
%Specifying Initial Conditions For Loops
%DV,DX,etc. Are Relative To Change In Theta (i.e. DV/Dtheta)
theta(1:360)=zeros;           %Starting Crank Angle [deg]
V(1:360)=zeros;               %Preallocate Volume Array
V(1)=V_BDC;                   %Starting Combustion Chamber Volume [m^3]
DV(1:360) = zeros;            %Preallocate Change In Volume Array
DV(1) = 0;                    %Specifying Initial Change In Volume [m^3}
P(1:360)=P_BDC;               %Preallocate Pressure Array
DP(1:360) = zeros;            %Specifying Initial Change In Pressure
T(1:360)=zeros;               %Preallocate Temperature Array
T(1) = T_BDC;                 %Inlet Temperature [K]
T_u(1)=T_BDC;                 %Initial Unburned Temperature[K]
DT(1:360) = zeros;            %Specifying Initial Change In Temperature
DT_u(1:360)=zeros;            %Preallocate Change In Unburned Temperature
gamma(1)=1.4;                 %Initial Gamma Input
gamma_u(1)=1.4;               %Initial Gamma Input
X(1:360) = 0;                 %Preallocate Mass Burn Array
DX(1:360) = zeros;            %Preallocate Change In Mass Burn Fraction
DQ(1:360) = zeros;            %Preallocate Heat Release Array
DQ2(1:360)=zeros;             %Preallocate Two Zone Heat Release Array
Q(1:360)=zeros;               %Preallocate Heat Array
Q2(1:360)=zeros;              %Preallocate 2 zone Heat Array
M_F(1:360) = 0; %Preallocate Mass In Combustion Chamber Array
rho(1:360) = zeros; %Preallocates Ideal Gas Law array
rho(1) = P(1)/(R_air*T(1)); %Initial Value Ideal Gas Array
mu(1:360)=zeros; %Preallocate Viscosity Array
mu(1)=7.457*10^(-6)+4.1547*10^(-8)*T_BDC-7.4793*10^(-12)*T_BDC^(2);
C_k(1:360)=zeros; %Preallocate Thermal Conductivity Array
C_k(1) = 6.1944*10^(-3)+7.3814*10^(-5)*T_BDC-1.2491*10^(-8)*T_BDC^(2);
C_R(1:360) = zeros; %Preallocate Radiation Coefficient Array
```

```
C_R(1) = 4.25*10^(-09)*((T(1)^4-T_w^4)/(T(1)-T_w)); %Initial Rad. Coeff
Re(1:360)=zeros; %Preallocate Reynolds Value Array
Re(1)=rho(1)*S_bar_p*B/mu(1); %Initial Reynolds Value
Nus(1:360)=zeros; %Preallocating Nusselt Number Array
Nus(1)=.49*Re(1)^(.7); %Initial Nusselt Number
h_g(1:360)=zeros; %Preallocate Heat Transfer Coefficient Array
h_g(1)=C_k(1)*Nus(1)/B; %Initial Heat Transfer Coefficient
s(1:360)=zeros; %Preallocates Distance Crank/Piston Axes Array
s(1) = -a*cosd(theta(1))+sqrt(l^2 - a^2*sind(theta(1))^2);%Initial Val.
W(1:360) = zeros; %Preallocate Work Array
W_dot(1:360) = zeros; %Preallocate Power Array
T_indicated(1:360) = zeros; %Preallocate Torque Array
Q_dot(1:360) = zeros; %Preallocate Heat Transfer Array
u(1:360) = zeros; %Preallocate Internal Energy Array
du(1:360) = zeros; %Preallocates Change In Internal Energy Array
cv(1:360) = zeros; %Preallocates Heat Capacity Array
DQ_w(1:360)=zeros; %Preallocate Convective Heat Loss Array
DQ_w2(1:360)=zeros; %Preallocate Convective Heat Loss Array 2 zone
m_b(1:360)= zeros; %Preallocate mass burned array
m_u(1:360)=m_c; %Preallocate unburned mass array
V_u(1:360)=zeros; %Preallocate unburned Volume Array
V_u(1) = V(1); %Initial Unburned Volume

%______________________________________________________________________
theta=1:360;
for i = 2:360
    %Specifies Distance Between Crank/Piston Axes As A Function Of theta
    s = -a*cosd(theta(i))+sqrt(l^2 - a^2*sind(theta(i))^2);
    %Specifies Volume As A Function Of Crank Angle
    V(i) = V_TDC +((pi/4)*B^2)*(l + a - s);
    %Specifies Change In Volume As A Function Of Crank Angle
    DV(i) = V(i)-V(i-1);
    %Calculates Density As A Function Of Crank Angle
    rho(i) = P(i-1)/(R_air*T(i-1));
    %Calculates Viscosity As A Function Of Temperature
    mu(i)=7.457*10^(-6)+4.1547*10^(-8)*T(i-1)-7.4793*10^(-12)*T(i-1)^(2);
    %Calculating Instantaneous Thermal Conductivity of Cylinder Gas
    C_k(i) = 6.1944*10^(-3)+7.3814*10^(-5)*T(i-1)-1.2491*10^(-8)*T(i-1)^(2);
    %Calculating The Radiation Heat Transfer Coefficient
    C_R(i) = 4.25*10^(-09)*((T(i-1)^4-T_w^4)/(T(i-1)-T_w));
    %Instantaneous Surface Area (For Heat Transfer)
    A = A_ch + A_p + pi*B*(l+a-s);
    if i<=2
        A_u=A;
    end
    %__________________________________________________________________

    %Specifies Mass Fraction Burn As A Function Of Crank Angle (Wiebe Fcn.)
    %Also Specifies Mass Of Fuel In Combustion Chamber As A Function Of Theta
    if theta(i)<theta_0
        X(i)=0;
    else
        X(i) = 1-exp(-5*((theta(i)-theta_0)/theta_b)^3);
        if theta(i) < theta_f
```

```
        M_F(i) = V(theta_0-1)*rho(theta_0-1)/(lambda*AF_ratio_mol);
    end
end
%Specifies Change In Mass Fraction Burn As A Function Of Crank Angle
DX(i) = X(i) - X(i-1);
%__________________________________________________________________

%Incorporating The Annand Method To Predict Heat Transfer
%Calculating Reynolds Number
Re(i)=rho(i)*S_bar_p*B/mu(i);
%Calculating Nusselt Number (constant=.26 two stroke, .49 4 stroke)
Nus(i)=.49*Re(i)^(.7);
%Calculating Heat Transfer Coefficient Using Annand Method
h_g(i)=C_k(i)*Nus(i)/B;
%Calculates Convective Losses Into Wall As A Function Of Crank Angle
DQ_w(i) = (h_g(i)+C_R(i))*A*(T(i-1)-T_w)*(60/(360*RPM));
%Calculates Change In Heat Transfer (total) As A Function Of Crank Angle
DQ(i) = eta_comb*LHV*M_F(i)*DX(i)-DQ_w(i);
%Calculates Total Heat Transfer (Per Cycle)
Q(i) = Q(i-1)+DQ(i);
%__________________________________________________________________
%Specifies Pressure and Temperature Increases Between Intake Valve
%Closing and Exhaust Valve Opening
if IVC< theta(i)
    DT(i)=T(i-1)*(gamma(i-1)-1)*((1/(P(i-1)*V(i-1)))*DQ(i)-(1/V(i-1))*DV(i));
    DP(i)=(-P(i-1)/V(i-1))*DV(i)+(P(i-1)/T(i-1))*DT(i);
    P(i) = P(i-1)+DP(i);
end
if EVO < theta(i)
    P(i) = P_atm;
end
if 200 < theta(i)
    if P(i)<=P_atm
        P(i)=P_atm;
    end
end
%__________________________________________________________________
%Calculate Burned, Unburned Mass Fractions
m_b(i) = m_b(i-1)+DX(i)*m_c; %Burned Mass
m_u(i) = m_u(i-1)-DX(i)*m_c; %Unburned Mass
%Calculating Burned, Unburned Volumes
if theta(i)<=theta_0
    V_u(i)=N_cyl*V(i);
end
if theta(i)>theta_0
    V_u(i)=((m_u(i)*V_u(i-1))/m_u(i-1))*(P(i)/P(i-1))^(-1/gamma_u(i-1));
end
V_b(i)=N_cyl*V(i)-V_u(i);
if V_b(i)<0
    V_b(i)=0;
end
%Calculating Burned, Unburned Temperatures
T_u(i)=P(i)*V_u(i)/(m_u(i)*R*1000);
if theta(i) <= theta_0+4
```

```
        T_b(i)=0;
    end
    if theta(i)>theta_0+4
        T_b(i)=P(i)*V_b(i)/(m_b(i)*R*1000);
    end
    %Calculate Unburned, Burned Areas Based On Volume Ratio
    A_u(i)=A*(1-sqrt(X(i)));
    A_b(i)=A*(X(i)/sqrt(X(i)));
    DT_u(i)=T_u(i)-T_u(i-1);

    %____________________________________________________________________
    %Returns Temperature Values To Beginning Of Loop
    %Assumes Temperature Drops Back To ATM Temp After Exhaust Is Extracted
    T(i) = T(i-1)+DT(i);
    %Calculate The Residual Gas Fraction Assume A Polytropic Constant Of 1.3
    R_frac = (1/C_r)*(P_BDC/P_atm)^(1/1.3)*(1/lambda);
    %Calculates Cylinder Work [J] As A Function Of Crank Angle
    %Treats Atmospheric Pressure As Reference State
    W(i) = W(i-1)+(P(i)-P_atm)*DV(i);
    %Calculates Power [kW] As A Function Of Crank Angle
    W_dot(i)=(N_cyl*W(i)*N/N_r)/1000;
    %Indicated Mean Effective Pressure
    imep = CF*W_dot(360)*N_r*1000/(V_d*1000*N);
    %Calculates Torque[N*m] As A Function Of Crank Angle
    T_indicated(i) = (W_dot(i)*1000)/(2*pi*N);
    %Calculates Heat Loss [kW] As A Function Of Crank Angle
    Q_dot(i) = (N_cyl*Q(i)*N/N_r)/1000;
    %____________________________________________________________________
    % The Following Section Of Code Calculates An Updated Value Of Gamma
    % Using The "Polynomial Method" Developed By Krieger-Borman
    % User Of This Code Must Be Careful Because Accuracy Of This Method
    % Drops As The Fuel Mixture Becomes Increasingly Rich

    %Calculates A,B Factors For Following Block Of Code
    A_t = a_1*T(i)+a_2*T(i)^2+a_3*T(i)^3+a_4*T(i)^4+a_5*T(i)^5;
    A_tu = a_1*T_u(i)+a_2*T_u(i)^2+a_3*T_u(i)^3+a_4*T_u(i)^4+a_5*T_u(i)^5;
    B_t = b_0+b_1*T(i)+b_2*T(i)^2+b_3*T(i)^3+b_4*T(i)^4;
    B_tu = b_0+b_1*T_u(i)+b_2*T_u(i)^2+b_3*T_u(i)^3+b_4*T_u(i)^4;
    %Calculates Factor "D" As A Function Of lambda
    D_lambda = d_0 + d_1*lambda^(-1)+ d_3*lambda^(-3);
    %Calculates Factor "F" As A Function Of Temperature,lambda
    E_TLambda = (e_0 + e_1*lambda^(-1)+ e_3*lambda^(-3))/T(i);
    E_TLambdau = (e_0 + e_1*lambda^(-1)+ e_3*lambda^(-3))/T_u(i);
    F_TPLambda = (f_0 + f_1*lambda^(-1) + f_3*lambda^(-3) + ...
        ((f_4 + f_5*lambda^(-1))/T(i)))*log(f_6*P(i));
    F_TPLambdau = (f_0 + f_1*lambda^(-1) + f_3*lambda^(-3) + ...
        ((f_4 + f_5*lambda^(-1))/T_u(i)))*log(f_6*P(i));
    %Calculates Correction Factor For Internal Energy
    u_corr = c_u*exp(D_lambda +E_TLambda + F_TPLambda);
    u_corr_u=c_u*exp(D_lambda +E_TLambdau + F_TPLambdau);
    %Calculates Internal Energy As A Function Of Crank Angle
    u(i) = A_t - B_t/lambda + u_corr;
    u_u(i) = A_tu - B_tu/lambda + u_corr_u;
    %Calculates Change In Internal Energy
    du(i) = u(i) - u(i-1);
    du_u(i) = u_u(i) - u_u(i-1);
```

```
    %Calculates Heat Capacity "C_v" As A Function Of Crank Angle
    cv(i) = du(i)/DT(i);
    cv_u(i)=du_u(i)/DT_u(i);
    %Calculates Correction Factor For "R" Value As A Function Of Crank Angle
    R_corr = c_r*exp(r_0*log(lambda) + (r_1+r_2/T(i) + ...
        r_3*log(f_6*P(i)))/lambda);
    R_corr_u = c_r*exp(r_0*log(lambda) + (r_1+r_2/T_u(i-1) + ...
        r_3*log(f_6*P(i)))/lambda);
    %Calculates Actual "R" Value
    R = .287 + .020/lambda + R_corr;
    R_u = .287 + .020/lambda + R_corr_u;
    %Calculates Actual Gamma Value And Returns To Beginning Of Code
    gamma_u(i)=1+R_u/cv_u(i);
    gamma(i) = 1 + R/cv(i);
    if gamma(i)<1.2
        gamma(i)=1.4;
        gamma_u(i)=1.4;
    end
    if theta(i)>=EVO
        gamma(i)=1.4;
        gamma_u(i)=1.4;
    end


%________________________________________________________________________

    %Calculate Temperature Of Exhaust Based On Polytropic Relations
    if EVO < theta(i)
        T(i)=T(EVO)*(P_BDC/P(EVO))^((gamma(i)-1)/gamma(i));
        T_b(i)=T_b(EVO)*(P_BDC/P(EVO))^((gamma(i)-1)/gamma(i));
    end
    %Calculates A Corrected Inlet Temperature Based On EGR
    %T_corr = R_frac*T(360)+(1-R_frac)*T_BDC;
    T_corr = T_BDC;
end

%________________________________________________________________________

%Specified Outputs (On MATLAB Screen)
W_dot_indicated=W_dot(360);
bmep = imep-fmep;
W_dot_ac = (bmep*V_d*1000*N/(N_r*1000));
T_ac = W_dot_ac/(2*pi*N*10^(-3));
%Calculated Mechanical Efficiency (Based On Previous Inputs)
eta_m = bmep/imep; %Calculates Mechanical Efficiency

%________________________________________________________________________

%Calculates Brake Specific Fuel Consumption
m_ta = P_BDC*V_d/(R_air*T_BDC); %Calculate Trapped Air In Cylinder
eta_v = CF*((m_ta)/(rho_a*V_d)); %Corrected Volumetric Efficiency
m_dot_f = N_cyl*M_F(theta_0)*(N/N_r); %Mass Flow Rate Of Fuel
m_dot_a = AF_ratio_ac*m_dot_f; %Mass Flow Rate Of Air
BSFC = (m_dot_f*1000*3600)/(W_dot_ac); %BSFC [g/kW*h]
eta_f = 3600/(BSFC*(LHV*10^(-6))); %Fuel Conversion Efficiency
end
```

6.4 correction.m

```
function [ CF ] = correction(Load,RPM)
if Load<=1
    %CF = (-3*10^(-9))*RPM^2+5*10^(-5)*RPM+.7088;
    CF=-8*10^(-9)*RPM^2+.000135*RPM+.31944;
    if Load<=.9
        CF=-8*10^(-9)*RPM^2+.000135*RPM+.31944;
        %CF = CF-(1-Load)/4;
    end
end
end
```