MARMARA UNIVERSITY

FACULTY OF ENGINEERING

1-D MODELLING OF A 1-CYLINDER ENGINE WITH GT-SUITE

Erdem Emir ÜLKÜ

GRADUATION PROJECT REPORT

Department of Mechanical Engineering

Supervisor

Doç.Dr. MUSTAFA YILMAZ

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Erdem Emir ÜLKÜ

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Signature of Author(s) ....

Department of Mechanical Engineering

Certified By ....

Project Supervisor, Department of Mechanical Engineering

Accepted By ....

Head of the Department of Mechanical Engineering

Preface

First of all, I would like to thank my thesis teacher, Assoc. Prof. Dr. Mustafa Yılmaz. It was extremely important for me that he gave the answer by teaching where I had difficulties in this thesis. I would also like to thank our teacher Prof. Dr. Zafer Gül, who did not hesitate to help me when he was not at school. He was extremely helpful in using the software in this thesis. Finally, I would like to thank Assistant Prof. Ramazan Şener, who I contacted with him via e-mail and helped me correct the errors I received in the software. In this thesis, we will see the modeling of a one-cylinder diesel engine with GT-Suite software. I will try to explain how to use the software and how to interpret the results when I can.

Jun 2022

Erdem Emir Ülkü

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Özet/Abstract

Bu tezde bir silindirli bir dizel motorun GT-Suite yazılımı ile nasıl modelleneceğini anlattım. Ardından alınan sonuçların ve çıkan grafiklerin nasıl yorumlanacağından bahsettim. Karşılaştığım problemleri elimden geldiğince açıklamaya çalıştım ve sebeplerini anlatmaya çalıştığım. Bulduğum ve çıkan hataları da sebepleriyle beraber açıklamaya çalıştım.

In this thesis, I explained how to model a one-cylinder diesel engine with GT-Suite software. Then I talked about how to interpret the results and the resulting graphics. I tried to explain the problems I encountered as best I could and I tried to explain the reasons. I tried to explain the errors I found and the errors with their reasons.

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1. INTRODUCTION

1.1 Modelling of an Engine

Internal combustion engines are still the main choice for automotive due to its engineering and production cost and durability. An internal combustion engine (ICE or IC engine) is a heat engine in which the combustion of a fuel occurs with an oxidizer (usually air) in a combustion chamber that is an integral part of the working fluid flow circuit. In an internal combustion engine, the expansion of the high-temperature and high-pressure gases produced by combustion applies direct force to some component of the engine. The force is typically applied to pistons (piston engine), turbine blades (gas turbine), a rotor (Wankel engine), or a nozzle (jet engine). This force moves the component over a distance, transforming chemical energy into kinetic energy which is used to propel, move or power whatever the engine is attached to. This replaced the external combustion engine for applications where the weight or size of an engine was more important.

Internal combustion engines are typically powered by fossil fuels like natural gas or petroleum products such as gasoline, diesel fuel or fuel oil. Renewable fuels like biodiesel are used in compression ignition (CI) engines and bioethanol or ETBE (ethyl tert-butyl ether) produced from bioethanol in spark ignition (SI) engines. As early as 1900 the inventor of the diesel engine, Rudolf Diesel, was using peanut oil to run his engines. Renewable fuels are commonly blended with fossil fuels. Hydrogen, which is rarely used, can be obtained from either fossil fuels or renewable energy.

Internal combustion engines also need optimizing, improving and increasing of performance and efficiency. While we do theese calculations, we need a calculation model and real life data for comparing results. We make a model for an engine in the designing process or after manufacturing process. Modelling in designing process can help us for deciding spesifications and measurements etc. For example we make an engine design just for a start, then we model and analize the design. For the results of the analizing step, we can make changes in the engine design. Theese two processes work like a cycle for making the optimum design.

We can make models after production step too. While we maket his type of modelling, we need to make an experiment on the engine to get data for power, torque, emissions, consumption etc. After that, we make a mathematical model with some softwares which are MatLab, ANSYS, GT-Suite for example. After we finish the model, we run the model and comparing results with real life datas. Then, we must equalize the results and datas with making small changes in the calculation model. After we equalize theese two, we can call it suitable model. Then, we can make changes on model for increasing efficiency or performance. That will work with the real engine, because we have a suitable model.

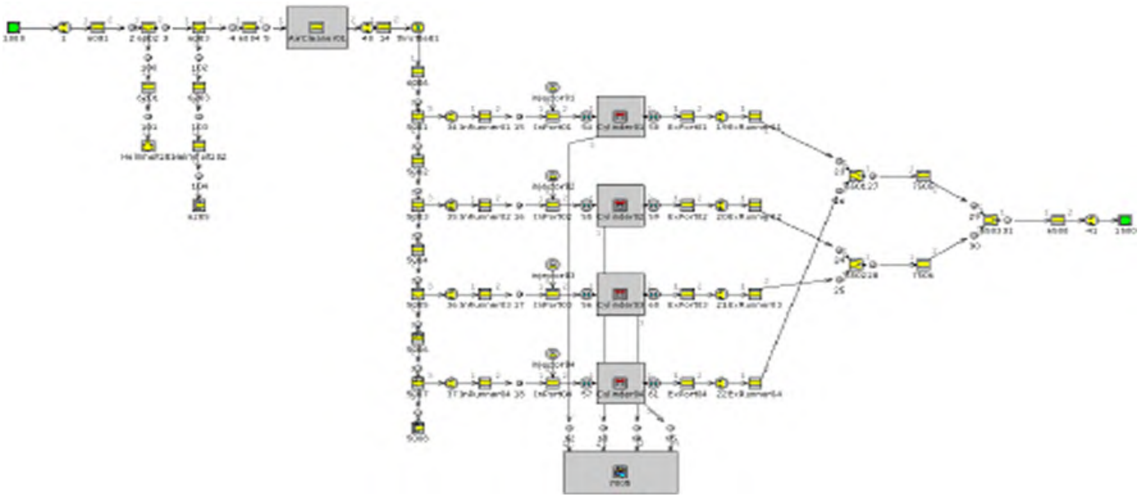


Figure 1: Modelling example of GT-Suite software

In the figure 1, we can see GT-Suite modelling interface. We can control every detail in this software. I will discuss details later on this article. There is an important thing, first and last component of this model are environment (green boxes). Engines are open systems, materials get into engine from environment and producted materials throwed into environment. There are two types of engine modelling which are 1D, 2D and 3D. We discuss in this article 1D and 3D.

1.2 Differance Between 1D and 3D

3D modelling gives us a practical way to analize and visualize how flowing fluids work in real life conditions. 3D analysis is being used where the nature of the flow and the und erstanding required makes 3D analysis the appropriate tool; 1D simulation is applied to examine the fluid flow conditions of the remaining system, which can be captured by a 1D calculation with inbuilt sub Boundmodels for specific components as required. 3D modelling is also the hard way of engine modelling. Because 3D model needs more and more input data than 1D modelling. One dimensional (1D) CFD (Computational Fluid Dynamics) allows us to understand the flow rates and pressures, and how they change, within a flow system of components such as cylinder, valves, pipes as the operating state of the components changes, for example, as pumps speed up or valves close.

In contrast, 3D CFD brings significant benefits to design engineers in understanding how detailed flows interact with all machinery and affect heat transfer and fluid flow, either increasing or decreasing pressure drop.

By comparison, 1D CFD calculations are typically much faster. When compared to 3D CFD calculations (can take hours). But 1D calculations may take only minutes to perform. 1D CFD simulation calculations are relatively quick and provide a system overview. In place of a detailed geometry and mesh, a comprehensive database of empirical and semi-empirical models, allows the analyst or designer to understand how a fluid system behaves. However without the detailed understanding of local behaviour that may come from a detailed with 3D CFD model.

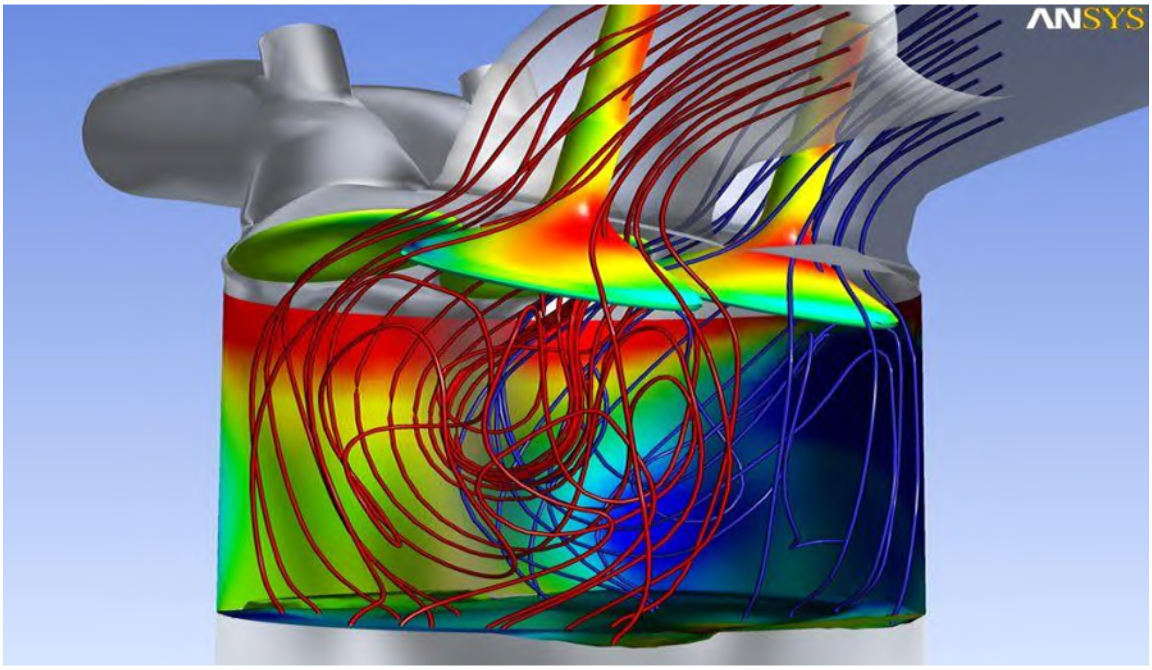


Figure 2: 3D model of an engine cylinder with ANSYS-Fluent

In figure 2, we can see how 3D model looks like after running the calculation. We can see every flow vector, temperature line, turbulant places etc. However, 3D CFD comes with some disadvantages, such as needed powerful PC components, CPU time, lots of input data. After all theese disadvantages, we always need 3D modelling for engines.

1.3 Why We Need 3D model

3D CFD (Computational Fluid Dynamics) is an engineering tool that using for simulation of thermal and fluid systems. It is used in many industries, such as aerospace, automotive, meteorology, health etc. CFD is best used in cases where the system behaviour cannot be calculated using conventional calculation (not necessarily because of the complexity of the maths theory) but because of the complexity of the overall system or its geometry. One example of complex geometry is blood vessels. We can simulate blood flow in body with using CFD software. That gives us very important datas for human life and health. Also the fluid flow in athmosphere is a very complex geometry. We can make weather forecast with modelling air flow and pressure movements in athmosphere.

With CFD software solutions vehicle designers can improve the aerodynamics of the vehicles that they produce. In addition, surgeons can cure arterial diseases through computation hemodynamics, while meteorologists can forecast the weather more accurately and warn us of impending natural disasters.

2. MODELLING OF AN ENGINE

In engineering, modeling a process means developing and using the necessary combination of assumptions and equations that understand critical features of the process to be simulated. The modeling of engine processes continues to improve as our basic understanding of the physics and chemistry of interest steadily expands and as the capability of computers to solve complex equations continues to increase. Modeling activities can make major contributions to engine engineering at different levels of generality or detail by:

• Developing a more complete understanding of the process under study from the discipline involved in formulating the model;

• Identifying key controlling variables that provide guidelines for better structured and therefore less costly experimental development efforts;

• Predicting engine behavior over a wide range of design and operating variables to screen concepts prior to major hardware programs, to determine trends and tradeoffs, and, if the model is sufficiently accurate, to optimize design and control;

• Providing a rational basis for design innovation.

Each of these contributions is valuable. Whether a model is ready to pass from one stage to the next depends on the accuracy with which it represents the actual process, the extent to which it has been tested and validated, and the time and effort required to use the model for extensive sets of calculations and interpret the results. This chapter reviews the types of models and their primary components that are used to describe engine performance, fuel consumption, and emissions characteristics. These models describe the thermodynamic, fluidflow, heat-transfer, combustion, and pollutant-formation phenomena that govern these aspects of engine performance. Many of the building blocks for these models have been described in previous chapters. The purpose here is to show how fluid dynamics, heat-transfer, thermodynamics, and kinetics fundamentals can be combined at various levels of sophistication and complexity to predict, with varying degrees of completeness, internal combustion engine combustion and emissions processes, and engine operating characteristics.

For the processes that govern engine performance and emissions, two basic types of models have been developed. These can be categorized as thermodynamic or fluid dynamic in nature, depending on whether the equations which give the model its predominant structure are based on energy conservation or on a full analysis of the fluid motion. Other labels given to thermodynamic energy-conservation-based models are: zerodimensional (since in the absence of any flow modeling, geometric features of the fluid motion cannot be predicted), phenomenological (since additional detail beyond the energy conservation equations is added for each phenomenon in turn), and quasi-dimensional (where specific geometric features, e.g., the spark-ignition engine flame or the diesel fuel spray shapes, are added to the basic thermodynamic approach). Fluid-dynamic-based models (referred to as computational fluid dynamics, CFD, models) are often called multi-dimensional, due to their inherent ability to provide detailed geometric information on the flow behavior, based on solution of the governing flow equations.

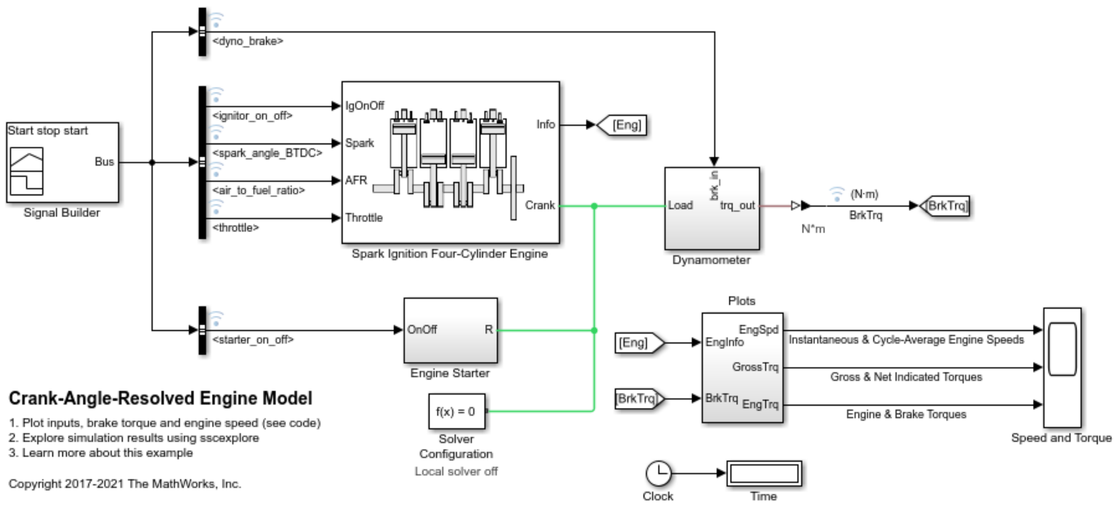


Figure 3: An example of an engine modelling with MatLab

We can model an engine with MatLab too. We can control every single equation with this software in 1D process like GT-Suite, but GT library is more suitable for engine and automobile modelling.

Some general observations about models of engine processes provide a context for the details that follow. The physical and chemical processes involved are extremely complex. While much is known about these processes, they are not fully understood at a fundamental level. At present, it is not possible to construct models that predict engine operation from the basic governing equations alone. Thus the objectives of any modeling effort should be clearly defined, and the structure and detailed content of the model should be appropriate to these objectives. It is not yet feasible to construct models that attempt to describe all important aspects of engine operation: more limited objectives are usually appropriate.

Due to this complexity of engine processes and our inadequate understanding at a fundamental level, most engine models are incomplete. Empirical relations and ad hoc approximations are often needed to bridge gaps in our understanding of critical phenomena. Since models will continue to develop greater completeness and generality, the emphasis in this chapter is on the basic relationships used in engine process models rather than the current status of these models.

Finally, an important issue in any overall engine model is balance in complexity and detail amongst the process submodels. A model is no more accurate than its weakest link. Thus it makes sense to describe the key phenomena involved at comparable levels of sophistication.

3. GT-SUITE MODELLING

When we open the software we can see it has a simple interface. There is a “new” button on the top-left side of the screen. When we click the button, software need to know what we want to do. In this article, I choose “Engine performance and acoustics” selection and click next button. After that, software asks us pre-loaded templates and what do we need. I do not make any changes in this part and click “finish” button. When we see the blank page, we can start data entering into the software.

3.1 Engine Crank Train

First of all, I choose “engine crank train” component from mechanical part from left side in the screen.

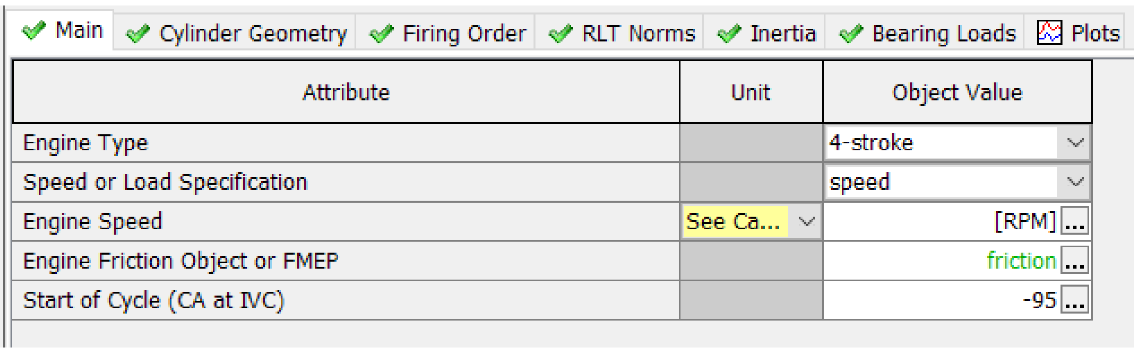


Figure 4: Engine crank train main page

In figure 4, we can see main informations of a crank train. Our engine is 1-cylinder, 4-stroke, diesel engine. Because of 1-cylinder engine, firing order is not important thing to us. RLT norms, inertia and bearing loads datas did not be given to me. That is why, I make theese parts with default informations.

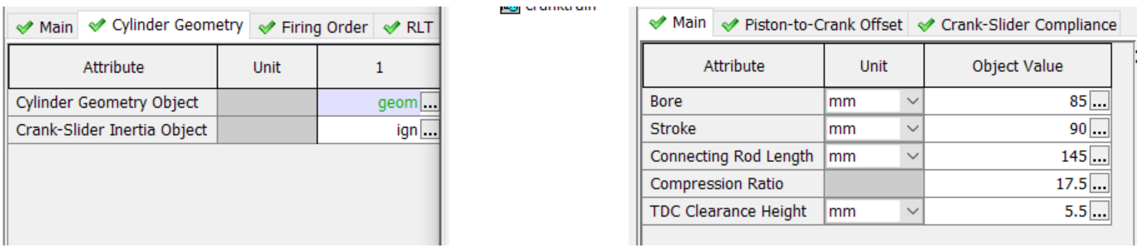


Figure 5: Cylinder geometry information in crank train component

Specification of cylinder geometry is a bit tricky part, because I must make some calculations and measuring from solidworks. Bore, stroke and compression ratio datas can be read from operating manual. We have 8.5cm bore, 9cm stroke and 17.5:1 compression ratio. Connecting rod length is in solidworks assembly file. I opened the file, selected the connecting rod and clicked "open part". Then we can measure the length with solidworks and we get 14.5cm length. We only need top dead center clearance height, we must make calculations for this. First of all, we need displacement volume. We can find it with bore and stroke datas.

$$V_d = \pi * \frac{(Bore)^2}{4} * Stroke$$

We get displacement volume (V_d) as 510.7 cm^3 and we know displacement volume is 510 cm^3 from operating manual. We need to find clearance volume (V_c) with using (V_d) and compression ratio.

$$\frac{V_c + V_d}{V_c} = 17.5$$

With this equation, we can find clearance volume as 30.95 cm^3 and we can find clearance height with previous equation. Let us assume clearance height as "x";

$$V_c = 30.95 = \pi * \frac{(Bore)^2}{4} * x$$

We can find clearance height value x=0.55 cm or 5.5 mm and we can make input to the software. In piston crank offset part we make values default or 1, because we do not have information for theese values.

3.2 Engine Cylinder

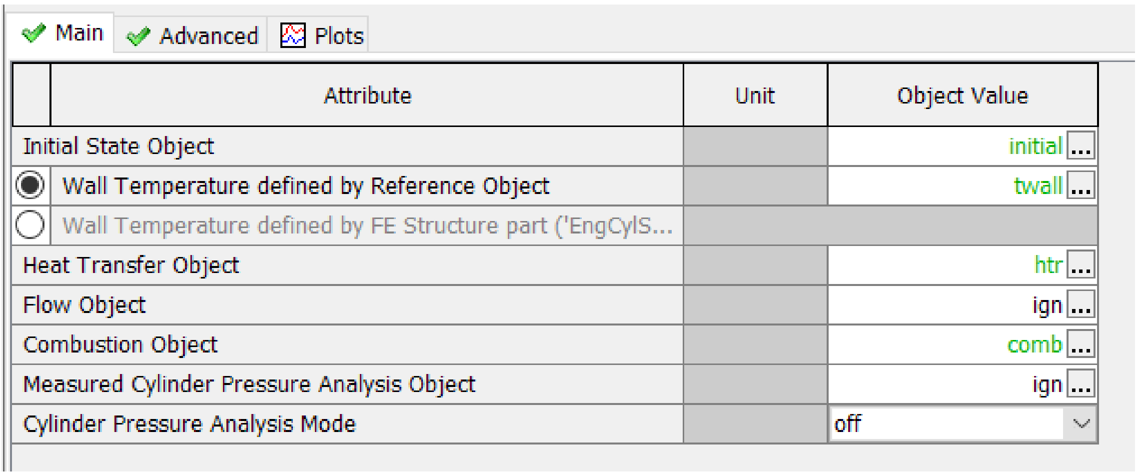


Figure 6: Engine cylinder component in GT-Suite

In figure 6, we can see engine cylinder component in the software, but I could not make changes in this page, because I do not have informations for this. I use default templates for necessary objects.

3.3 Engine Valves

For valves, I have two data files for lifting informations, but I do not know which one is intake or which one is exhaust. To know that, I made an excel graph for valve lifting. In the graph, x-axis is crank angle, y-axis is lifting value;

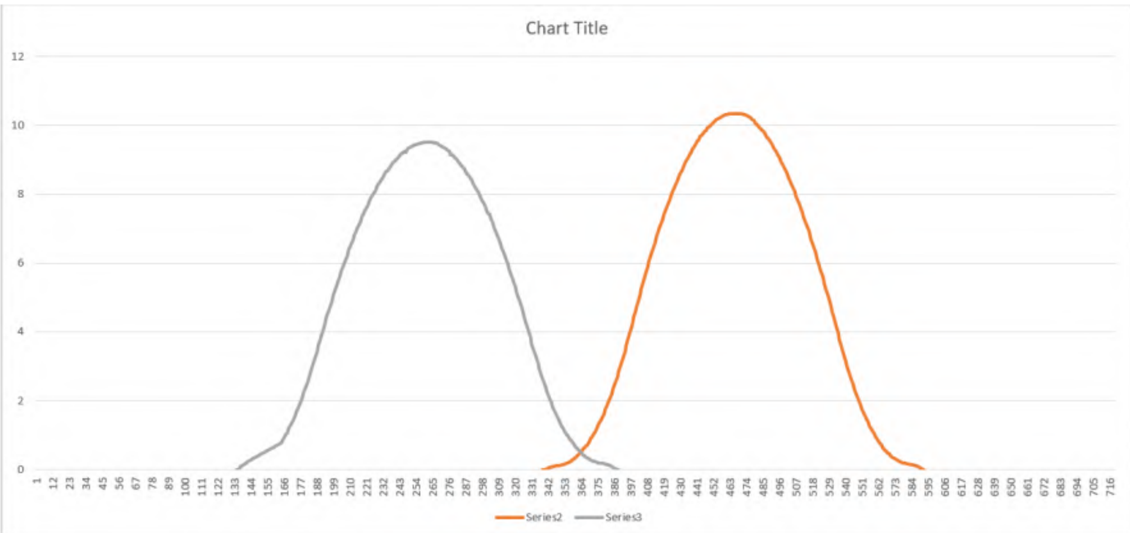


Figure 7: Valve lift graph with crank angle

In figure 7, we can see the valve overlap phenomenon. We know the overlap is a situation that both valves are open for a while. For positive overlap, while exhaust valve closing, intake valve starts opening movement. In that case, we can see the grey line is exhaust valve and orange line is intake valve.

3.3.1 Intake valve

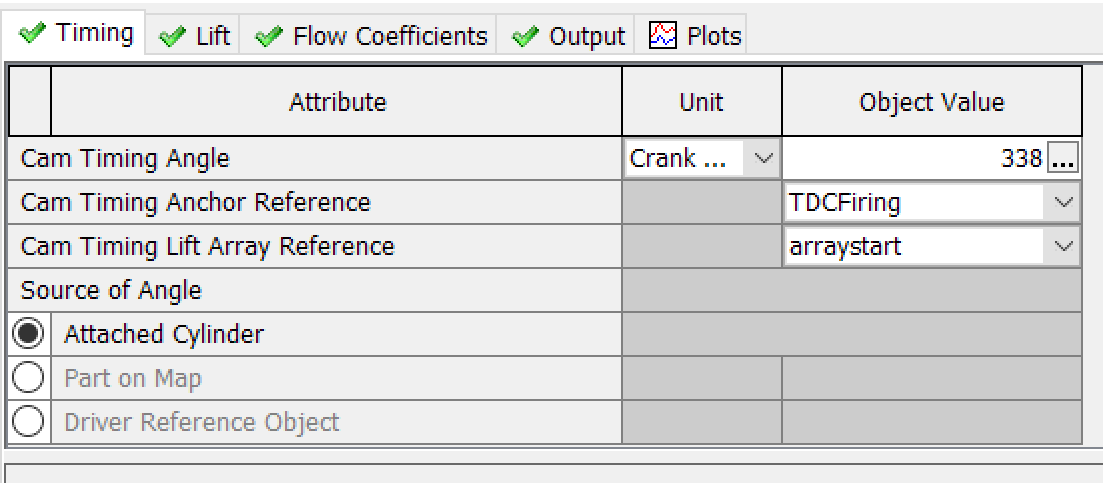


Figure 8: Timing values of intake valve

From previous part, we know orange line data set is intake valve. In the data set, we know lifting starts in 338 degree crank angle. I arrenged values as timing angle is 338 degree, top dead center firing as we know. The lift array reference is obtained as "arraystart"

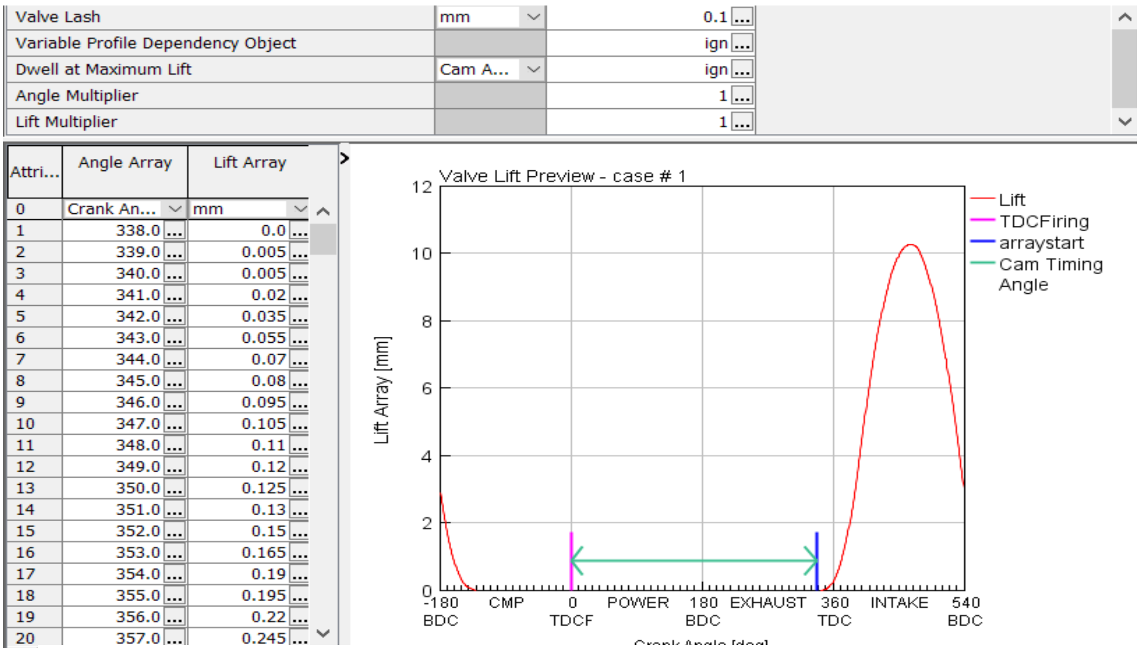


Figure 9: Lift array and graph of intake valve

because lifting graph stands in a logical place with this option. Attached cylinder selection comes with component selection and I did not make any change. After we obtained values in this part, we can click “lift” button. In figure 9, we can see lifting graph of intake valve. I obtained valve lash value as 0.1 mm for production tolerances. In lifting graph, we see some overflowing into the compression area. That is because production safety in cam profile. This is unpredictable phenomenon for cam profile design.

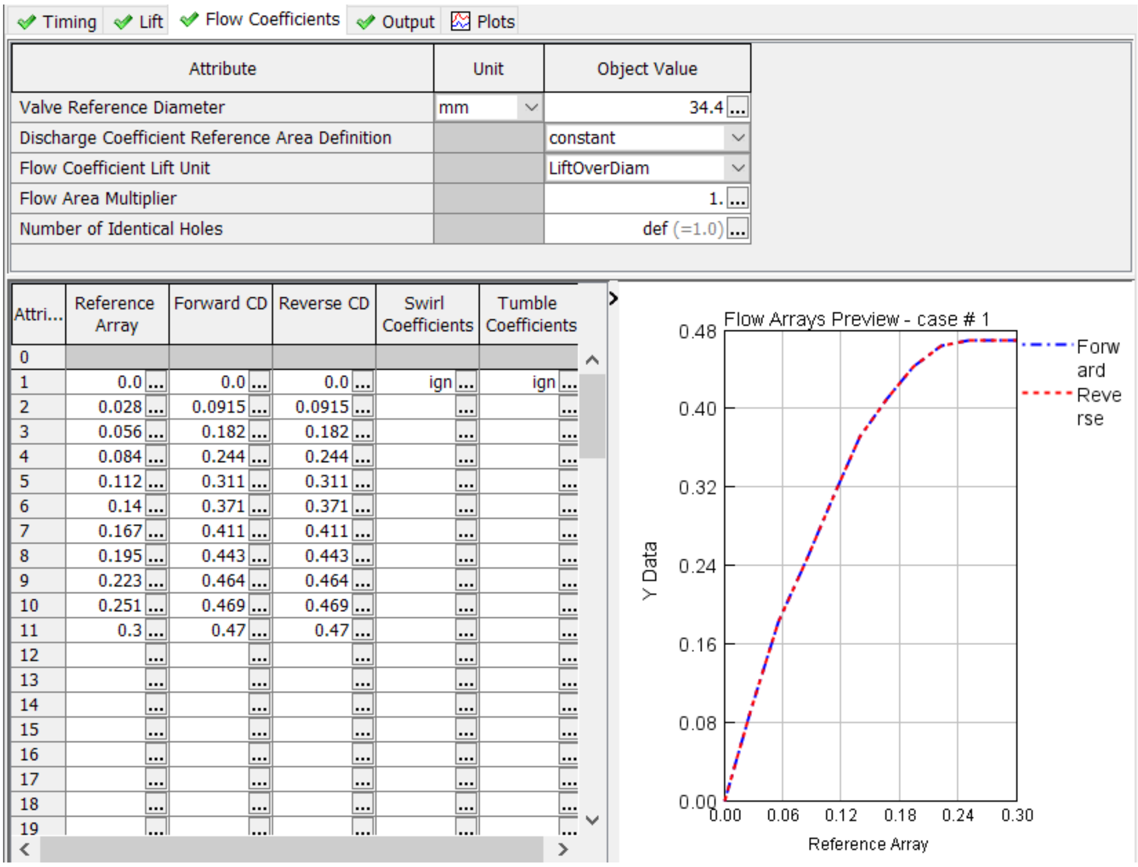


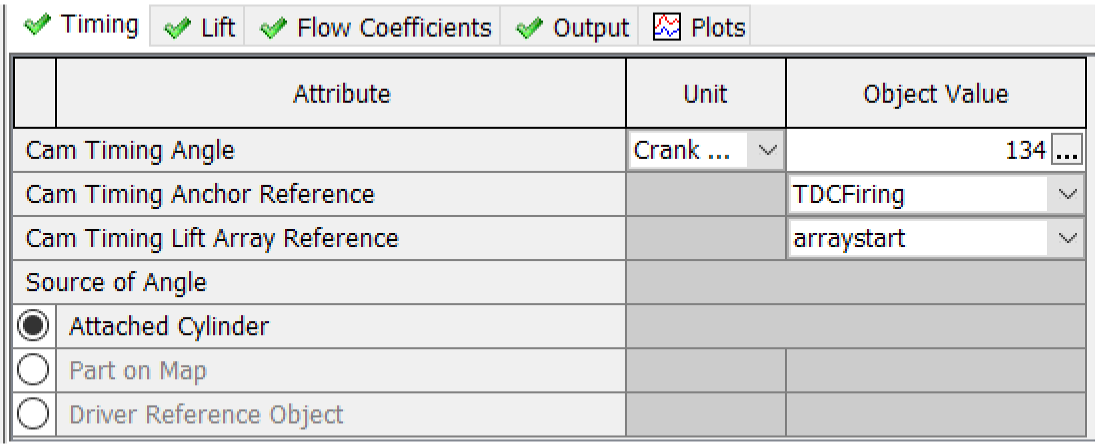
Figure 10: Flow coefficient values of intake valve

In figure 10, reference array is Lift/Diameter ratio. I find the diameter value from solidworks assembly file. We can select some crank angles and read the lift value, then we can calculate L/D ratios. But maximum L/D value can be calculated as;

$$\left(\frac{L}{D}\right)_{max} = \frac{Max\ lift}{Diameter} = \frac{10.345}{34.4} = 0.30$$

Forward and reverse CD values can be calculated with experiment values. CD value is test flow over potential orifice flow. Then I use default values from software. Forward and reverse CD values are same because our cam profile is symmetric.

3.3.2 Exhaust valve

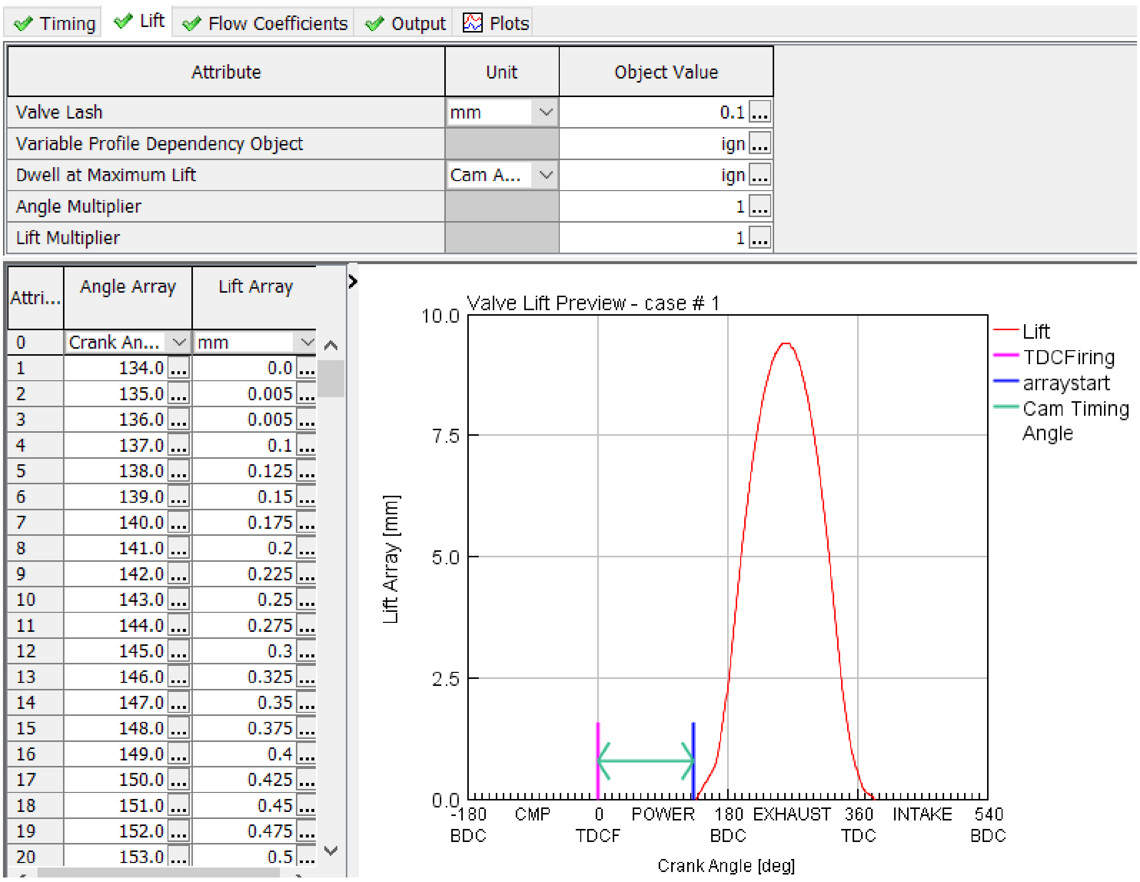


Timing | Lift | Flow Coefficients | Output | Plots

Attribute	Unit	Object Value
Cam Timing Angle	Crank ...	134
Cam Timing Anchor Reference		TDCFiring
Cam Timing Lift Array Reference		arraystart
Source of Angle		
Attached Cylinder		
Part on Map		
Driver Reference Object		

Figure 11: Exhaust valve timing informations

From previous parts, we know grey line is exhaust valve and in the data set we can see lifting starts in 134 degree crank angle. The other datas are same with intake valve. After this arrangement we click the lift selection.



Timing | Lift | Flow Coefficients | Output | Plots

Attribute	Unit	Object Value
Valve Lash	mm	0.1
Variable Profile Dependency Object		ign
Dwell at Maximum Lift	Cam A...	ign
Angle Multiplier		1
Lift Multiplier		1

Attri...	Angle Array	Lift Array
0	Crank An...	mm
1	134.0	0.0
2	135.0	0.005
3	136.0	0.005
4	137.0	0.1
5	138.0	0.125
6	139.0	0.15
7	140.0	0.175
8	141.0	0.2
9	142.0	0.225
10	143.0	0.25
11	144.0	0.275
12	145.0	0.3
13	146.0	0.325
14	147.0	0.35
15	148.0	0.375
16	149.0	0.4
17	150.0	0.425
18	151.0	0.45
19	152.0	0.475
20	153.0	0.5

Figure 12: Lifting graph of exhaust valve

In figure 12, we see the lifting array of exhaust valve. Valve lash is same with intake valve as 0.1 mm. There is also an overflowing into the power stroke area. That is for production safety again. Exhaust valve is usually smaller than intake valve, because taking air is much more important thing for an engine.

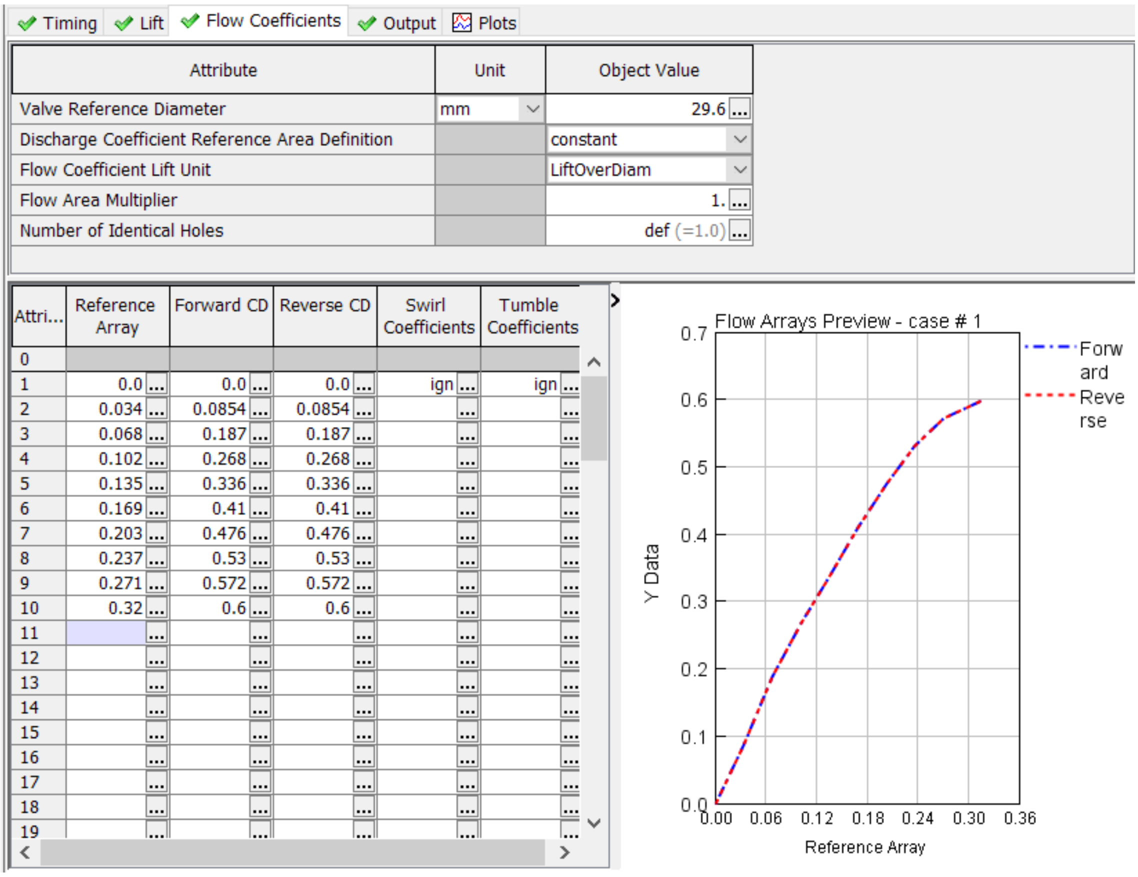


Figure 13: Flow cofficients of exhaust valve

Another difference between intake and exhaust valve is lift value. For reference array value, we need to make another calculation. In that case, maximum lift value is 9.505 mm and diameter is 29.6 mm.

$$\left(\frac{L}{D}\right)_{max} = \frac{Max\ lift}{Diameter} = \frac{9.505}{29.6} = 0.321$$

We can see a little difference between calculation value and software value, because the calculation value is the maximum value for reference array. We can not input a value bigger than $(L/D)_{max}$ result. I choose 0.32 to stay in safety place for reference array value. For reverse and forward CD values, I use default values again, because I do not have data for it.

3.4 Intake and Exhaust Ports

In this part, I used Solidworks to measure pipe diameters. However, I could not measure pipe length, discretization length and radius of bend. Because the Solidworks file could not open effectively by my computer. I could not convert engine head part from mesh file to part file. Then I used default values for theese length and radius values.

Main | Thermal | Pressure Drop | Plots

	Attribute	Unit	Object Value
	Basic Geometry and Initial Conditions		
	Diameter at Inlet End	mm	40
	Diameter at Outlet End	mm	def (=Diameter at I...
	Length	mm	80
	Discretization Length	mm	40
	Initial State Name		initial
	Surface Finish		
○	Smooth		
◉	Roughness from Material		cast_iron
○	Sand Roughness	mm	
	Additional Geometry Options		
	Radius of Bend	mm	ign
	Angle of Bend	deg	ign
	Pipe Elevation Change or 3D Acceleration Object	mm	ign
	Number of Identical Pipes		def (=1.0)

Figure 14: Intake port values for the engine

Main | Thermal | Pressure Drop | Plots

	Attribute	Unit	Object Value
	Basic Geometry and Initial Conditions		
	Diameter at Inlet End	mm	30
	Diameter at Outlet End	mm	def (=Diameter at I...
	Length	mm	60
	Discretization Length	mm	55
	Initial State Name		initial
	Surface Finish		
○	Smooth		
◉	Roughness from Material		cast_iron
○	Sand Roughness	mm	
	Additional Geometry Options		
	Radius of Bend	mm	ign
	Angle of Bend	deg	ign
	Pipe Elevation Change or 3D Acceleration Object	mm	ign
	Number of Identical Pipes		def (=1.0)

Figure 15: Exhaust port for the engine

Diameter at the inlet port is bigger than exhaust port due to same thing with valve diameters. Engines are focused to take more air, more oxygen and make efficient combustion. Effective combustion means more and efficient power. Also I do not have material information for engine parts. You can see “cast iron” selection, it is default selection.

3.5 Diesel Injector

The injector is the part with errors, because needed fuel mass and given consumption value from the manual is not same. Also the value in software is different to get acceptable results. Because, there are three different power values for different cases and one consumption value in manual. We can make a calculation for diesel combustion.

$$4\ C_{12}H_{23} + 71\ O_2 \rightarrow 48\ CO_2 + 46\ H_2O$$

When we make mass calculations, we need 2272 grams of oxygen to burn 668 grams of diesel fuel. I made calculations with respect to this combustion equation. Due to displacement volume of engine, we have 510 cm^3 = 0,00051 m^3 air in cylinder. We know air density is 1,184 kg/m^3 at 25^oC.

$$0{,}00051\ m^3 * 1{,}184 \frac{kg}{m^3} = 6{,}0384 * 10^{-4}\ kg\ air\ in\ cylinder$$

Oxygen mass in air;

$$6{,}0384 * 10^{-4} * \frac{21}{100} = 1{,}268 * 10^{-4}\ kg\ oxygen\ in\ cylinder$$

$1.268*10^{-4}$ kg oxygen is equal to 0.1268 grams of oxygen. Then, we can use combustion quation values to get needed diesel mass for combustion in cylinder.

$$Comb\ diesel\ mass = 0{,}1268 * \frac{668}{2272} = 0{,}03728\ grams\ of\ diesel$$

I use miligram values in software, so 0.03728 gram is equal to 37.28 miligrams of diesel. With this values, we get AFR (air fuel ratio) is equal to 16.2 in maximum load condition. We need to make consumption calculation for an hour in maximum load condition.

In 4-stroke engines, we know there is one combustion in two revolutions of crankshaft. Maximum revolution number is 3000 rpm (revolution per minute) for this engine. It means 1500 combustion in a minute. We have 0,03728 grams of diesel for one combustion.

$$0{,}03728 * 1500 = 55{,}92 \; grams\ of\ diesel\ per\ minute$$

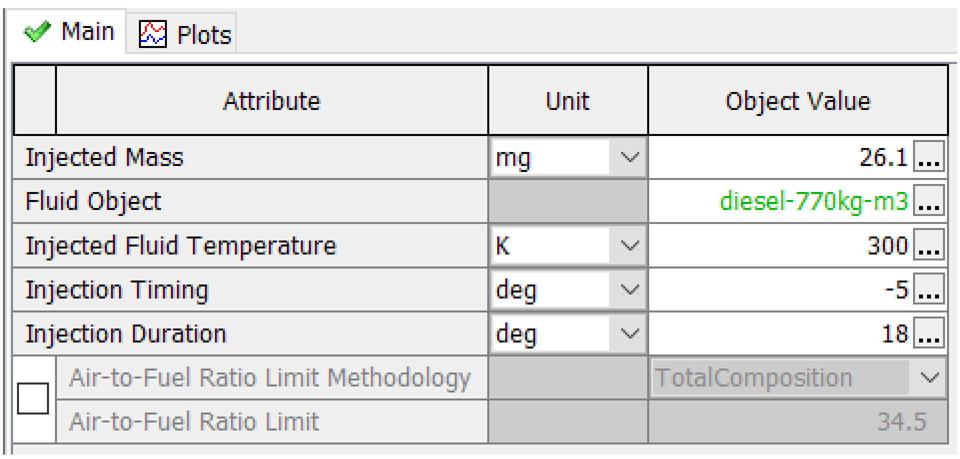
$$\frac{55{,}92}{1000} * 60 = 3{,}36 \; kg\ of\ diesel\ per\ hour$$

Fuel density of $C_{12}H_{23}$ is equal to 772,88 kg/m^3 and we can calculate liter/hour value. First of all we need kg/liter value for diesel fuel. Then, we get the density 0,77288 kg/liter.

$$\frac{3{,}36\,\frac{kg}{hour}}{0{,}77288\,\frac{kg}{liter}} = 4{,}34 \; liter\ per\ hour\ in\ maximum\ load\ condition$$

We have three conditions in manual as using types.

- N (DIN 70020) [Automotive rating]: For intermittent duty operation at variable speed and load. 12 hp @3000 rpm
- NB (DIN 6270) [Rating with no overload capacity]: For continuous light duty operation with constant speed and variable load. 10 hp @3000 rpm
- NA (DIN 6270) [Continuous rating with overload capacity]: For continuous heavy duty operation with constant speed and load. 9 hp @3000 rpm



	Attribute	Unit	Object Value
	Injected Mass	mg	26.1
	Fluid Object		diesel-770kg-m3
	Injected Fluid Temperature	K	300
	Injection Timing	deg	-5
	Injection Duration	deg	18
	Air-to-Fuel Ratio Limit Methodology		TotalComposition
	Air-to-Fuel Ratio Limit		34.5

Figure 16: Simple diesel injector component in GT-Suite

Our calculated value is 37,28 miligrams of diesel for N condition. However, I need to write into the software 26,1 miligrams to get 12 hp @3000 rpm result. With this value, we have AFR=23,1 which is also acceptable. In manual, the consumption value is 1,9 liter/h in NB condition. I could not compare our calculated result and manual data because of the condition difference.

3.6 Environment

Internal combustion engines are open systems. Power generation process starts with getting air from environment, burning fuel with air and making piston movement. That is why we need environment components in first and last edges in our model.

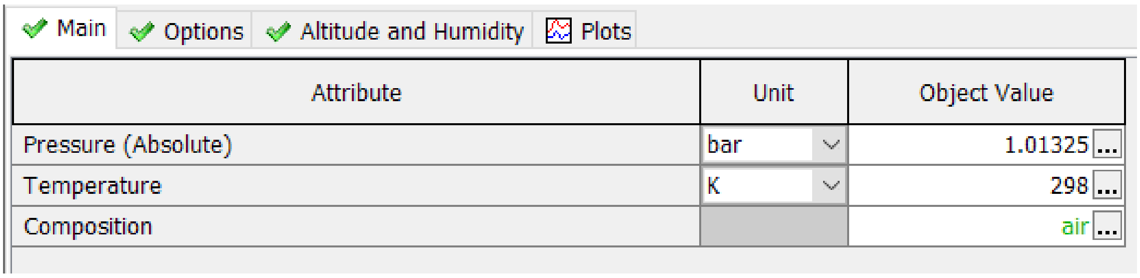


Figure 157: Environment component in GT-Suite

As we know, athmospheric pressure at sea level and 25°C is 1,01325 bar. I use same values for two environment components. If we have turbo compressor or supercharger, we can enter pressure value as compressor output value while we want to model only engine not compressor part. But our engine is naturally aspirated 1-cylinder diesel engine. I use air with athmospheric pressure at 25°C.

3.7 Case and Run Setup

With those components, we finish our model. Then, we need to define our cases. In this article, we have one case with respect to rpm value. I started with 600 rpm and increased by 100 rpm. Then, we have 25 cases to run. In every case, software starts to solve the model until it reaches steady state. When it reached steady state, software finishes the calculation and starts to next case. For divergence situations, we have another option for it as run setup. In run setup, there is an option that is maximum simulation duration with cycle number.

Main | Folder1 | All

Parameter	Unit	Description	Case 8	Case 9	Case 10
Case On/Off		Check Box to Turn Case On	☑	☑	☑
Case Label		Unique Text for Plot Legends	Speed = 1300	Speed = 1400	Speed = 1500
RPM	RPM	Engine Speed	1300	1400	1500

Figure 18: Case setup in GT-Suite

In figure 17, we can see some cases from this setup and their rpm values. When we make changes in model, we can turn off or turn on cases to get results quickly. Because, the CPU time increases with case number increment.

TimeControl | Initialization | FlowControl | ODEControl | SignalControl | ThermalControl

Attribute	Unit	Object Value
Time Control Flag		periodic
◉ Maximum Simulation Duration (Cycles)		10
Minimum Simulation Duration (Cycles)		ign
○ Maximum Simulation Duration (Time)	s	
Minimum Simulation Duration (Time)	s	ign
Automatic Shut-Off When Steady-State		on
Main Driver (Defines Periodic Frequency)		
◉ Automatic		
○ Part Name		
○ Reference Object		
Improved Solution Sequence for Multi-Circuit Models		☐

Figure 19: Run setup window in GT-Suite

We can see "Automatic shut-off when steady state" option to solve the model quickly. This setup combination helps us to solve model quickly and to avoid from divergence situations. If a case diverged, software will stop to solve the case after 10 cycle. After we prepared all this components and setups, we can run this model.

4. RESULTS AND DISCUSSION

4.1 Power and Torque

For an internal combustion engine, we need to see some main graphs, such as power and torque with respect to rpm, break spesific fuel consumption, pressures, emissions etc. After the software finished to solve model, I clicked wiew results button to open GT-Post application.

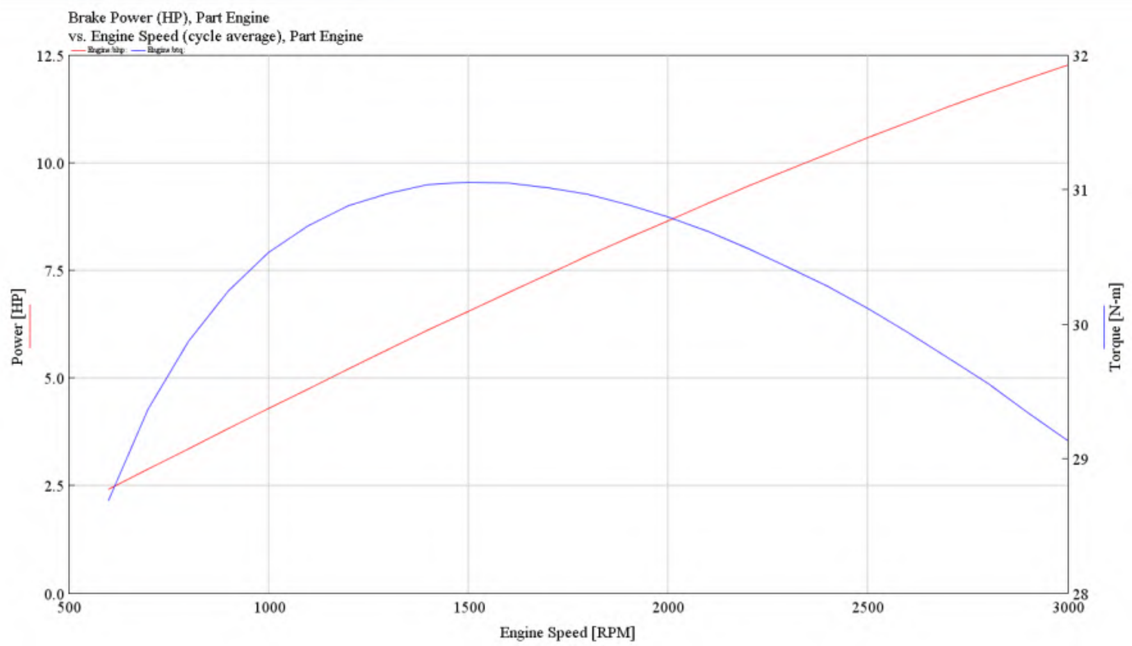


Figure 20: Power an torque graph with respect to rpm

In figure 19, we can see the main graph of an internal combustion engine that is power and torque diagram. I solved the model in N condition. Here we can see red line as power and blue line as torque values at given rpm.

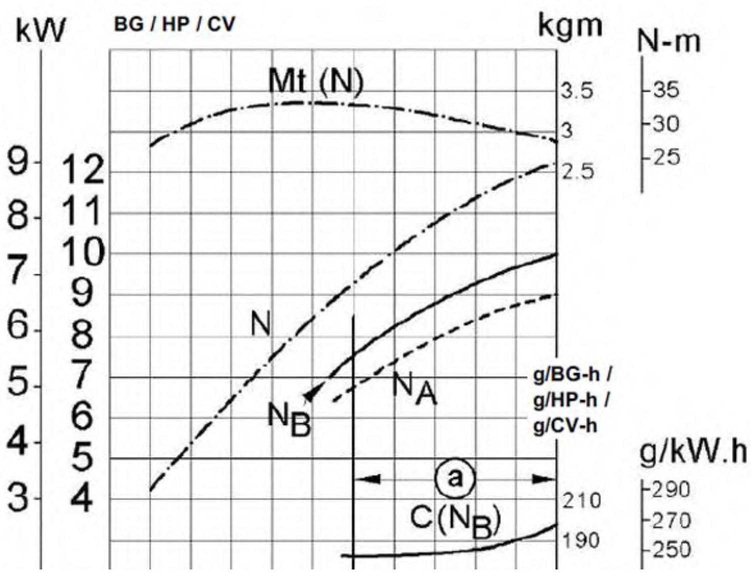


Figure 21: Experimental data of engine from user manual

The maximum torque of the engine is 32 N.m at 1800 rpm but in our model, maximum torque value 32 N.m occurs at 1500 rpm. I do not make different cases for diesel injector, may be this error occured by this. Intersection of lines occurs at manual graph too, but there is a scale difference between left and right side. Torque values are acceptable with manual data. We can see a very similar shape in power lines. For example at 1000 rpm we get same value with manual.

4.2 Break Specific Fuel Consumption

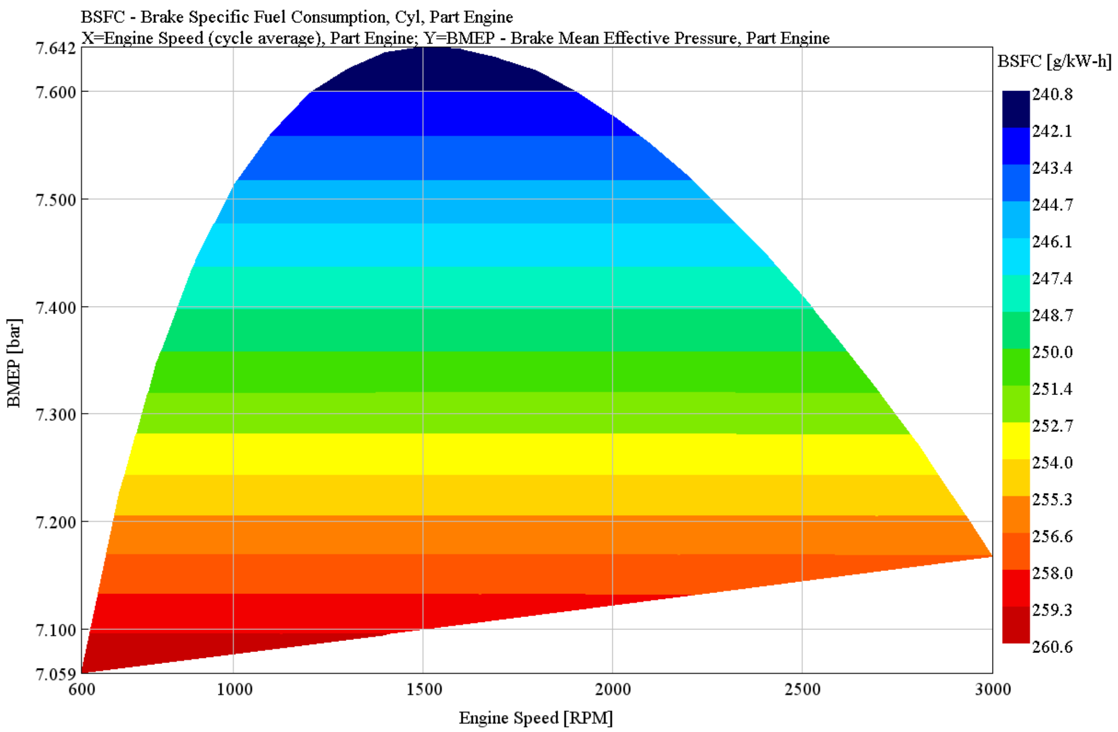


Figure 22: BSFC (break specific fuel consumption graph of the engine

Brake-specific fuel consumption (BSFC) is a measure of the fuel efficiency of any prime mover that burns fuel and produces rotational, or shaft power. It is typically used for comparing the efficiency of internal combustion engines with a shaft output. It is the rate of fuel consumption divided by the power produced. In traditional units, it measures fuel consumption in pounds per hour divided by the brake horsepower, lb/(hp·h); in SI units, this corresponds to the inverse of the units of specific energy, kg/J = s^2/m^2. It may also be thought of as power-specific fuel consumption, for this reason. BSFC allows the fuel efficiency of different engines to be directly compared. The term "brake" here as in "brake horsepower" refers to a historical method of measuring torque.

Brake Mean Effective Pressure (BMEP) is another very effective value for comparing the performance of an engine of a given type to another of the same type, and for evaluating the reasonableness of performance claims or requirements.

4.3 Pressure at Crank Angle

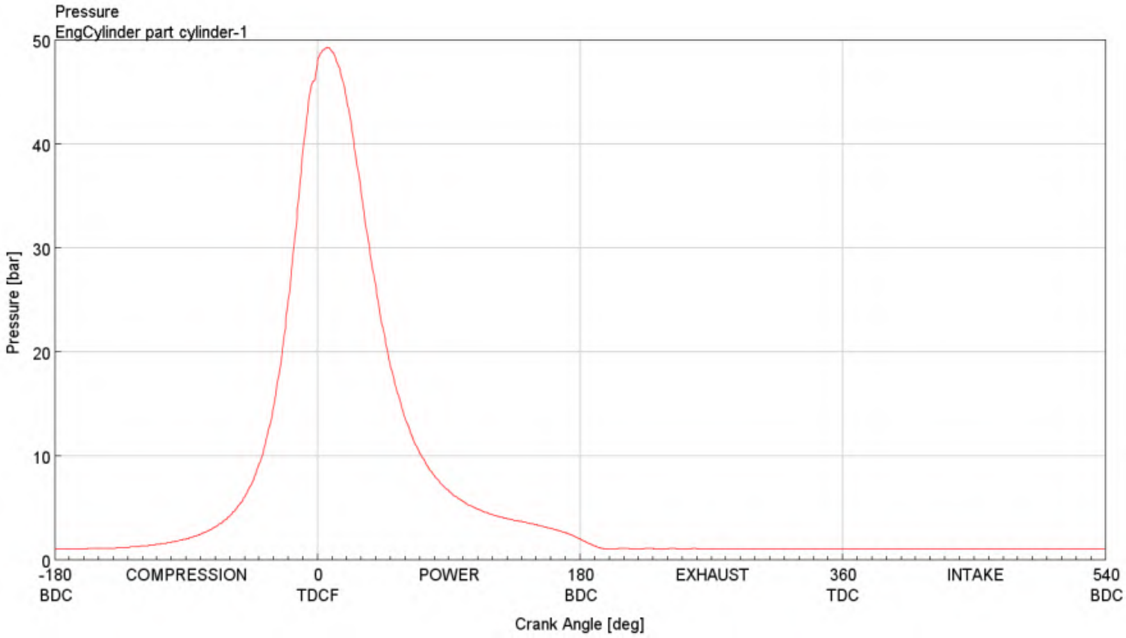


Figure 23: Pressure value (bar) at given crank angle

Figure 22 gives us pressure value at one comlete cycle. Complete cycle duration is two revolution of crank or 720° angle. We can see the pressure rises at compression phase and there is a jump at 0° angle. Our engine works with TDCF (top dead center firing), so when the engine burns the fuel, there will be a pressure increment at firing step 0° angle.

4.4 Temperature at Crank Angle

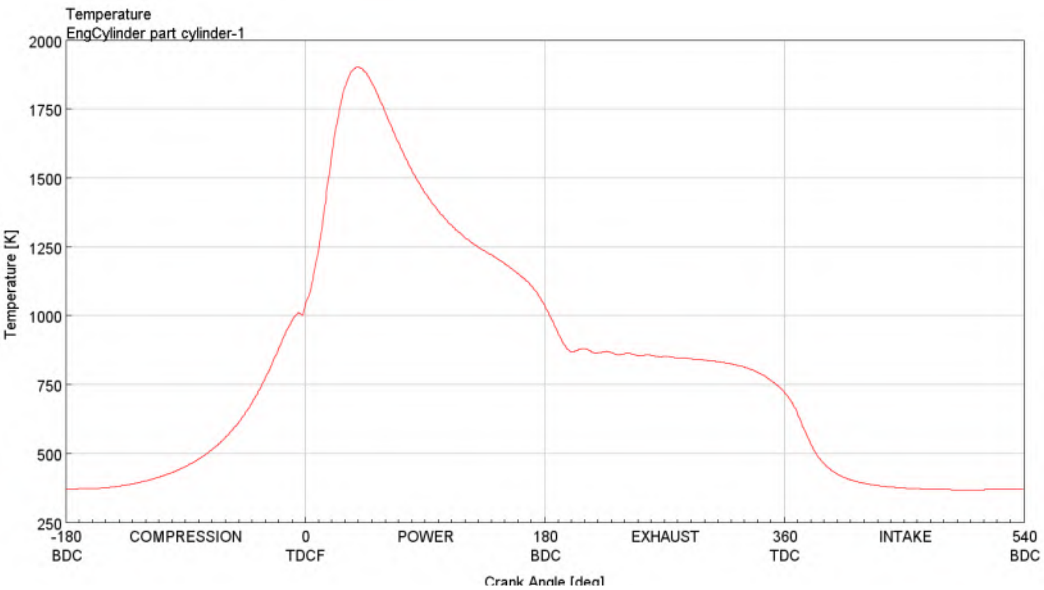


Figure 24: Temperature (K) values at given crank angle

In figure 23, we see the temperature values at given crank angle for a complete cycle. In the beginning of compression, temperature starts rising according to Gay Lussac rule. There is a jump point at 0° angle again, because fuel burning is exhotermic reaction. Temperature decrease at power stroke is meaningful, because when volume increases at same pressure, temperature will decrease too.

4.5 P-V Diagram

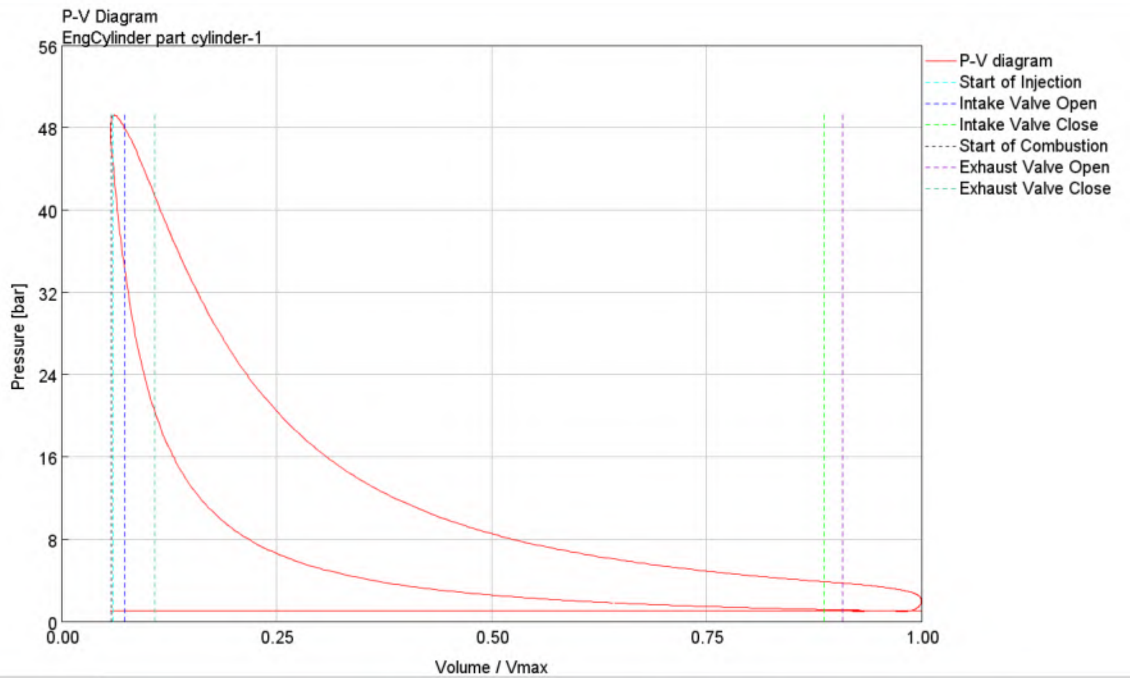


Figure 25: P-v diagram of the engine at 600 rpm

We can see at figure 25, there is a big difference between real cycle and ideal cycle. The image shows a p-V diagram for the ideal Diesel cycle; where p is pressure and V the volume or v the specific volume if the process is placed on a unit mass basis. The idealized Diesel cycle assumes an ideal gas and ignores combustion chemistry, exhaust and recharge procedures and simply follows four distinct processes:

1→2: isentropic compression of the fluid

2→3: reversible constant pressure heating

3→4: isentropic expansion

4→1: reversible constant volume cooling

4.6 T-s Diagram

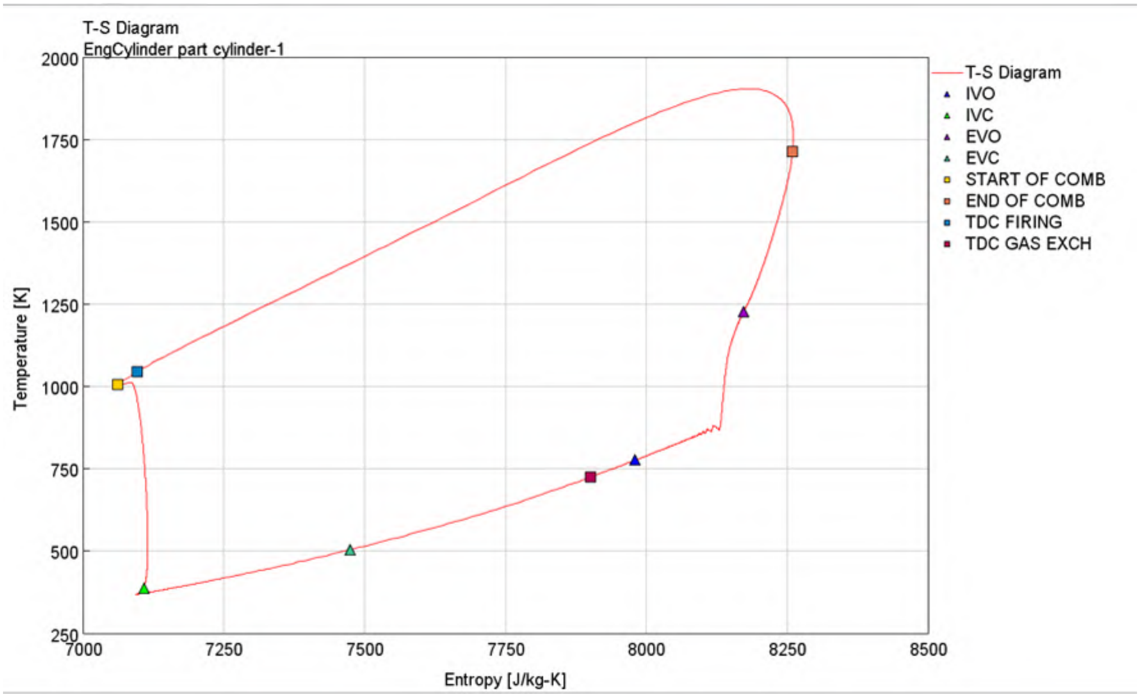


Figure 26: T-s diagram of the engine at 600 rpm

We can see much more difference between ideal cycle and our result in figure 25.

4.7 Heat Release Diagram

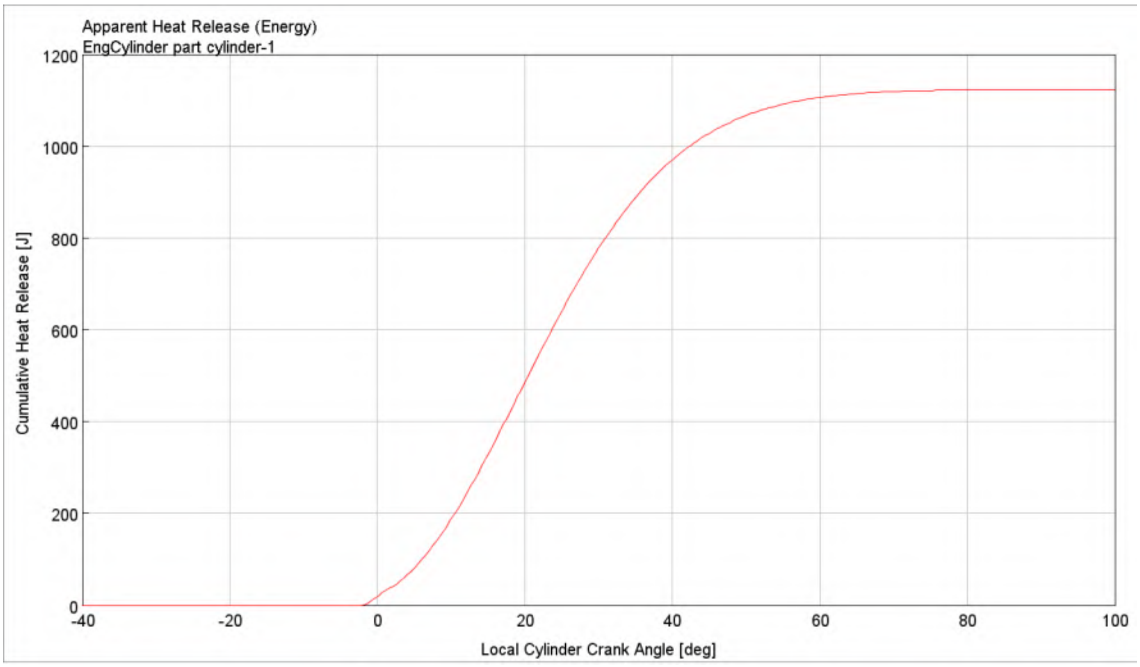


Figure 27: Heat release at given crank angle

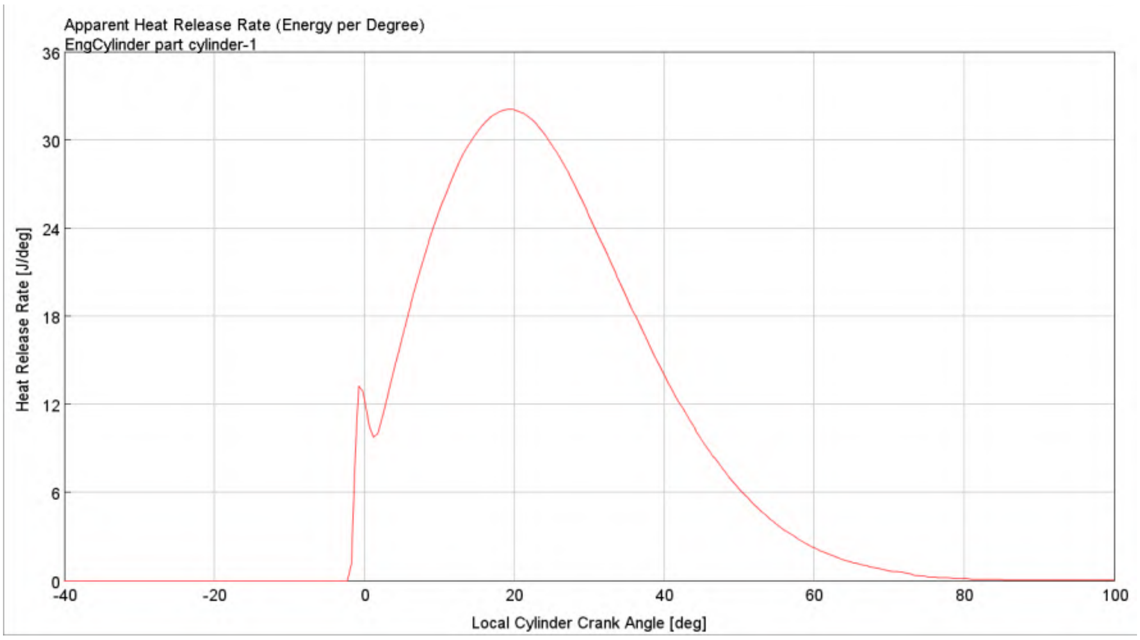


Figure 28: Heat release rate per crank angle

In this type of diagram, the combustion is characterized by four phases. The first phase characterizes the ignition delay, which corresponds to the period between when the fuel injection starts and the first fuel burn signal. The last stage, late combustion phase, has a low combustion rate that decreases until the phase is completed.

4.8 Emissions

4.8.1 CO emission

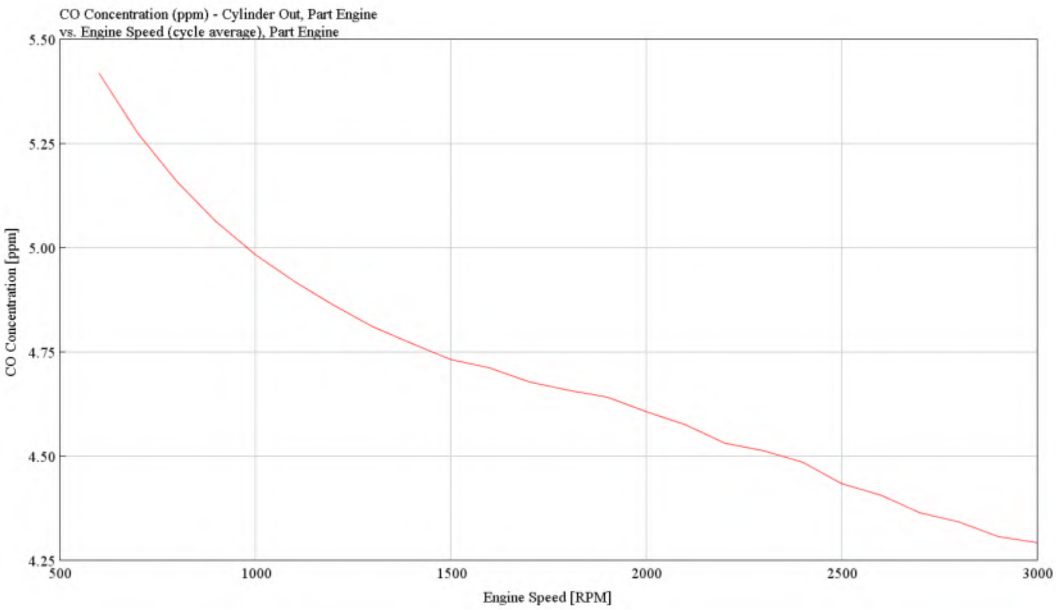


Figure 29: Carbon-monoxide emission at given rpm

When diesel fuel burns completely, production chemical is CO_2. However, when the fuel did not burn exactly, there is carbon-monoxide. I think its reason is I did not make cases for fuel injection mass at rpm value. So at low rpm, there is more fuel for combustion in our model.

4.8.2 Hydrocarbon emission

Diesel engines normally emit low levels of hydrocarbons. Diesel hydrocarbon emissions occur principally at light loads. The major source of light-load hydrocarbon emissions is lean air–fuel mixing. In lean mixtures, flame speeds may be too low for combustion to be completed during the power stroke, or combustion may not occur, and these conditions cause high hydrocarbon emissions.

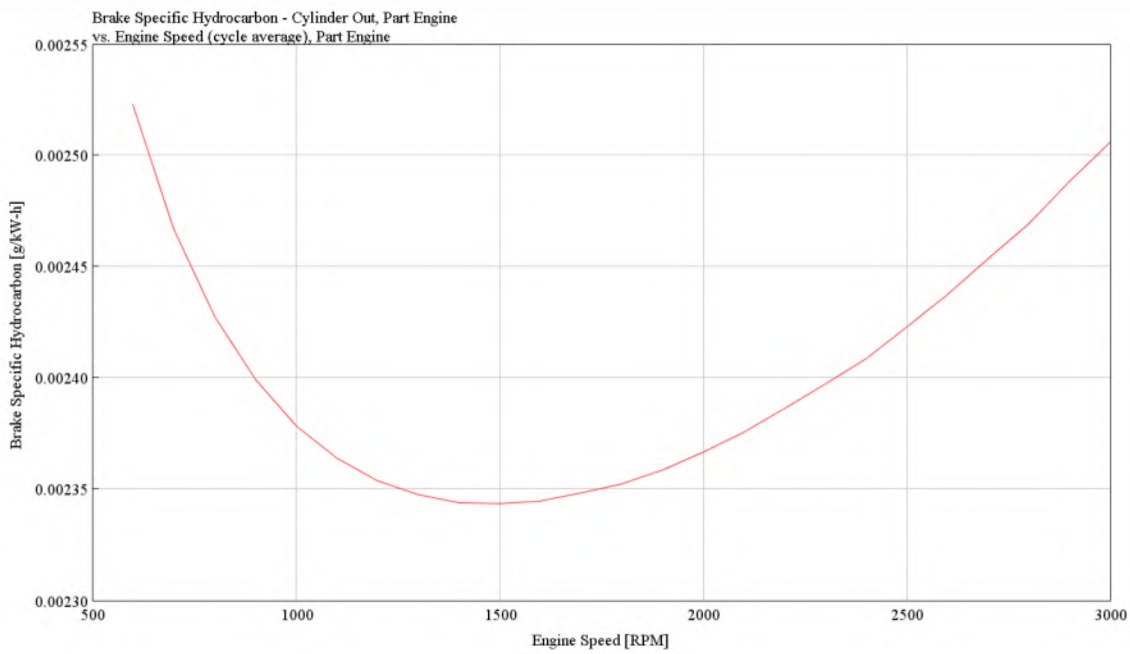


Figure 30: HC (hydrocarbon) emission at given rpm

Hydrocarbon emission is like reverse of torque diagram. We can see the unit g/kW-h and I think the unit is effective for that diagram shape. On the other hand, the engine can be more efficient at maximum torque rpm.

4.8.3 NO_x concentration at crank angle

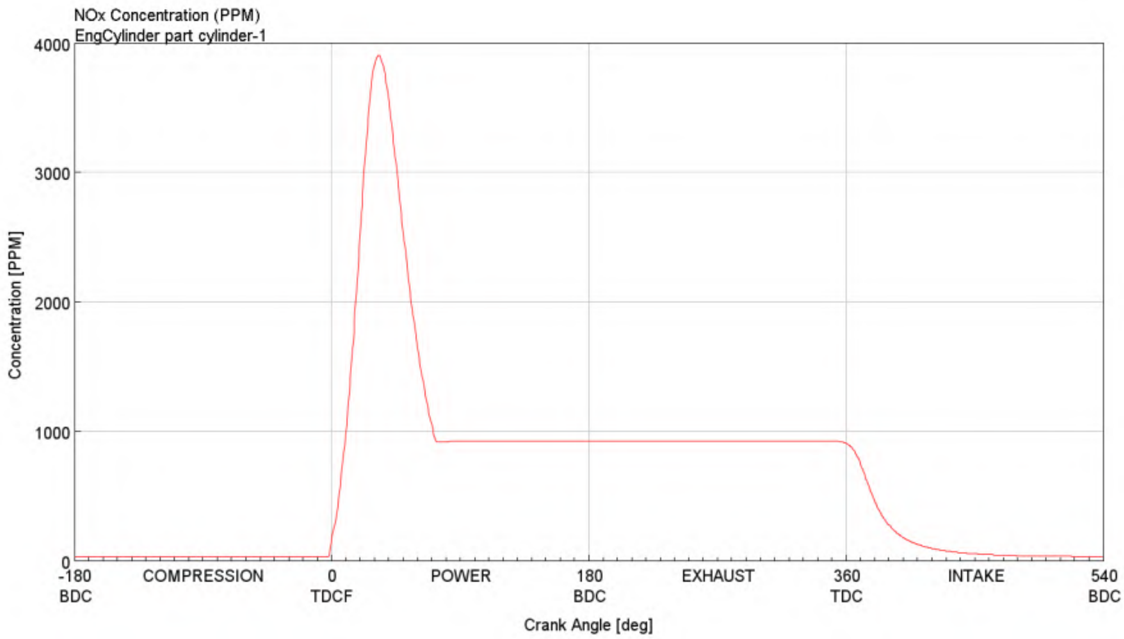


Figure 31: NOx concentration (ppm) at crank angle

5. CONCLUSSION

While doing this thesis, I gained detailed information about diesel engines as well as how to model a diesel engine. I learned how an engineer should approach a problem, especially at some points where I made an error and fixed that error. I used Excel and Solidworks as support software. These were software I already knew, but I learned a lot about using GT-Suite by starting from scratch. Where I needed to calculate, I extracted formulas and found results using them. I learned a lot about valve lift and camshaft design.

As a result, I learned a lot in this thesis, most importantly, I enjoyed it. Later, I would like to experiment with GT-Suite software for hobby purposes.

6. REFERENCES

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