BACHELOR THESIS – ME1841038

ANALYSIS OF COMBUSTION PERFORMANCE ON DUAL FUEL DIESEL ENGINE BASED ON SIMULATION

THERESIA DIANITA
NRP. 04211641000004

SUPERVISOR :
Prof. Semin, S. T., M. T., Ph. D.
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SKRIPSI - ME 1841038

ANALISA PERFORMA PEMBAKARAN MESIN DIESEL BERBAHAN BAKAR GANDA DENGAN METODE SIMULASI

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Submitted to Comply One of Requirements to Obtain a Bachelor Engineering Degree
In
Laboratory of Marine Power Plant (MPP)
Double Degree of Marine Engineering Program
Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember

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Surabaya, January 2020

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DECLARATION OF HONOUR

With signed below, I declare that:
This has been written and developed independently without any plagiarism act. All contents and ideas drawn directly from internal and external sources are indicated such as citied sources, literatures, and other professional sources.

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Surabaya, January 30th, 2020

Theresia Dianita

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PREFACE

First of all, I would like to say thank to the Almighty God who has given guidance, pleasure, and opportunities so author can finish this bachelor thesis report well. In the process of making this bachelor thesis report, the author would like to say thank those who have helped to complete this bachelor thesis report, including:

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The author realizes that the bachelor thesis report far from perfection. Therefore, every constructive and supportive suggestion and idea to improve this bachelor thesis are accepted by author so that the author can further do her best.

Finally, the author hope this bachelor thesis report can useful fir the readers. In the further, the author will always improve herself and may all become a person who is blessed and give many benefits to the others.

Surabaya, January 2020

Theresia Dianita

Analysis of Combustion Performance on Dual Fuel Diesel Engine Based On Simulation

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ABSTRACT

The viable alternative fuel for internal combustion is natural gas. Dual fuel engine that used natural gas produced high torque and a better thermal efficiency and generate less emission than conventional diesel engine. Researchers revealed that combustion efficiency on dual fuel engine decrease in low load condition. On this research discuss combustion performance using GT-POWER simulation with diesel and gas consumption variation, that are 70% of diesel:30% of CNG, 65% of diesel:35% of CNG, 60% of diesel:40% of CNG, 55% of diesel:45% of CNG; load variation, that is 1000-4000 watt with interval 1000; and engine speed variation, that is 1800-2200 rpm with interval 100. The analysis is carried out to discuss gas consumption variation effect to cylinder pressure, cylinder temperature, heat release rate, and volumetric efficiency. From 3 variable variations on this research, the biggest cylinder pressure difference is 4% and the smallest is 0.01%. Cylinder temperature increased with increasing gas consumption. The biggest cylinder temperature difference is 1.41% and the smallest is 0.08%. Heat release rate result increased with increasing gas consumption. Volumetric efficiency decreased with increasing gas consumption. The biggest volumetric efficiency difference is 1.88% and the smallest is 0.28%.

Keywords: cylinder pressure, cylinder temperature, dual fuel diesel engine, gas consumption variation, heat release rate, volumetric efficiency

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Analisa Performa Pembakara Mesin Diesel Berbahan Bakar Ganda dengan Metode Simulasi

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ABSTRAK

Bahan bakar alternatif yang layak untuk pembakaran dalam adalah gas alam. Mesin berbahan bakar ganda yang menggunakan gas alam menghasilkan torsi yang tinggi dan efisiensi termal yang baik serta menghasilkan emisi yang lebih rendah dibandingkan mesin diesel konvensional. Beberapa peneliti menyatakan bahwa efisiensi pembakaran mesin berbahan bakar ganda memburuk pada kondisi beban rendah. Pada penelitian ini membahas performa pembakaran menggunakan simulasi GT-POWER dengan variasi substitusi gas dan diesel, yaitu 70% diesel:30% CNG, 65% diesel:35% CNG, 60% diesel:40% CNG, dan 55% diesel:45% CNG; variasi pembebanan, yaitu 1000-4000 watt dengan interval 1000; dan variasi kecepatan mesin, yaitu 1800-2200 rpm dengan interval 100. Analisa yang dilakukan mengenai pengaruh variasi konsumsi bahan bakar terhadap tekanan silinder, suhu silinder, laju pelepasan panas, dan efisiensi volumetrik. Dari 3 variasi variabel pada penelitian ini, selilisih tekanan silinder paling besar adalah 4% dan paling kecil adalah 0.01%. Hasil suhu silinder meningkat seiring dengan meningkatnya konsumsi gas dengan selisih suhu silinder paling besar adalah 1.41% dan paling kecil adalah 0.08%. Hasil laju pelepasan panas adalah meningkat seiring meningkatnya konsumsi gas. Hasil efisiensi volumetrik menurun seiring meningkatnya konsumsi gas dengan selisih efisiensi volumetrik paling besar adalah 1.88% dan paling kecil adalah 0.28%.

Kata kunci: efisiensi volumetrik, laju pelepasan panas, mesin diesel berbahan bakar ganda, suhu silinder, tekanan silinder, variasi konsumsi gas

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CHAPTER I
INTRODUCTION

1.1 Background

Nowadays, the total of diesel-fuelled motor vehicles in the world is increasing. The increasing of diesel-fuelled motor vehicle also was happened in Indonesia that is along with the increasing of Indonesian population. The increasing of diesel-fuelled motor vehicle can increase the demand of diesel fuel. However, the demand of diesel fuel cannot be fulfilled. The production of crude oil for the past 10 years decreases significantly, from 346 million barrels in 2009 become around 283 million barrels in 2018. The decreasing of the crude oil production is caused by the main wells productivity of crude oil production has decreased. For decreasing the diesel fuel consumption, it needs to divert the used of diesel fuel into alternative fuel (Indonesia Energy Outlook, 2019).

The researchers found the some alternative which is viable alternative to replacing the conventional fossil fuels. One of viable alternative for conventional fossil fuels is natural gas. Natural gas has been emerged as fuel in transportation sector. The advantages of natural gas use in internal combustion engines are generating less carbon dioxide emissions, attractive cost, better combustion efficiency, renewability through the biomass production process, etc. Natural gas is also a clean fuel that generates less particulate matter emissions than diesel combustion. Therefore, replacing diesel fuel into natural gas in internal combustion is a great interest of industries (Cheenkachorn et al., 2013).

Natural gas is an inexpensive low carbon fuel that produces less carbon dioxide emissions than diesel combustion. Natural gas on transportation sector can use in the form of compressed natural gas (CNG) and liquefied natural gas (LNG). CNG is more popular for automobiles, while LNG is mainly use for transportation and electricity production. To apply natural gas in internal combustion, dual fuel combustion is more practical and an efficient way.

Dual fuel engine is a diesel engine that can run on both gaseous and liquid fuels. That gas is injected to the intake port, then the premixed air and natural gas is injected to in-cylinder. Therefore, it needs a gas injector to inject gas to the intake port. There is required less modification job to convert a conventional diesel engine into a dual fuel engine. Converted of conventional diesel engine into a dual fuel engine can be implemented with install gas injector which the gas will be injected to intake port. The dual-fuel combustion system features essentially a homogeneous gas-air mixture compressed rapidly below its auto-ignition conditions, and ignited by the injection of pilot liquid fuel near the top dead centre position.

Interest in natural gas replacement of diesel fuel in compression ignition (CI) has substantially increased in recent years. Natural gas in dual fuel engines produce high output torque and a better thermal efficiency. It also reduces NO_x emission by lowering the charge temperature, which reduces the NO_x formation. Methane is the main component in natural gas and has a high octane number, which makes it knock resistant and suitable for high compression ratio engines. Although the use of dual fuel engines with natural gas is more efficient, there are still some problems that need to be improved.

Wan Nurdiyana Wan Mansor, 2014, natural gas is mixed with air intake, the amount of diesel used is reduce. The maximum natural gas substitution is limited by knock or emission of carbon monoxide and total hydrocarbon.

Febrian Rohiim, 2019 shows that is decreased performance of dual-fuel engine and there is methane slip. Methane slip is unburned methane when the combustion stage in combustion chamber. Thus, the unburned methane is wasted with exhaust gas. The right gas-diesel fuel energy share ratio is important at dual-fuel engine so can decrease the unburned methane.

Dimitriou et al., Tsujimura et al., and Suzuki et al. shows the poor combustion efficiency at low-load condition of dual fuel engine. To improve the combustion efficiency is conducted by controlling combustion phasing by changing the diesel injection pattern, changing the compression ratio, or implementing low temperature combustion. Semin, 2009 shows that conversion diesel engine into CNG and diesel engine can be decreased the maximum combustion temperature. In the low speed, the conversion CNG engine can be increase the combustion temperature, but the increasing engine speed can be decrease the combustion temperature. So, this statement is become problems. When the engine operation at low speed it can be increased the combustion temperature but it will reduce the combustion efficiency. While at high speed operation it can be decrease the combustion temperature and the combustion efficiency can be increase. But with the decreasing of combustion temperature, it will increase the methane slip that the methane is unburned because the low temperature of combustion. The increasing of methane slip means the exhaust gas emissions is increase.

According to Mateos Kassa and Carrie Hall, dual fuel combustion strategies become advantageous on both spark-ignited (SI) and compression-ignited (CI). On SI engines, dual-fuel technologies can be leveraged to combat knock. Knock can damage the engine and is most prevalent at high loads where the efficiency reaches its peak. On CI engines, dual-fuel technologies utilize of an alternative power source. The research of dual-fuel was conducted on single cylinder engine. If the dual-fuel technology is implemented on multi-cylinder, combustion variations and phasing challenges begin to dominate. One such challenge is the occurrence of more significant cylinder-to-cylinder variations that can lead to inconsistent power production and potentially damaging engine conditions.

In this study has been conducted simulation of combustion process using GT-POWER software to comparing a varying gas composition over several cases with the cylinder temperature, cylinder pressure, heat release rate, and volumetric efficiency. GT-POWER software will draw the result of combustion process. It is expected that can define the proper energy share ratio which can get a good combustion process.

1.2 Problem Statement

Based from the background explanation previously, the problem of this research stated as follows:

1. What is the effect of gas composition variation to the cylinder temperature on dual-fuel engine?

2. What is the effect of gas composition variation to the cylinder pressure on dual-fuel engine?
3. What is the effect of gas composition variation to the heat release rate of combustion on dual-fuel engine?
4. What is the effect of gas composition variation to the volumetric efficiency on dual-fuel engine?

1.3 Problem Limitation

For ease of study, preliminary design and analysis, the limitation within this research stated as follows:

1. The simulation software is GT-POWER.
2. The simulation use engine of dual fuel diesel engine Yanmar TF 85 MH Direct Engine Single Cylinder model.
3. This experiment focuses on analyse of combustion process.
4. The combustion parameters that tested are temperature, pressure, and heat release when the combustion occur and the volumetric efficiency on dual fuel engine
5. Dual-fuel engine use 2 fuels, that are diesel fuel and compressed natural gas (CNG).
6. The variation of experiment is the amount of diesel fuel and CNG with modified the fuel and gas injector.

1.4 Research Objectives

Based from the problem statement previously, the objective of this research stated as follows:

1. To analyse the effect of gas composition variation to the cylinder temperature on dual-fuel engine.
2. To analyse the effect of gas composition variation to the cylinder pressure on dual-fuel engine.
3. To analyse the effect of gas composition variation to the heat release rate of combustion process on dual-fuel engine.
4. To analyse the effect of gas composition variation to the volumetric efficiency on dual-fuel engine.

1.5 Research Benefits

The benefit of this simulation is the result of the effect of gas variation composition to combustion process on dual-fuel diesel engine, especially for cylinder temperature, cylinder pressure, heat release rate, and volumetric efficiency. Hopefully, it can be study to improve the combustion process to produce better power, that can know the dual fuel diesel engine effectiveness, especially for cylinder temperature, cylinder pressure, heat release rate, and volumetric efficiency.

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CHAPTER II
LITERATURE STUDY

2.1 Natural Gas

During most of the 19th century, natural gas was used almost exclusively as a source of light, but there is a new opportunity to use natural gas. Natural gas applications have expanded to home appliances such as stoves, clothes dryers, and furnaces and industrial applications such as engines for electrical power generation, pumping, compression and boilers for process heating.

Natural gas is most energy efficient fossil fuel to used instead of oil or coal. Natural gas can be used as fuel. It is also a source of hydrocarbons for petrochemical feed stocks and a major source of element sulphur, an important industrial chemical. Natural gas can be used in compressed or liquid form (Mokhatab et al., 2018) .

Natural gas is a complex mixture of hydrocarbons and non-hydrocarbon constituent and under atmospheric condition. Natural gas is colourless, odourless, tasteless, shapeless, and lighter than air. While natural gas is formed primarily of methane (CH_4). It can also include significant quantities of ethane (C_2H_6), propane (C_3H_8), butane (C_4H_{10}), pentane (C_5H_{12}), and heavier hydrocarbons. Many natural gases often contain nitrogen (N_2), carbon dioxide (CO_2), hydrogen sulphide (H_2S), and other sulphur components (Mokhatab et al., 2018).

The natural gas composition varies based on geographic location where the gas taken and the processing of the gas as shown in Table 2.1. Typically natural gas contains methane, ranging between 85-95% and a small percentage of ethane and propane (Mansor, 2014).

Table 2. 1 Variation of pipeline quality gas compositions from around the world (National Energy Technology, 2005)

Origin	Methane (%)	Ethane (%)	Propane (%)	Butane (%)	Nitrogen (%)
Malaysia	89.8	5.2	3.3	1.4	0.3
Qatar	89.9	6	2.2	1.5	0.4
Australia	89.3	7.1	2.5	1	0.1
USA	94	3.14	0.45	0.1	1
Algeria	87.6	9	2.2	0.6	0.6

Nigeria	91.6	4.6	2.4	1.3	0.1
Oman	87.7	7.5	3	1.6	0.2
Tinidad & Tobago	96.9	2.7	0.3	0.1	0

Most of natural gases are formed by two mechanism, that are biogenic and thermogenic. Biogenic gases are formed by methanogenic organisms in swamps, garbage dumps, and shallow sediment. Whereas thermogenic gasses are formed from organic matter buried in deeper parts of the earth at greater temperature and pressure (Clarke, 2012).

2.2.1 Natural Gas Treatment

Crude natural gas will go through processing. Processing of crude natural gas from wells until meet quality as dry natural gas flowed through a highly complex transmission pipeline and usually involves several processes to remove oil, water condensation and impurities such as sulphur, mercury, and carbon dioxide. In addition to the four processes, it is often necessary to install a scrubber and heater on the wellhead. The main function of the scrubber is to remove sand and other impurities that have a large size. While the heater is used to ensure the temperature of natural gas does not decrease too low and forms a hydrate in the presence of water vapour (H_2O) in the flow of natural gas. Hydrates are crystalline solids like ice that can block the path of natural gas through valves and pipes (Yuhanes, 2011).

The natural gas treatment process generally begins with an initial separation process based on the specific gravity of each fluid, so that oil, water, and gas are separated. Then, the gas conditioning process will be carried out in which the gas will be conditioned in such a way that the impurities contained in natural gas that can interfere in the gas process further can be removed. This gas conditioning process consists of several stages, that are separation of acid gas, separation of sulphur, separation of water vapour content, separation of nitrogen, and separation of mercury. The next process is to process the gas in the fractionation column to separate the components as needed (Yuhanes, 2011).

2.2.2 Liquefied Natural Gas

When natural gas is cooled to approximately -260°F (-162°C) at atmospheric pressure, a condensed liquid forms which is termed liquefied natural gas (LNG). LNG typically contains more than 90% methane. It also contains small amounts of ethane, propane, butane, and some heavier alkanes. The volume reduction is about 1/600th the volume of natural gas at the burner tip. The density of LNG is roughly 0.41 to 0.5 kg/L, depending on temperature, pressure, and composition. The heat value depends on the source of gas and the process that is used to liquefy the gas. The higher heating value of

LNG is estimated to be 24 MJ/L at 162°C. LNG is odorless, colorless, non-toxic, and non-corrosive (DR. Buyon Guo, 2005).

2.2.3 Compressed Natural Gas

Compressed natural gas (CNG) has a lower energy density and therefore needs a larger additional tank. CNG has a higher octane rating of up to 130 (petrol is typically 92-98) and a slight emissions. CNG is stored at 200 bar, burns cleanly and is readily available in many parts of the world, with a more equal distribution than oil.

Table 2. 2 Average emissions of petrol and CNG (% compared with petrol at a 100)

Emission	Petrol	CNG
CO	100	30
HC	100	10 (NHMC)*
NO_x	100	25-50
CO_2	100	80

*Non-methane hydrocarbons, a way of measuring CNG emissions which would be disadvantaged as it is largerly methane

Semin et al., 2008 was revealed CNG has long been used for machinery, including for engine developed in transportation. The CNG as fuel characteristics is shown in 2.

Table 2. 3 CNG fuel characteristics (Semin, et al., 2008)

CNG Characteristics	Value
Vapour density	0.68
Auto ignition	700 °C
Octane rating	130

Boiling point (at atmosphere pressure)	-162 °C
Air-fuel ratio	17.24
Chemical reaction with rubber	No
Storage pressure	20.6 MPa
Fuel air mixture quality	Good
Pollution CO-HC-NO_x	Very low
Flame speed (m/s)	0.63
Combustion ability with air	4 – 14%

CNG is can be used as transportation fuel. Economically, the use of CNG is less cost than using diesel fuel. The most of CNG chemical element is methane and the others are ethane, propane, butane, pentane, and carbon monoxide. There are many variant of CNG composition and characteristics, which can affect the stoichiometric conditions. CNG has disadvantage, that its phase in room temperature making CNG difficult to storage and mobility (Dzahab, 2019).

2.2 Dual-Fuel Diesel Engine

A dual-fuel engine is an engine designed to burn predominantly natural gas but with a small percentage of diesel as a pilot fuel to start ignition. The engines operate on a cross between the diesel and Otto cycles. A natural gas-air mixture is admitted to the cylinder during the intake stroke, then compressed during the compression stroke. At the top of the compression stoke the pilot diesel fuel is admitted and ignites spontaneously, igniting the gas-air mixture to create the power expansion. Care has to be taken to avoid spontaneous ignition of the natural gas-air mixture, but with careful design the engine can operate to close to the compression conditions of a diesel engine, with a high-power output and high efficiency, yet with the emissions close to those of a gas-fired spark-ignition engine. However, efficiency tends to fall and emissions of unburned hydrocarbons and carbon monoxide rise at part load (Breeze, 2019).

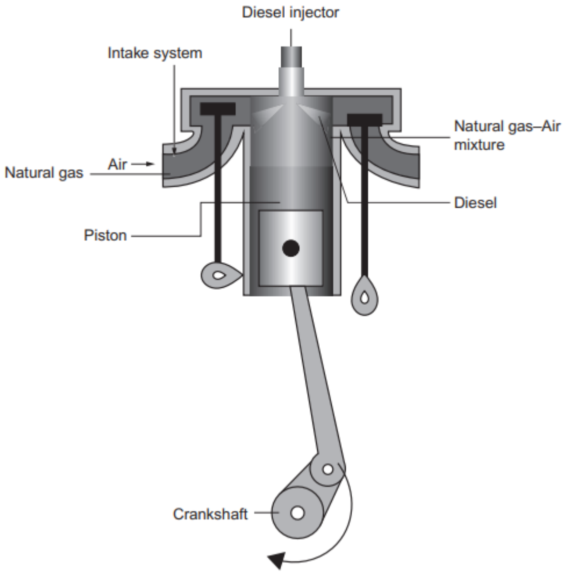

Figure 2. 1Dual Fuel Engine (Breeze, 2019)

Typical dual-fuel engines operate with between 1% and 15% diesel fuel. A dual fuel engine must be equipped with diesel injectors, exactly as if it were a diesel engine, a dual-fuel engine can also burn 100% diesel if necessary, though with the penalty of much higher emissions (Breeze, 2019).

Dual fuel engine is a diesel engine that can run on both gaseous and liquid fuels. That gas is injected to the intake port, then the premixed air and natural gas is injected to in-cylinder. Therefore, it needs a gas injector to inject gas to the intake port. There is required less modification job to convert a conventional diesel engine into a dual fuel engine. Converted of conventional diesel engine into a dual fuel engine can be implemented with install gas injector which the gas will be injected to intake port. The dual-fuel combustion system features essentially a homogeneous gas-air mixture compressed rapidly below its auto-ignition conditions, and ignited by the injection of pilot liquid fuel near the top dead centre position (Ghazi A. Karim, 1980).

Dual fuel diesel engine should be operated on fuel oil mode at any load automatically. During on fuel oil mode operation, dual fuel diesel engine will be operated according to conventional diesel engine process. Basically dual fuel system is a conventional diesel engine, so if there is interference with gas supply, the system will be switched automatically from dual fuel mode to fuel oil mode (Wijaya, 2014).

Before operate the dual fuel system, diesel engine will operated first with fuel oil for a certain period of time. The load is given until around a half of the maximum of load. This is because at low load condition, dual fuel engine will occur knock. That load is maintained until achieve the normal operation condition (idle). Furthermore to operate on dual fuel mode, maintain the engine condition at the same condition when only need fuel oil. Then, reduce the quantity of fuel oil and at a time open CNG control valve (Ehsan, 2009).

The dual-engine works using a compression ignited (CI) engine but operates on a combustion process with characteristics from both spark ignited (SI) and compression ignited (CI). To illustrate the process of the SI and CI engine operating cycle, a pressure-volume (*p*-V) diagram associated with an ideal air-standard cycle is considered. A heat addition term is used to replace the combustion process since air alone cannot combust. The Otto cycle is the ideal air cycle for SI engine which is differs from the actual cycle. The cycle is divided into intake, compression, combustion, expansion, and exhaust process. Figure 2.2 shows the pressure-volume diagram for Otto cycle. This cycle consists of an isentropic compression process 1-2, which compresses the mixture of air and gad and brings the piston to top dead centre (TDC), a constant volume heat addition process 2-3 which raises the temperature of the mixture at constant volume to point 3, an isentropic expansion process 3-4 to the bottom dead centre (BDC), and constant volume heat rejection 4-1 (Mansor, 2014).

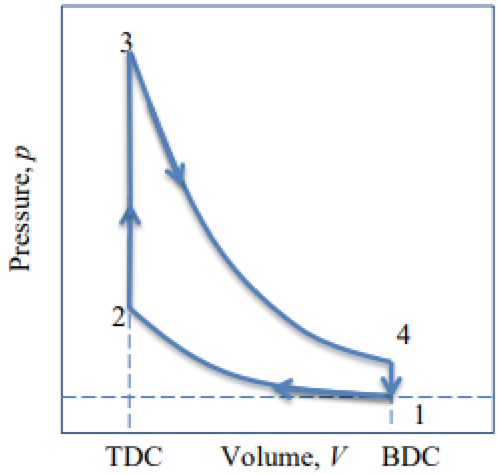

Figure 2. 2 Otto cycle p-V diagram (Mansor, 2014)

The working processes for Diesel cycle is presented in Figure 2.3 The processes for this cycle are isentropic compression 1-2, constant pressure heat addition 2-3, isentropic expansion 3- 4, and constant volume heat rejection 4-1. All processes are the same as the Otto cycle except the heat addition process 2-3 (Mansor, 2014).

The dual cycle contains features of both Otto and Diesel cycles. This can be considered a model for dual fuel combustion since it contains features of the ideal cycle for the SI engine (Otto cycle) and the ideal cycle for the CI engine (Diesel cycle). The process of Dual cycle is illustrated in Figure 2.4. The isentropic compression process 1-2, the isentropic expansion process 4-5, and the constant volume heat rejection process 5-1 are the same as in the Otto and Diesel cycles with different numbering of states. The heat addition process is different for the dual cycle. The heat addition process is composed of a dual mechanism, which is where the dual cycle name comes from. Heat addition consists of constant volume heat addition 2-3 followed by constant pressure heat addition 3-4 (Mansor 2014).

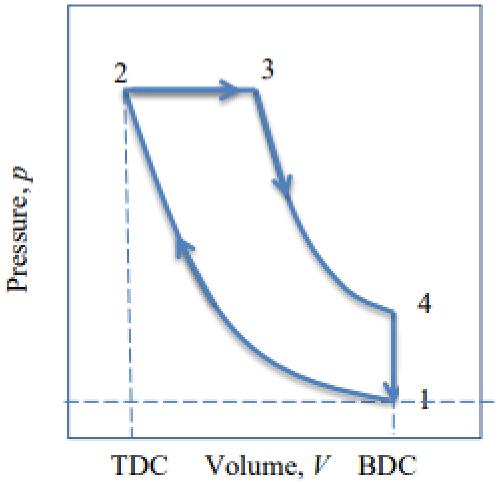

Figure 2. 3 Diesel cycle p-V diagram (Mansor, 2014)

Comparing the Otto, Diesel and Dual cycles at the same inlet conditions and compression ratio (CR) shows that Otto has the highest work output and thermal efficiency (ηt), while Diesel has the lowest (Mansor, 2014).

$$(\eta_t)Otto > (\eta_t)Dual > (\eta_t)Diesel$$

However, since CI engines have much higher compression ratio values than SI engine, CI engines typically have higher efficiencies than SI engines (Mansor, 2014).

$$(\eta_t)Diesel > (\eta_t)Dual > (\eta_t)Otto$$

From these concepts, it is suggested that the most efficient engine would have combustion close to constant volume as in the Otto cycle but operates at high compression ratios as with the Diesel cycle (Mansor, 2014).

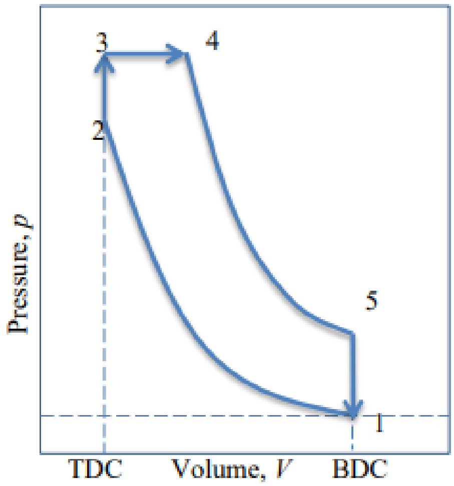

Figure 2. 4 Dual cycle p-V diagram (Mansor, 2014)

2.3 Combustion Performance on Dual-Fuel Engine

The chemical reaction from dual fuel combustion with fuel oil (1) and compressed natural gas (CNG) (2) which the methane as a major compound is as follows:

$$C_{12}H_9 + 18.5O_2 \rightarrow 12CO_2 + 12H_2O \quad (1)$$
$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \quad (2)$$

The combustion process of a dual-fuel engine is described with five stages (Nwafor, 2001). The data is taken on a single cylinder pre-chamber diesel engine with a pumpline-injector mechanical diesel injection system. The combustion phases of a dual-fuel engine are illustrated with a cylinder pressure trace in Figure 2.5. After diesel fuel is injected at point A, a longer ignition delay period AB is observed in dual fuel combustion than in conventional diesel engines due to the reduction in oxygen concentration resulting from the introduction of natural gas to the intake charge. The premixed combustion phase BC in dual-fuel is slower compared to conventional diesel premixed combustion. This is because the dual-fuel engine is injecting a smaller amount of liquid fuel, therefore a smaller amount of burning mixture is added to the fuel. Period CD shows a decrease in pressure until it rises at period DE. Period CD is described as the primary fuel (premixed air and natural gas fuel) delay period. The DE phase is the actual combustion of the natural gas fuel starting with the flame propagation initiated by the spontaneous liquid fuel ignition. Nwafor described the period EF as a diffusion combustion stage starting at the end of gaseous fuel combustion. It is unclear why this phase of combustion is characterized as diffusion combustion. It may be more appropriate to characterize it as the late combustion phase.

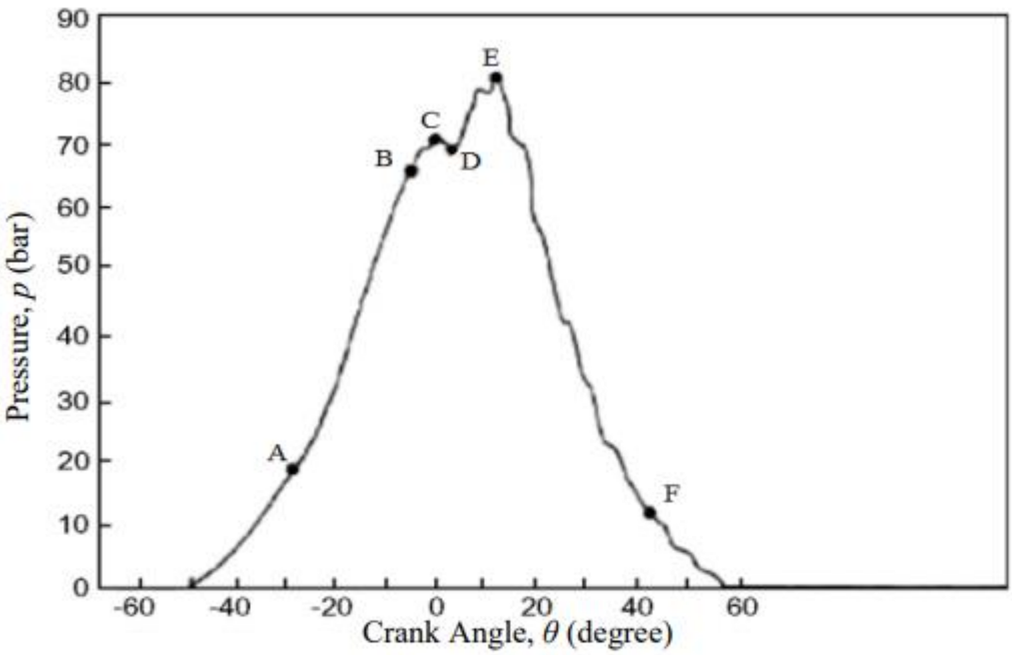

Figure 2. 5 Combustion process of a dual-fuel engine

On dual-fuel operation, the flame temperature is lowered due to the quantity of gaseous fuel and the air-fuel mixture, therefore reducing the formation of NO_x. As the mixture goes leaner, increasing excess air concentration reduces the gas temperature. At low temperature, atmospheric nitrogen is a stable diatomic molecule (N_2) and less oxides of nitrogen are produced. In contrast, at high temperatures some of N_2 breaks down to reactive monotonic nitrogen radicals. At high temperatures with excess oxygen available oxygen can combine with the nitrogen to form nitric oxide (Nwafor,2001):

$$O + N_2 \rightarrow NO + N$$
$$N + O_2 \rightarrow NO + O$$

Oxygen radicals (O) are produced from the dissociation of oxygen molecules (O_2) or from a collision between the H radical and O_2. These oxygen radicals combine with N_2 to start the simple chain shown, called Zeldovich mechanism. Nitric oxide can further oxidize to form NO_2. In order to minimize NO_x formation, reducing flame temperature is crucial, which slows the reaction rates of NO_x (Nwafor, 2001).

Some researches show that dual-fuel engine has a lower thermal efficiency than diesel engine. This suggests that the thermal efficiency comparison is dependent on engine design and operating conditions. The cause of lower thermal efficiency is the reduced combustion duration, which also reduces cylinder charge temperature (Papagiannakis et al., 2003). At part load some of air and gaseous fuel mixture is burned but some escape from the combustion process due to valve overlap. Increased valve overlap causes the intake charge to flow directly into the exhaust leading to poor combustion especially during idling. Abdelaal et al., 2012 define better fuel utilization in dual fuel engine at high loads. This is due to larger amount of natural gas introduced leading to better combustion and higher brake thermal efficiency at high loads. In contrast, at high loads diesel engine combustion sees increment in heat loss to the cylinder wall, therefore negatively impacting the thermal efficiency. As a consequence, more power may be produced in dual fuel combustion.

2.3.1 Cylinder Temperature

Semin et al., 2009 revealed the results of the cylinder temperature research on the modified dual fuel diesel engine experiments with varying speeds, that CNG cylinder temperature is higher than the diesel engine cylinder temperature at low engine speed. Whereas at high engine speed, the diesel engine cylinder temperature is higher than CNG cylinder temperature as shown in Figure 2.6.

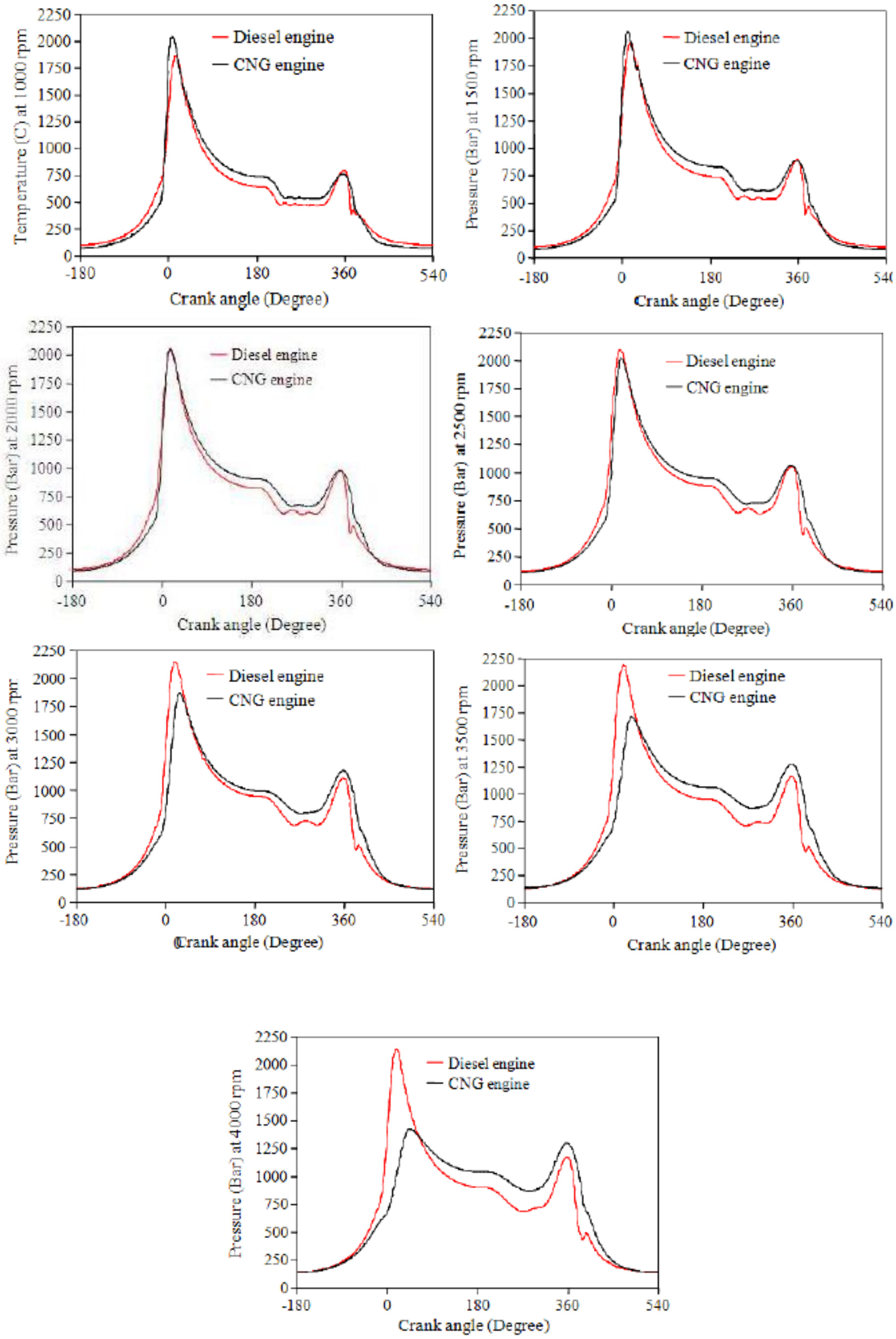

Figure 2. 6 Combustion temperature in the engine cylinder at 1000 - 4000 rpm engine speed (Semin et al., 2009)

Decreasing engine speed of diesel engine will be decrease the maximum combustion temperature in the engine cylinder. Decreasing engine speed of CNG engine will be increase the maximum combustion temperature in the engine cylinder as shown on Figure 2.7 (Semin et al., 2009).

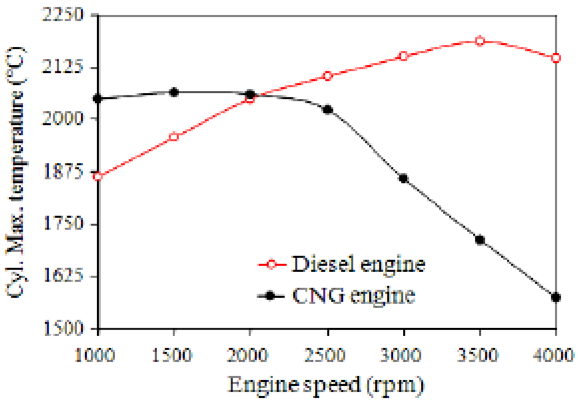

Figure 2. 7Maximum combustion temperature in the engine cylinder (Semin et al., 2009)

2.3.2 Cylinder Pressure

Cylinder pressure versus crank angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion (Heywood, 1988).

From experiments conducted by Rohiim, 2019 on dual fuel diesel engine with variations in injection duration, variations in load, and at constant engine speed, the cylinder pressure decreases with increasing injection duration as shown in Figure 2.8. This is caused by the length of gas injection duration so that the injected gas mass increases. The increasing injected gas mass can reduce the heating effect of diesel fuel and gas so that it affects the formation of an imperfect air-fuel mixture. The imperfect air-fuel mixture causes imperfect combustion (Rohiim, 2019).

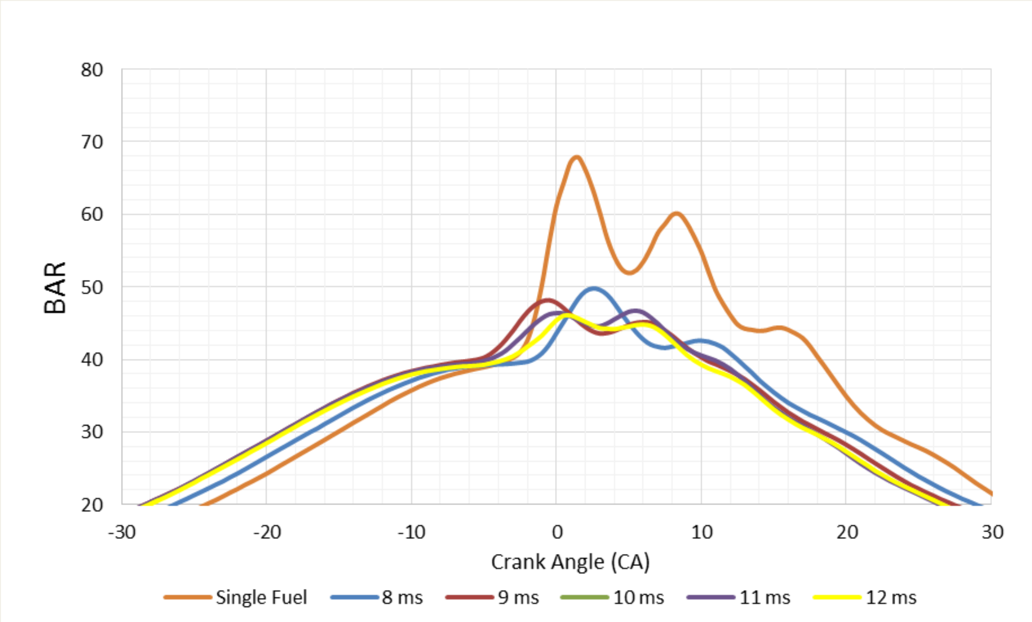

Figure 2. 8 The graph cylinder pressure over crank angle with injection duration variant (Rohiim, 2019)

2.3.3 Heat Release Rate

Another parameter to determining the engine's performance is the heat release rate (HRR). The HRR is calculated by computing the amount of energy release from the fuel obtain the experimentally observed pressure, while combustion reaction extent is evaluated through the released fraction of the total fuel chemical energy. Abdelaal et al. shows that longer ignition delays in the dual-fuel engine aggravate the HRR. Generally, the diesel engine exhibits better HRR trends than dual fuel engine. In the diesel engine, a large amount of diesel is utilized especially at high load resulting in high in-cylinder temperatures. The reduction of HRR in dual-fuel operation is mainly due to the very lean mixture of air and gaseous fuel. This affects the combustion in dual fuel, a portion of natural gas usually escapes the combustion process resulting in low combustion efficiency (Abdelaal et al., 2012).

Figure 2.9 presents a graph of heat release rate between combustion of diesel engine and dual fuel diesel engine. It can be seen heat release graphic on the combustion process of dual fuel diesel engine starts faster. This is caused by the pilot premixed combustion, the proceed to premixed combustion. The diesel engine only occurs once combustion while in dual fuel diesel engine occurs pilot premixed combustion after that combustion again occurs which is caused by gas as shown on the Figure 2.9 there is a grade incline after diesel fuel is injected (Budiyanto, 2012).

Whereas the diesel engine heat release starts late, this is caused by the cooling effect of the injection of liquid fuel. In dual fuel combustion, pre-oxidation of gas occurs before the fuel is injected by the pilot. Figure 2 shows the initial heat release caused by the injection pilot and also illustrates that dual fuel combustion continues well untul the expansion stage (Budiyanto, 2012).

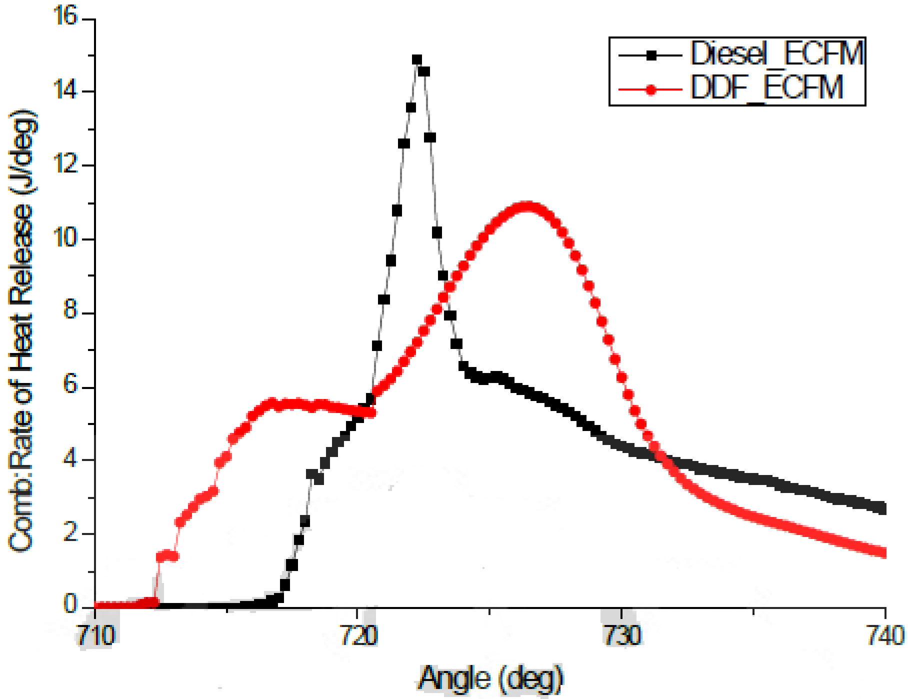

Figure 2. 9 Comparison graph of heat release rate of diesel engine and dual fuel diesel engine (Budiyanto, 2012)

2.3.4 Volumetric Efficiency

The intake system-the air filer, intake manifold, intake port, intake valve-restricts the amount of air which an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the volumetric efficiency. Volumetric efficiency is only used with four-stroke cycle engines which have a distinct induction process. It is define as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston (Heywood,1988).

Heywood, 1988 revealed that volumetric efficiency is affected by the following fuel, engine design, and engine operating variables:

1. Fuel type, fuel-air ratio, fraction of fuel vaporized in the intake system, and fuel heat of vaporization
2. Mixture temperature as influenced by heat transfer
3. Ratio of exhaust to inlet manifold pressures
4. Compression ratio
5. Engine speed
6. Intake and exhaust manifold and port design
7. Intake and exhaust valve geometry, size, lift, and timings

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CHAPTER III
METHODOLOGY

This research was conducted to analyse the combustion performance on dual fuel diesel engine with variation of gas composition percentage. The research methodology that used on this research is covering all the activities that conduct to solve and do analyse process of research problem. This research methodology has several steps with some problem limitation which is explained before. The research steps are explained as follows:

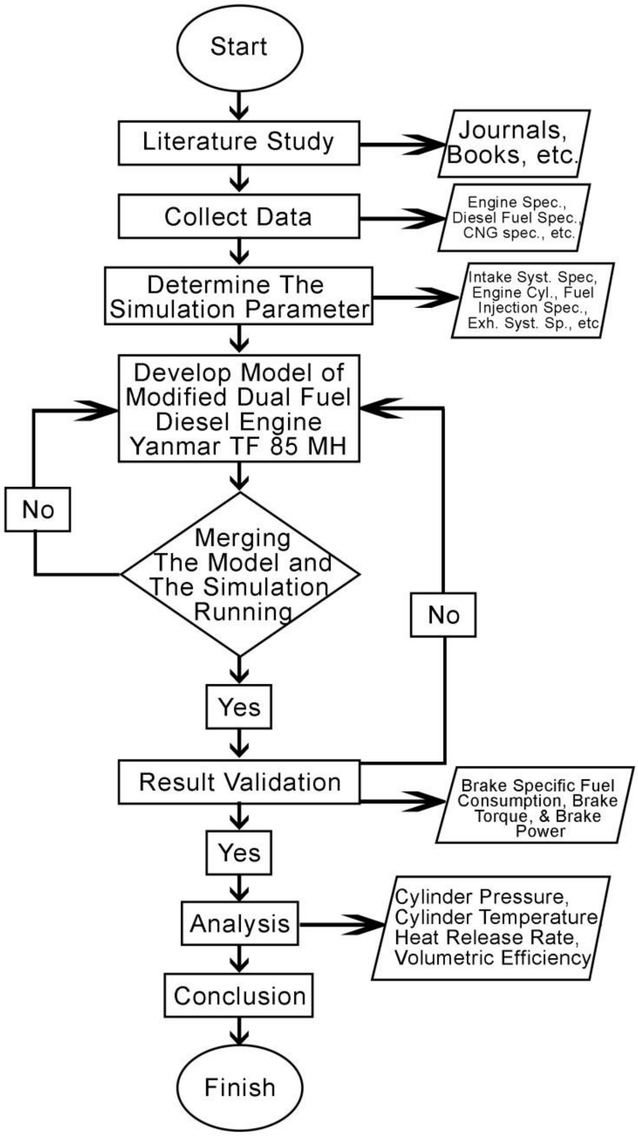

Figure 3. 1 Research flowchart

The focused of this experiment is the effect of quantity of gas injection variation to the combustion process. The following is the data from the engine used :

Engine (Four stroke cycle)	: Yanmar TF 85 MH
Number of cylinder	: 1
Bore × Stroke	: 85 mm × 87 mm
Displacement	: 493 cc
Compression Ratio	: 18 : 1
Max. Engine speed at full load	: 2200 RPM
Continuous power output	: 7.5 kW
Specific fuel consumption	: 229.31 g/kWh

The research is carried out in several steps to get desired research result. In this methodology is already categorized according the desired subject so this research can be done efficiently. The detail explanation of method on this research is:

1. Literature Study
2. Collect Data
3. Determine The Simulation Parameter
4. Construct The Model
5. Simulation
6. Data Analysis

3.1 Literature Study

Literature study is an early learning step to learn the basic theories from the problems that will be analysed in this research. This literature studies aim to obtain basic knowledge and data from the previous researches that can used as supporting references for next research. At this step, the study of references in the journals, thesis, internet, and other supporting books was conducted. The information needed are the basic theory of dual fuel diesel engine system, characteristic of fuel oil and natural gas, and the combustion process of diesel engine.

3.2 Collect Data

After finish to collect the materials for arrange the basic theories, the next step is collecting engine and fuel that is used for making simulation model.

- Engine Specification

Table 3. 1 Engine specification

No.	Description	Detail
1.	Merk	Yanmar

2.	Type	TF85MH-di
3.	Operation	Diesel 4 stroke
4.	Total of cylinder	1 cylinder
5.	Bore	85 mm
6.	Stroke	87 mm
7.	Injection timing	18° BTDC
8.	Cylinder volume	493 cc
9.	Continuous power	7.5 HP / 2,200 RPM
10.	Maximum power	8.5 HP / 2,200 RPM
11.	Maximum torque	3.44 kgm / 1,600 RPM
12.	Compression ratio	18
13.	Direction of shaft rotation	Counter clockwise
14.	Specific fuel consumption	171 g/HPh

- High Speed Diesel Specification

Table 3. 2 High speed diesel specification

No.	Description	Detail
1.	Density	830 kg/m^3

2.	Low heating value	43250 kJ/kg
3.	Pour point	18 °C
4.	Flash point	55 °C
5.	Boiling Point	370 °C
6.	Carbon atom per molecule	0.3 %m/m

- Compressed Natural Gas Specification

Table 3. 3 Compressed Natural Gas specification

No.	Description	Detail
1.	Low heating value	48000 kJ/kg

3.3 Determine Simulation Parameter

In this step is conducted measurement of each diesel engine component from intake system until exhaust system that was created as the simulation parameter with GT-Power.

3.4 Develop The Model

The model is created on variation of gas composition that is injected and the variation of load. Previously, the Yanmar diesel engine model was created on single fuel condition. After that, the single fuel diesel engine was modified into dual fuel diesel engine by adding gas injector through one of air intake pipe. The execution steps can be seen in the Appendix A.

Picture 3.2 is the model of dual fuel diesel engine. This model is created appropriate to the original engine. The components were created appropriate to the original component. The way to create the model is measure the dimension of all combustion process main components of dual fuel diesel engine from intake port until exhaust port, then the enter the value to the template as shown in Appendix A.

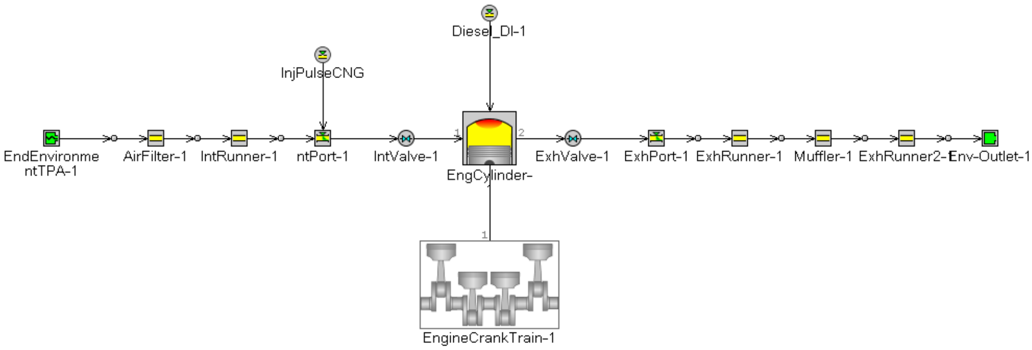

Figure 3. 2 Dual Fuel Diesel Engine Model

3.5 Simulation and Result Validation

The model that already created can be run with the variation of fuel quantity that has previously calculated. The fuel quantities are shown in Table 3.4. The result of simulation can be graphic or data. The result from simulation is validated with experiment results. The validated data is Brake Specific Fuel Consumption (BSFC), brake torque, and brake power.

Table 3. 4 Fuel consumption variation

Engine speed (rpm)	Load (watt)	70% Diesel 30% CNG		65% Diesel 35% CNG		60% Diesel 40% CNG		55% Diesel 45% CNG	
		Diesel	Gas	Diesel	Gas	Diesel	Gas	Diesel	Gas
1800	1000	6.166489	2.642780823	5.726025	3.083244293	5.285562	3.523707763	4.845098	3.964171234
	2000	7.711633	3.304985636	7.160802	3.855816576	6.609971	4.406647515	6.05914	4.957478454
	3000	9.599878	4.114233542	8.914173	4.799939132	8.228467	5.485644722	7.542761	6.171350312
	4000	11.39059	4.881681574	10.57698	5.695295169	9.763363	6.508908765	8.94975	7.32252236
1900	1000	5.117388	2.193166131	4.75186	2.55869382	4.386332	2.924221509	4.020805	3.289749197
	2000	7.327495	3.140355178	6.804103	3.663747708	6.28071	4.187140237	5.757318	4.710532767
	3000	9.119466	3.908342478	8.468075	4.559732891	7.816685	5.211123304	7.165295	5.862513717
	4000	11.5487	4.949441585	10.72379	5.774348516	9.898883	6.599255447	9.073976	7.424162378
2000	1000	6.695146	2.869348444	6.216922	3.347573185	5.738697	3.825797926	5.260472	4.304022667
	2000	8.439986	3.617137042	7.83713	4.219993216	7.234274	4.822849389	6.631418	5.425705563
	3000	10.22639	4.382739109	9.495935	5.113195627	8.765478	5.843652145	8.035022	6.574108663
	4000	13.1765	5.64707027	12.23532	6.588248649	11.29414	7.529427027	10.35296	8.470605405
2100	1000	7.374612	3.160547945	6.847854	3.687305936	6.321096	4.214063927	5.794338	4.740821918
	2000	9.582896	4.106955435	8.898403	4.791448008	8.213911	5.47594058	7.529418	6.160433153
	3000	10.79349	4.625783092	10.02253	5.39674694	9.251566	6.167710789	8.480602	6.938674638
	4000	14.27447	6.1176303	13.25487	7.13723535	12.23526	8.1568404	11.21566	9.17644545
2200	1000	5.9013	2.5291286	5.479779	2.950650034	5.058257	3.372171467	4.636736	3.793692901
	2000	9.539915	4.088535092	8.858493	4.769957607	8.17707	5.451380122	7.495648	6.132802638
	3000	11.98859	5.137967914	11.13226	5.9942959	10.27594	6.850623886	9.419608	7.706951872
	4000	19.86156	8.512095894	18.44287	9.930778543	17.02419	11.34946119	15.60551	12.76814384

3.6 Data Analysis

The result data from the simulation can be used to analyse and compare with the data from the previous experiment of modified dual fuel diesel engine. The result data can be analysed are cylinder pressure, cylinder temperature, heat release rate, and volumetric efficiency.

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CHAPTER IV
RESULTS AND DISCUSSION

In chapter 4 contains the results and discussion of dual fuel diesel engine simulations. The fuels used are high speed diesel (Pertamina-Dex) and Compressed Natural Gas (CNG). The observations made to observe combustion performance, including cylinder pressure, combustion temperature, heat release rate, and volumetric efficiency. The main discussions in chapter 4 are the calculation for supporting data in the simulation process, validation of simulation model by comparing the simulation results data and experimental results data with maximum difference is 5%, analysis and discuss of combustion performance.

4.1 Validation of Simulation and Experimental Results

After creating the simulation model will be conducted data validation process between simulation and experimental results In the following is the experiment calculation with load at 1000 – 4000 watt and at engine speed 1800 rpm. The following is the calculation:

4.1.1 Break Specific Fuel Consumption (BSFC) Validation.

As example the brake specific fuel consumption (BSFC) experimental results will be compared with the simulation results. The following is the calculation of BSFC with load at 1000 watt:

- Calculation of Pertamina-Dex BSFC
  - Fuel Consumption Rate (FCR) = $7.41 \times 10^{-8}\ m^3/s$
  - Density of diesel ρ = $830\ kg/m^3$
  - $FCR = 7.41\ \times 10^{-8}\ m^3/s\ \times 830\ kg/m^3$

    $FCR = 6.1503 \times 10^{-5}\ kg/s$
  - $Brake\ Specific\ Fuel\ Consumption\ (BSFC) = \frac{FCR}{Power}$

$$BSFC = \frac{6.1503 \times 10^{-5}\ kg/s}{0.656\ kW}$$

$$BSFC = 9.37546\ \times 10^{-5} kg/kW\ s$$

$$BSFC = 0.33751656\ kg/kW\ h$$

$$BSFC = 337.51656\ g/kW\ h$$

- Calculation of CNG BSFC
  - Fuel Consumption Rate (FCR) = $3.33284 \times 10^{-5}\ m^3/s$
  - Density of diesel ρ = $2.12\ kg/m^3$

    $FCR = 3.33284\ \times 10^{-5}\ m^3/s\ \times 2.12\ kg/m^3$

    $FCR = 7.06562 \times 10^{-5}\ kg/s$
  - $Brake\ Specific\ Fuel\ Consumption\ (BSFC) = \frac{FCR}{Power}$

$$BSFC = \frac{7.06562 \times 10^{-5}\ kg/s}{0.656\ kW}$$

$$BSFC = 1.077076 \times 10^{-4}\ kg/kW\ s$$
$$BSFC = 0.38774736\ kg/kW\ h$$
$$BSFC = 387.74736\ g/kW\ h$$

From the experimental results was obtained BSFC with load at 1000 watt and at engine speed 1800 rpm $337.51656\ g/kW\ h + 387.74736\ g/kW\ h = 725.26392\ g/kW\ h$.

- BSFC simulation results with load at 1000 watt and at engine speed 1800 rpm is $714.36053\ g/kW\ h$.

There is error margin of simulation and experimental results and in the following is error margin calculation:

$$Validation = \frac{Experimental\ result - Simulation\ result}{Experimental\ result} \times 100\%$$
$$Validation = \frac{725.26392 - 714.36053}{725.26392} \times 100\%$$
$$Validation = 1.5\ \%$$

From the experimental result and simulation results were obtained that will be compared in the graphic, that is the result:

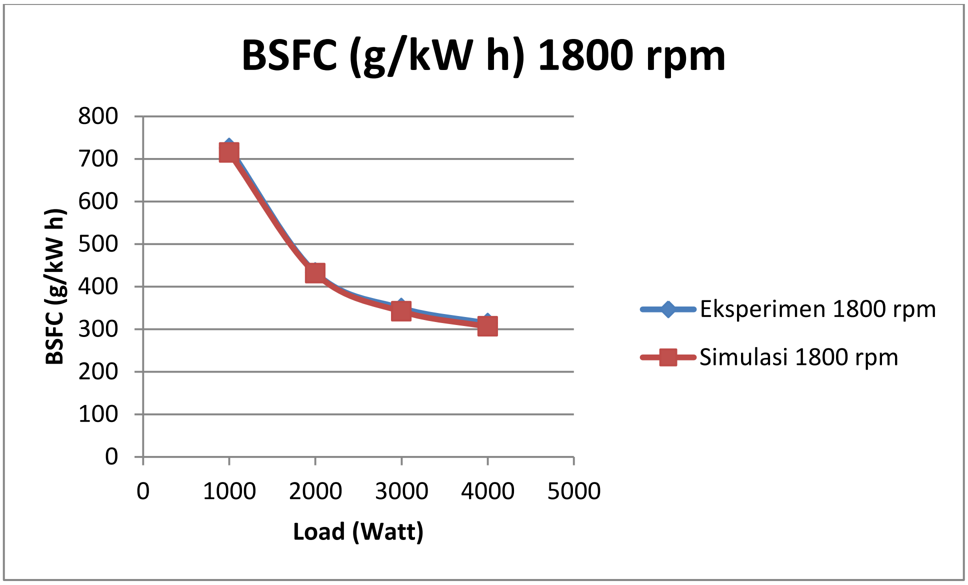

Figure 4. 1 Comparison graphic of Brake Specific Fuel Consumption (BSFC) with load variation and at engine speed 1800 rpm

As shown in Figure 4.1, the comparison of BSFC experimental and BSFC simulation shown the trend that the points are almost at the same point. The percentage of the comparison of BSFC from experiment and BSFC from simulation is 0.5% - 2.4%.

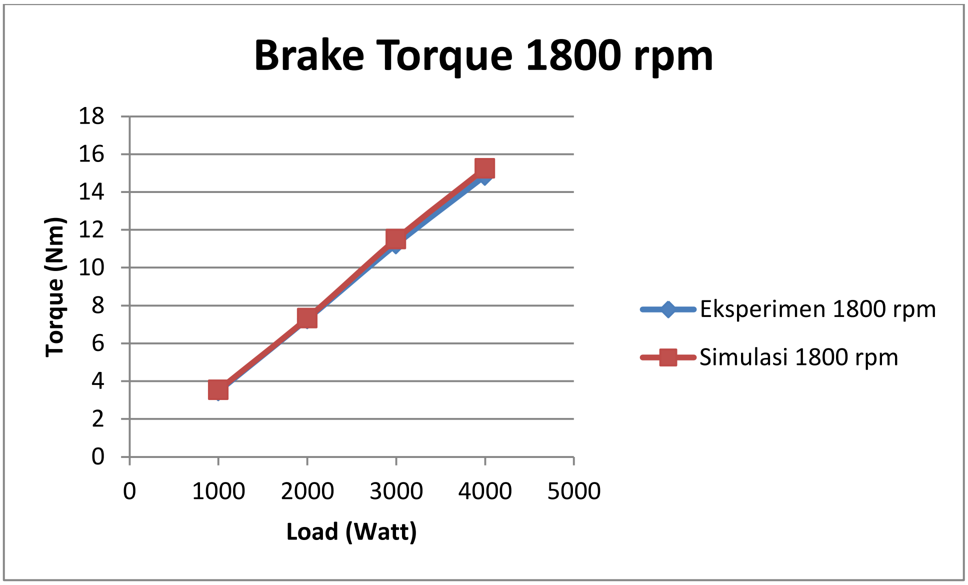

Figure 4. 2 Comparison graphic of Break Torque with load variation and at engine speed 1800 rpm

As shown in Figure 4.2, the comparison of brake torque experimental and brake torque simulation shown the trend that the points are almost at the same point. The percentage of the result comparison of brake torque from experiment and brake from simulation is 0.2% - 2.5%.

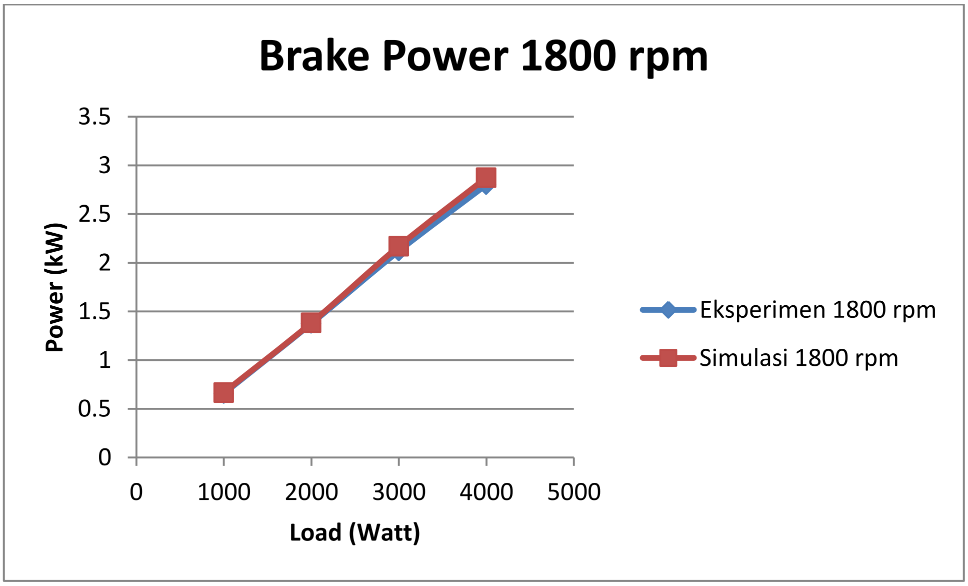

Figure 4. 3 Comparison graphic of Brake Power with load variation and at engine speed 1800 rpm

As shown in Figure 4.3, the comparison of brake power experimental and brake power simulation shown the trend that the points are almost at the same point. The percentage of the results comparison of brake power from experiment and brake power from simulation is 0.5% - 2.6%.

4.2 Combustion Performance Analysis

The discussion in this chapter is about combustion performance, including cylinder pressure, combustion temperature, heat release rate, and volumetric efficiency.

4.2.1 The Effect of Gas Consumption Variation to Cylinder Pressure

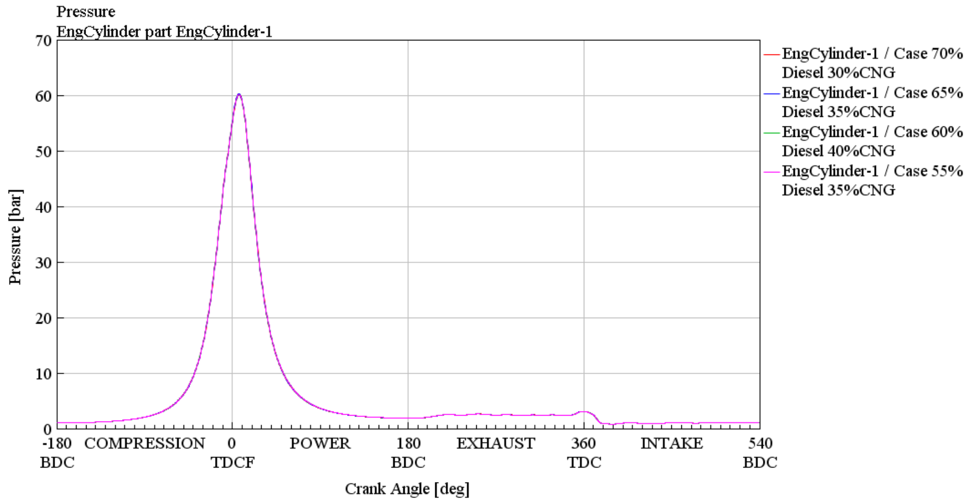

Figure 4. 4 Cylinder pressure graph with variation of gas consumption at load 1000 watt and at constant engine speed 1800 rpm

Figure 4.4 is a graphic of cylinder pressure with crank angle for modified dual fuel diesel engine with gas consumption variation at engine load 1000 watt and at engine speed 1800 rpm. At engine load 1000 watt and engine speed 1800 rpm, the maximum cylinder pressure for each gas consumption variations are 60.200584 bar with ratio 70% of diesel and 30% of CNG, 60.32564 bar with ratio 65% of diesel and 35% of CNG, 60.049095 bar with ratio 60% of diesel and 40% of CNG, and 60.088943 bar with ratio 55% of diesel and 45% of CNG.

Figure 4.4 shows cylinder pressure is not changing significantly. This result can indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG. As shown in Figure 4.8 that the highest maximum cylinder pressure is cylinder pressure with 65% of diesel and 35% of CNG.

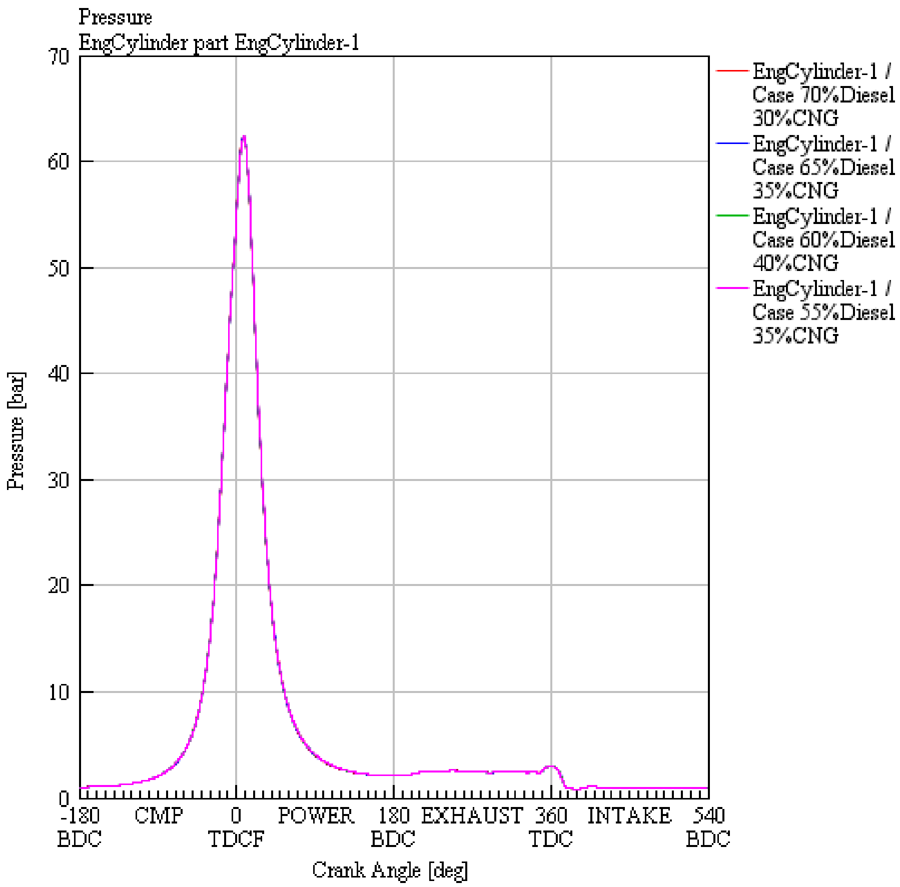

Figure 4. 5 Cylinder pressure graph with variation of gas consumption at load 2000 watt and at engine speed 1800 rpm

Figure 4.5 is a graphic of cylinder pressure for modified diesel dual fuel engine with gas consumption variation at engine load 2000 watt and at engine speed 1800 rpm. The maximum cylinder pressure for both of gas consumption variations are 62.45269 bar with ratio 70% of diesel and 30% of CNG, 62.432953 bar with ratio 65% of diesel and 35% of CNG, 62.306683 bar with ratio 60% of diesel and 35% of CNG, and 62.35087 bar with ratio 55% of diesel and 45% of CNG.

From Figure 4.5 can be seen that cylinder pressure is not changing significantly. This result can indicate that the engine is good because there is not changing result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the fuel ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG. As shown in Figure 4.8 that the highest maximum cylinder pressure is cylinder pressure with 65% of diesel and 35% of CNG.

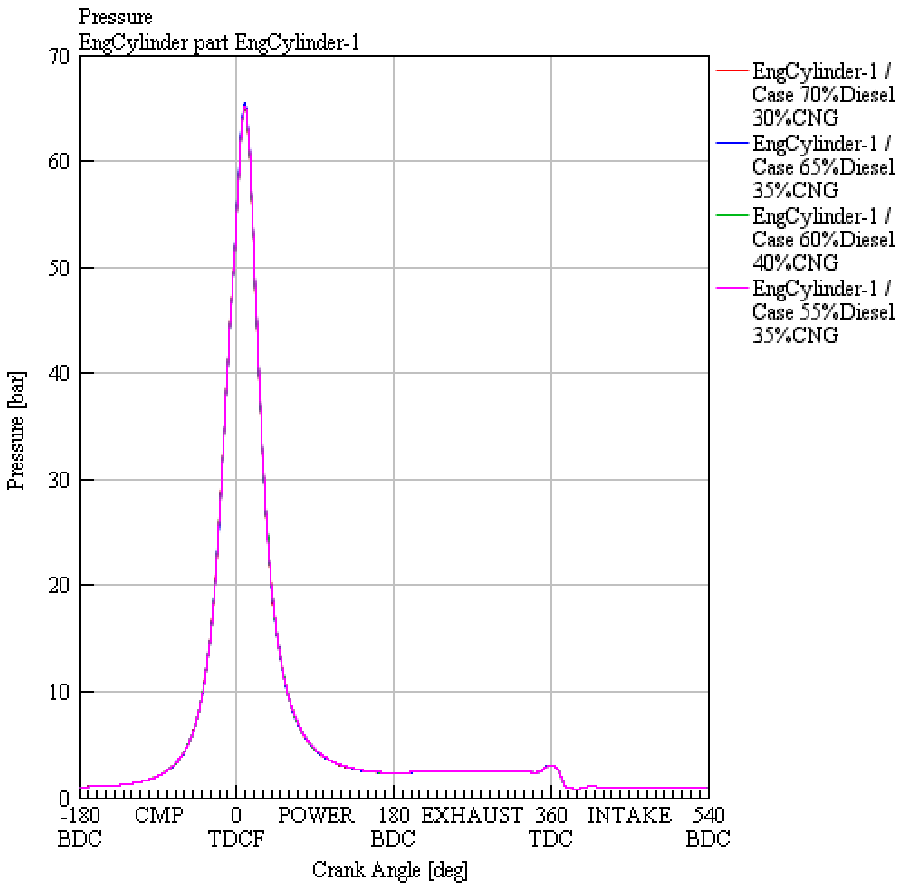

Figure 4. 6Cylinder pressure graph with variation of gas consumption at load 3000 watt and at constant engine speed 1800 rpm

Figure 4.6 shows the graph of cylinder pressure for modified dual fuel diesel engine with gas consumption variation at engine load 3000 watt and at engine speed 1800 rpm. The results for maximum cylinder pressure with gas consumption variation are 65.09703 bar with ratio 70% of diesel and 30% of CNG, 65.429794 bar with ratio 65% of diesel and 35% of CNG, 65.103676 bar with ratio 60% of diesel and 30% of CNG, and 65.1774 bar with ratio 55% of diesel and 45% of CNG.

Figure 4.6 shows cylinder pressure is not changing significantly. This result can indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with fuel ratio 55% of diesel and 45% of CNG.

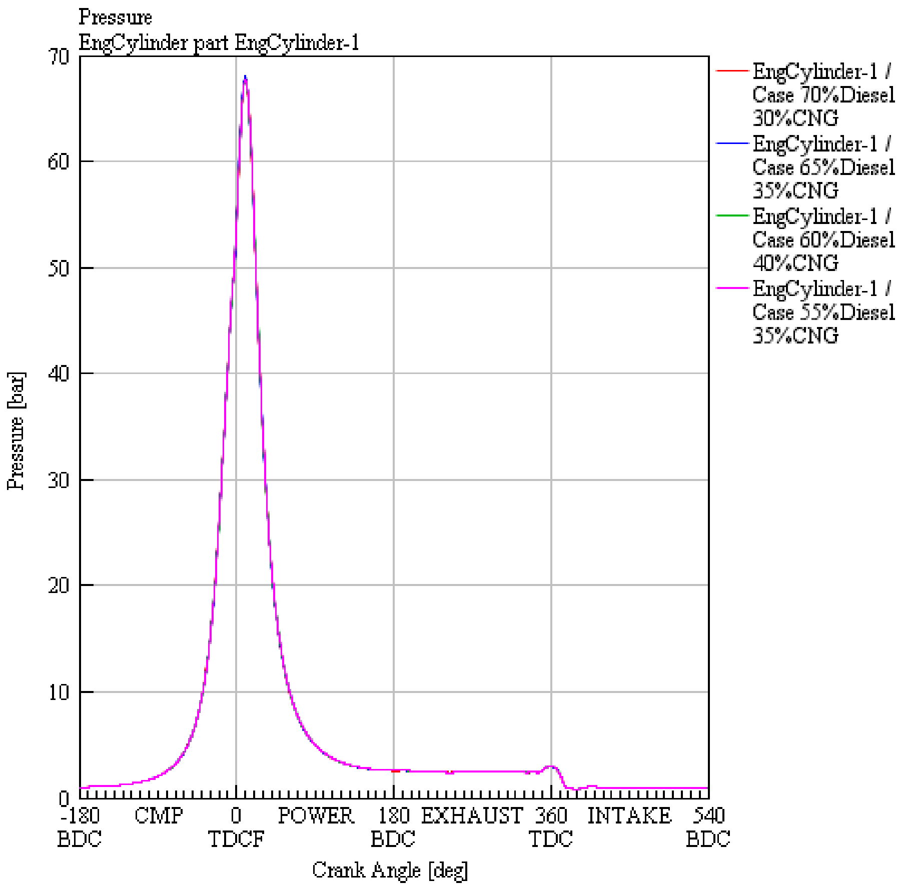

Figure 4. 7 Cylinder pressure graph with variation of gas consumption at load 4000 watt and at engine speed 1800 rpm

Figure 4.7 shows that the graph of cylinder pressure with gas consumption variation at engine load 4000 watt and at engine speed 1800 rpm. The maximum cylinder pressure at engine load 4000 watt and engine speed 1800 rpm with gas consumption variations are 67.72769 bar with ratio 70% of diesel and 30% of CNG, 67.953575 bar with ratio 65% of diesel and 35% of CNG, 67.66242 bar with ratio 60% of diesel and 40% of CNG, and 67.74599 bar with ratio 55% of diesel and 45% of CNG.

From Figure 4.7 shows cylinder pressure for converted diesel engine into dual fuel diesel engine with variation of gas consumption is not changing significantly. This result can indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG.

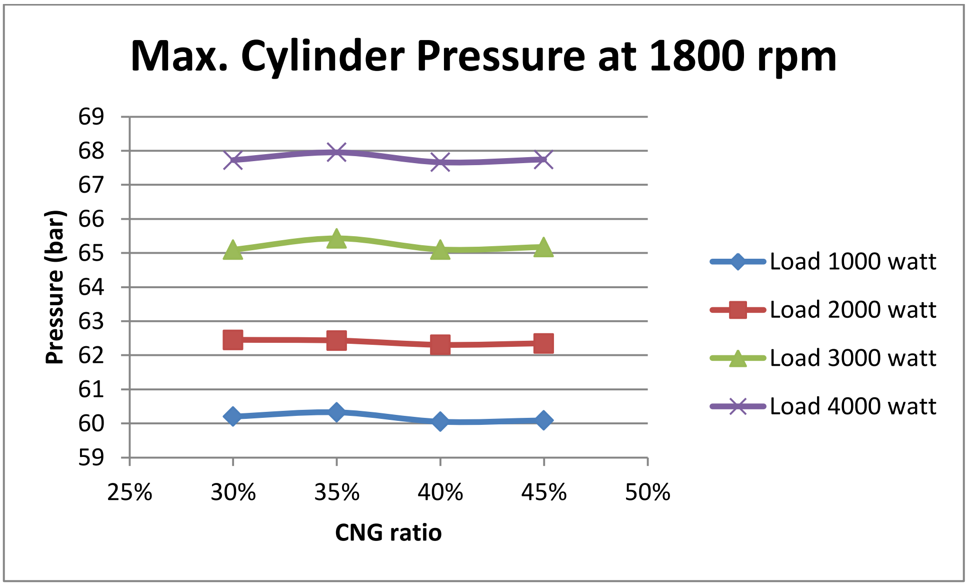

Figure 4. 8 Maximum cylinder pressure graph over CNG ratio with variation of load and at constant engine speed 1800 rpm

From Figure 4.8 show that the graph of cylinder pressure for dual fuel diesel engine with variation of gas consumption at engine load 1000 – 4000 watt with interval 1000 and at constant engine speed 1800 rpm. The results are obtained the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG. As shown in Figure 4.8 that the highest maximum cylinder pressure is cylinder pressure with 65% of diesel and 35% of CNG.

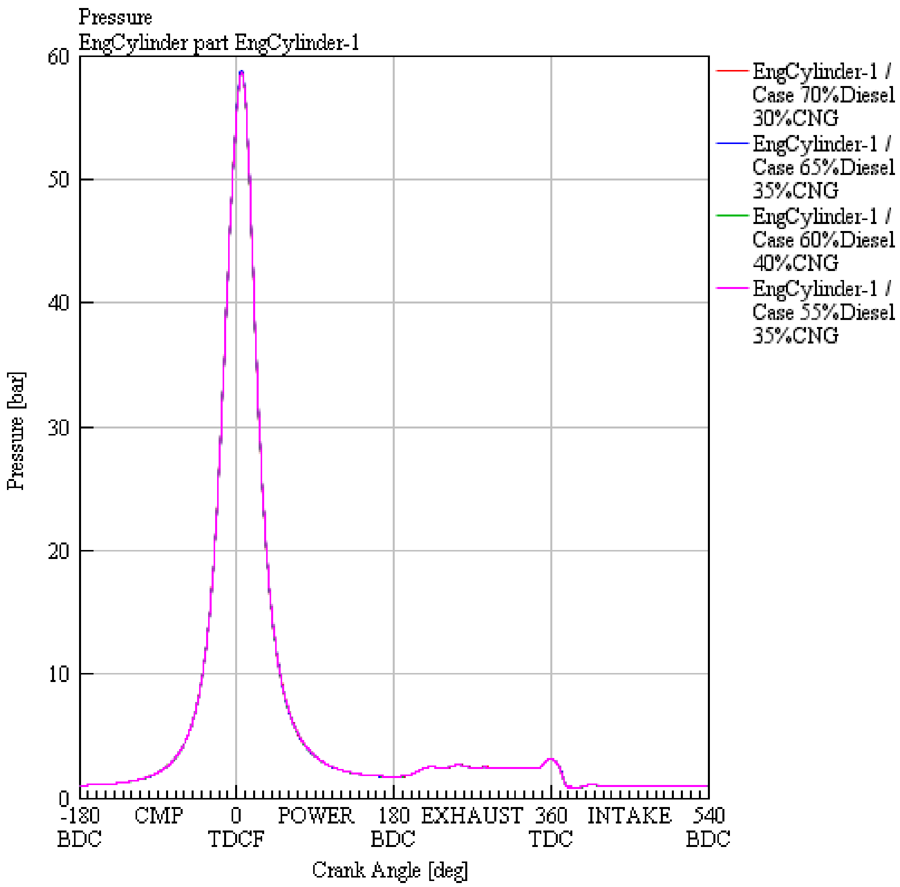

Figure 4. 9 Cylinder pressure graph with variation of gas consumption at load 1000 watt and at engine speed 1900 rpm

Figure 4.9 is a graphic of cylinder pressure with crank angle for modified dual fuel diesel engine with gas consumption variation at engine load 1000 watt and at engine speed 1900 rpm. At engine load 1000 watt and at engine speed 1900 rpm, the maximum cylinder pressure results are different for each gas consumption variations. In 70% of diesel and 30% of CNG, the maximum cylinder pressure is 58.686966 bar. In 65% of diesel and 35% of CNG, the maximum cylinder pressure is 58.791355 bar. In 60% of diesel and 40% of CNG, the maximum cylinder pressure is 58.55329 bar. In 55% of diesel and 45% of CNG, the maximum cylinder pressure is 58.584167.

Figure 4.9 shows cylinder pressure is not changing significantly. This result indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG.

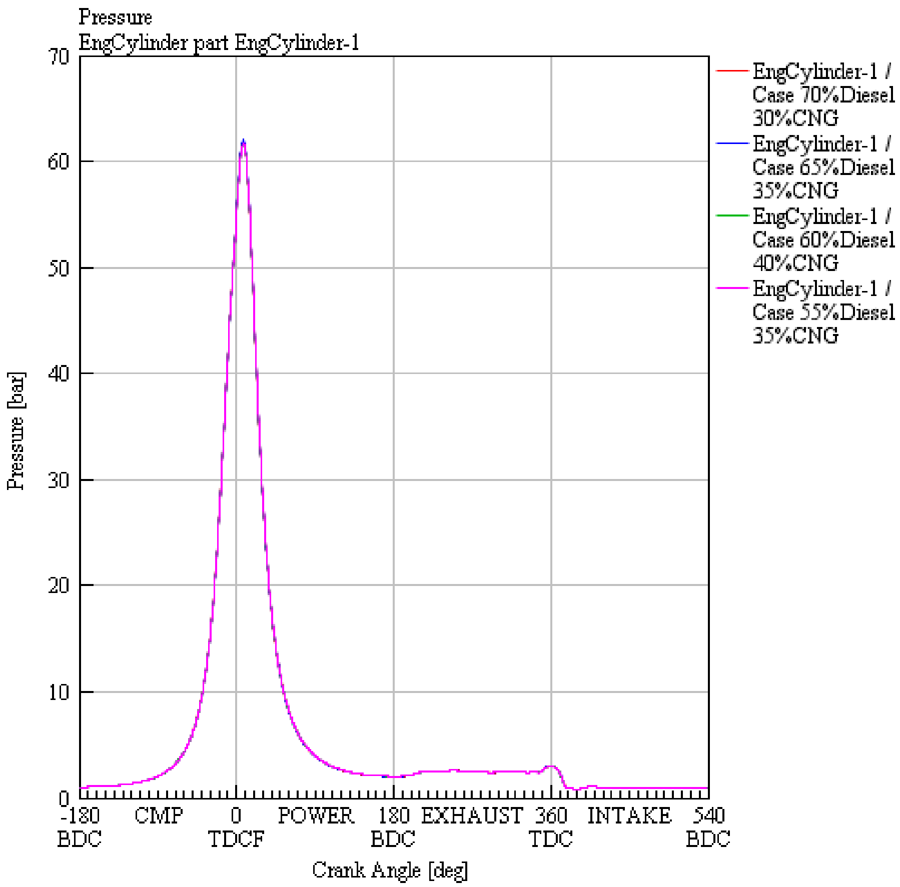

Figure 4. 10 Cylinder pressure graph with variation of gas consumption at load 2000 watt and at constant engine speed 1900 rpm

Figure 4.10 is a graphic of cylinder pressure with crank angle for modified dual fuel diesel engine with gas consumption variation at engine load 12000 watt and at engine speed 1900 rpm. At engine load 2000 watt and engine speed 1900 rpm, the maximum cylinder pressure for each gas consumption variations are 61.80475 bar with ratio 70% of diesel and 30% of CNG, 61.9882784 bar with ratio 65% of diesel and 35% of CNG, 61.68353 bar with ratio 60% of diesel and 40% of CNG, and 61.724102 bar with ratio 55% of diesel and 45% of CNG.

As shown in Figure 4.10, the cylinder pressure is not changing significantly. This result can indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG.

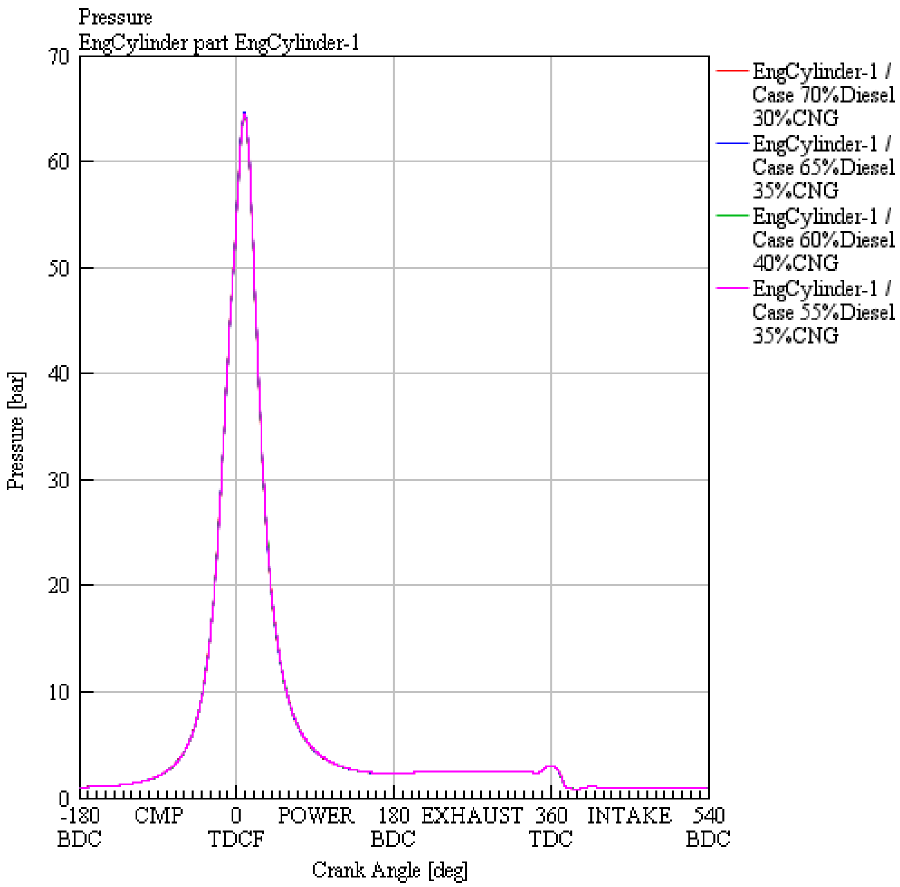

Figure 4. 11 Cylinder pressure graph with variation of gas consumption at load 3000 watt and at engine speed 1900 rpm

Figure 4.11 shows that the graph of cylinder pressure with gas consumption variation at engine load 3000 watt and at engine speed 1900 rpm. The maximum cylinder pressure at engine load 3000 watt and engine speed 1900 rpm with gas consumption variations are 64.50463 bar with ratio 70% of diesel and 30% of CNG, 64.5415 bar with ratio 65% of diesel and 35% of CNG, 64.35801 bar with ratio 60% of diesel and 40% of CNG, and 67.74599 bar with ratio 55% of diesel and 45% of CNG.

From Figure 4.11 shows cylinder pressure for converted diesel engine into dual fuel diesel engine with variation of gas consumption is not changing significantly. This result can indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. In 55% of diesel and 45% of, the maximum cylinder pressure increases again.

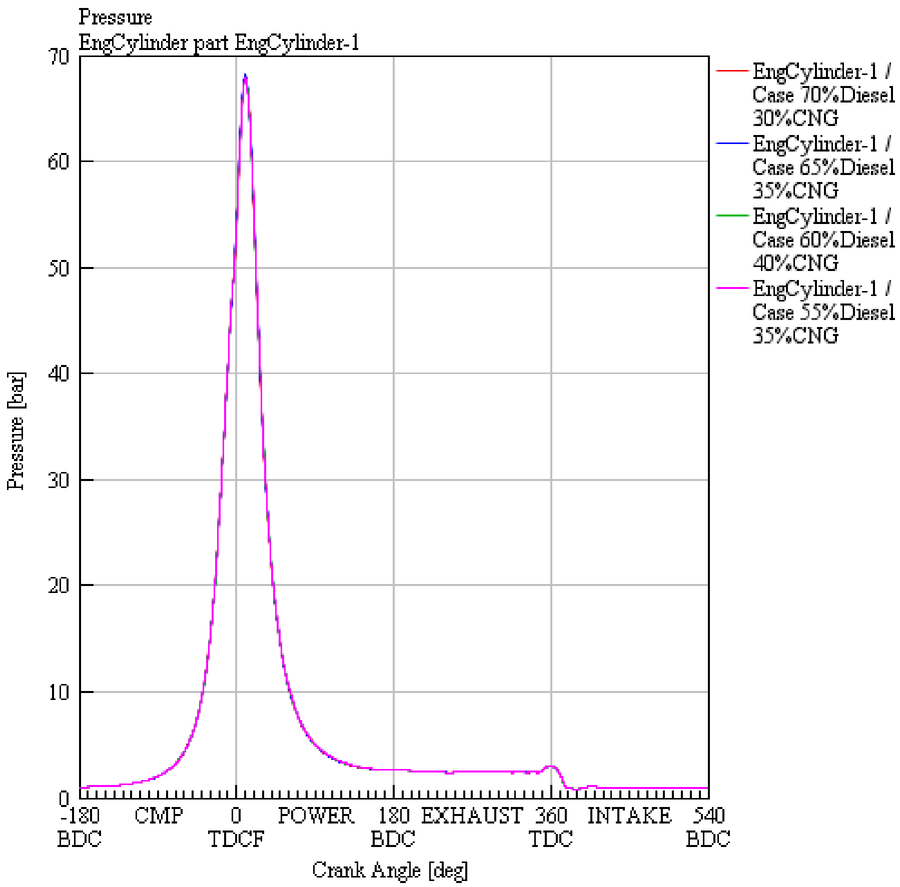

Figure 4. 12 Cylinder temperature graph with variation of gas consumption at load 4000 watt and at engine speed 1800 rpm

Figure 4.12 shows that the graph of cylinder pressure with gas consumption variation at engine load 4000 watt and at engine speed 1900 rpm. The maximum cylinder pressure at engine load 4000 watt and engine speed 1900 rpm with gas consumption variations are 67.8898 bar with ratio 70% of diesel and 30% of CNG, 68.164986 bar with ratio 65% of diesel and 35% of CNG, 67.794914 bar with ratio 60% of diesel and 40% of CNG, and 67.89265 bar with ratio 55% of diesel and 45% of CNG.

From Figure 4.12 can be seen the cylinder pressure for converted diesel engine into dual fuel diesel engine with variation of gas consumption is not changing significantly. This result can indicate that the engine is good because there is not change result significantly. If we see from the result of maximum cylinder pressure, the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with in ratio 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG.

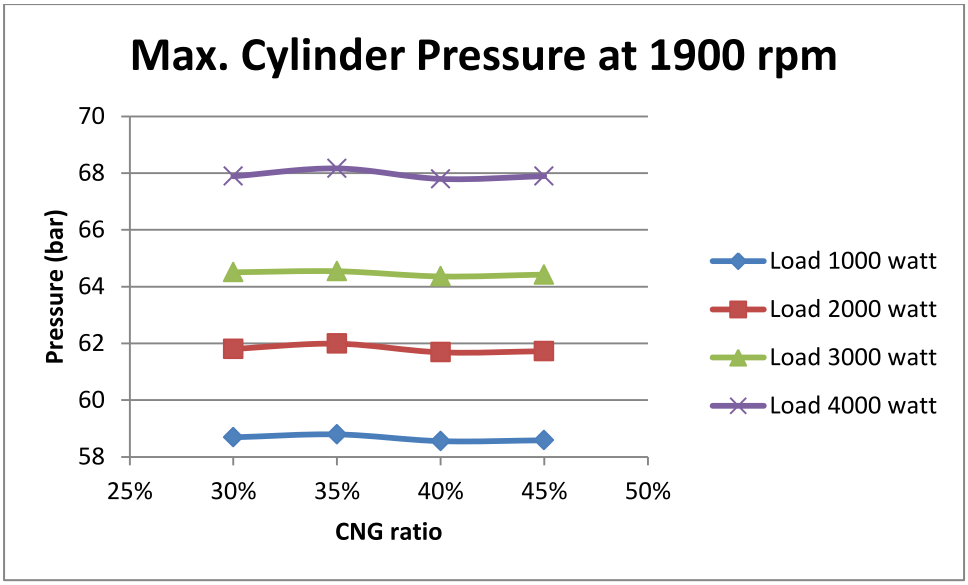

Figure 4. 13 Maximum cylinder pressure graph over CNG ratio with variation of load and at constant engine speed 1900 rpm

As shown in Figure 4.13 show that the graph of maximum cylinder pressure for dual fuel diesel engine with variation of gas consumption at engine load 1000 – 4000 watt with interval 1000 and at constant engine speed 1900 rpm. The results are obtained the maximum cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 30% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG. As shown in Figure 4.13 that the highest maximum cylinder pressure is cylinder pressure with 65% of diesel and 35% of CNG. Also, Figure 4.13 shows that the maximum cylinder pressure increases with increasing the load engine.

4.2.2 The Effect of Gas Consumption Variation to Cylinder Temperature

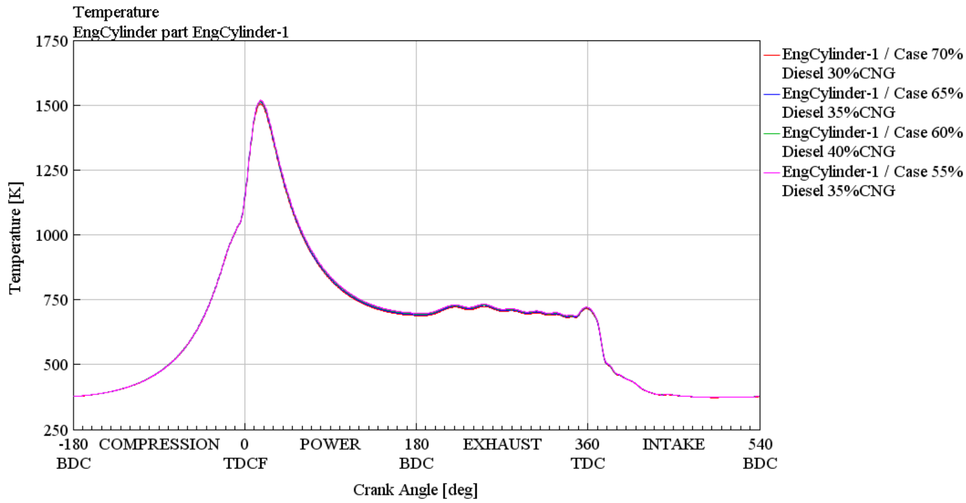

Figure 4. 14 Cylinder temperature graph with variation of gas consumption at load 1000 watt and at engine speed 1800 rpm

Figure 4.14 is a graphic of cylinder temperature with crank angle for modified dual fuel diesel engine with gas consumption variation at engine load 1000 watt and at engine speed 1800 rpm. At engine load 1000 watt and engine speed 1800 rpm, the maximum cylinder temperature for each gas consumption variations are 1504.5746 K with ratio 70% of diesel and 30% of CNG, 1511.9913 K with ratio 65% of diesel and 35% of CNG, 1514.2017 K with ratio 60% of diesel and 40% of CNG, and 1518.5028 K with ratio 55% of diesel and 45% of CNG.

From Figure 4.14 can be seen that the variation of gas consumption is not significant effect to cylinder temperature. The changing not significantly indicates the engine is good because the variation of gas and diesel consumption is not effect to the engine. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 1000 watt and at engine speed 1800 rpm obtained that cylinder temperature increases with increase the composition of CNG.

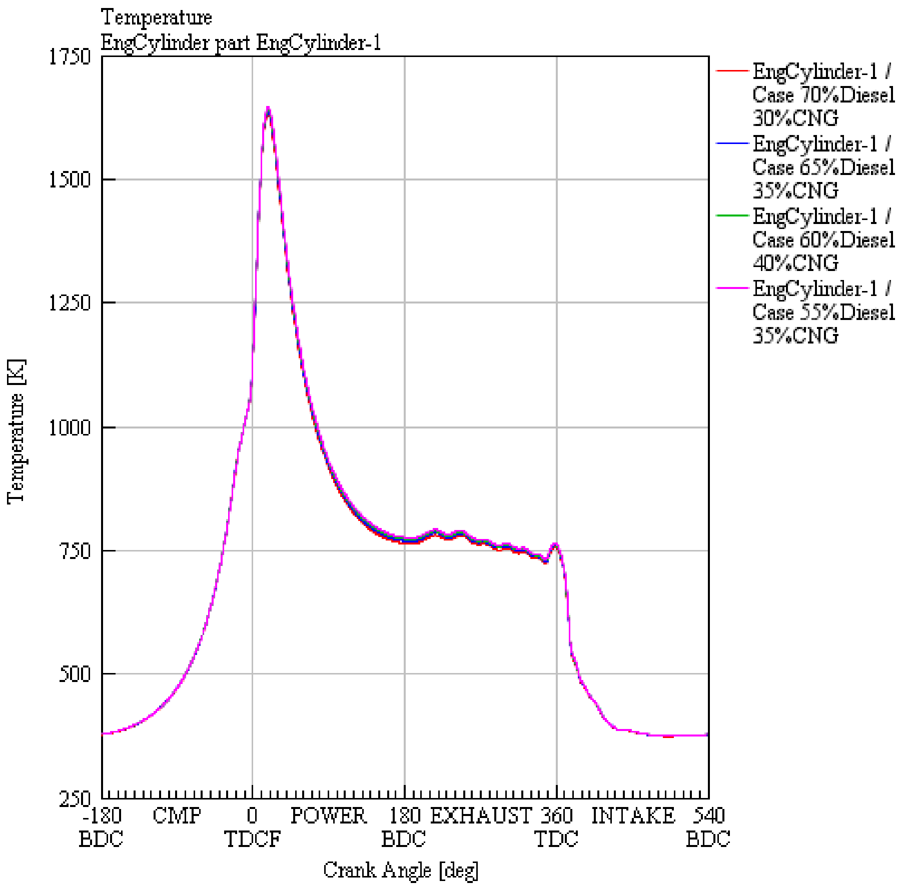

Figure 4. 15 Cylinder temperature graph with variation of gas consumption at load 2000 watt and at constant engine speed 1800 rpm

Figure 4.15 is a graphic of cylinder temperature with gas consumption variation at engine load 2000 watt and at engine speed 1800 rpm. The maximum cylinder temperature for both of gas consumption variations is different. The maximum cylinder temperature are 1628.6544 K with ratio 70% of diesel and 30% of CNG, 1636.5555 K with ratio 65% of diesel and 35% of CNG, 1641.1438 K with ratio 60% of diesel and 35% of CNG, and 1646.2673 K with ratio 55% of diesel and 45% of CNG.

From Figure 4.15 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 2000 watt and at engine speed 1800 rpm obtained that cylinder temperature increases with increase the composition of CNG.

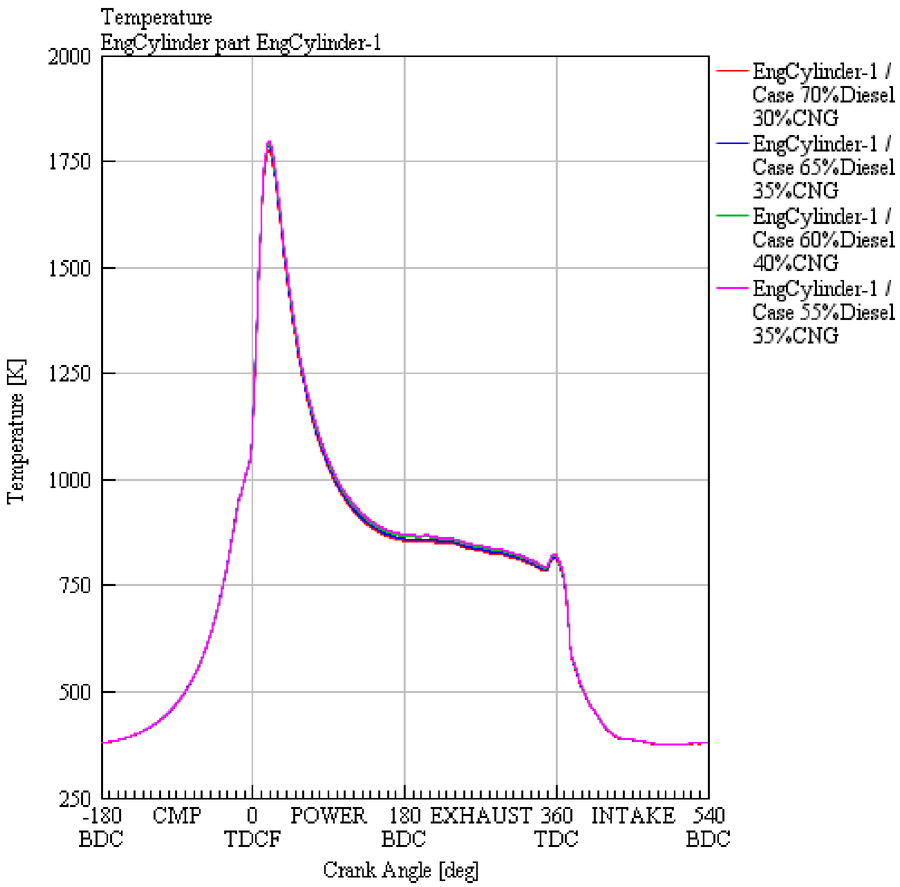

Figure 4. 16 Cylinder temperature graph with variation of gas consumption at load 3000 watt and at constant engine speed 1800 rpm

As shown in Figure 4.16 the graph of cylinder temperature for dual fuel diesel engine with gas consumption variation at engine load 3000 watt and at engine speed 1800 rpm. The results for maximum cylinder temperature with gas consumption variation at engine load 3000 watt and at engine speed 1900 rpm are 1774.9119 K with ratio 70% of diesel and 30% of CNG, 1785.9011 K with ratio 65% of diesel and 35% of CNG, 1790.7441 K with ratio 60% of diesel and 30% of CNG, and 1797.064 K with ratio 55% of diesel and 45% of CNG.

From Figure 4.16 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 3000 watt and at engine speed 1800 rpm obtained that cylinder temperature increases with increase the composition of CNG.

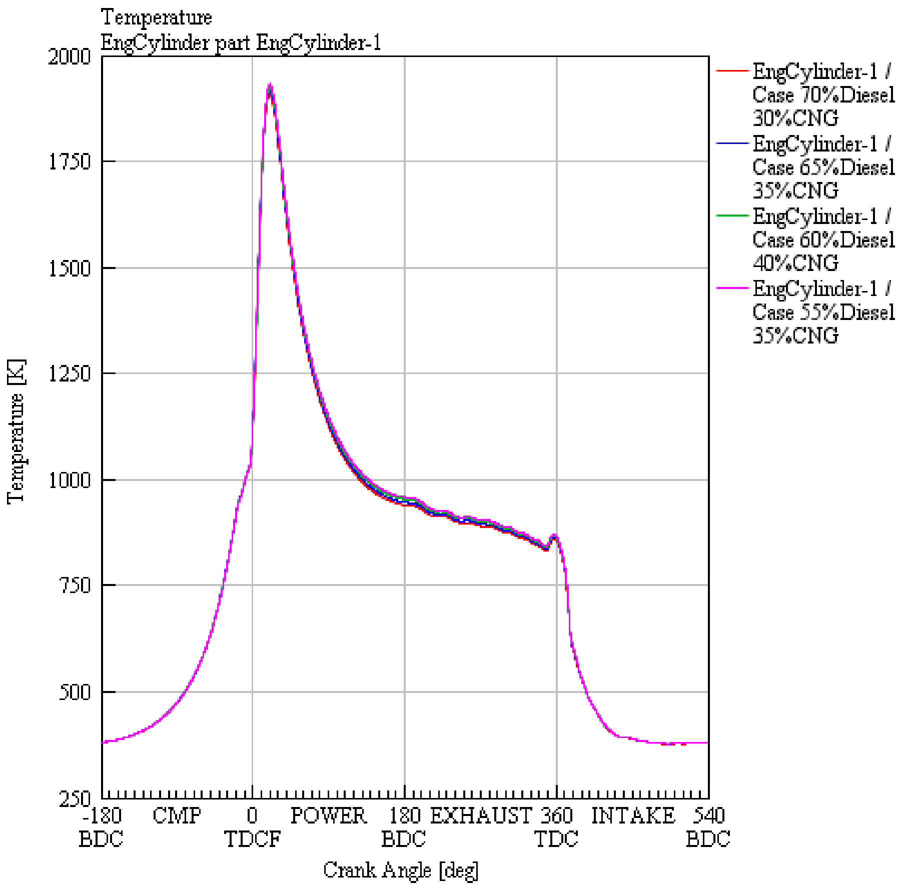

Figure 4. 17 Cylinder temperature graph with variation of gas consumption at load 4000 watt and at constant engine speed 1500 rpm

Figure 4.17 shows that the graph of cylinder temperature with gas consumption variation at engine load 4000 watt and at engine speed 1800 rpm. The maximum cylinder temperature at engine load 4000 watt and engine speed 1800 rpm with gas consumption variations are 1907.4908 K with ratio 70% of diesel and 30% of CNG, 1918.9354 K with ratio 65% of diesel and 35% of CNG, 1925.0597 K with ratio 60% of diesel and 40% of CNG, and 1932.3835 K with ratio 55% of diesel and 45% of CNG.

From Figure 4.17 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 4000 watt and at engine speed 1800 rpm obtained that cylinder temperature increases with increase the composition of CNG.

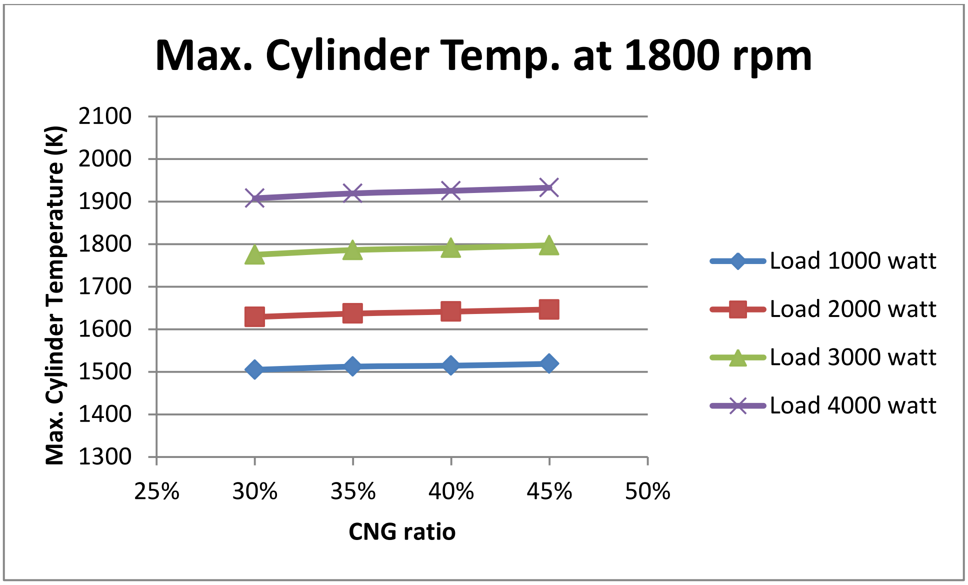

Figure 4. 18 Maximum cylinder temperature graph with variation of gas consumption and engine load at a constant engine speed 1800 rpm

From Figure 4.18 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 1000 - 4000 watt and at engine speed 1800 rpm obtained that cylinder temperature increases with increase the composition of CNG. Also from Figure 4.18 can be seen the maximum cylinder temperature increases with increasing the engine load. It is because the quantity of fuel burned increase. The increasing of fuel quantity cause the heat energy release which resulted in an increased the cylinder temperature.

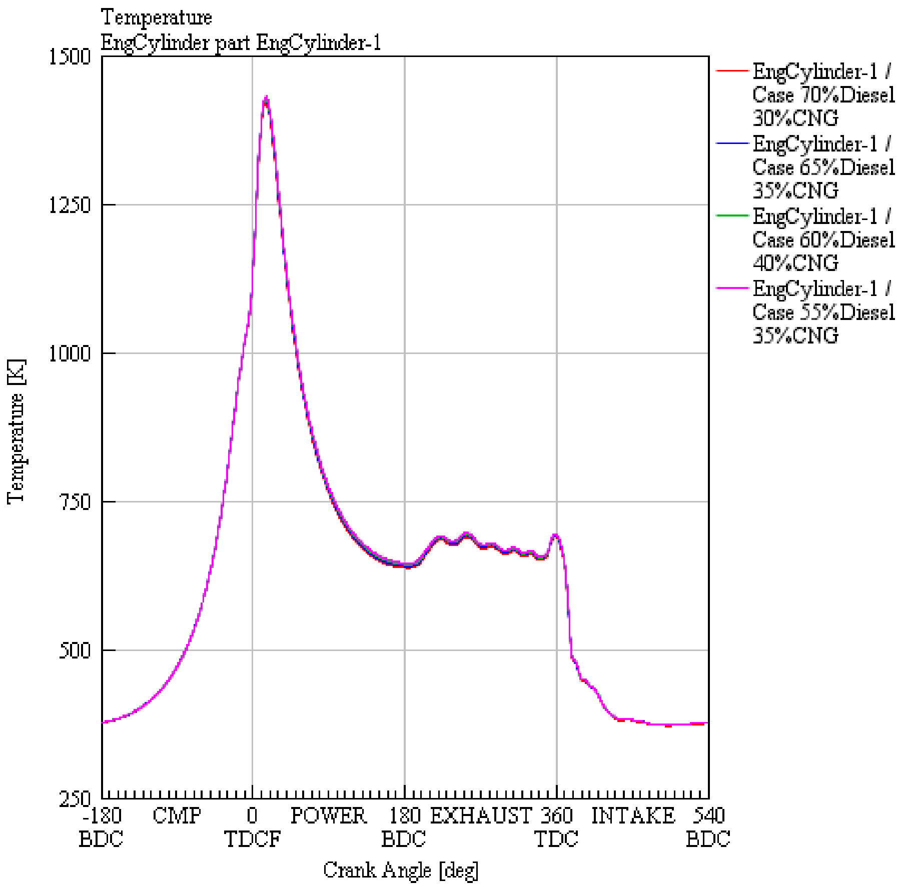

Figure 4. 19 Cylinder temperature with variation of gas consumption at engine load 1000 watt and at engine speed 1900 rpm

Figure 4.19 shows that the graph of cylinder temperature with gas consumption variation at engine load 1000 watt and at engine speed 1900 rpm. The maximum cylinder temperature at engine load 1000 watt and engine speed 1900 rpm occurred changing but not significant. In 70% of diesel and 30% of CNG, the maximum cylinder temperature is 1418.2893 K. In ratio 65% of diesel and 35% of CNG, the maximum cylinder temperature is 1425.1888 K. In ratio 60% of diesel and 40% of CNG, the maximum cylinder temperature is 1925.0597 K. In ratio 55% of diesel and 45% of CNG, the maximum cylinder temperature is 1932.3835 K with

From Figure 4.19 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 1000 watt and at engine speed 1900 rpm obtained that cylinder temperature increases with increase the composition of CNG.

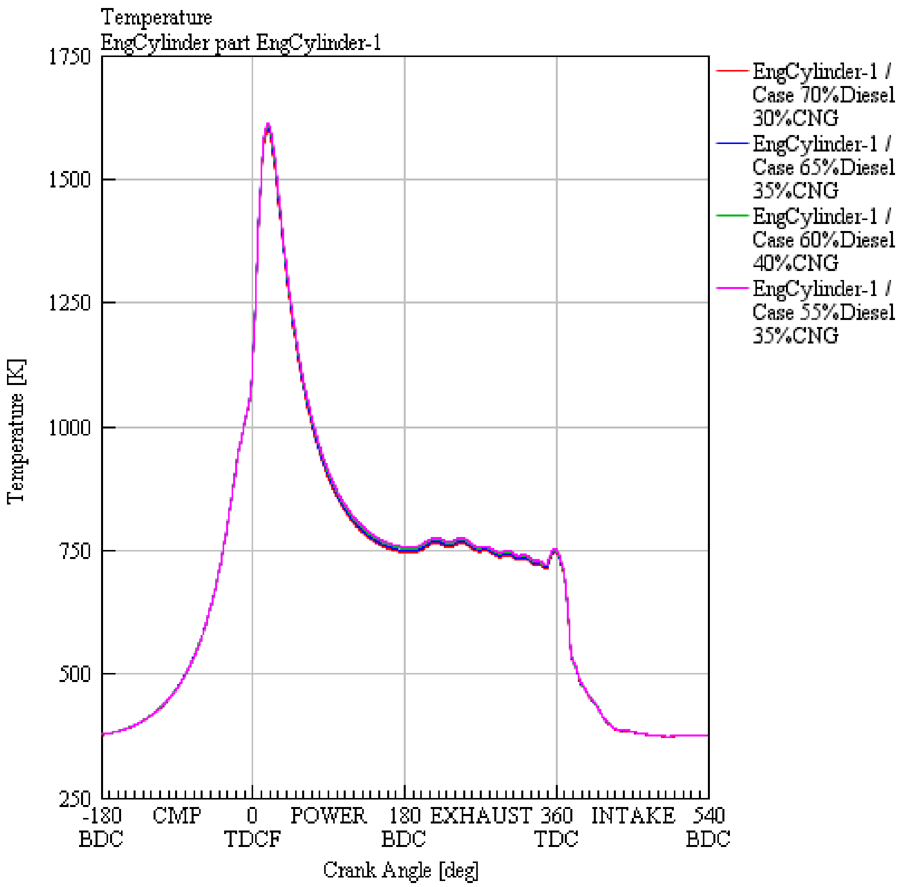

Figure 4. 20 Cylinder temperature graph with variation of gas consumption at engine load 2000 watt and at engine speed 1900 rpm

Figure 4.20 shows that the graph of cylinder temperature for modified dual fuel diesel engine with gas consumption variation at engine load 2000 watt and at engine speed 1900 rpm. The maximum cylinder temperature at engine load 2000 watt and engine speed 1900 rpm with gas consumption variations are 1597.5261 K with ratio 70% of diesel and 30% of CNG, 1606.0605 K with ratio 65% of diesel and 35% of CNG, 1609.3745 K with ratio 60% of diesel and 40% of CNG, and 16140995 K with ratio 55% of diesel and 45% of CNG.

From Figure 4.20 can be seen that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 2000 watt and at engine speed 1900 rpm obtained that cylinder temperature increases with increase the composition of CNG.

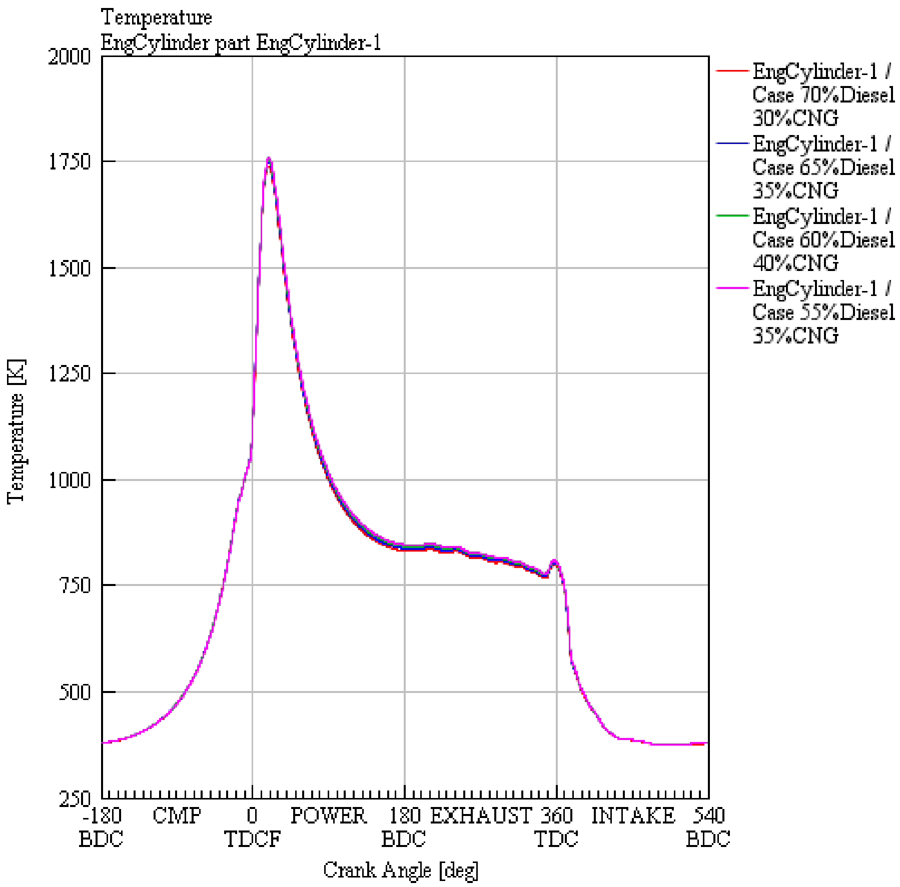

Figure 4. 21 Cylinder temperature graph with variation of gas consumption at engine load 3000 watt and at engine speed 1900 rpm

A shown in Figure 4.21 that the graph of cylinder temperature for converted diesel engine into dual fuel diesel engine with gas consumption variation at engine load 3000 watt and at engine speed 1900 rpm. The maximum cylinder temperature at engine load 3000 watt and engine speed 1900 rpm with gas consumption variations are 1738.6411 K with ratio 70% of diesel and 30% of CNG, 1747.8799 K with ratio 65% of diesel and 35% of CNG, 1752.8445 K with ratio 60% of diesel and 40% of CNG, and 1759.1095 K with ratio 55% of diesel and 45% of CNG.

From Figure 4.21 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 3000 watt and at engine speed 1900 rpm obtained that cylinder temperature increases with increase the composition of CNG.

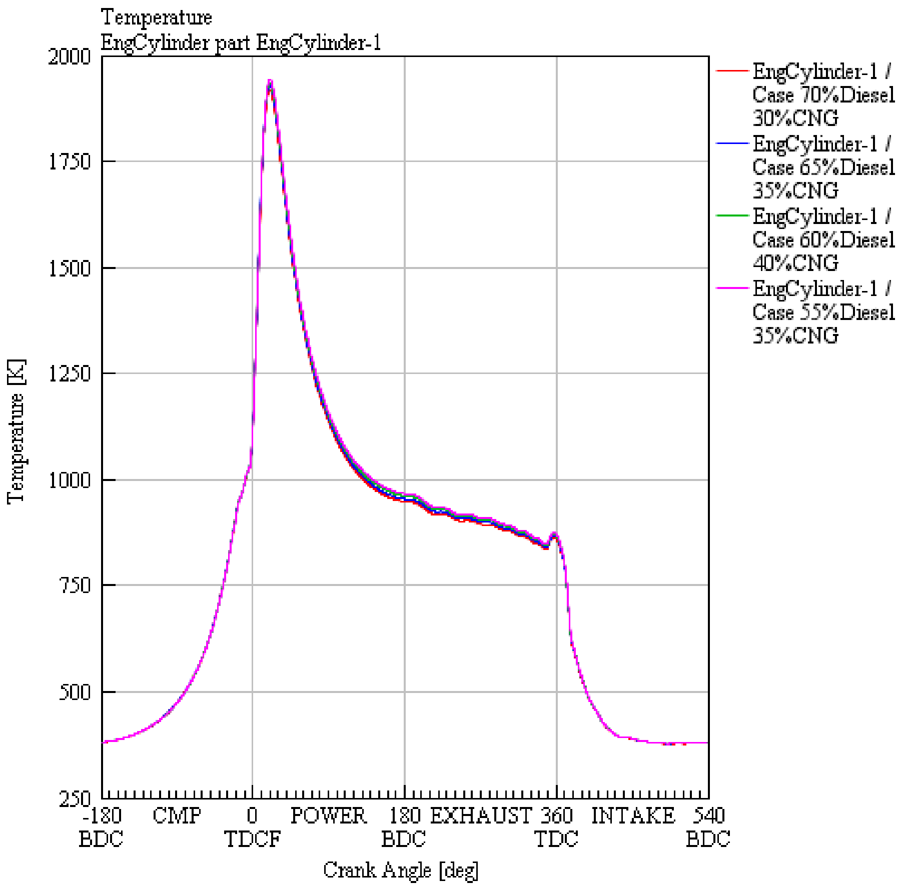

Figure 4. 22 Cylinder temperature graph with variation of gas consumption at engine load 4000 watt and at engine speed 1900 rpm

Figure 4.22 shows that the graph of cylinder temperature with gas consumption variation at engine load 4000 watt and at engine speed 1900 rpm. The maximum cylinder temperature at engine load 4000 watt and engine speed 1900 rpm with gas consumption variations are not significant changed. In ratio 70% of diesel and 30% of CNG, the maximum cylinder temperature is 1918.3148 K. In ratio 65% of diesel and 35% of CNG, the maximum cylinder temperature is 1930.0857 K. In ratio 60% of diesel and 40% of CNG, the maximum cylinder temperature is 1935.8936 K. In ratio 55% of diesel and 45% of CNG, the maximum cylinder temperature is 1943.5807 K.

From Figure 4.23 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 4000 watt and at engine speed 1900 rpm obtained that cylinder temperature increases with increase the composition of CNG.

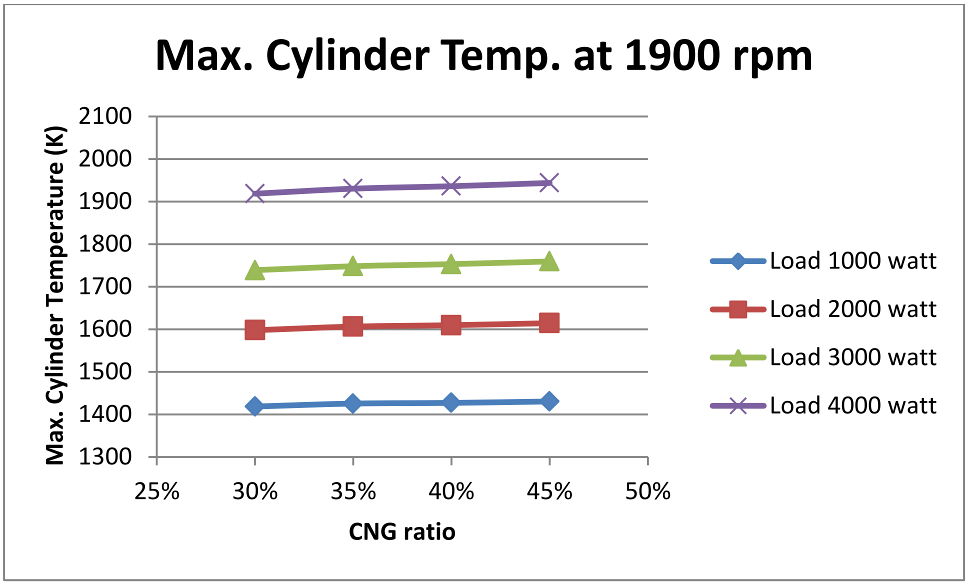

Figure 4. 23 Maximum cylinder temperature graph with variation of gas consumption and engine load at a constant engine speed 1900 rpm

From Figure 4.23 shows that the variation of gas consumption is not significant effect to cylinder temperature. From the results of maximum cylinder temperature for both of gas consumption variation at engine load 1000 - 4000 watt and at engine speed 1900 rpm obtained that cylinder temperature increases with increase the composition of CNG. Also from shown Figure 4.23 the maximum cylinder temperature increases with increasing the engine load. It is because the quantity of fuel burned increases with increasing the engine load. The increasing of fuel quantity cause the heat energy release which resulted in an increased the cylinder temperature.

4.2.3 The Effect of Gas Consumption Variation to Heat Release Rate

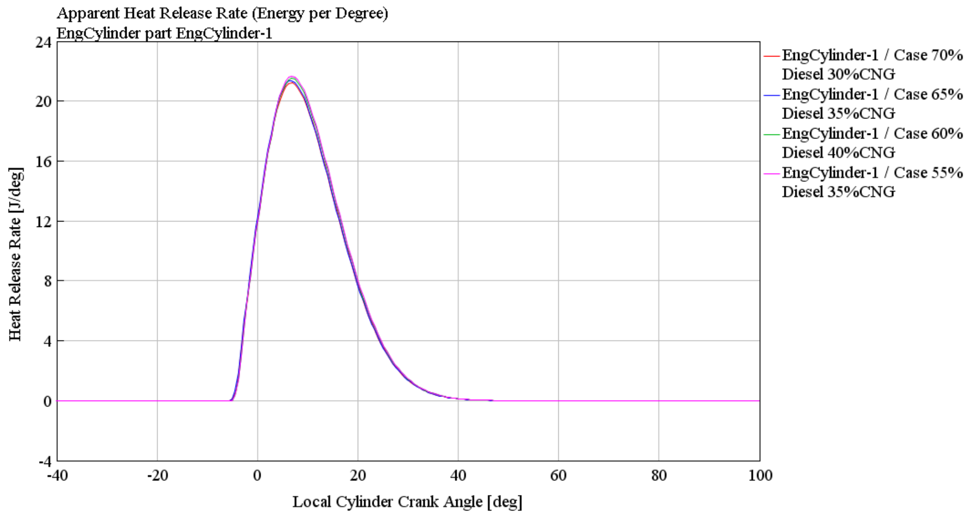

Figure 4. 24 Heat release rate graph with variation of gas consumption at engine load 1000 watt and at engine speed 1800 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

Figure 4.24 is a heat release rate graphic with gas consumption variation at load 1000 watt and at engine speed 1800 rpm. As shown in Figure 4.24, the peak heat release is not significant increasing with increase the gas consumption injected. But if we see in the result of maximum cylinder temperature which is increased with the increasing of gas consumption, it can be concluded that the heat release rate also increased with increasing the gas consumption.

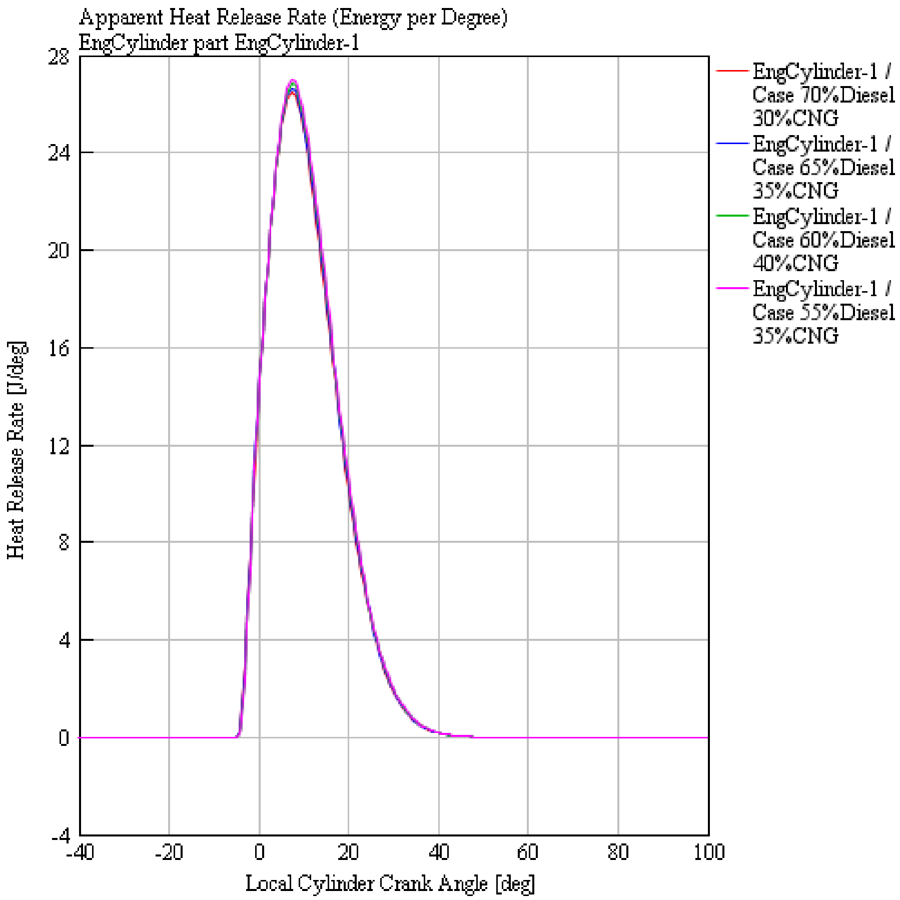

Figure 4. 25 Heat release rate graph with variation of gas consumption at engine load 2000 watt and at engine speed 1800 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

From Figure 4.25 can be seen the heat release rate graph for modified dual fuel diesel engine with gas consumption variation at engine load 2000 watt and at engine speed 1800 rpm. As shown in Figure 4.25, the peak heat release is not significant increasing with increase the gas consumption injected. But if we see in the result of maximum cylinder temperature which is increased with the increasing of gas consumption, it can be implied that the heat release rate also increased with increasing the gas consumption.

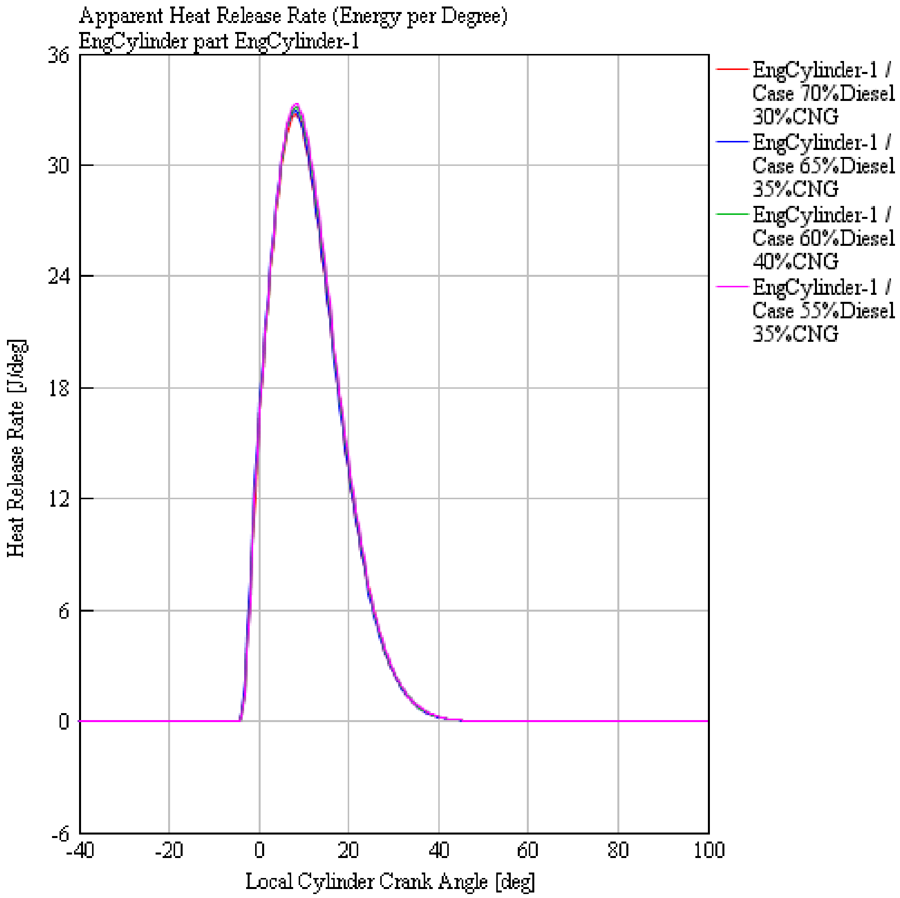

Figure 4. 26 Heat release rate graph with variation of gas consumption at engine load 3000 watt and at engine speed 1800 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

As shown in Figure 4.26, there is the heat release rate for converted diesel engine into dual fuel diesel engine with variation of gas consumption at engine load 3000 watt and at engine speed 1800 rpm. Figure 4.26 shows the peak heat release is not significant increasing with increase the gas consumption injected.

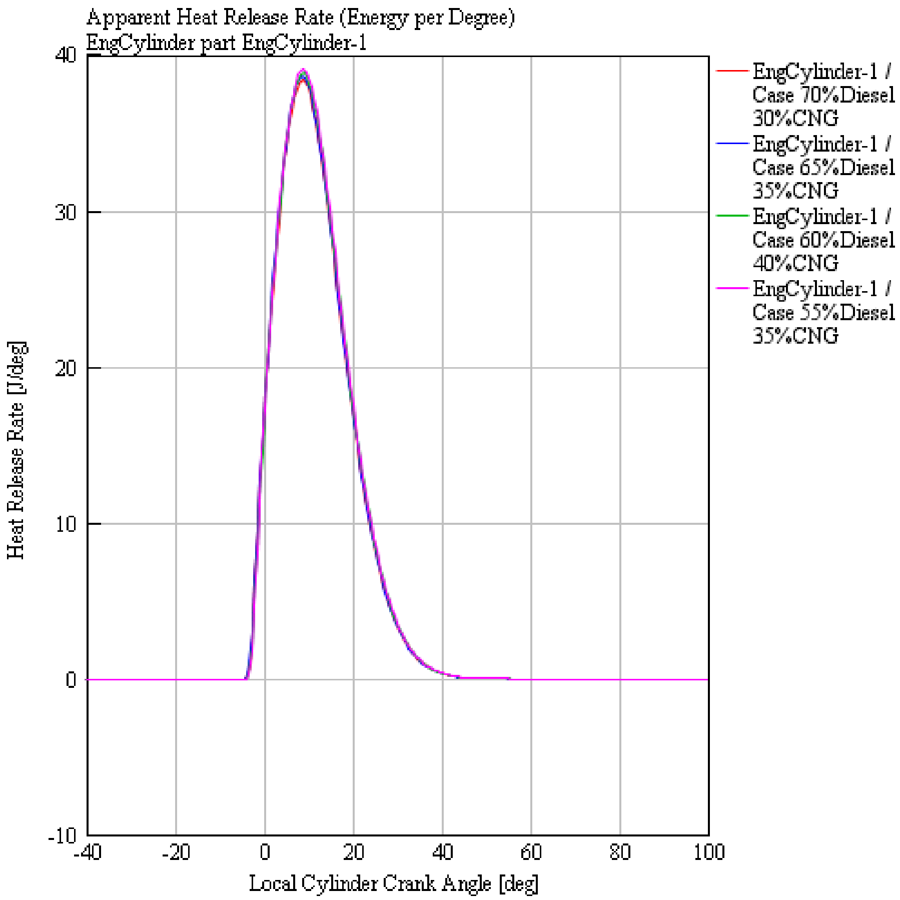

Figure 4. 27 Heat release rate graph with variation of gas consumption at engine load 4000 watt and at engine speed 1800 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

Figure 4.27 is a graph of heat release rate with gas consumption variation at engine load 4000 watt and at engine speed 1800 rpm. As shown in Figure 4.27, the heat release is not significant increasing with increase the gas consumption injected.

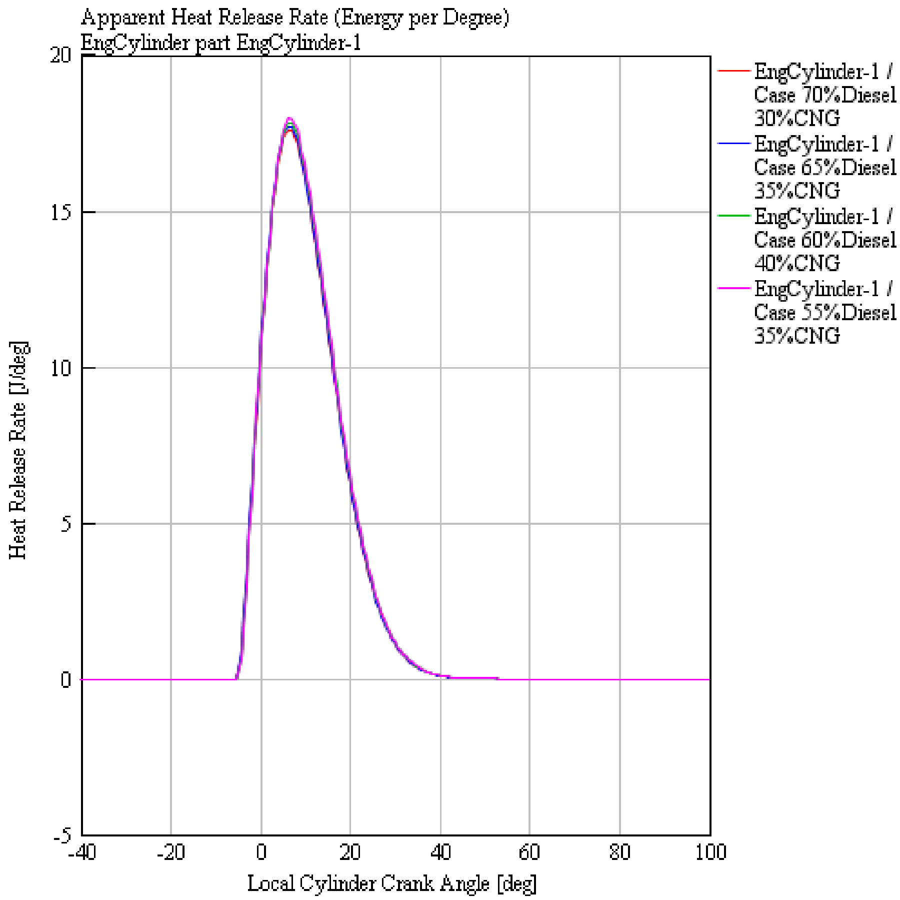

Figure 4. 28 Heat release rate graph with variation of gas consumption at engine load 1000 watt and at engine speed 1900 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

Figure 4.28 is a heat release rate graphic with gas consumption variation at load 1000 watt and at engine speed 1900 rpm. As shown in Figure 4.28, the peak heat release is not significant increasing with increase the gas consumption injected. But if we see in the result of maximum cylinder temperature which is increased with the increasing of gas consumption, it can be concluded that the heat release rate also increased with increasing the gas consumption.

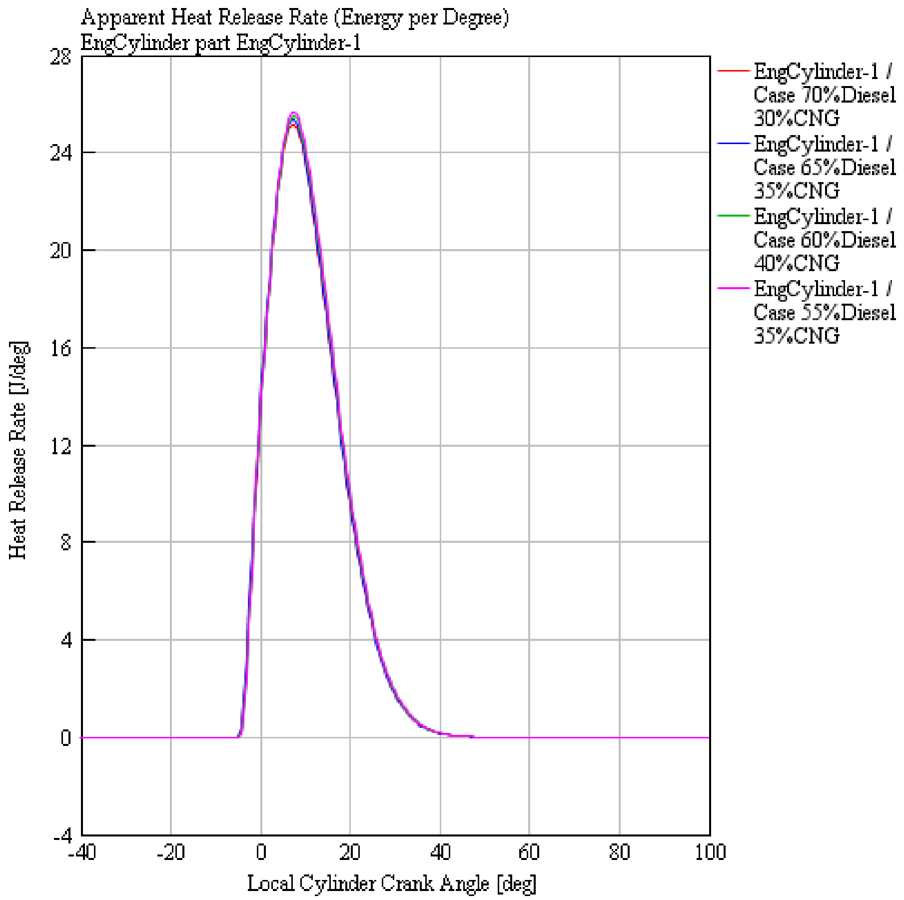

Figure 4. 29 Heat release rate graph with variation of gas consumption at engine load 2000 watt and at engine speed 1900 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

From Figure 4.29 can be seen the heat release rate graph for modified dual fuel diesel engine with gas consumption variation at engine load 2000 watt and at engine speed 1900 rpm. As shown in Figure 4.29, the peak heat release is not significant increasing with increase the gas consumption injected. But if we see in the result of maximum cylinder temperature which is increased with the increasing of gas consumption, it can be implied that the heat release rate also increased with increasing the gas consumption.

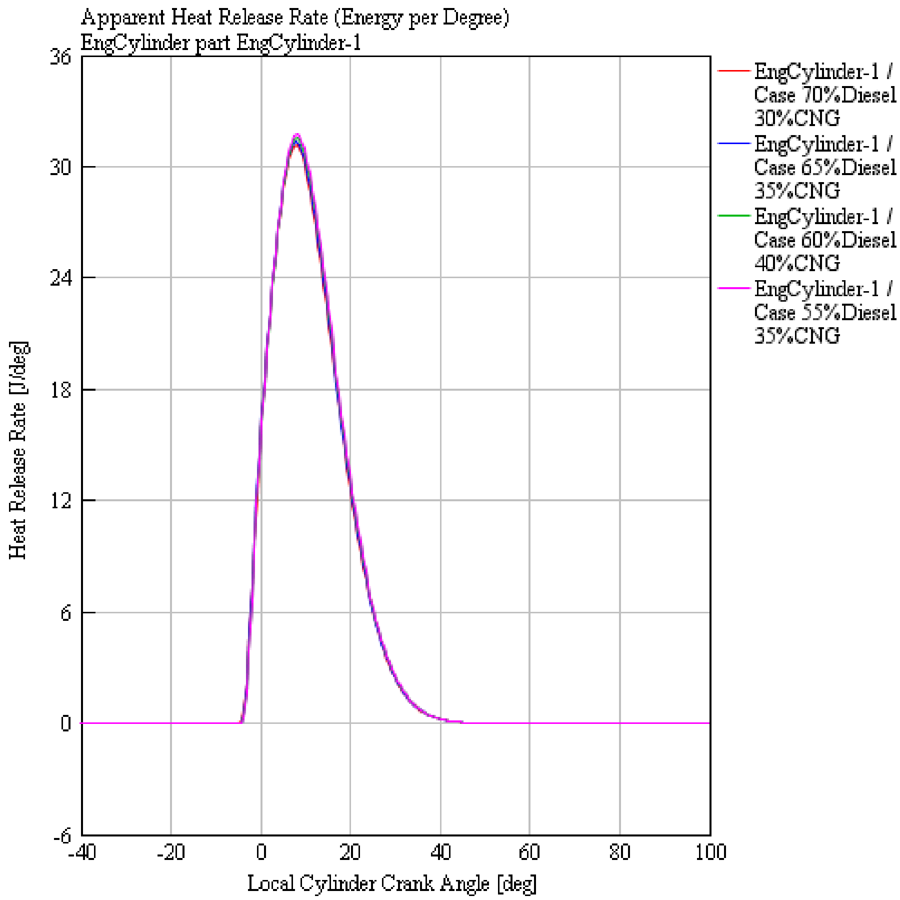

Figure 4. 30 Heat release rate graph with variation of gas consumption at engine load 3000 watt and at engine speed 1900 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

As shown in Figure 4.30, there is the heat release rate for converted diesel engine into dual fuel diesel engine with variation of gas consumption at engine load 3000 watt and at engine speed 1900 rpm. Figure 4.30 shows the peak heat release is not significant increasing with increase the gas consumption injected.

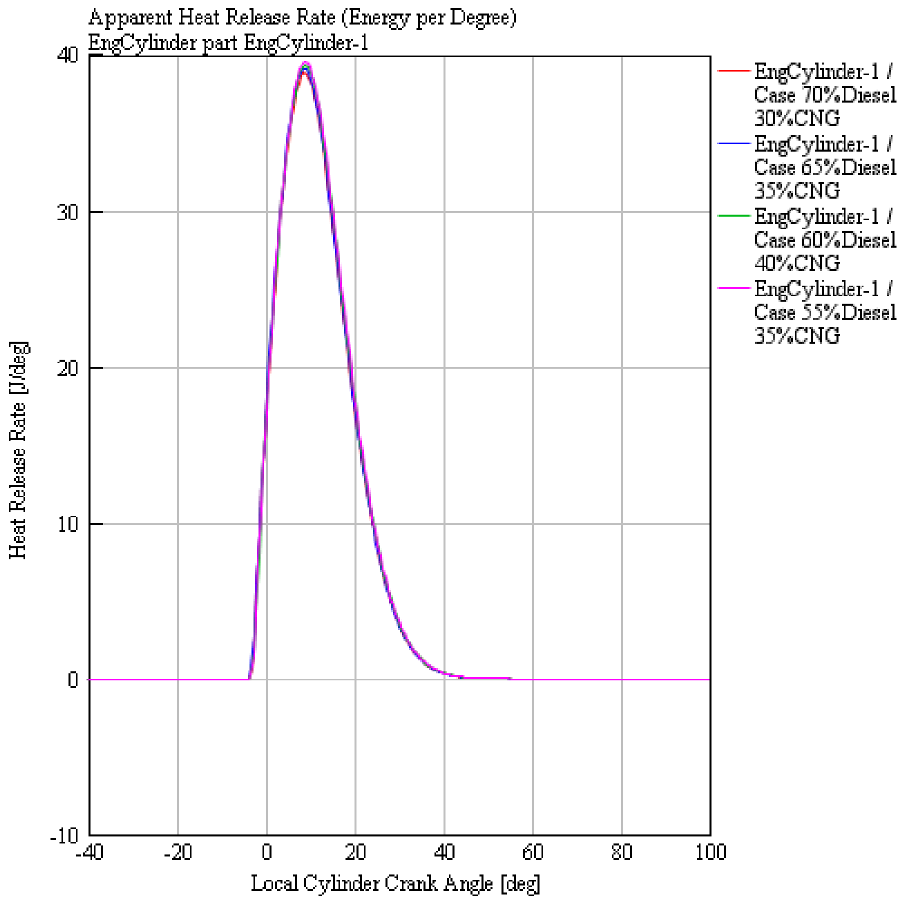

Figure 4. 31 Heat release rate graph with variation of gas consumption at engine load 4000 watt and at engine speed 1900 rpm

The heat release rate is used to identify the start of combustion, the fraction of fuel burned, and differences in combustion rates of fuel. The heat rate curves exhibited a drop due to charge cooling effect shortly after the onset of the first injection. Subsequently, the heat release rate rapidly increased due to premixed combustion. The peak heat release rate increased as the timing of the first injection was further retarded, this was mainly due to improved fuel evaporation and mixing processes whereby the greater fuel quantity injected was the primary factor.

Figure 4.31 is a graph of heat release rate with gas consumption variation at engine load 4000 watt and at engine speed 1900 rpm. As shown in Figure 4.31, the heat release is not significant increasing with increase the gas consumption injected.

4.2.4 The Effect of Gas Consumption Variation to Volumetric Efficiency

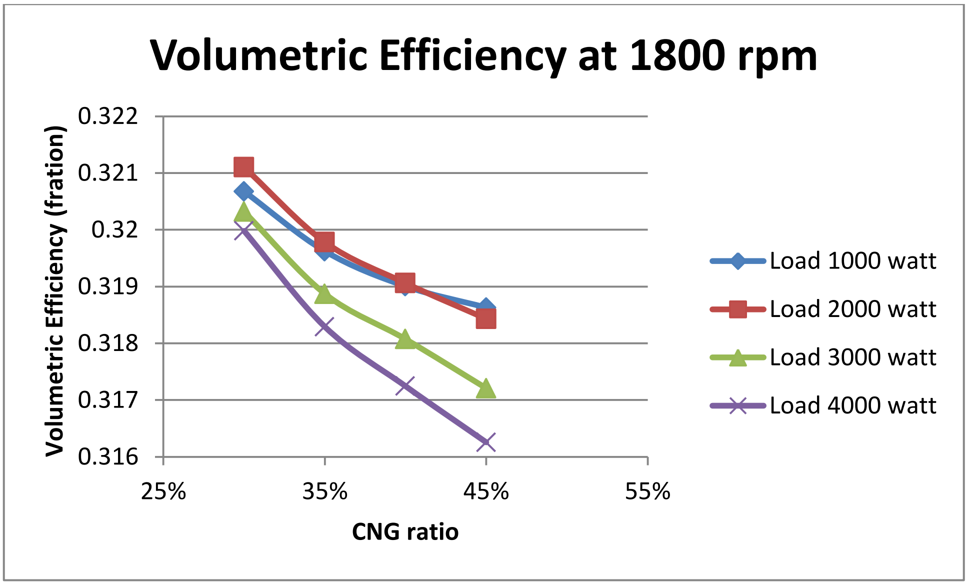

Figure 4. 32 The volumetric efficiency graph with variation of gas consumption and engine load variation at engine speed 1800 rpm

In Figure 4.32 shows a volumetric efficiency graphic for modified dual fuel diesel engine with engine load variation 1000 – 4000 watt and at constant engine speed 1800 rpm. As shown in Figure 4.32, the increased engine load can increase the volumetric efficiency with at constant engine speed. Figure 4.32 also shows volumetric efficiency decreases with increasing the gas consumption. The volumetric efficiency at engine load 1000 watt is 0.3206773 with 70% of diesel and 30% of CNG, 0.31962332 with 65% of diesel and 35% of CNG, 0.31900427 with 60% of diesel and 40% of CNG, and 0.31862968 with 55% of diesel and 45% of CNG. The volumetric efficiency at engine load 2000 watt is 0.32110205 with 70% of diesel and 30% of CNG, 0.3197827 with 65% of diesel and 35% of CNG, 0.31906024 with 60% of diesel and 40% of CNG, and 0.31843084 with 55% of diesel and 45% of CNG. The volumetric efficiency at engine load 3000 watt is 0.32031888 with 70% of diesel and 30% of CNG, 0.31887323 with 65% of diesel and 35% of CNG, 0.31807294 with 60% of diesel and 40% of CNG, and 0.3172073 with 55% of diesel and 45% of CNG. The volumetric efficiency at engine load 4000 watt is 0.31998044 with 70% of diesel and 30% of CNG, 0.3182913 with 65% of diesel and 35% of CNG, 0.3172442 with 60% of diesel and 40% of CNG, and 0.3162548 with 55% of diesel and 45% of CNG.

Volumetric efficiency defines as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston. So if the volumetric efficiency value rises, this is due to the increasing volume flow rate of air into the intake system. From Figure 4.32 can be seen the volumetric efficiency decrease with increased the gas consumption. This is due to decreasing the volume flow rate of air into the intake system with increasing the engine load.

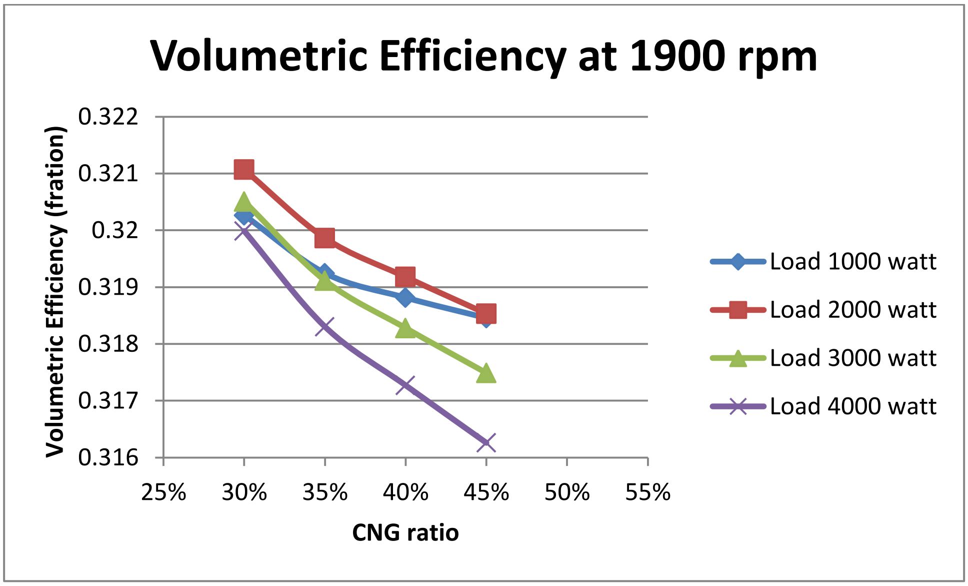

Figure 4. 33 The volumetric efficiency graph with variation of gas consumption and engine load variation at engine speed 1900 rpm

Figure 4.33 shows a volumetric efficiency graphic for converted diesel engine into dual fuel diesel engine with engine load variation 1000 – 4000 watt and at constant engine speed 1900 rpm. As shown in Figure 4.33, the increased engine load can increase the volumetric efficiency with at constant engine speed.

Figure 4.33 also shows volumetric efficiency decreases with increasing the gas consumption. The volumetric efficiency at engine load 1000 watt with ratio 70% of diesel and 30% of CNG is 0.3202658. The volumetric efficiency at engine load 1000 watt with ratio 65% of diesel and 35% of CNG is 0.31924543. the volumetric efficiency at engine load 1000 watt with ratio 60% of diesel and 40% of CNG is 0.31881398. The volumetric efficiency at engine load 1000 watt with ratio 55% of diesel and 45% of CNG 0.318456. The volumetric efficiency at engine load 2000 watt with ratio 70% of diesel and 30% of CNG is 0.32106915. The volumetric efficiency at engine load 2000 watt with ratio 65% of diesel and 35% of CNG is 0.3198629. the volumetric efficiency at engine load 2000 watt with ratio 60% of diesel and 40% of CNG is 0.31917846. The volumetric efficiency at engine load 2000 watt with 55% of diesel and 45% of CNG is 0.31853217. The volumetric efficiency at engine load 3000 watt is 0.32050782 with 70% of diesel and 30% of CNG, 0.3191158 with 65% of diesel and 35% of CNG, 0.318278 with 60% of diesel and 40% of CNG, and 0.3174882 with 55% of diesel and 45% of CNG. The volumetric efficiency at engine load 4000 watt is 0.31998706 with 70% of diesel and 30% of CNG, 0.31830266 with 65% of diesel and 35% of CNG, 0.31726947 with 60% of diesel and 40% of CNG, and 0.3162578 with 55% of diesel and 45% of CNG.

Volumetric efficiency defines as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston. So if the volumetric efficiency value rises, this is due to the increasing volume flow rate of air into the intake system. From Figure 4.33 can be seen the volumetric efficiency decrease with increased the gas consumption. This is due to decreasing the volume flow rate of air into the intake system with increasing the engine load.

CHAPTER V
CONCLUSION

Based on the results of simulation that have been carried out using GT-Power regarding combustion performance for dual fuel diesel engine can be summed up:

1. The cylinder pressure is no significant changes occurred. The maximum cylinder pressure for both of gas consumption variation at every engine load and at constant engine speed obtained the cylinder pressure increases with added the gas ratio become 65% of diesel and 35% of CNG from 70% of diesel and 35% of CNG. Then the maximum cylinder pressure decreases with added the gas consumption become 60% of diesel and 40% of CNG. The maximum cylinder pressure increases again with consume 55% of diesel and 45% of CNG. The highest maximum cylinder pressure is cylinder pressure with 65% of diesel and 35% of CNG.
2. The cylinder temperature is not significant increase with the increasing gas substitution at constant engine load and at engine speed.
3. The peak heat release rate is no significant increase with increase the gas consumption at constant engine load and at constant engine speed.
4. The volumetric efficiency defines as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston. So if the volumetric efficiency value rises, this is due to the increasing volume flow rate of air into the intake system. The volumetric efficiency will decrease with increased the gas ratio. This is due to decreasing the volume flow rate of air into the intake system with increasing the gas ratio.

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BIBLIOGRAPHY

Abdelaal, M., & Hegab, A. (2012). Combustion and Emission Characteristics of A Natural Gas-Fueled Diesel Engine with EGR. *Energy Conversion and Management*.

Breeze, P. (2018). Diesel Engines. In P. Breeze, *Piston Engine - Based Power Plants.* London: Academic Press.

Budiyanto, M. A. (2012). *Simulasi Pembakaran Mesin Diesel Bahan Bakar Ganda (Solar-Gas).* Depok: Perpustakaan Universitas Indonesia.

Castro , N., Toledo, M., & Amador, G. (2019). An Experimental Investigation of The Performance and Emissions of Hydrogen-Diesel Dual Fuel Compression Ignition Internal Combustion Engine. *Applied Thermal Engineering*.

Cheenkachorn, K., Poompipatpong, C., & Ho, C. G. (2013). Performance and Emissions of A heavy-Duty Diesel Engine Fuelled with Diesel LNG (Liquid Natural Gas). *Energy*.

Departement of Energy United States of America. (2005). *Natural Gas: Understanding the Basic Facts.* America: U. S. Departement of Energy Office of Fossil Energy.

Dimitriou, P., Tsujimura, T., & Suzuki, Y. (2019). Low-Load Hydrogen-Diesel Dual-Fuel Engine Operation - A Combustion Efficiency Improvement Approach. *International Journal of Hydrogen Energy*, 1-13.

Dzahab, I. S. (2019). *Analisa Esperimen Pengaruh Gas Split Injection pada Mesin Bahan Bakar Ganda terhadap Emisi Gas Buang.* Surabaya.

Ghazal, O. (2019). Combustion Analysis of Hydrogen-Diesel Dual Fuel Engine with Water Injection Technique. *Case Studies in Thermal Engineering*.

H., R. Y., & Ariana, I. (2009). Analisa Pembakaran Pada Dual Fuel DIesel Engine Berbahan Bakar Utama Compressed Natural Gas (CNG) Dengan Metode Pemodelan. *Digital Library ITS*.

Herfatmanesh, M., Zhao, H., & Ganippa, L. (2011). In-Cylinder Studies of Split Injection in A Single CYlinder Optical Diesel Engine. *Internal Combustion Engines: Improving Performance, Fuel Economy, and Emission*.

Heywood, J. (1988). *Internal Combustion Engine Fundamentals.* United States of America: McGraw-Hill, Inc.

Kassa, M., & Hall, C. (2018). Dual-Fuel Combustion. *IntechOpen*, 2-18.

Kharim, G. A. (1980). A Review of Combustion Processes in The Dual Fuel Engine - The Gas Diesel Engine. *Pergamon Press*.

Mansor, W. N. (2014). *Dual Fuel Engine Combustion And Emissions - An Experiemental Investigation Coupled With Computer Simulation.* Colorado: Sematicscholar.

Mokhatab, S., Poe, W., & Mak, J. (2018). *Hanbook of Natural Gas Transmission an Processing.* USA: Gulf Professional Publishing.

Mustafi, N., Raine, R., & Verhelst, S. (2013). Combustion and Emissions Characteristics of A Dual Fuel Engine Operated on Alternative Gaseous Fuels. *Fuel*, 1-10.

Nabi, M. N., Rasul, M., & Gudimetla, P. (2018). Modelling and Simulation of Performance and Combustion Characteristics of Diesel Engine. *Energy Procedia*, 1-8.

National Energy Council. (2019). Outlook Energi Indonesia 2019. *Outlook Energi Indonesia 2019*.

Ouchikh, S., Lounici, M., Tarabet, L., Loubar, K., & Tazerout, M. (2019). Effect of Natural Gas Enrichment with Hydrogen on Combustion Characteristics of A Dual Fuel Diesel Engine. *Interrnational Journal of Hydrogen Energy*.

Papagiannakis, R., Hountalas, D., & Rakopouluos, C. (2007). Theoretical Study of The Effects of Pilot Fuel Quantity and Its Injection Timing on The Performance and Emissions of A Dual Fuel Diesel Engine. *Energy Conversion and Management*, 1-11.

Raheman, H., & Padhee, D. (2014). Combustion Characteristics of Diesel Engine Using Producer Gas and Blends ofJatropha Methyl Ester with Diesel in Mixed Fuel Mode. *Int. Journal of Renewable Energy Development*.

Rohiim, F. (2019). *Kajian Eksperimental Performa, Combustion, Methane Slip pada Modifikasi Diesel Engine menjadi Dual Fuel Engine.* Surabaya.

Semin, & Bakar, R. A. (2008). A Technical of Compressed Natural Gas as An Alternative Fuel for Internal Combustion Engines. *American J. of Engineering and Applied Sciences*.

Semin, Bakar, A., & Bakar, R. A. (2008). Comparative Performance of Direct Injection Diesel Engines Fuelled Using Compressed Natural Gas and Diesel Fuel Based on GT_POWER Simulatiion. *American Journl of Applied Sciences*.

Semin, Idris, A., & Bakar, R. A. (2009). An Overview of Compressed Natural Gas as an Alternative Fuel and Malaysian Scenario. *European Journal of Scientific Research*.

Semin, Ismail, A. R., & Bakar, R. A. (2009). Combustion Temperature Effect of Diesel Engine Convert to Compressed Natural Gas Engine. *American J. of Engineering and Applied Sciences*.

Semin, Ismail, A. R., & Bakar, R. A. (2009). Effect of Diesel Engne Converted to Sequential Port Injection Compressed Natural Gas Engine on The Cylinder Pressure vs Crank Angle in Variation Engine Speeds. *American J. of Engineering and Applied Sciences*.

Wei, L., & Geng, P. (2015). A Review on Natural Gas/Diesel Dual Fuel Combustion, Emissions, and Performance. *Fuel Processing Technology*.

Wijaya, T. A. (2019). *ANalisa Emisi Mesin Dual Fuel menggunakan Injektor Multi Diameters Hole berbasis Simulasi.* Surabaya.

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APPENDIX

APPENDIX A

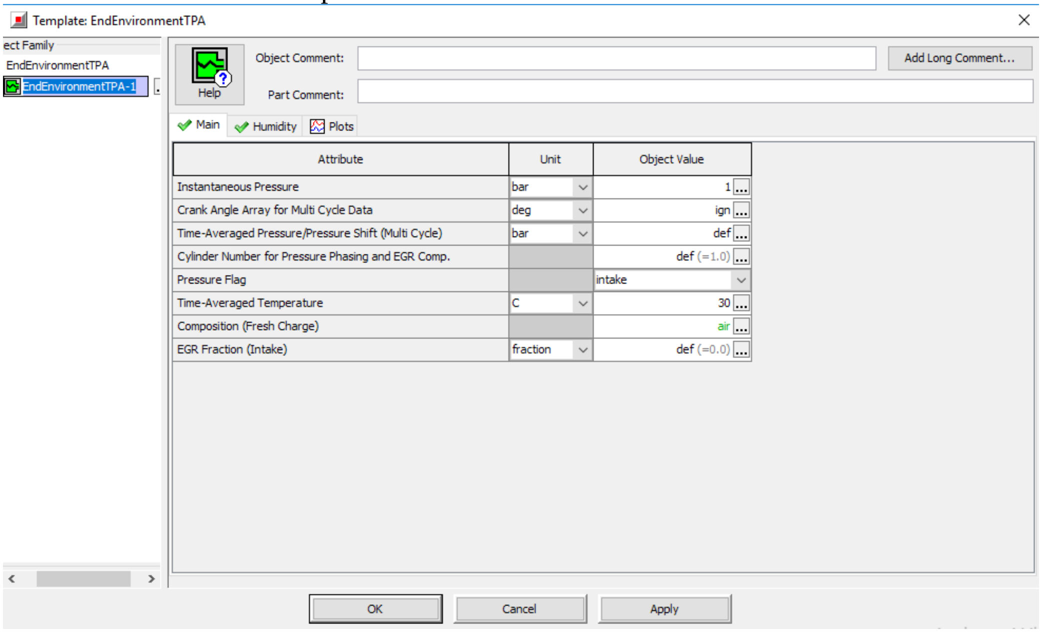
1. Entering Data and Creating Diesel Engine Simulation Model

System was created for diesel engine,

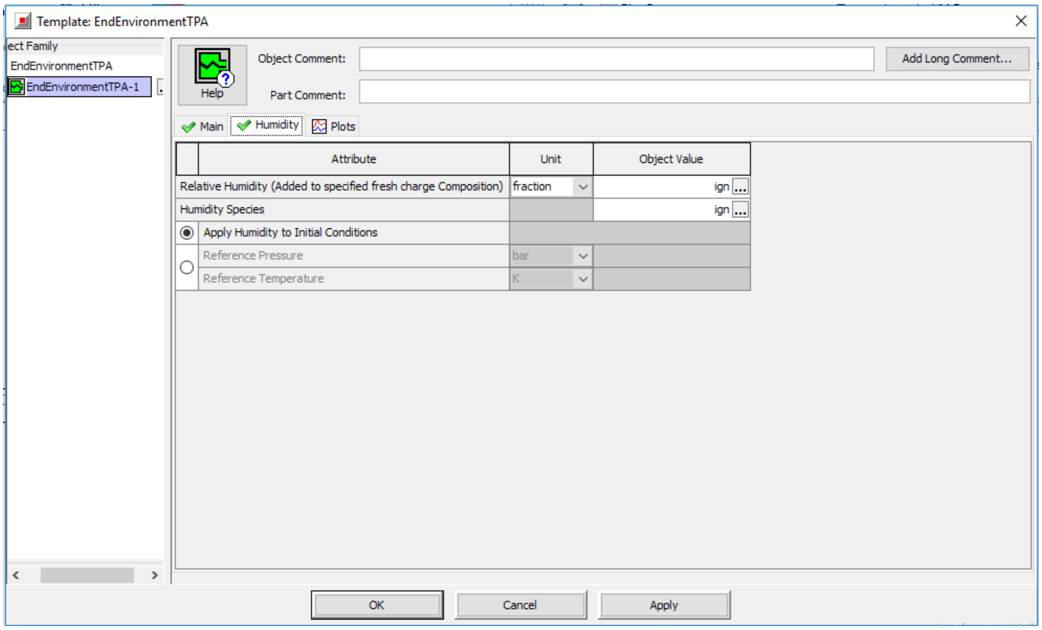
- Intake system
- Engine cylinder
- Fuel injection system
- Exhaust system

For the first step to create one cylinder diesel engine model is to determine environment condition with use EndEnvironmentTPA and fill with value that determined before as in the following figure:

- EndEnvironmentTPA template



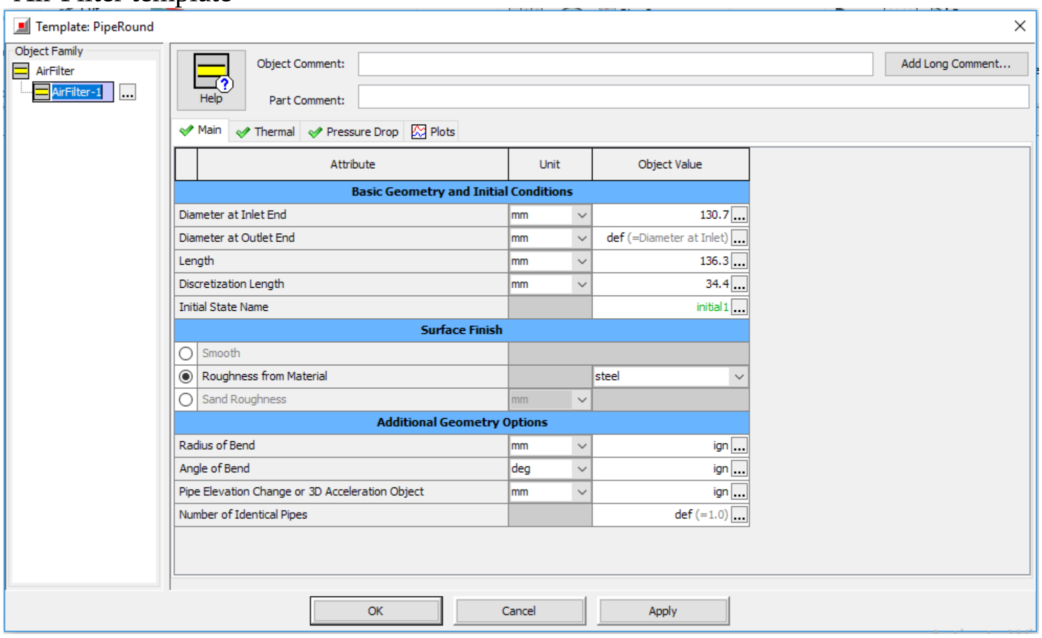
Main folder of EndEnvironmentTPA template



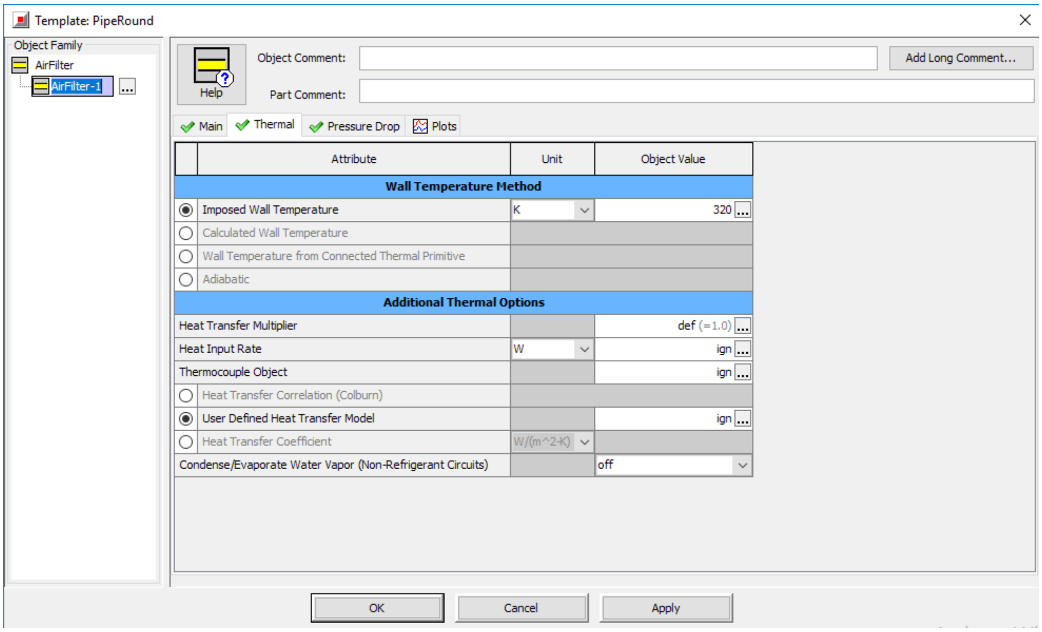
Humidity folder of EndEnvironmentTPA template

Then create an air filter component to connect the End environment component to intake runner component.

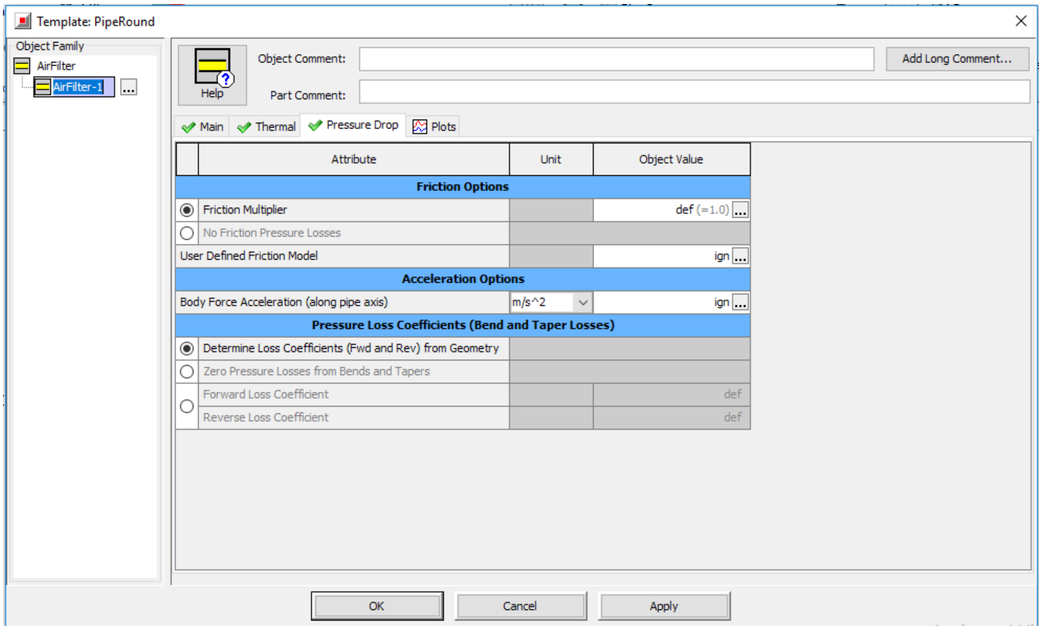
- Air Filter template



Main folder of air filter template

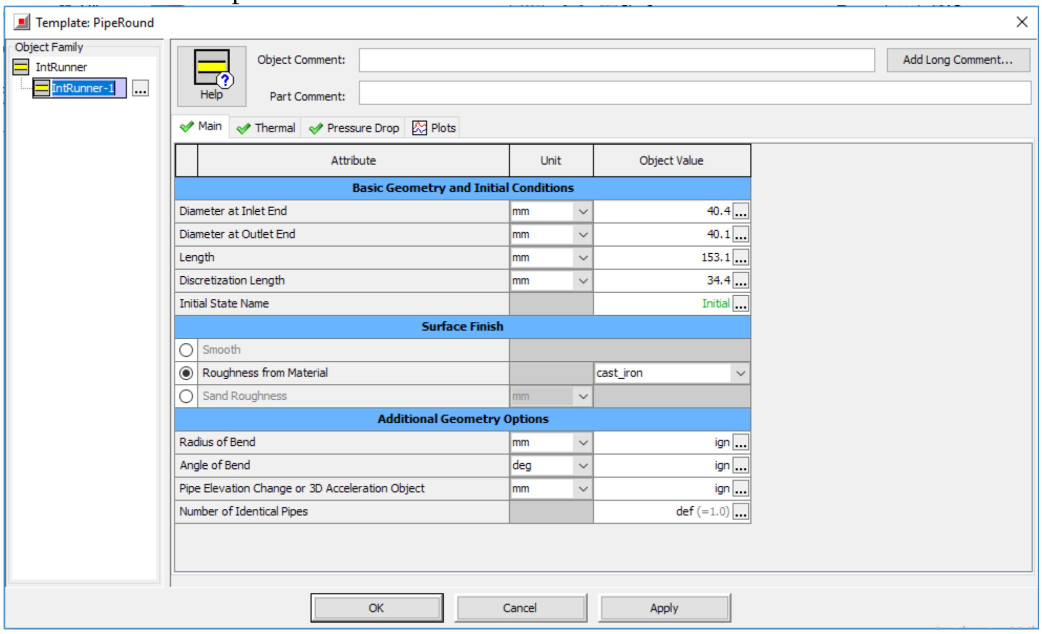


Thermal folder of air filter template

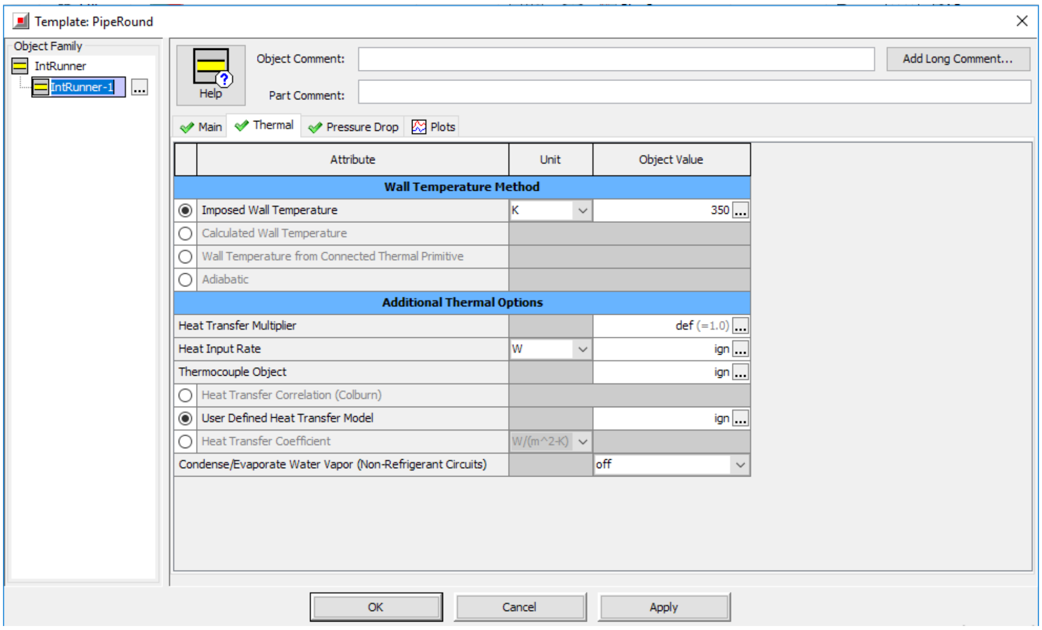


Pressure drop folder of air filter template

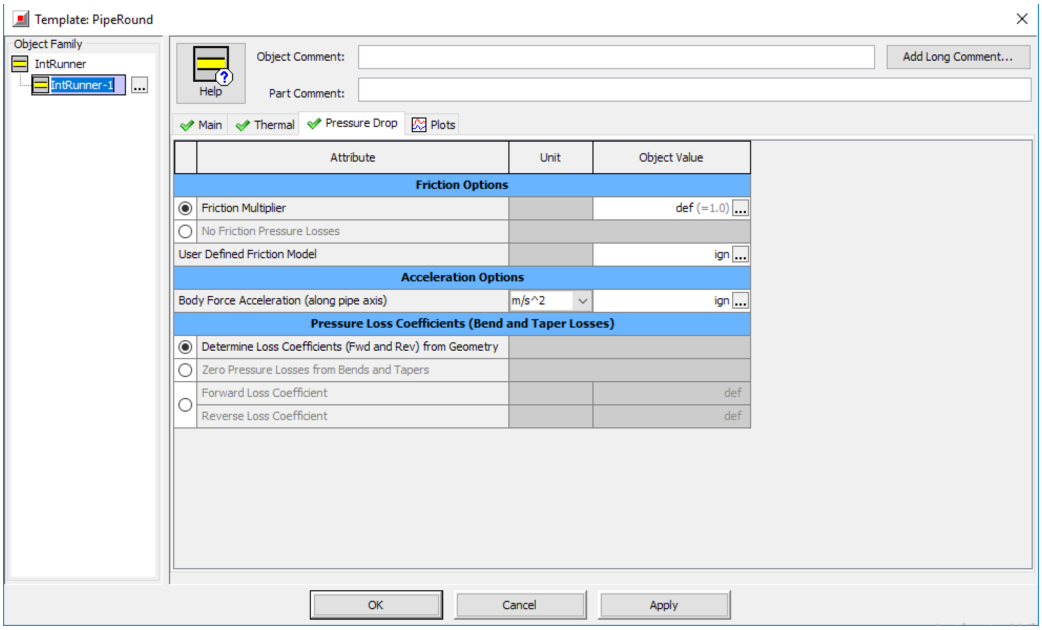
- Intake Runner template



Main folder of intake runner template

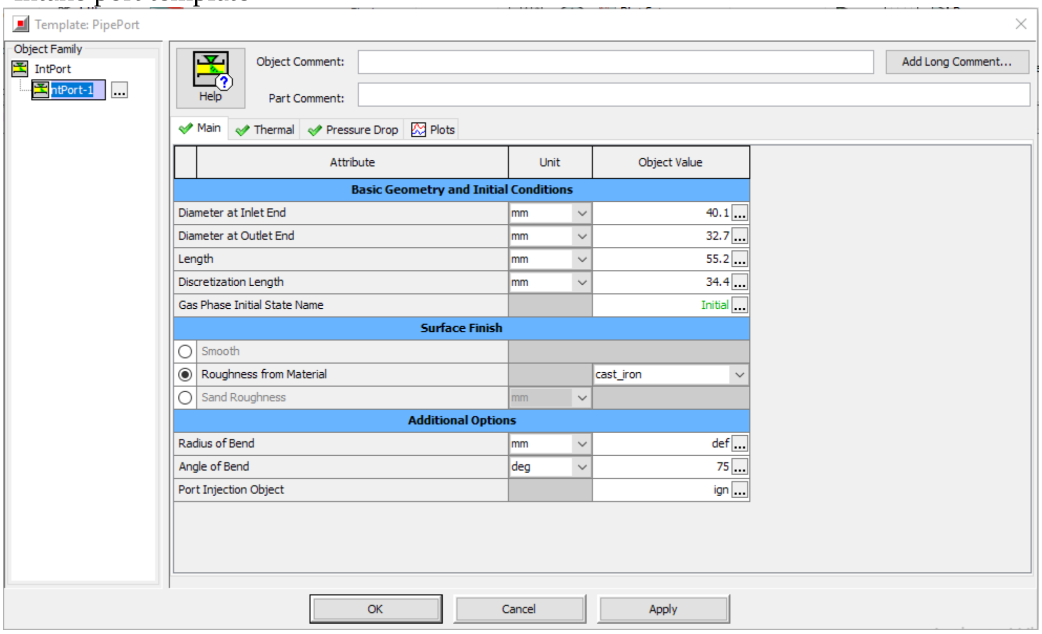


Thermal folder of intake runner template

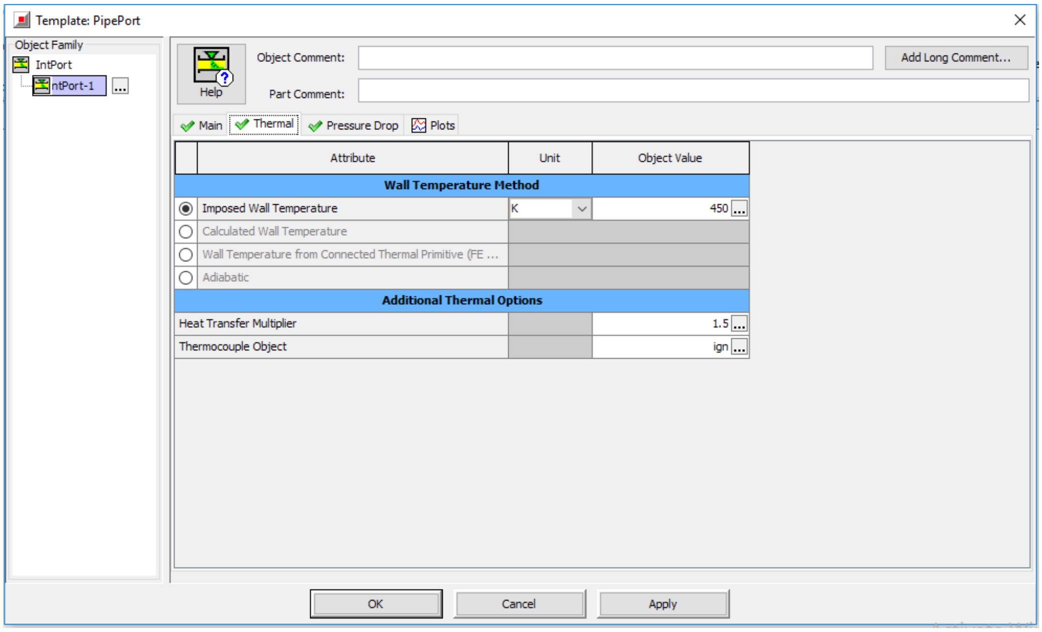


Pressure drop folder of intake runner template

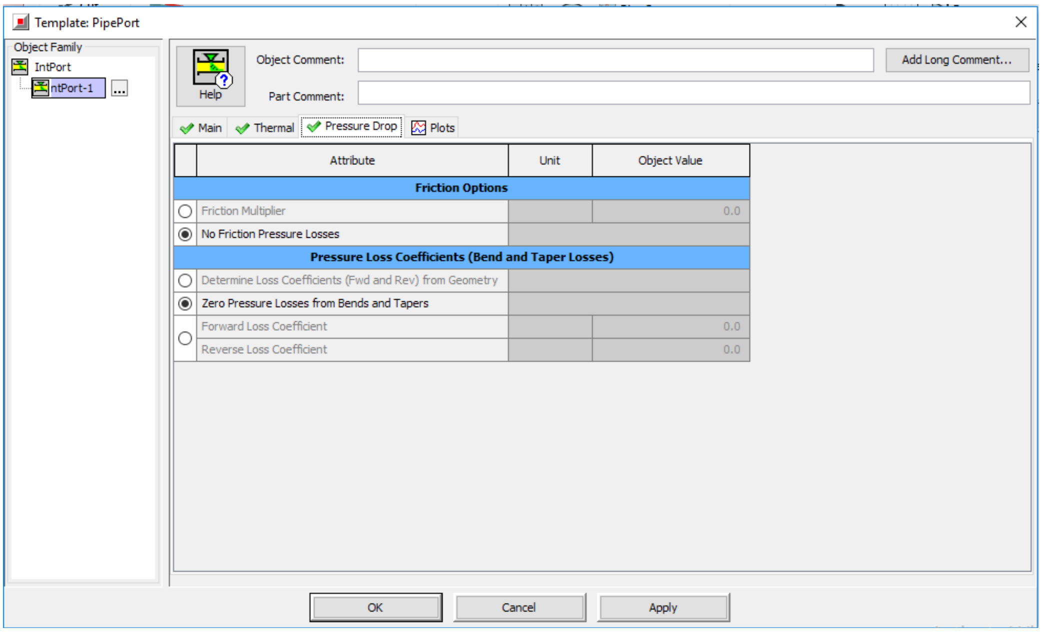
- Intake port template



Main folder of intake port template

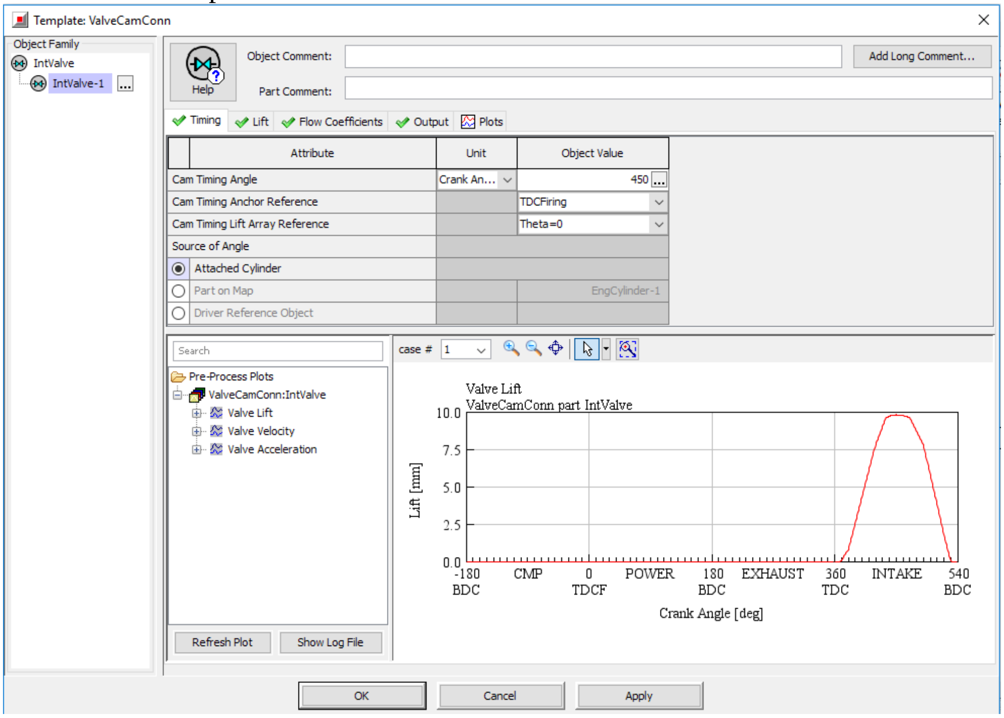


Thermal folder of intake port template

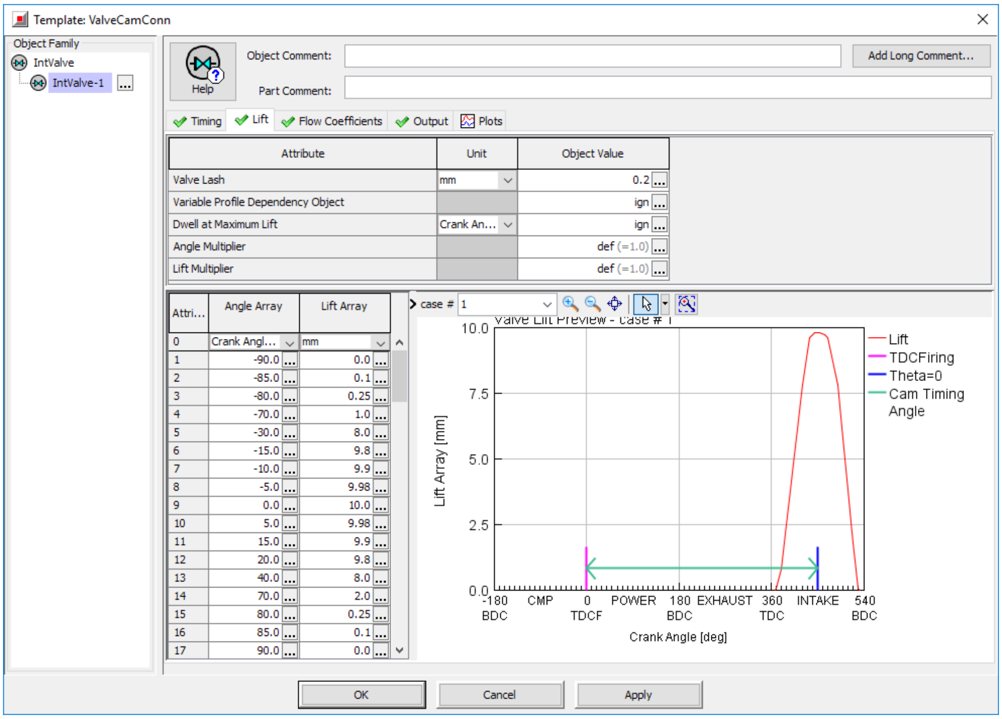


Pressure drop folder of intake port template

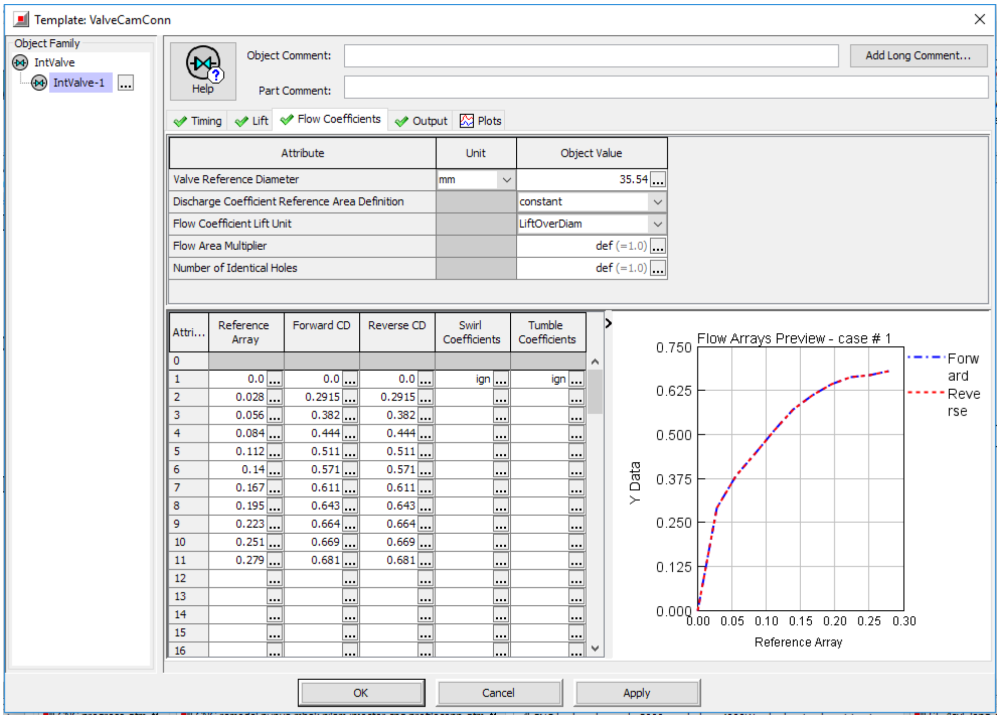
- Intake valve template



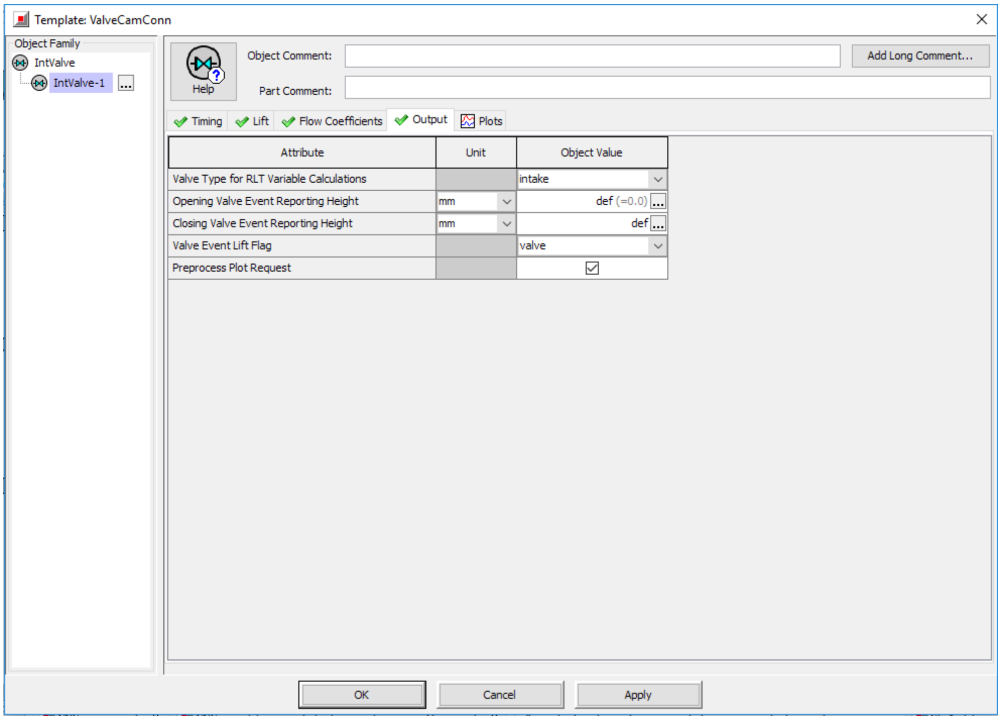
Timing folder of intake valve template



Lift folder of intake valve template



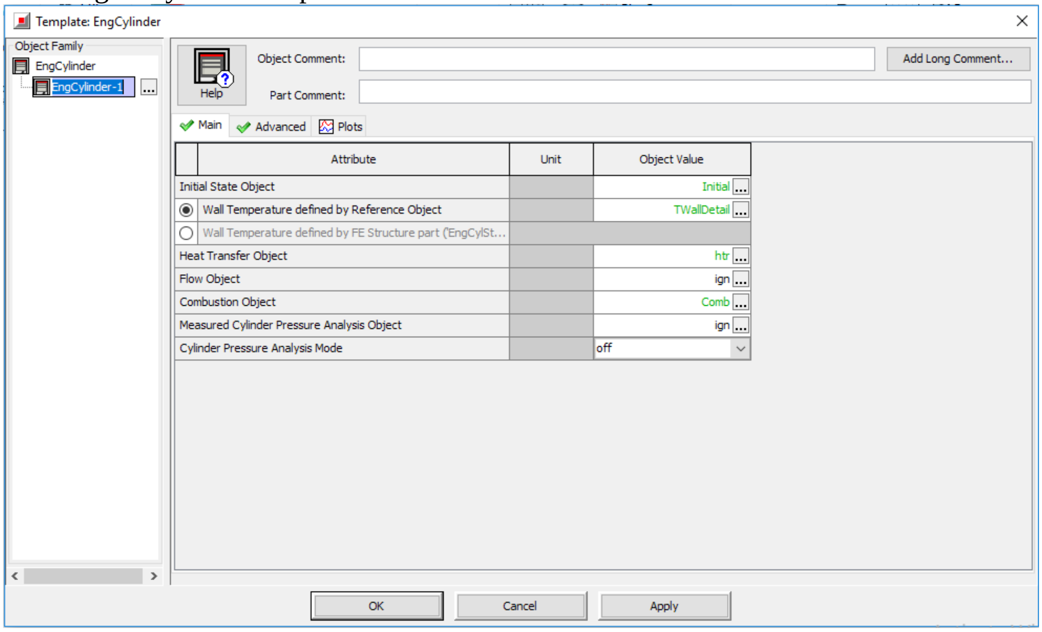
Flow coefficients folder of intake valve template



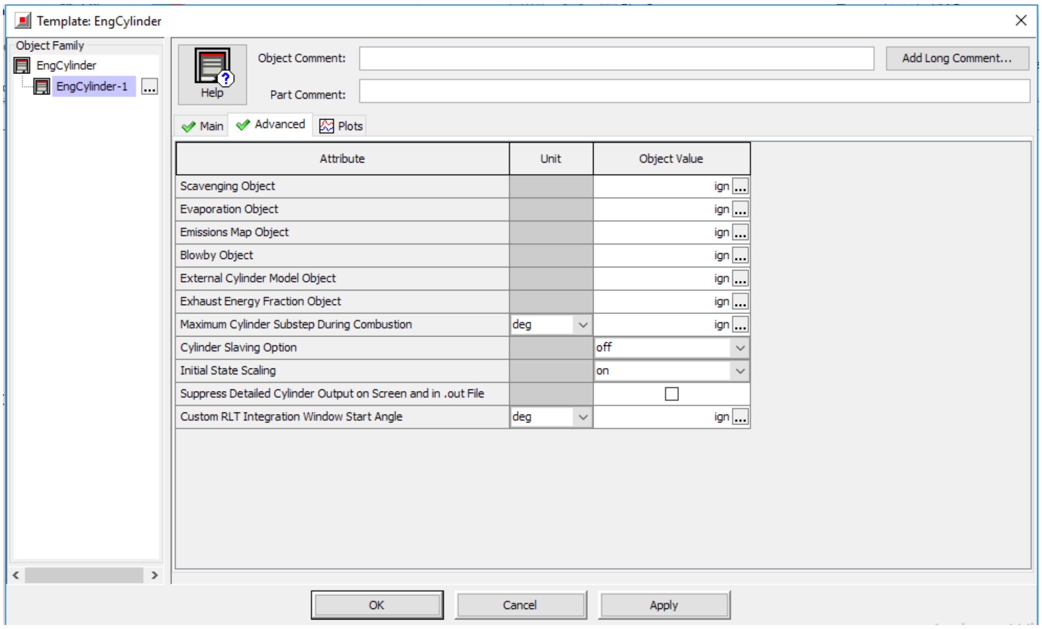
Output folder of intake valve template

Then create the engine cylinder component.

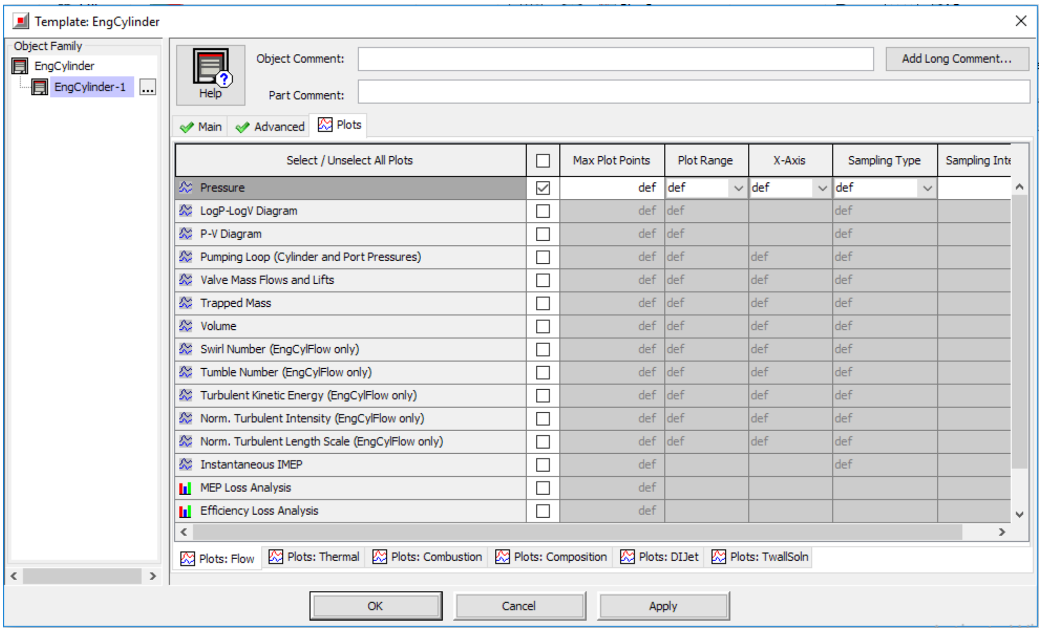
- Engine cylinder template



Main folder of engine cylinder template

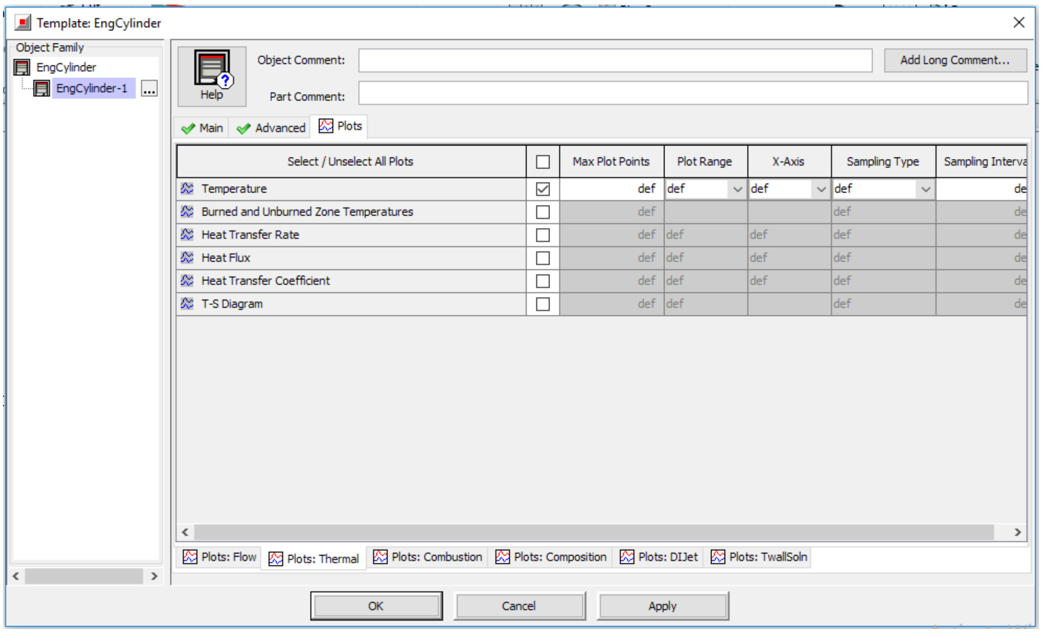


Advanced folder of engine cylinder template

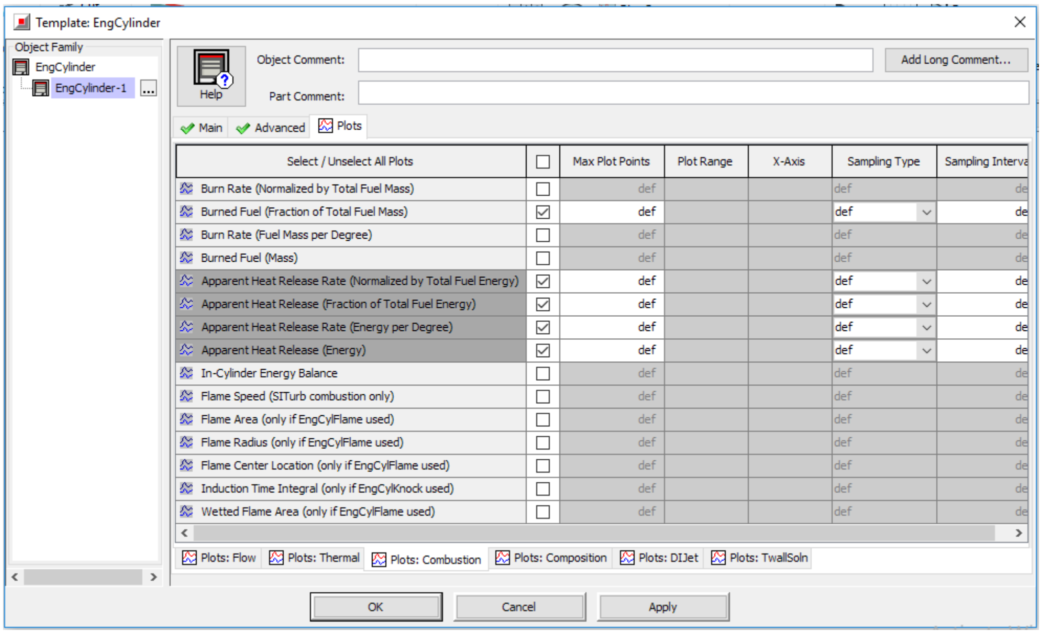


Plot: flow folder of engine cylinder template

In the plot: flow folder select the pressure list. The selection list use for to know the result of that selection.

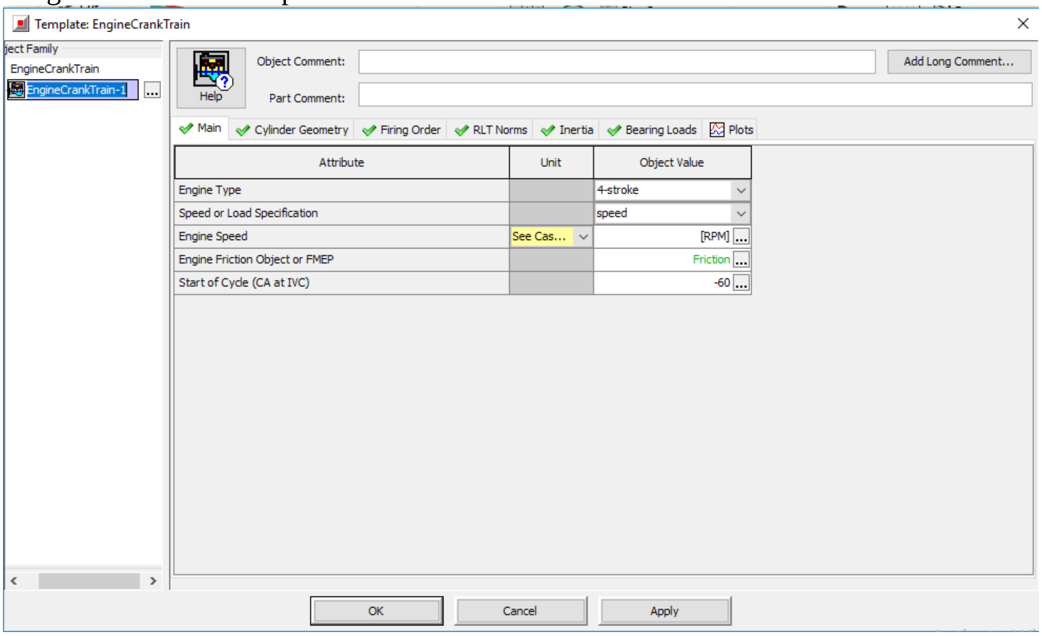


Plot: thermal folder of engine cylinder template

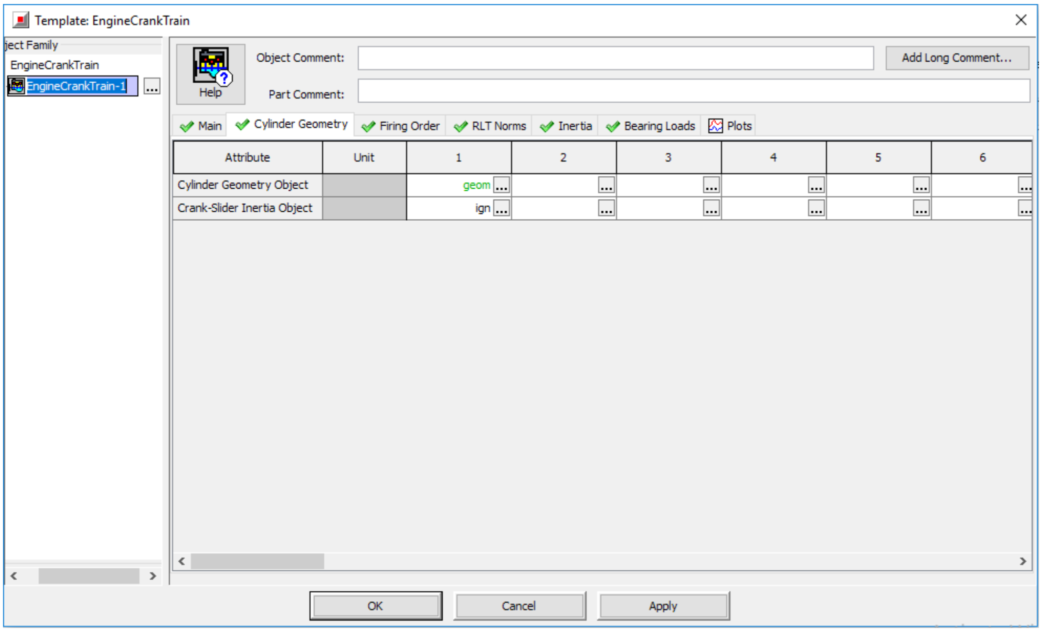


Plot: Combustion folder of engine cylinder template

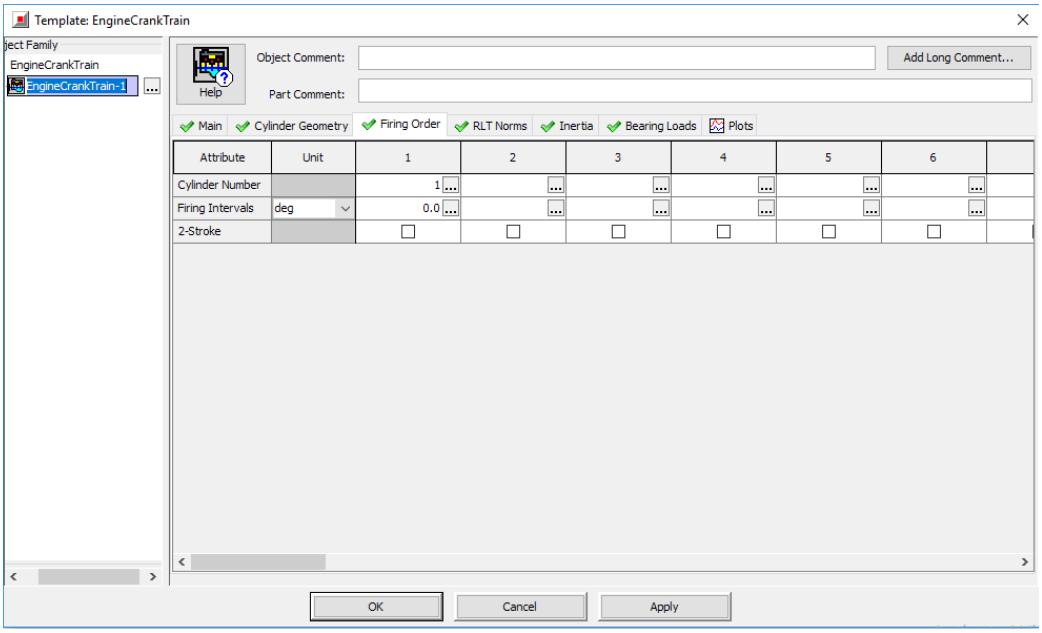
- Engine crank train template



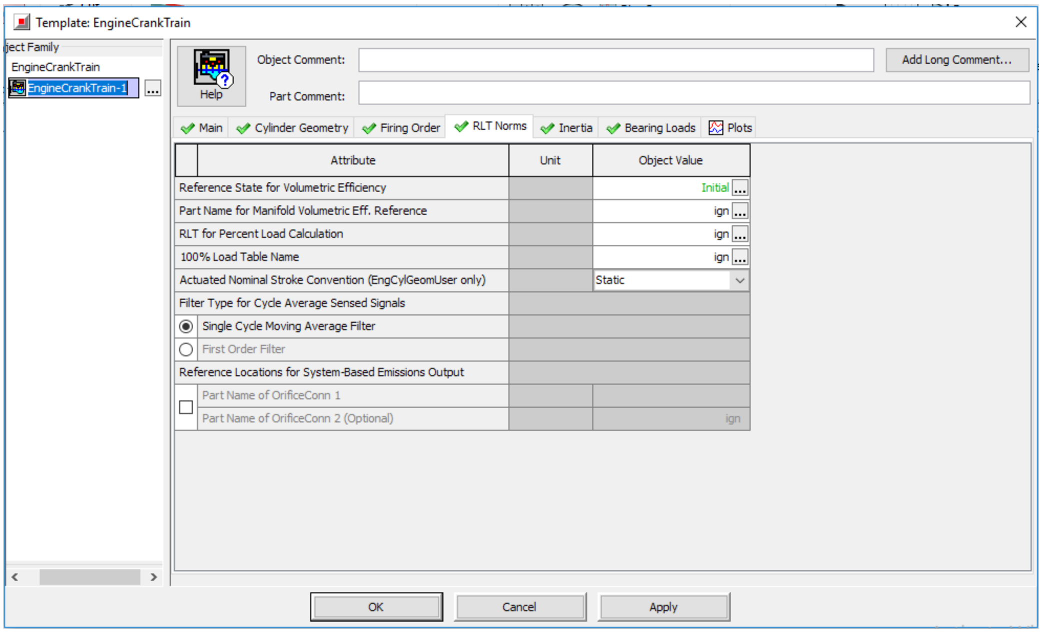
Main folder of engine crank train template



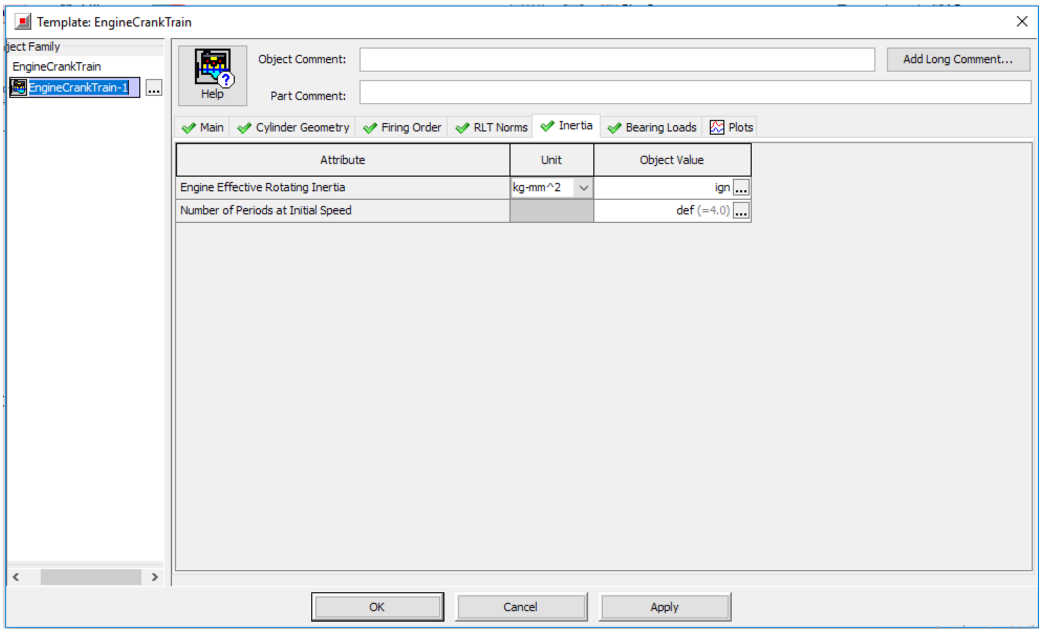
Cylinder geometry folder of engine crank train template



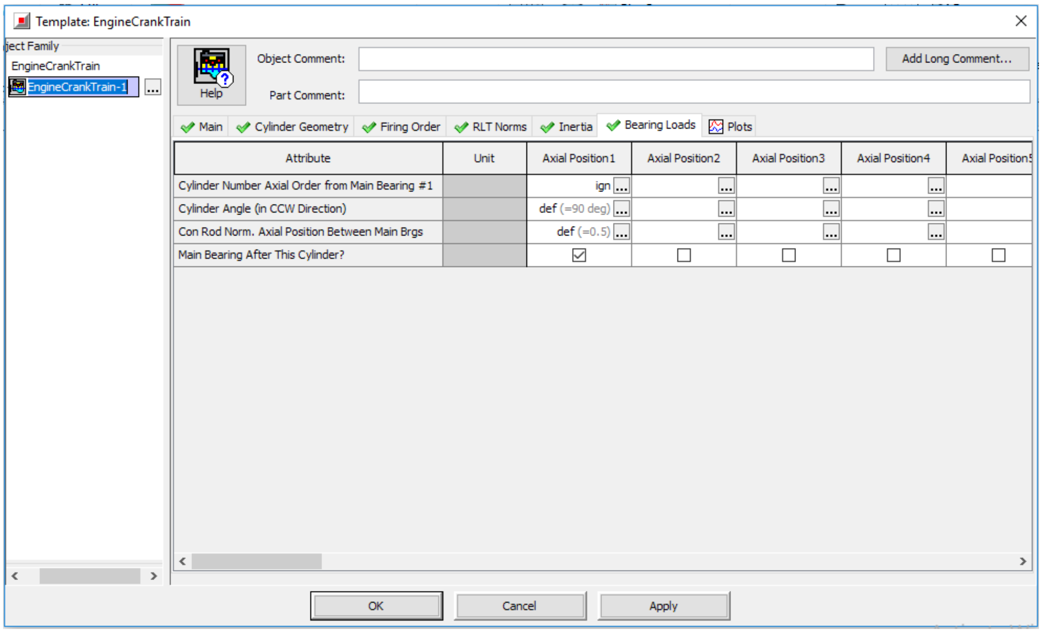
Firing order folder of engine crank train template



RLT norms folder of engine crank train template



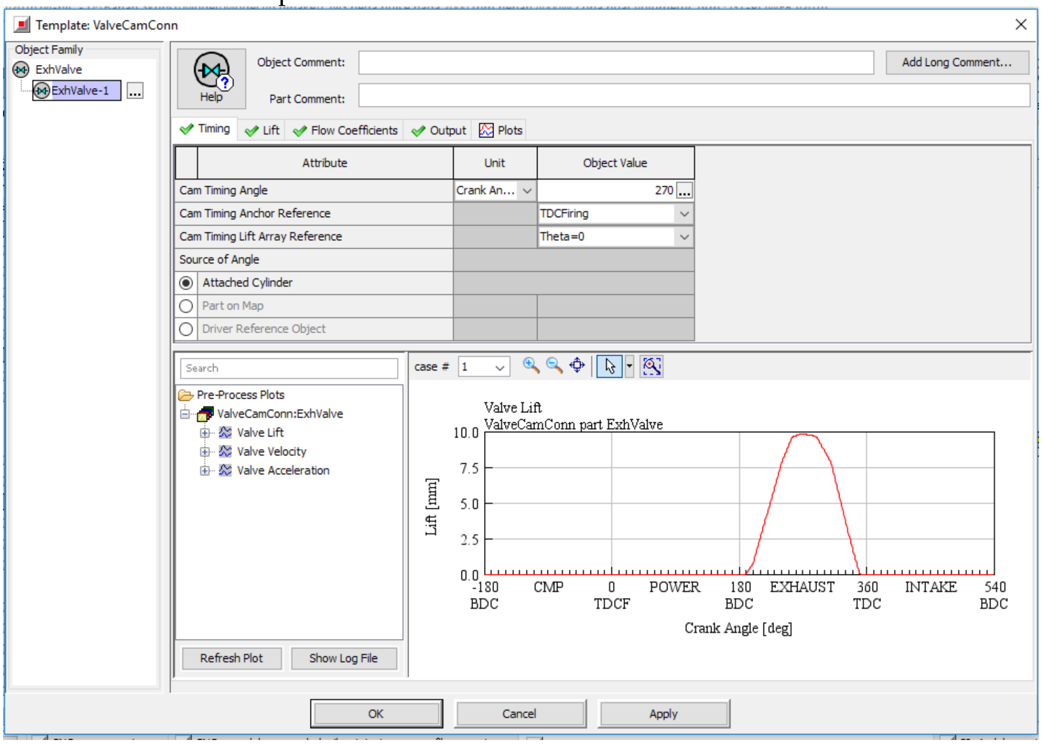
Inertia folder of engine crank train template



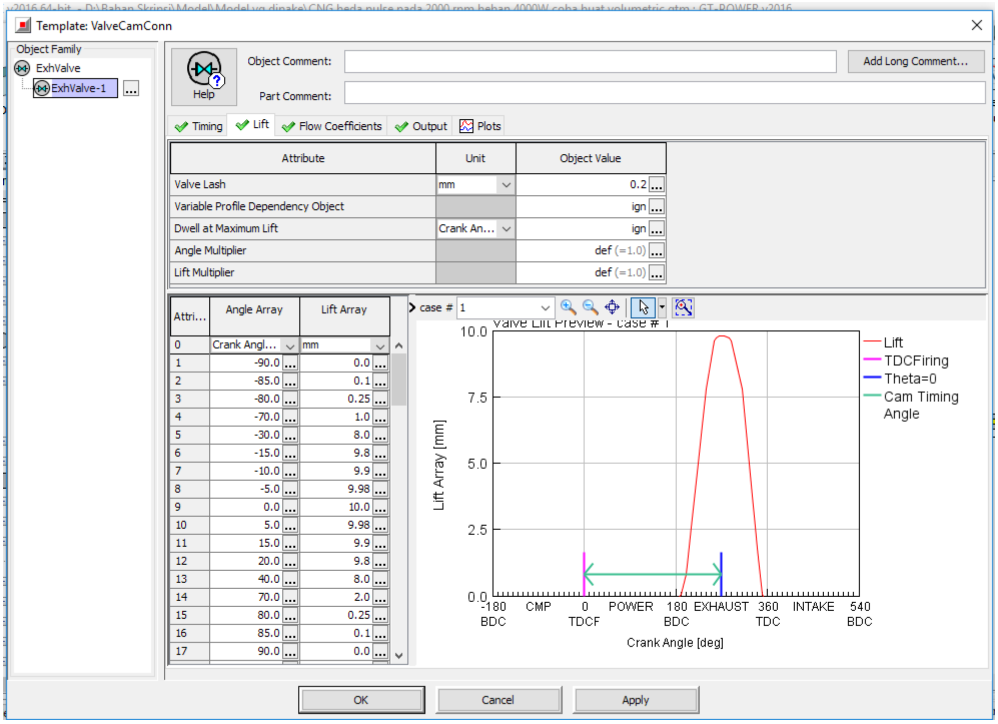
Bearing loads folder of engine crank train template

Then to create the components for exhaust system, that are exhaust valve, exhaust port, muffler, exhaust runner, and outlet end environment.

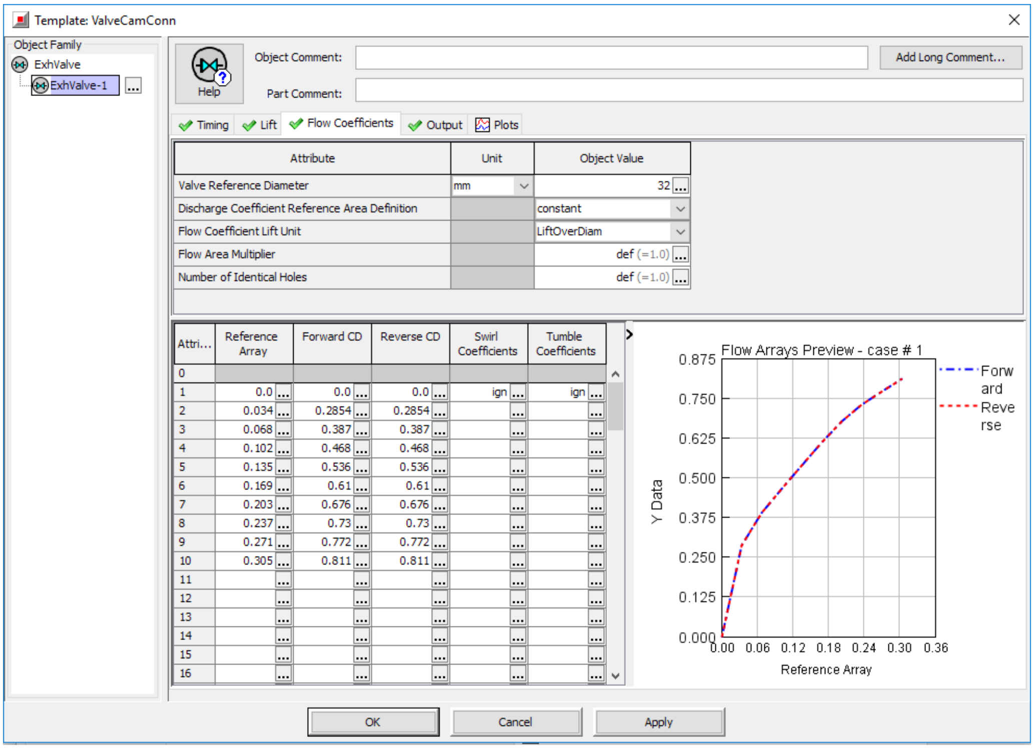
- Exhaust valve template



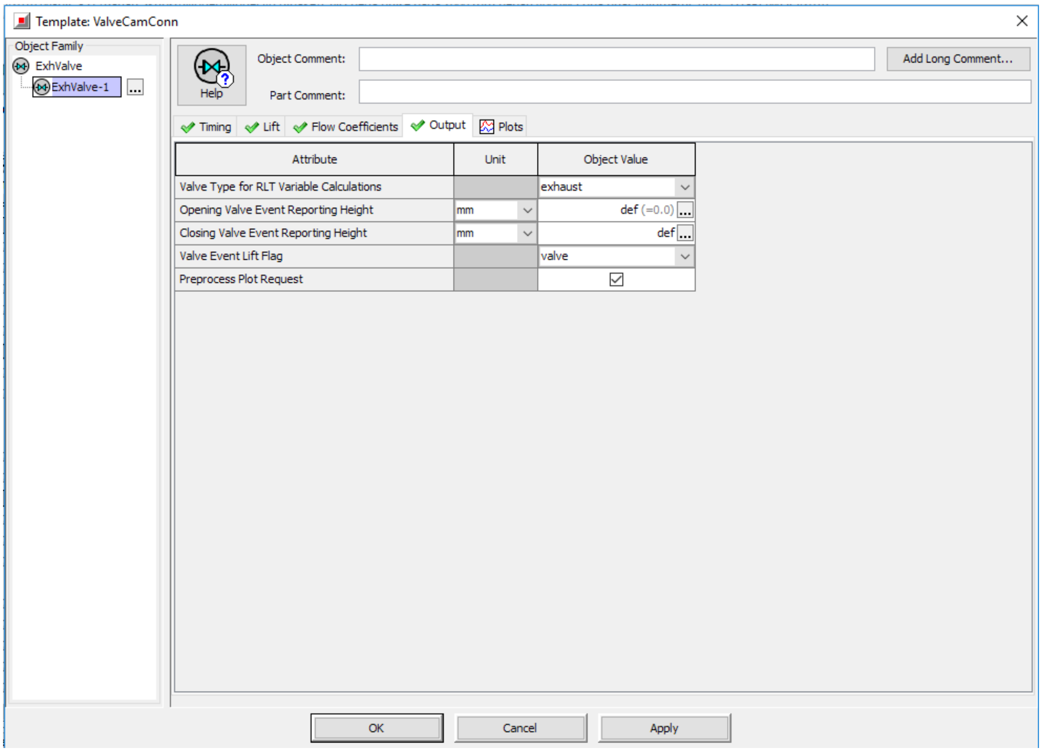
Timing folder of exhaust valve template



Lift folder of exhaust valve template



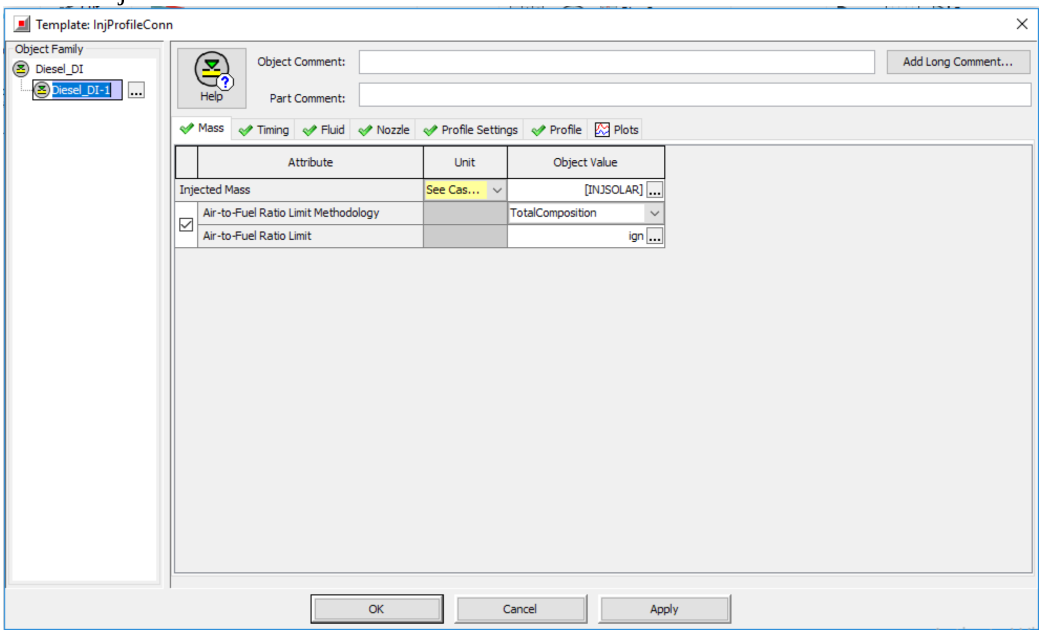
Flow coefficient folder of exhaust valve template



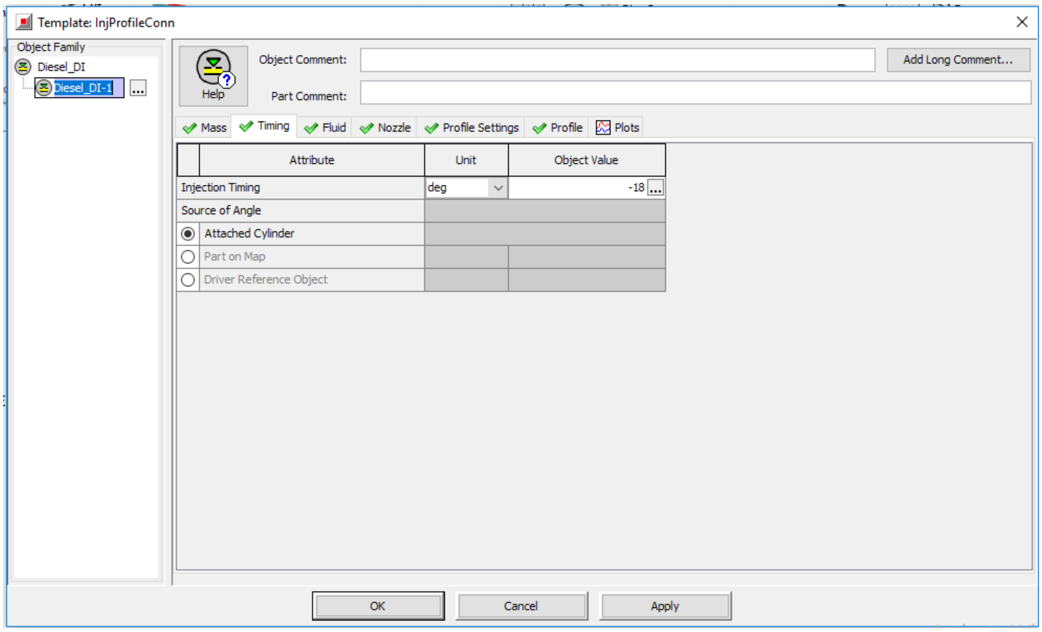
Output folder of exhaust valve template

Then create the diesel injector which is connected to the engine cylinder.

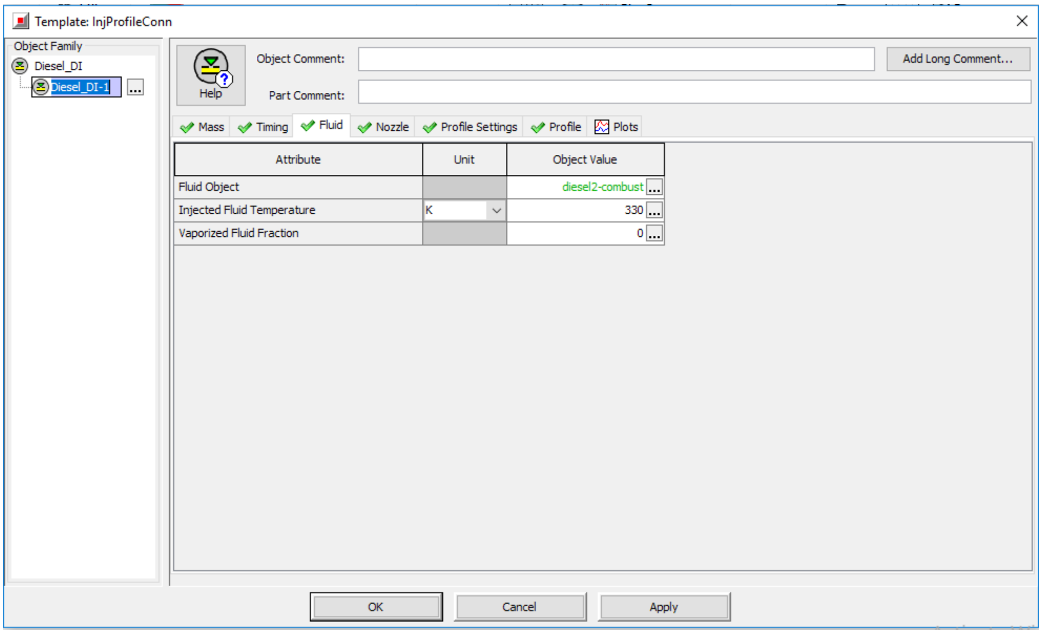
- Diesel injector



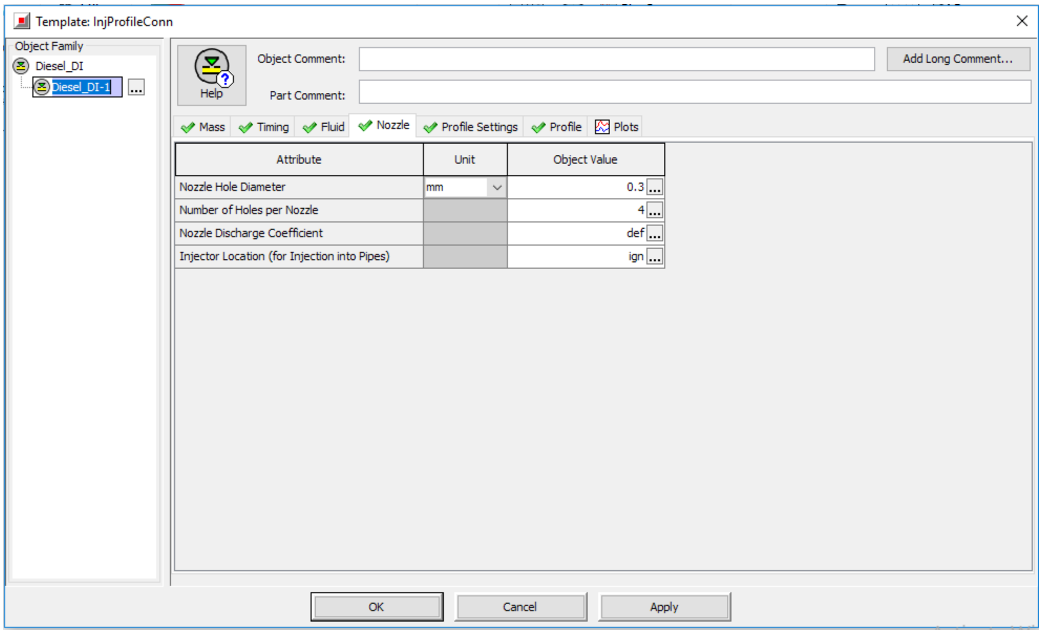
Main folder of diesel injector template



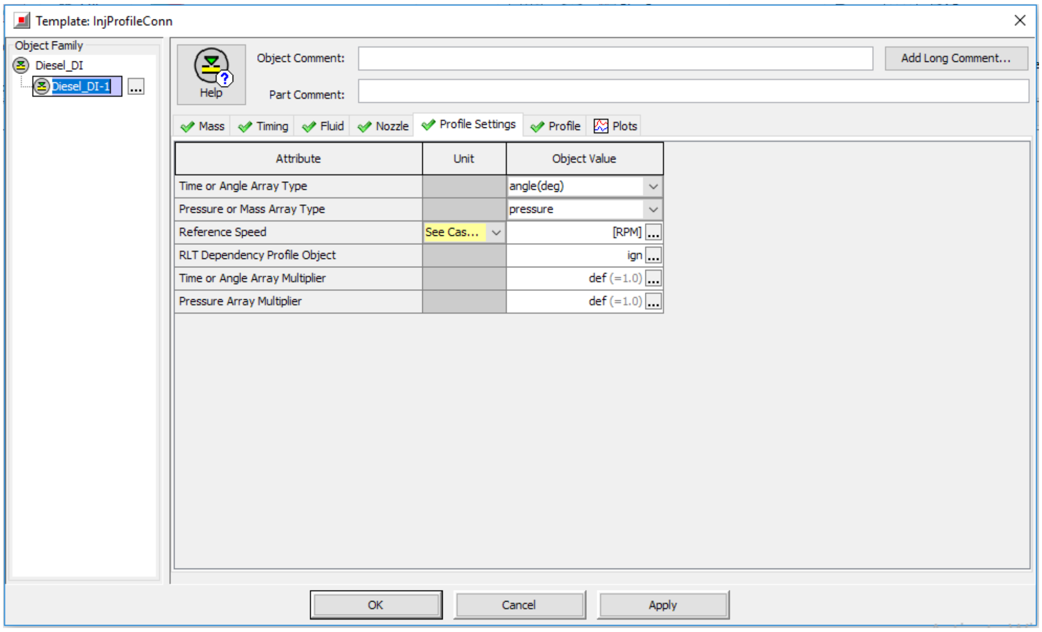
Timing folder of diesel injector template



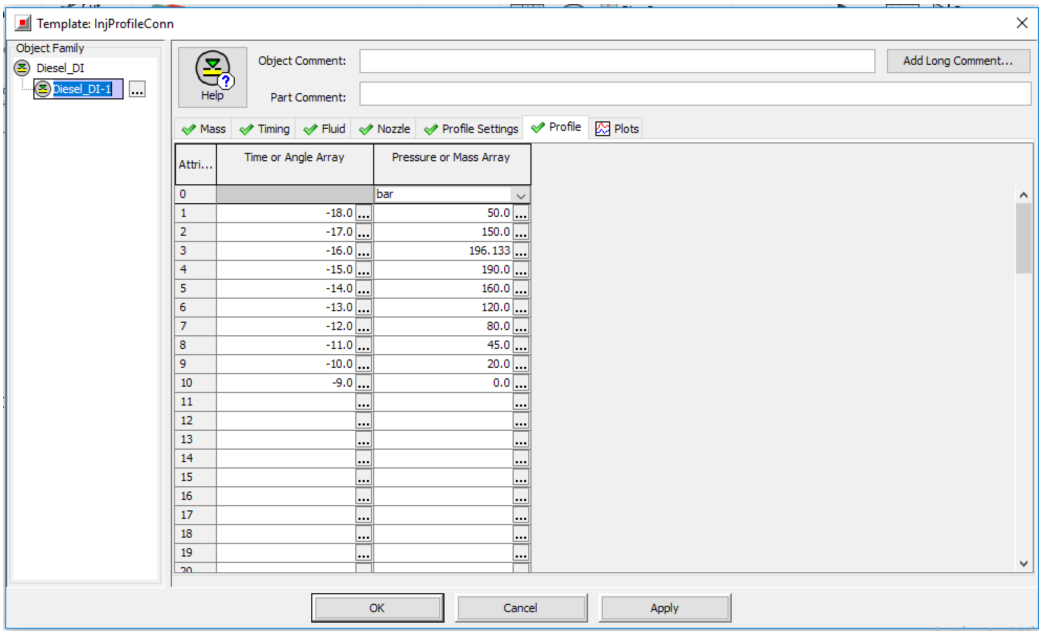
Fluid folder of diesel injector template



Nozzle folder of diesel injector template

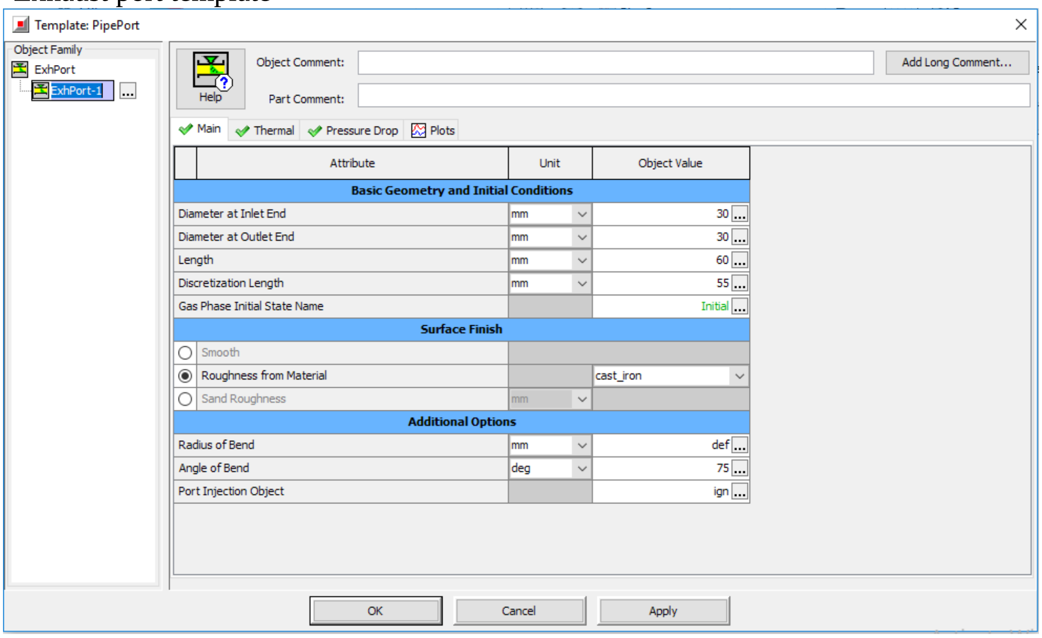


Profile settings folder of diesel injector template

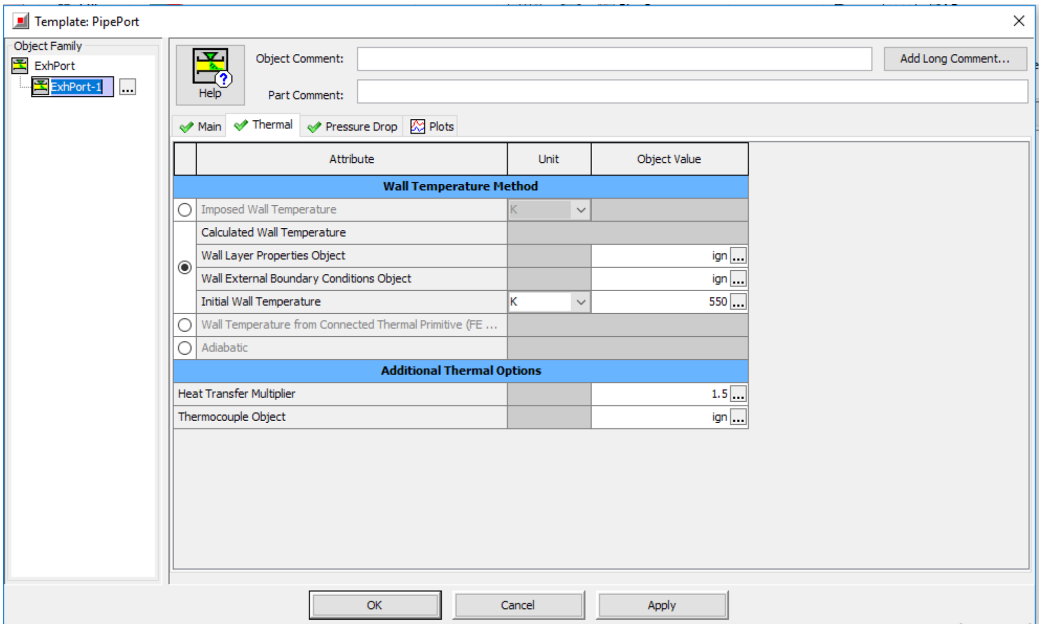


Profile folder of diesel injector template

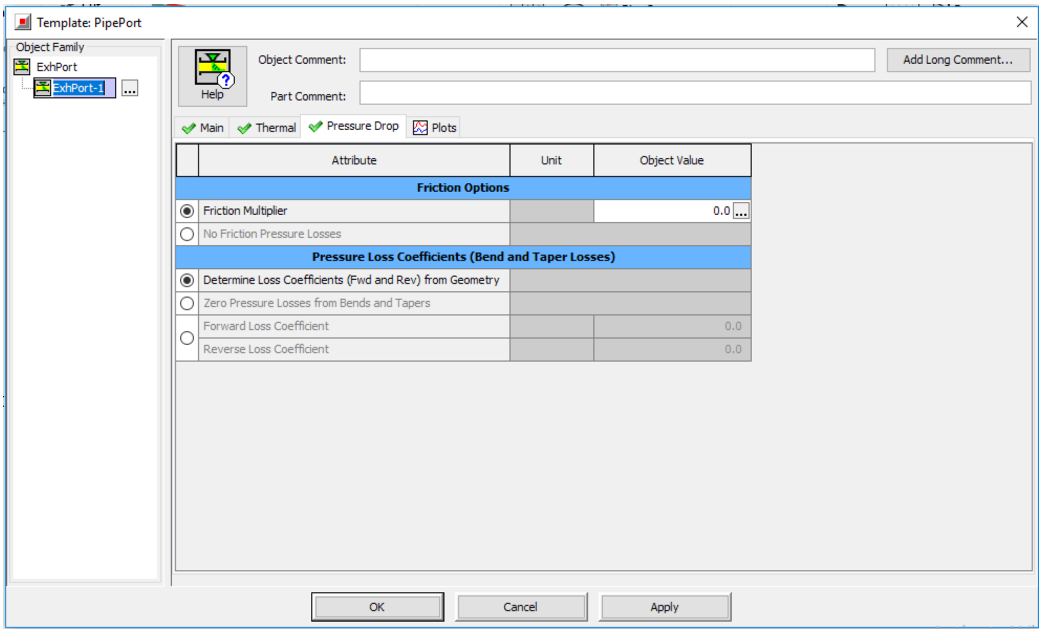
- Exhaust port template



Main folder of exhaust port template

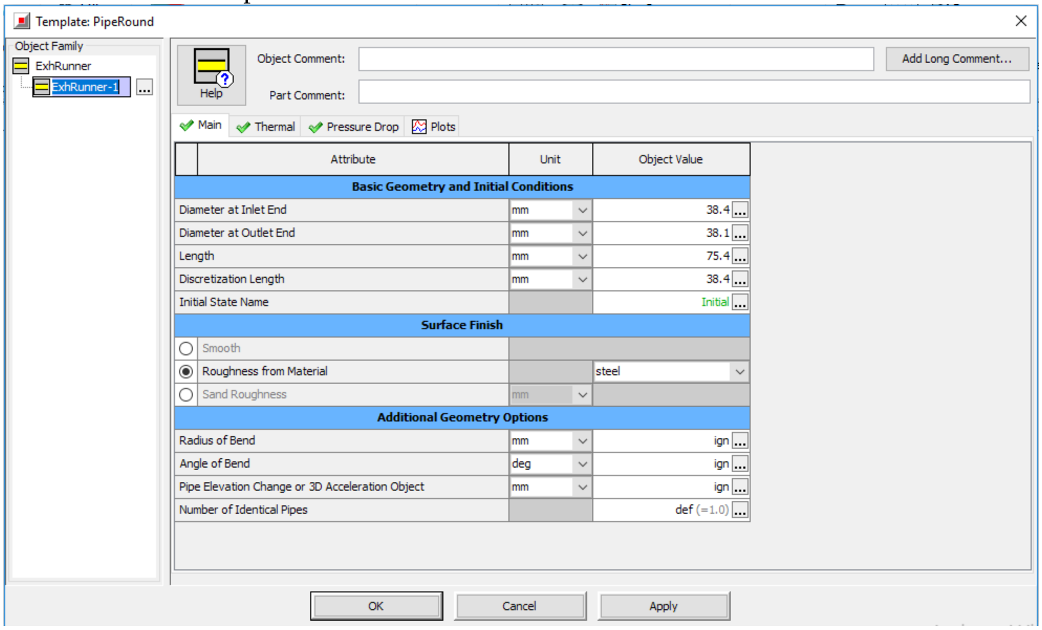


Thermal folder of exhaust port template

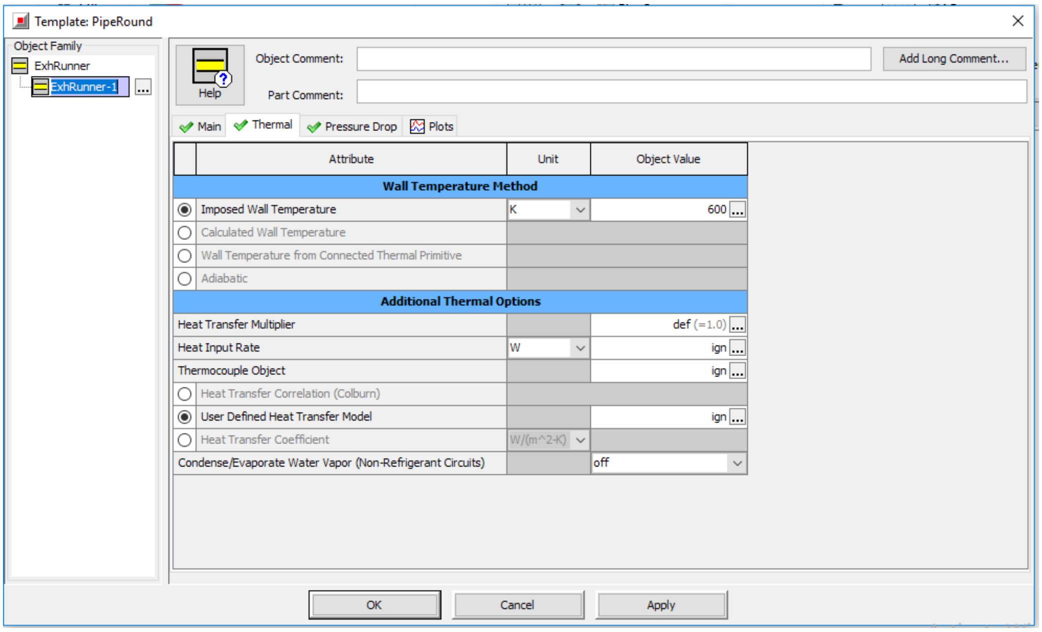


Pressure drop folder of exhaust port template

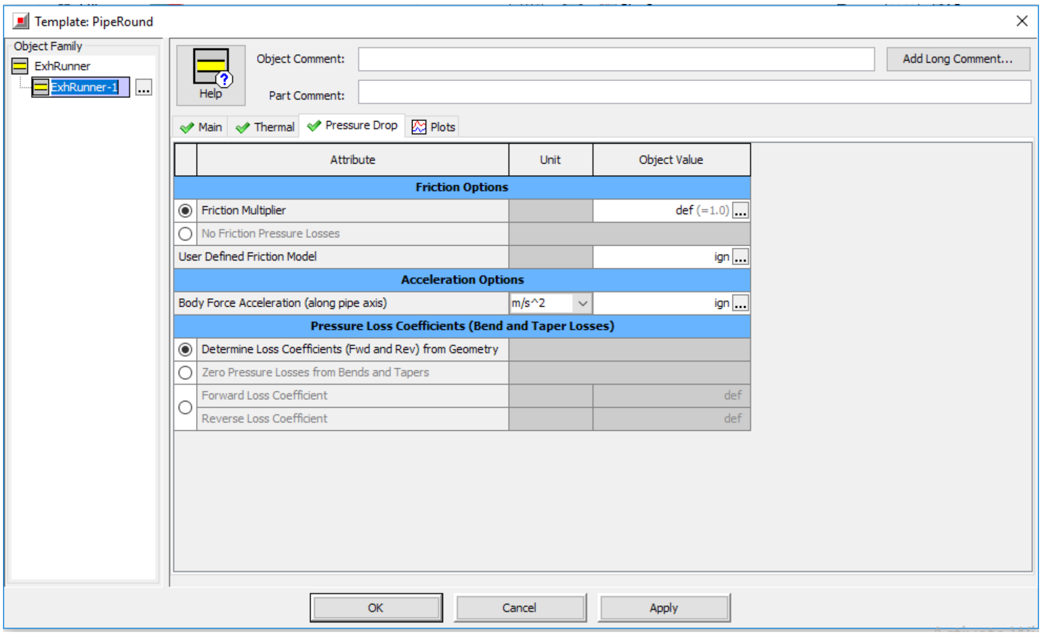
- Exhaust runner template



Main folder of exhaust runner template

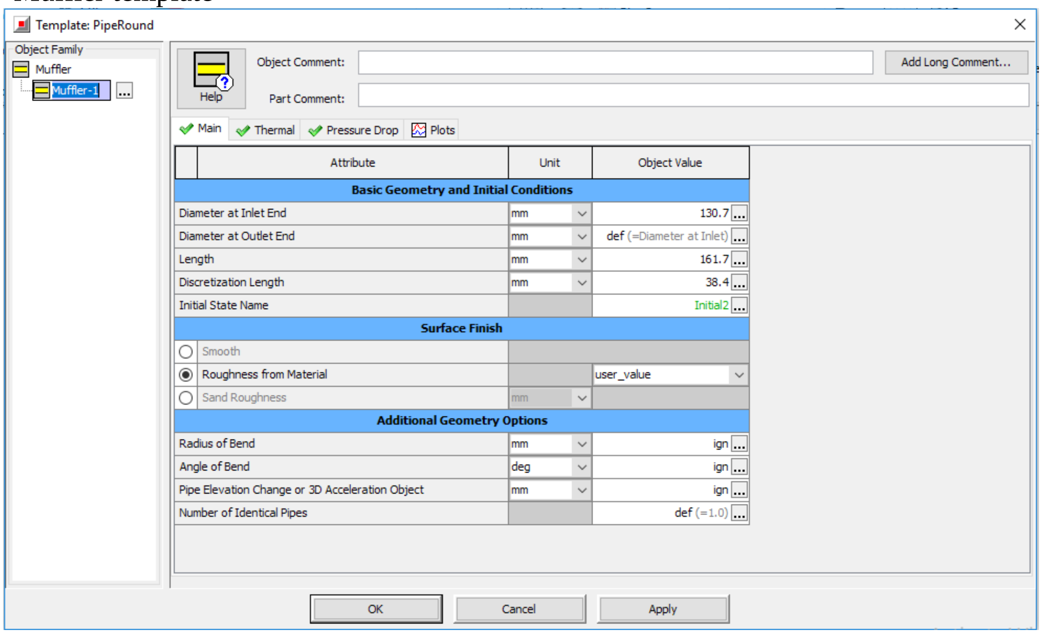


Thermal folder of exhaust runner template

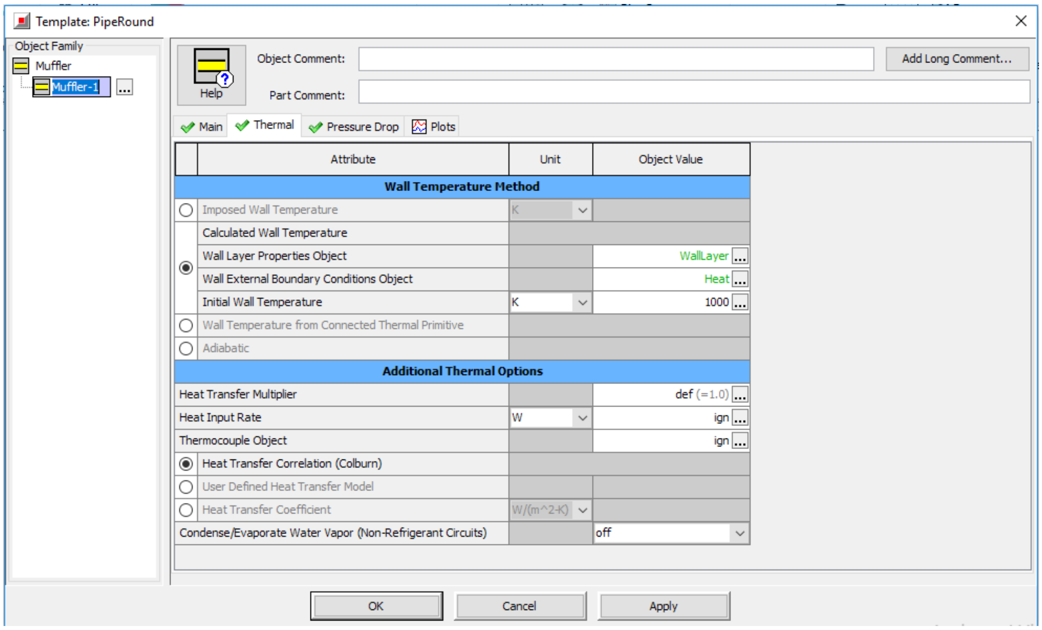


Pressure drop folder of exhaust runner template

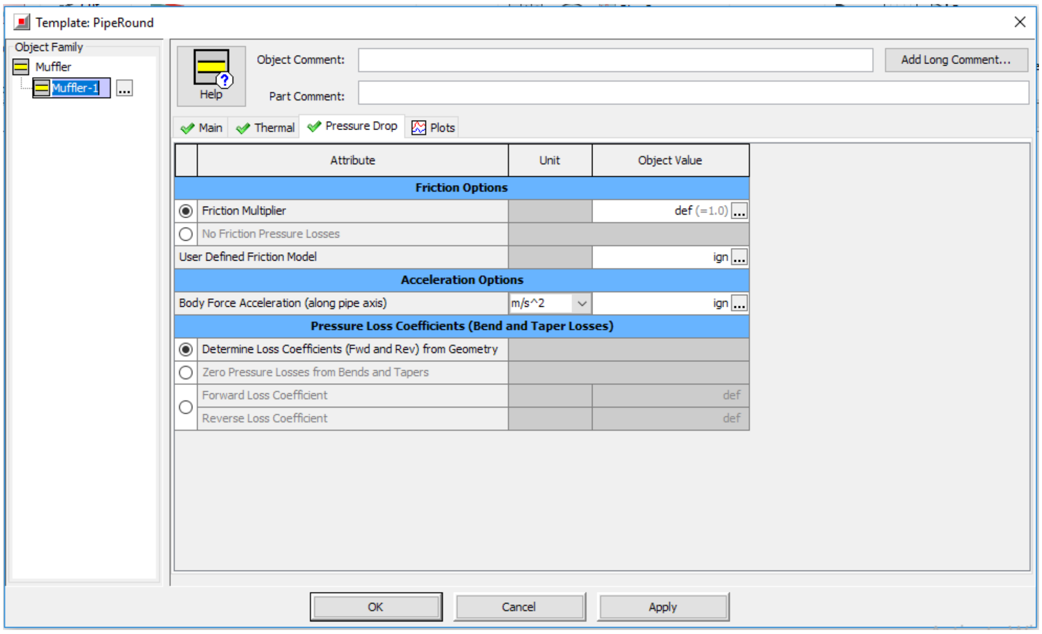
- Muffler template



Main folder of muffler template

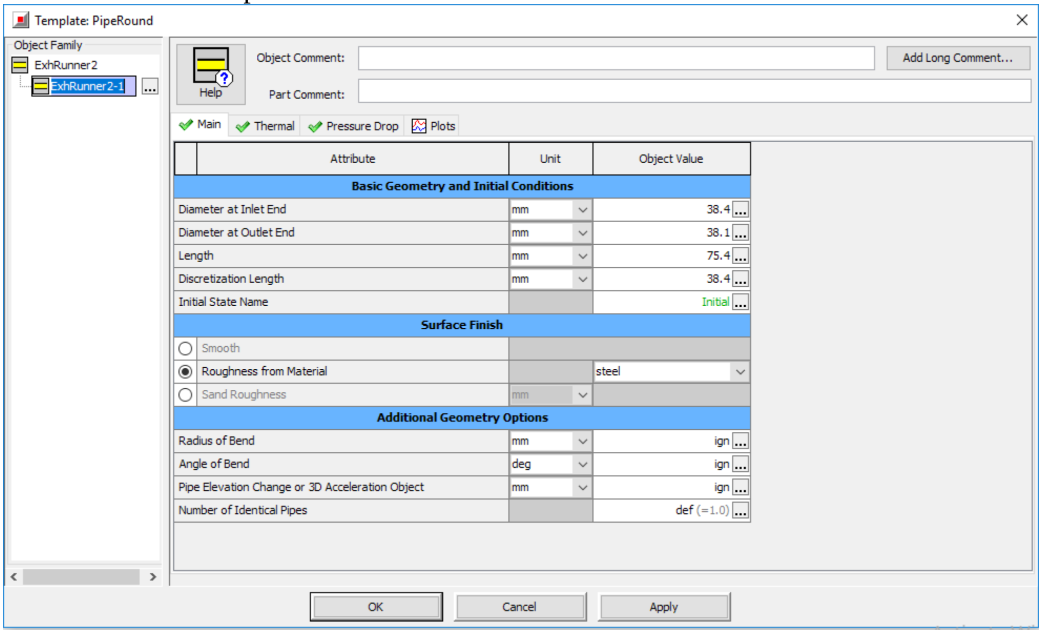


Thermal folder of muffler template



Pressure drop folder of muffler template

- Exhaust runner template



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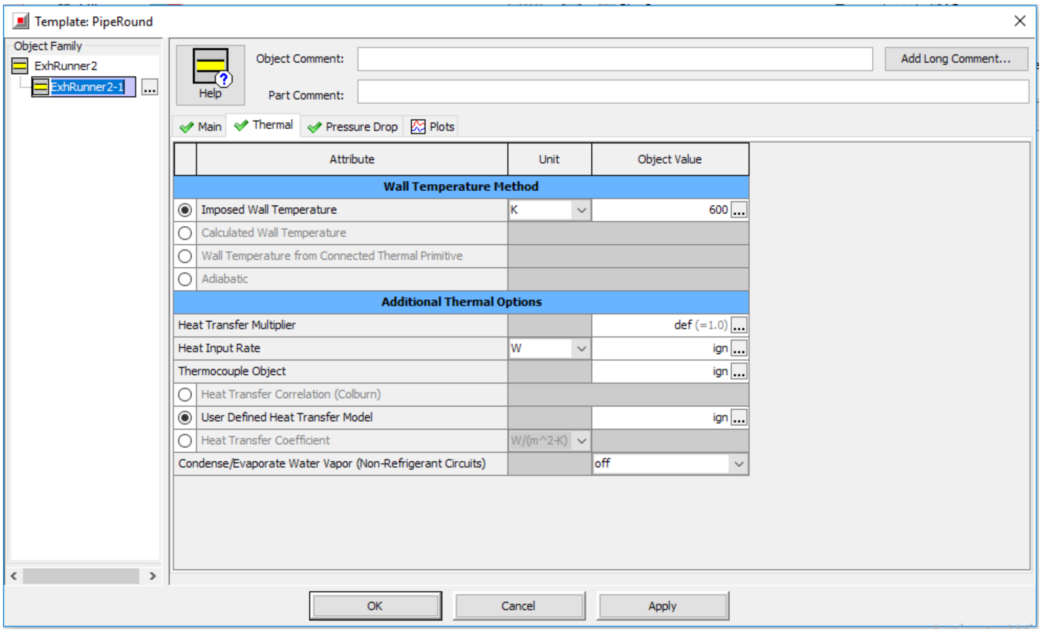

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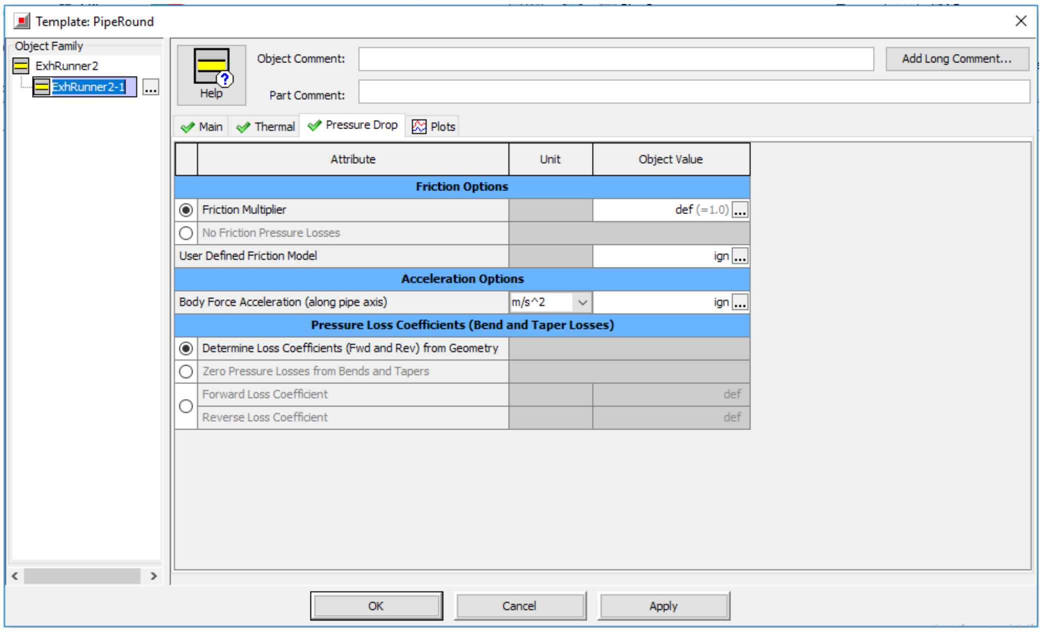
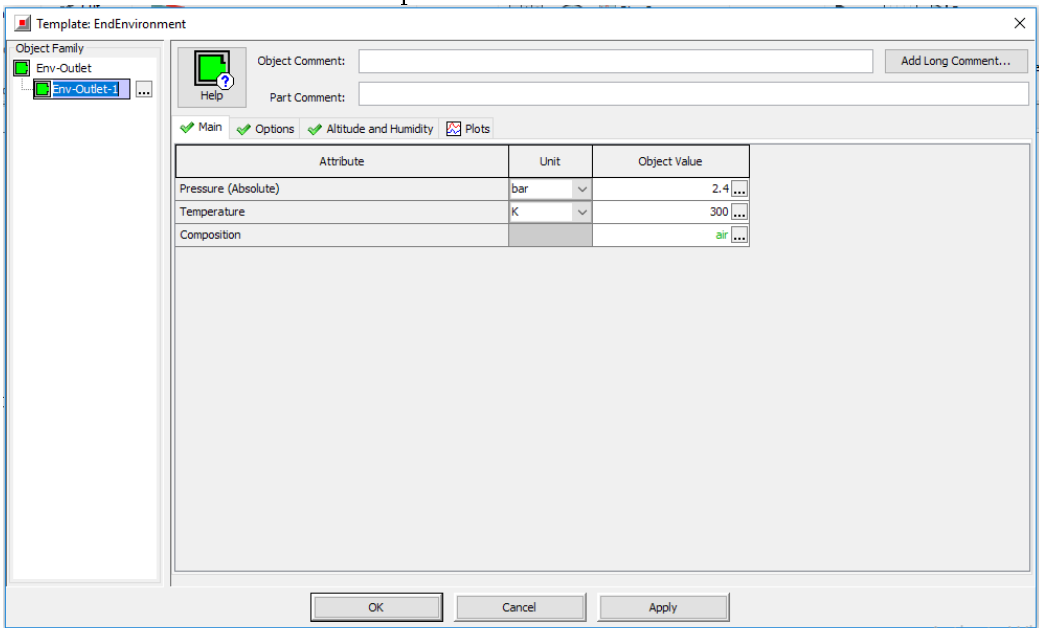
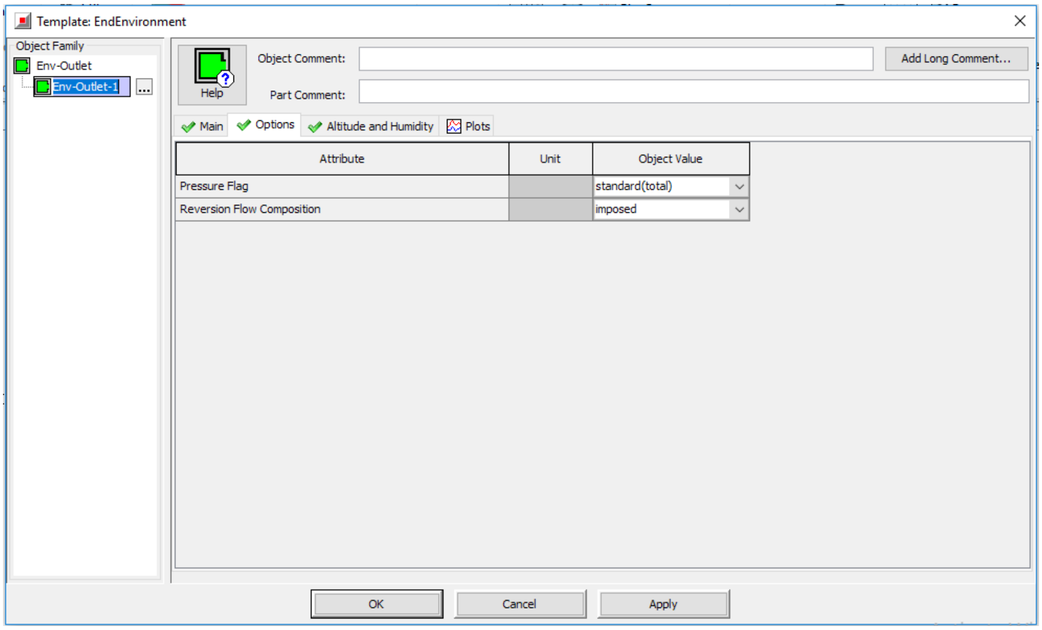


Pressure drop folder of exhaust runner template

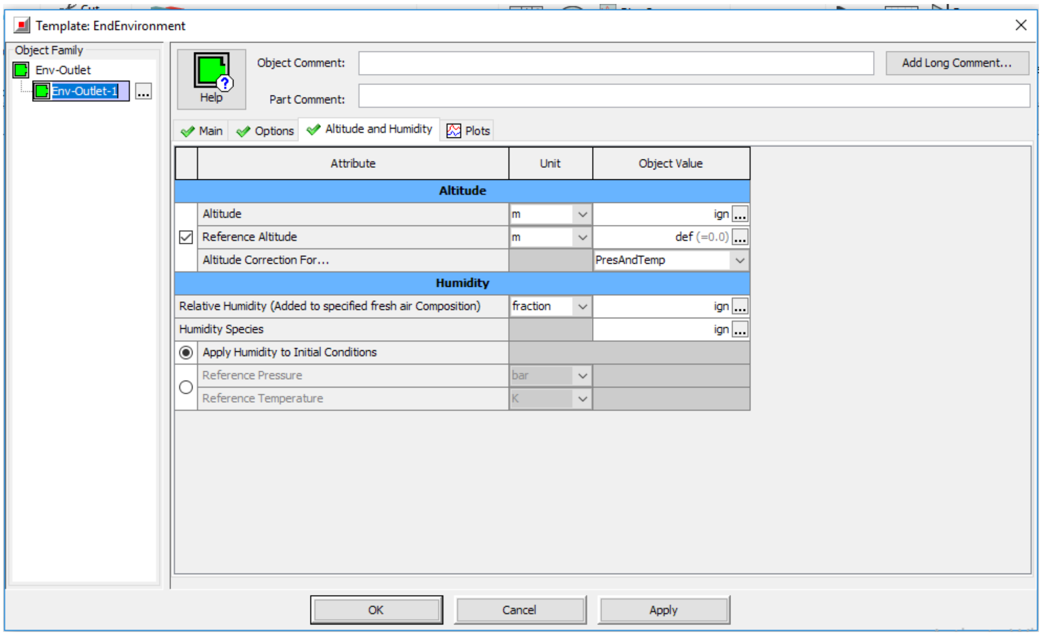
- Outlet End Environment template



Main folder of outlet end environment template



Options folder of outlet end environment template



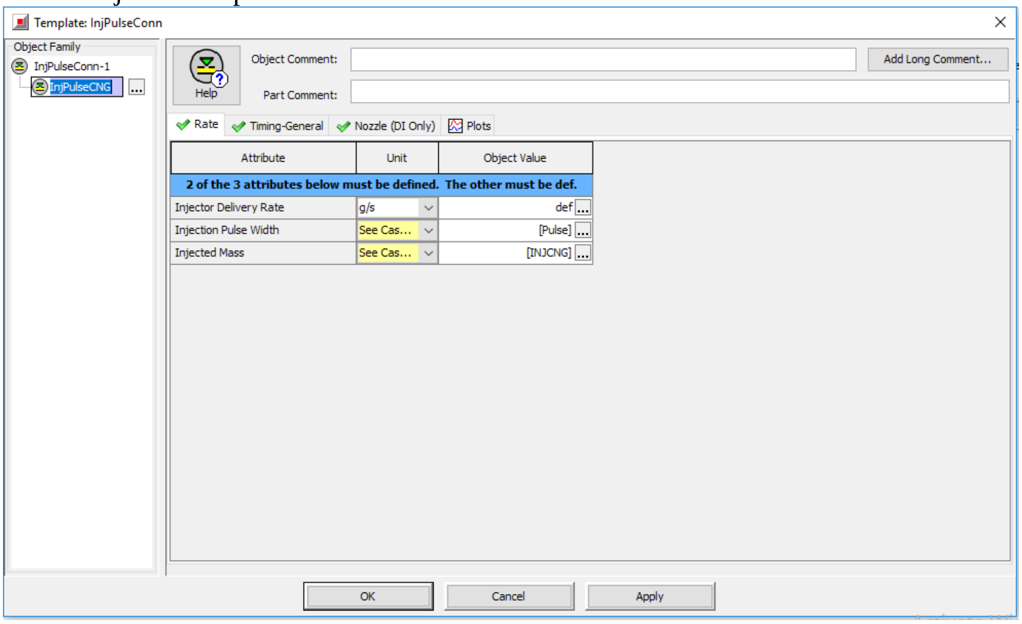
Altitude and humidity folder of outlet end environment template

Every components are linked with use create links.

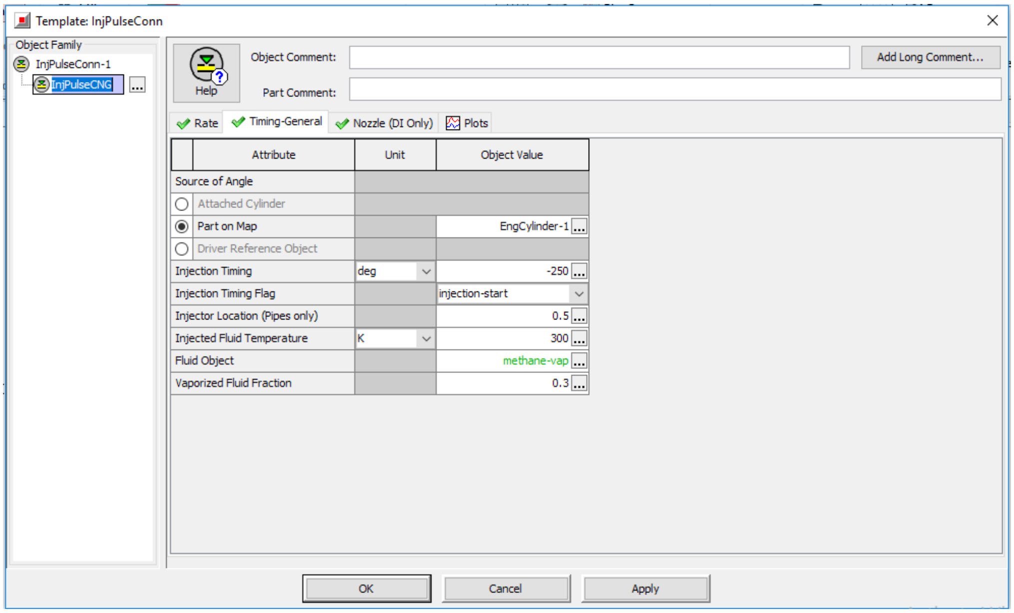
2. Entering Data and Creating Modification Dual Fuel Diesel Engine Simulation Model

For create the dual fuel diesel engine model, we need to add the CNG injector. The CNG injector is connected to intake port. The value that is needed for CNG injector are showed in the following figure.

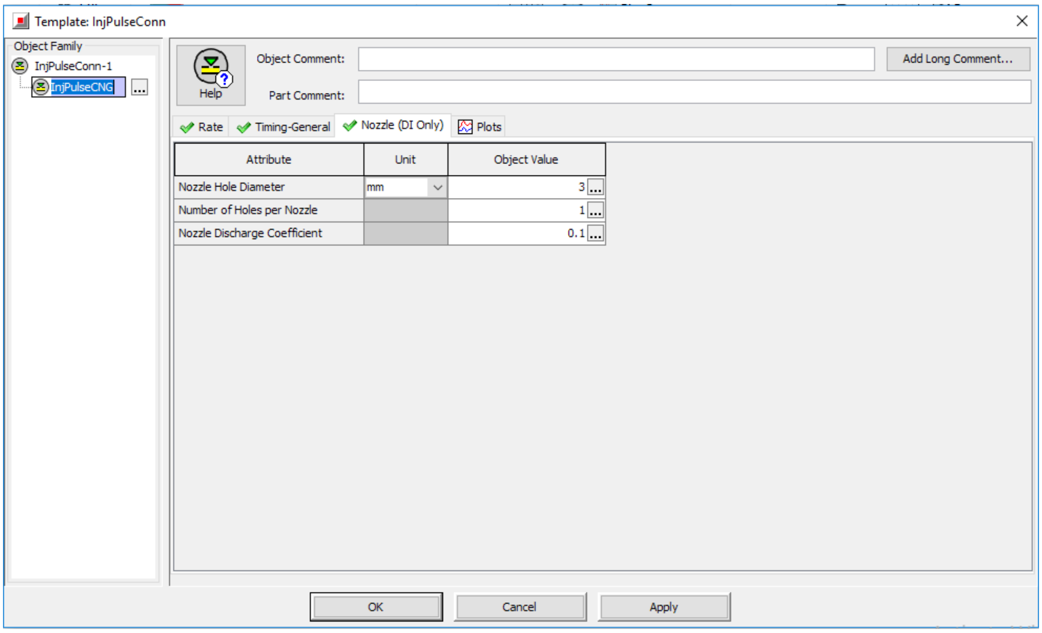
- CNG injector template



Rate folder of CNG injector template



Timing-general folder of CNG injector template



Nozzle folder of CNG injector template

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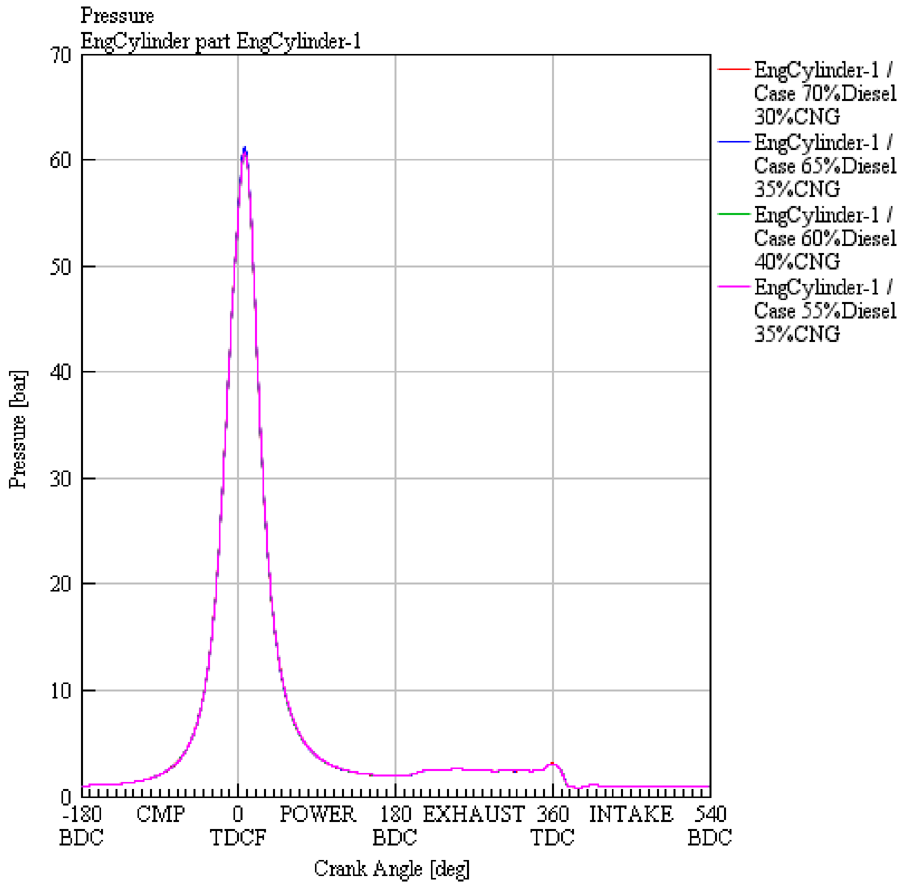
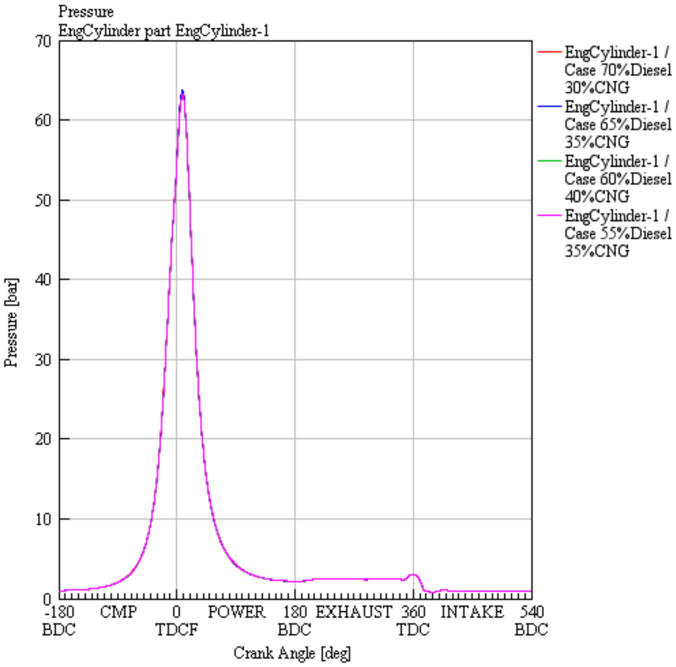
APPENDIX B

Following are the result of cylinder pressure, cylinder temperature, heat release rate, and volumetric efficiency with gas consumption variation at load 2000 - 4000 watt and at engine speed 1900 – 2200 rpm with interval 1000.

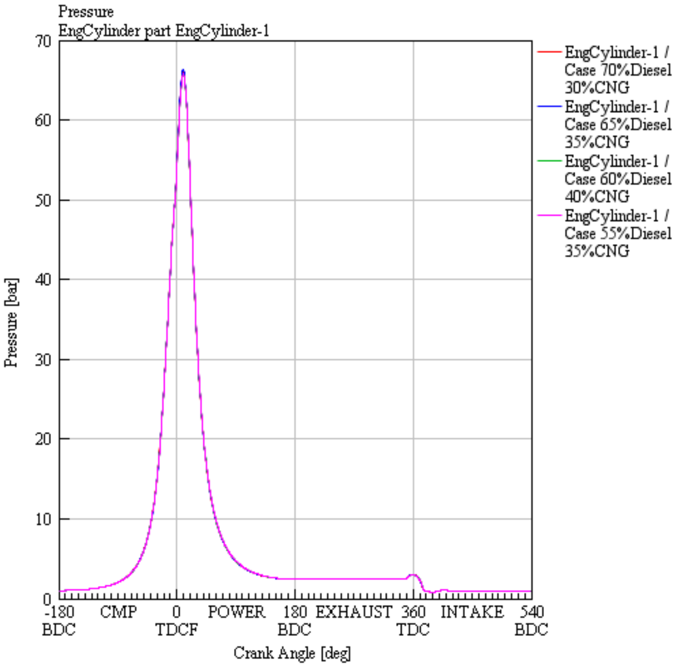
1. Cylinder Pressure



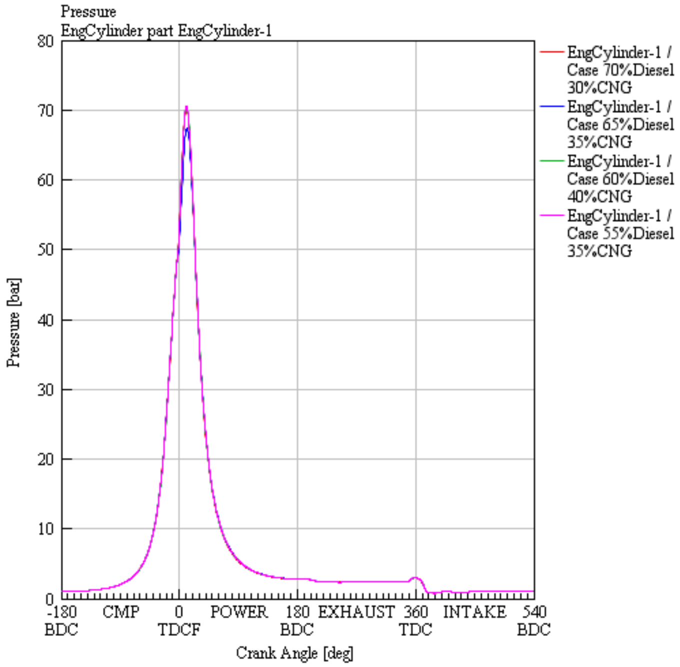
Cylinder pressure with gas consumption variation at load 1000 watt and speed 2000 rpm



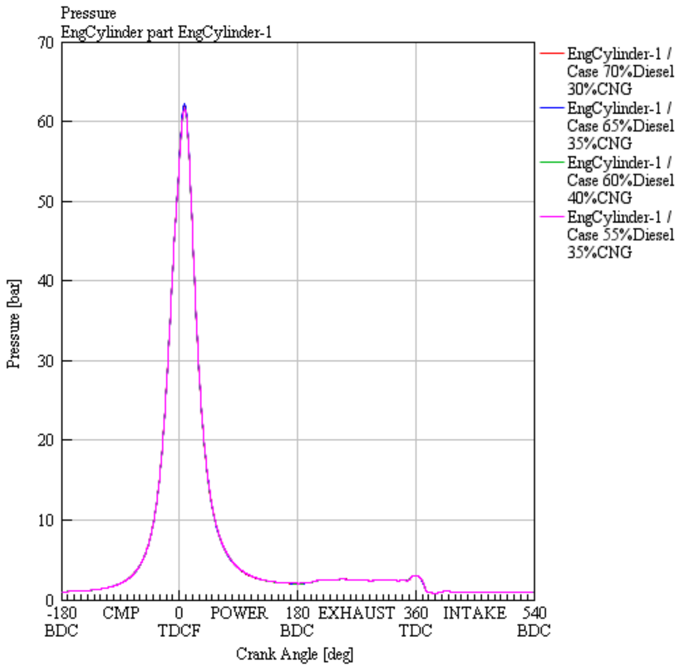
Cylinder pressure with gas consumption variation at load 2000 watt and speed 2000 rpm



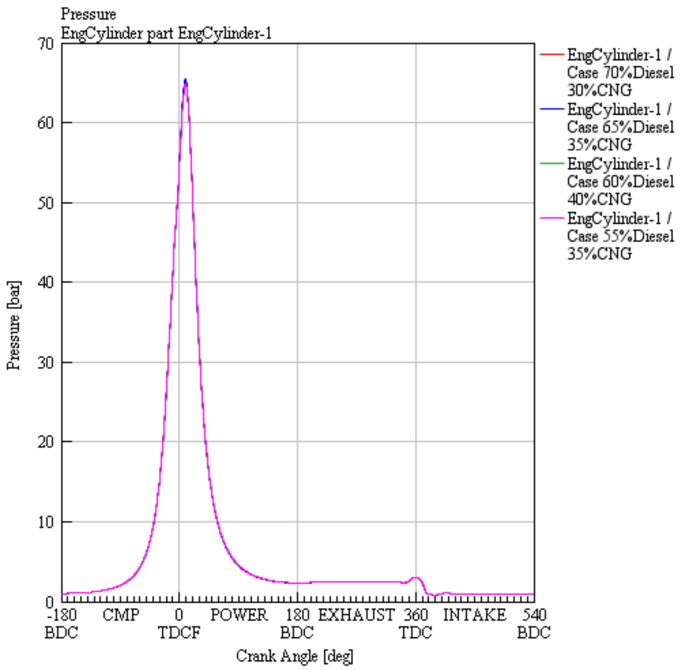
Cylinder pressure with gas consumption variation at load 3000 watt and speed 2000 rpm



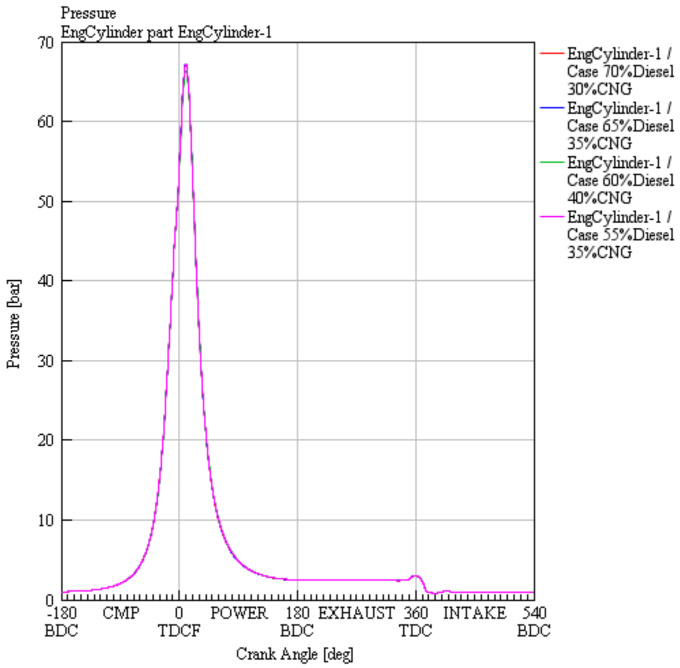
Cylinder pressure with gas consumption variation at load 4000 watt and speed 2000 rpm



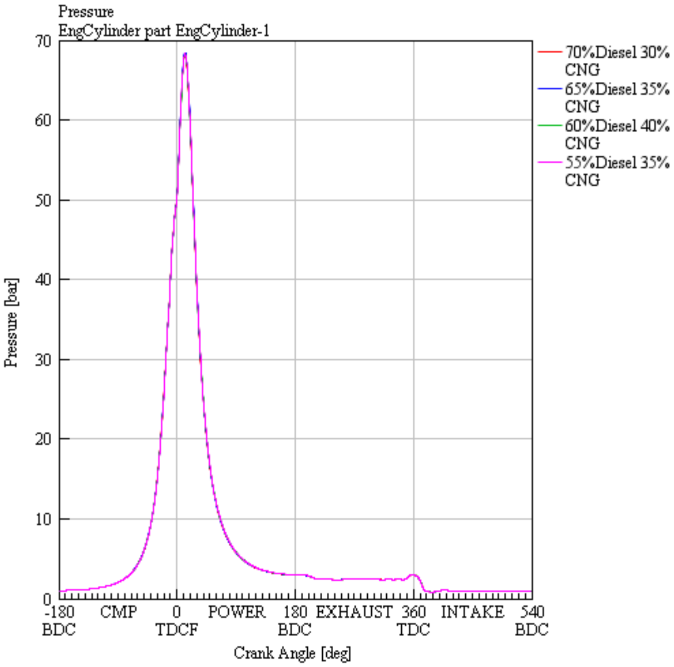
Cylinder pressure with gas consumption variation at load 1000 watt and speed 2100 rpm



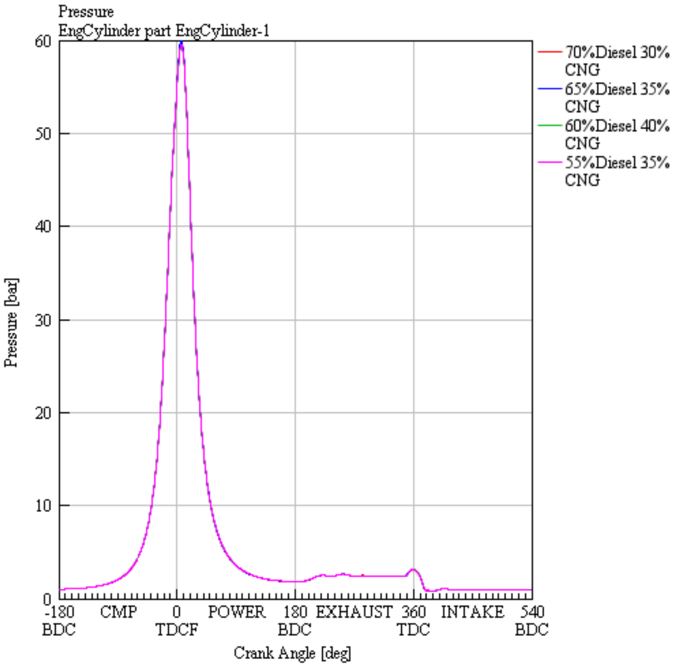
Cylinder pressure with gas consumption variation at load 2000 watt and speed 2100 rpm



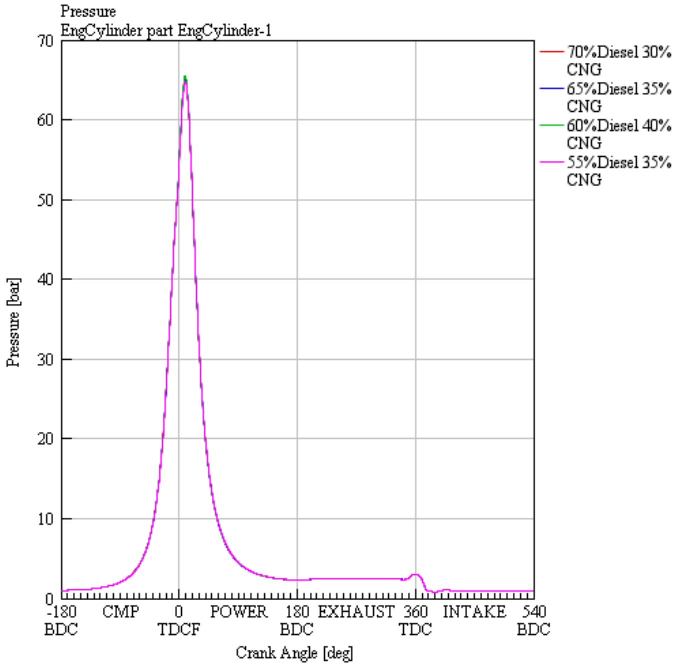
Cylinder pressure with gas consumption variation at load 3000 watt and speed 2100 rpm



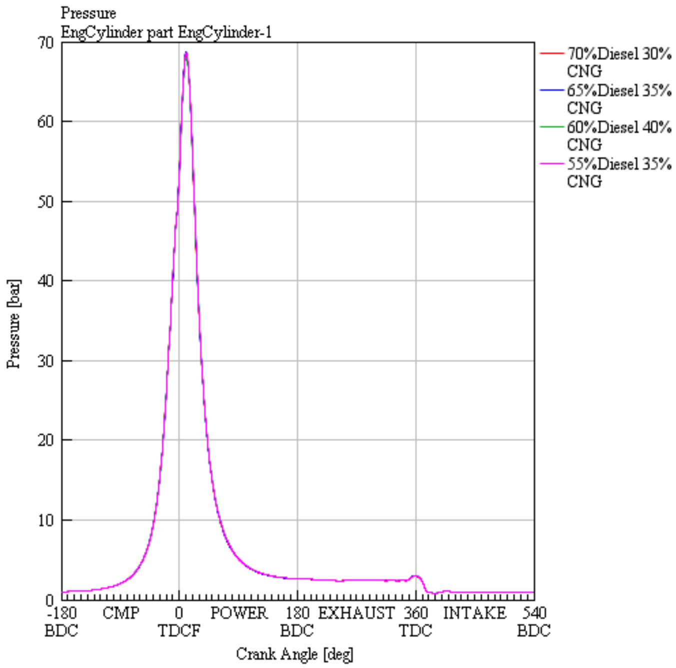
Cylinder pressure with gas consumption variation at load 4000 watt and speed 2100 rpm



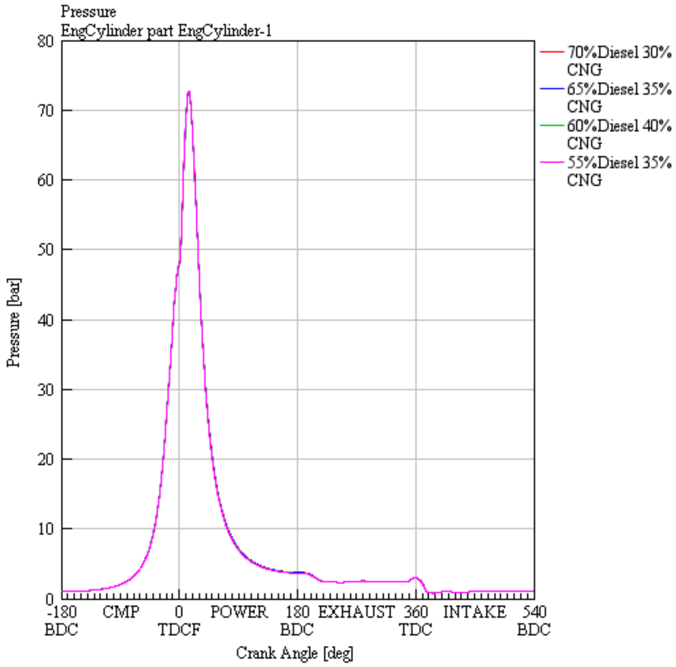
Cylinder pressure with gas consumption variation at load 1000 watt and speed 2200 rpm



Cylinder pressure with gas consumption variation at load 2000 watt and speed 2200 rpm

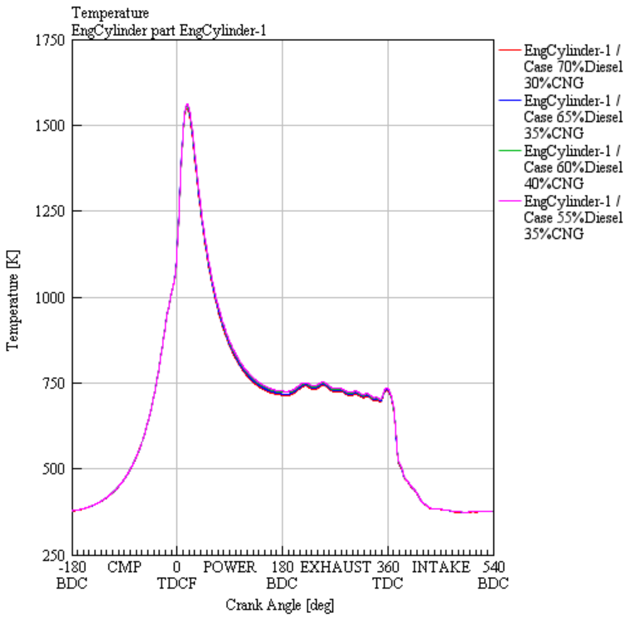


Cylinder pressure with gas consumption variation at load 3000 watt and speed 2200 rpm

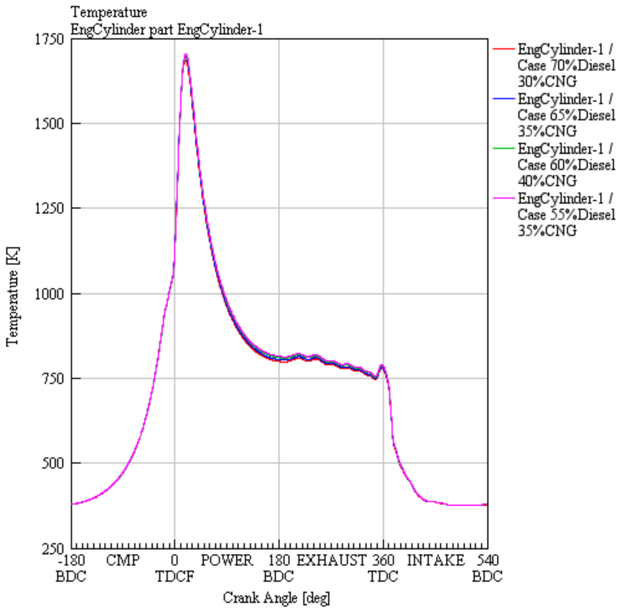


Cylinder pressure with gas consumption variation at load 4000 watt and speed 2200 rpm

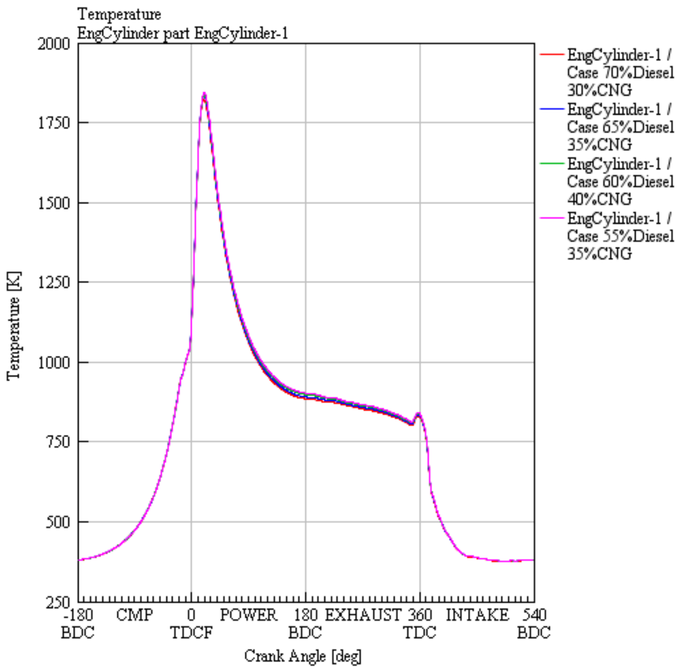
2. Cylinder Temperature



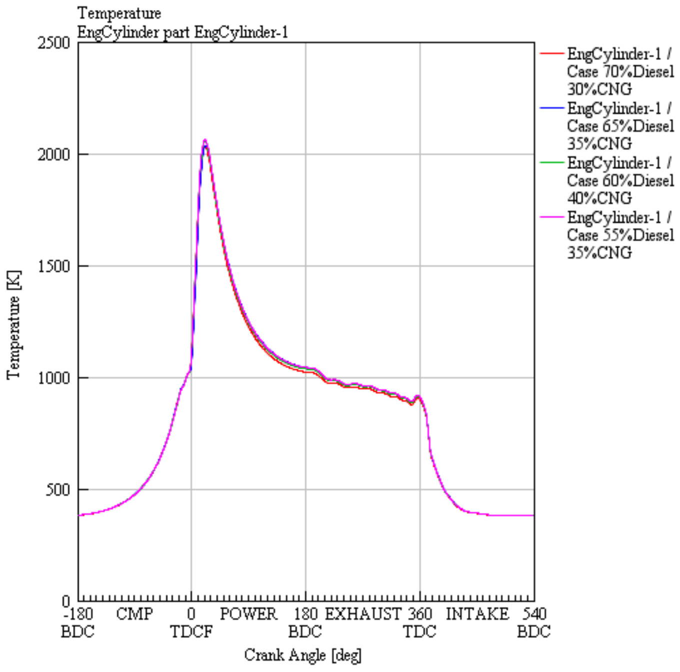
Cylinder temperature with gas consumption variation at load 1000 watt and speed 2000 rpm



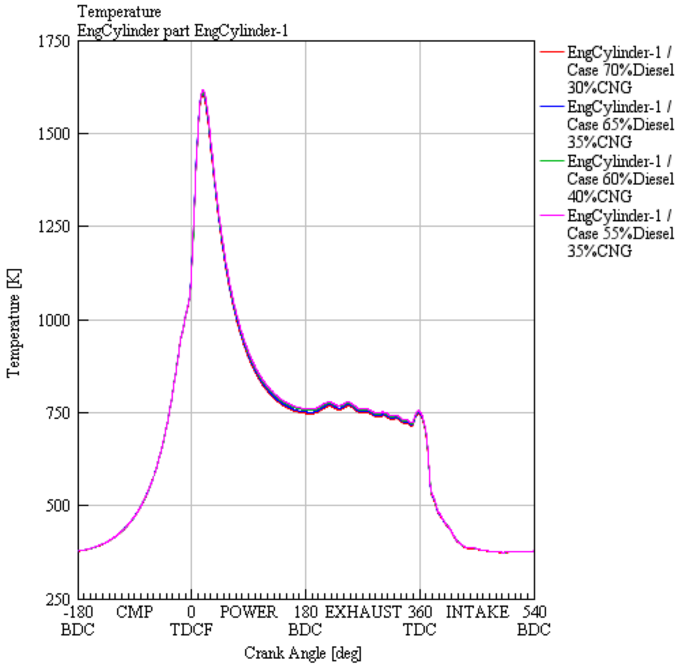
Cylinder temperature with gas consumption variation at load 2000 watt and speed 2000 rpm



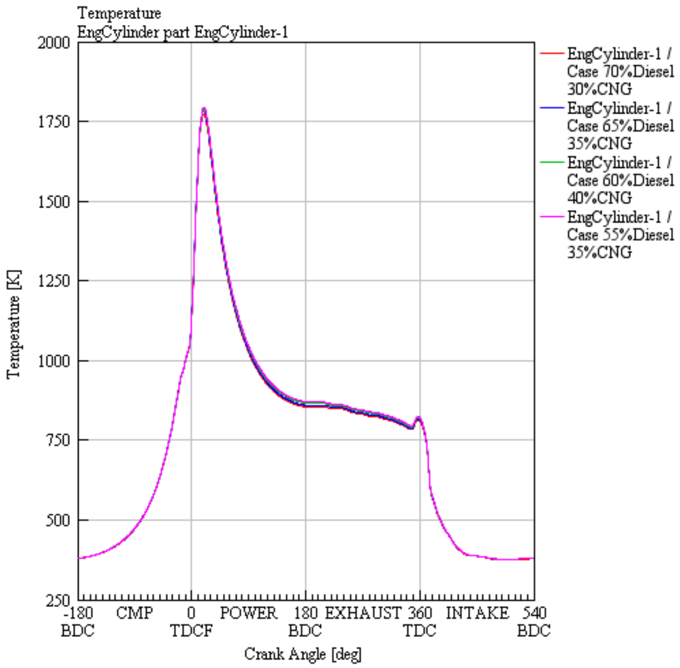
Cylinder temperature with gas consumption variation at load 3000 watt and speed 2000 rpm



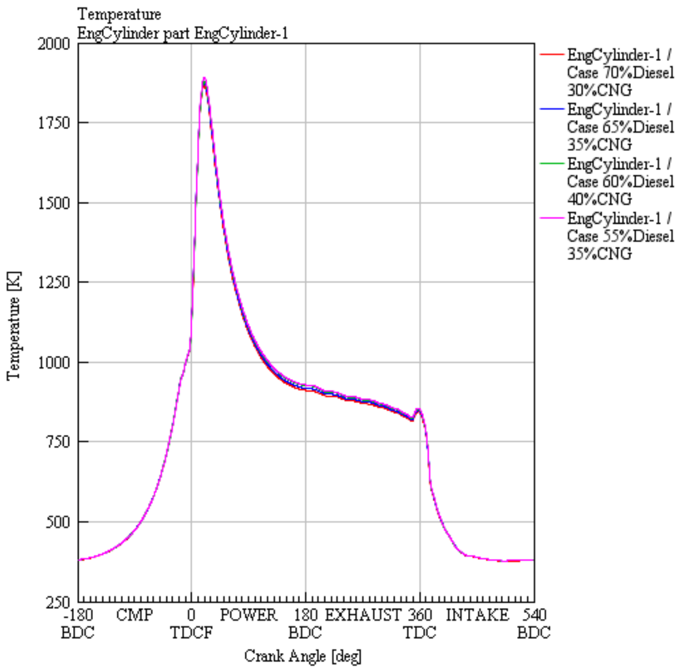
Cylinder temperature with gas consumption variation at load 4000 watt and speed 2000 rpm



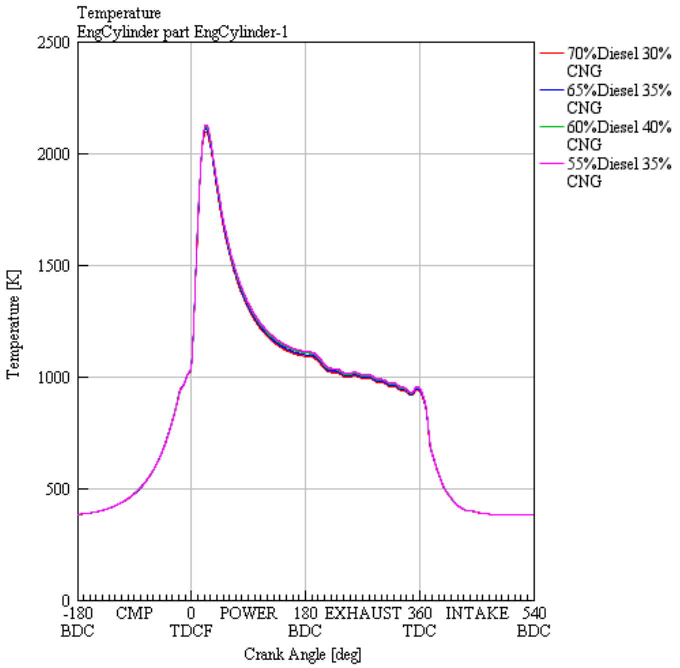
Cylinder temperature with gas consumption variation at load 1000 watt and speed 2100 rpm



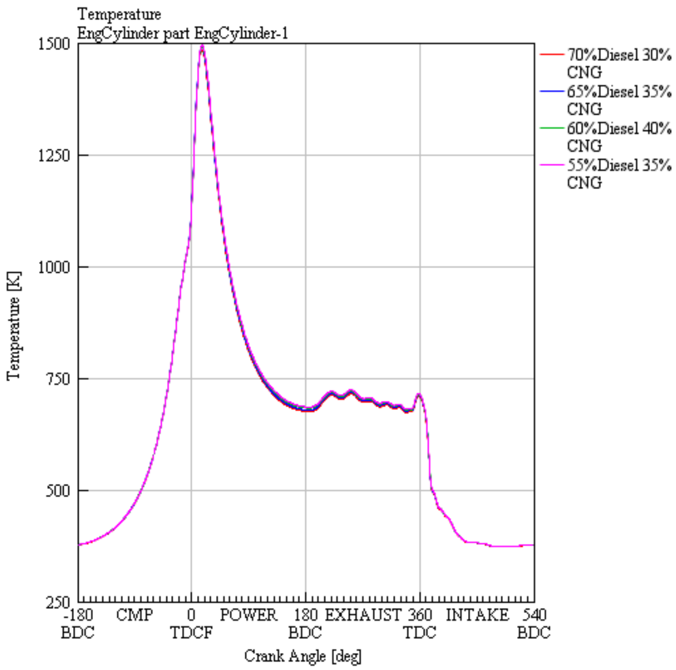
Cylinder temperature with gas consumption variation at load 2000 watt and speed 2100 rpm



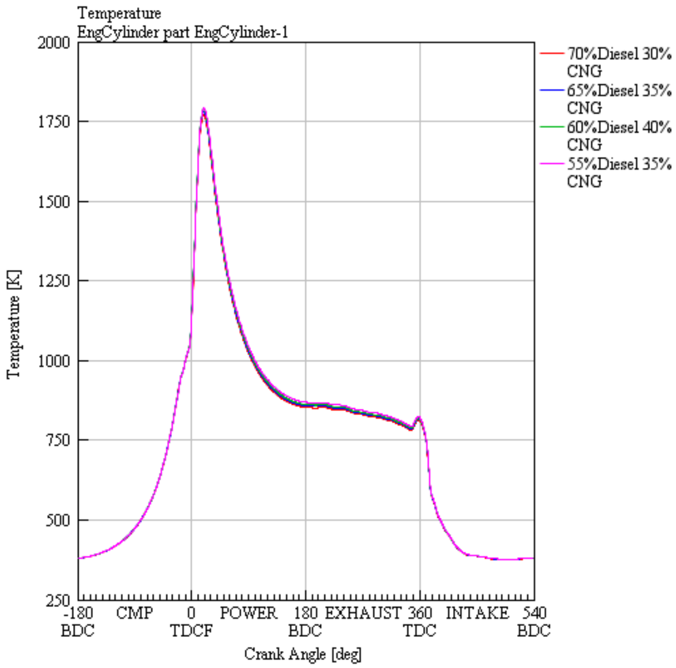
Cylinder temperature with gas consumption variation at load 3000 watt and speed 2100 rpm



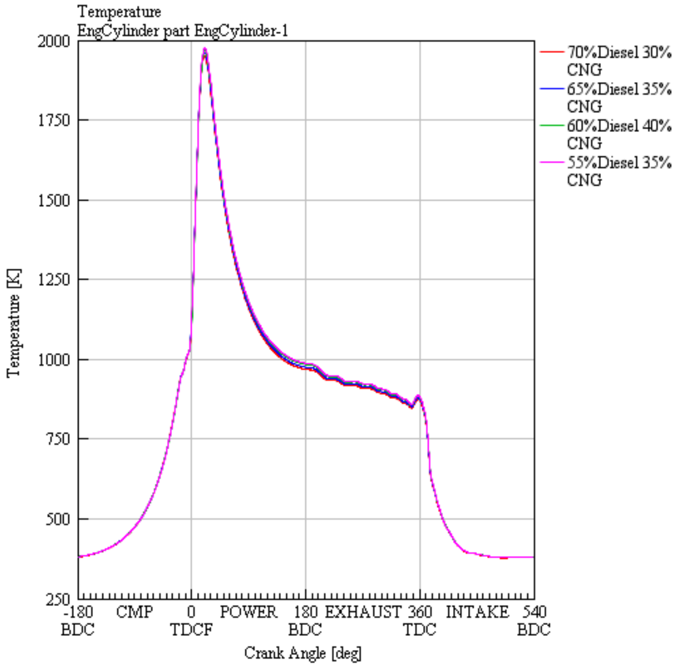
Cylinder temperature with gas consumption variation at load 4000 watt and speed 2100 rpm



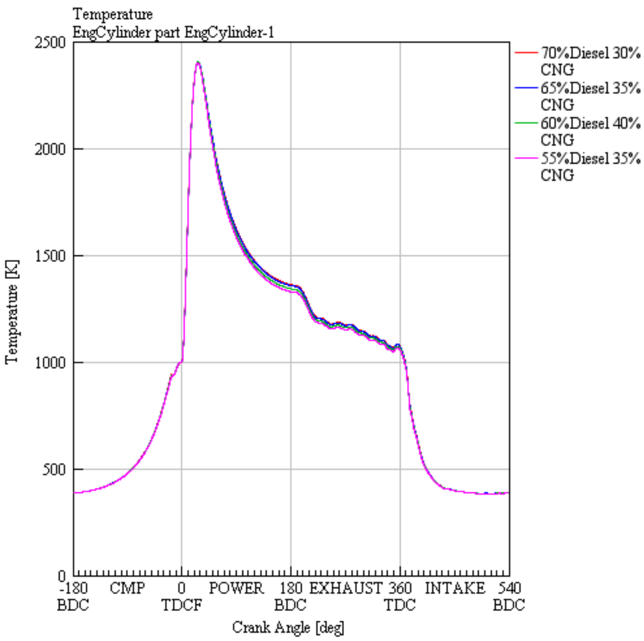
Cylinder temperature with gas consumption variation at load 1000 watt and speed 2200 rpm



Cylinder temperature with gas consumption variation at load 2000 watt and speed 2200 rpm

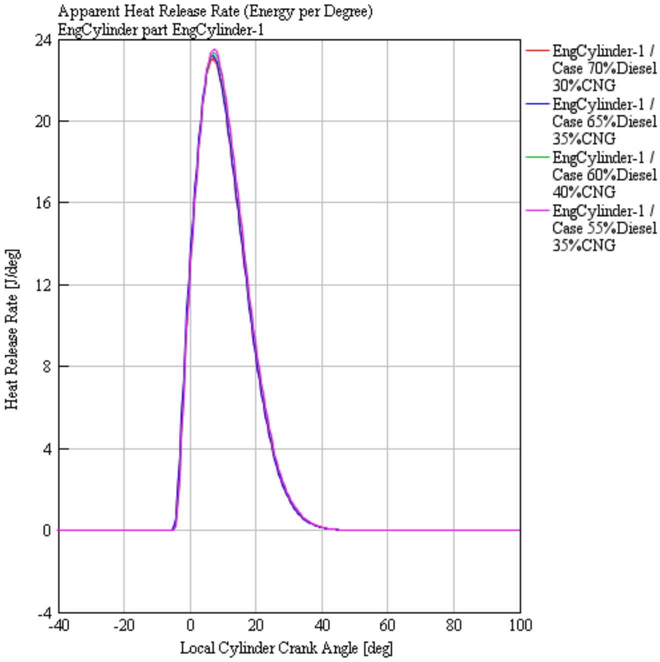


Cylinder temperature with gas consumption variation at load 3000 watt and speed 2200 rpm

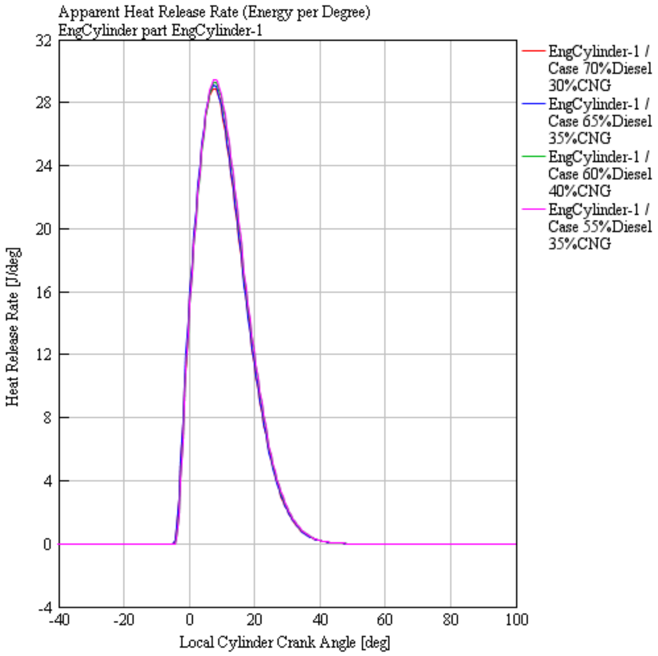


Cylinder temperature with gas consumption variation at load 4000 watt and speed 2200 rpm

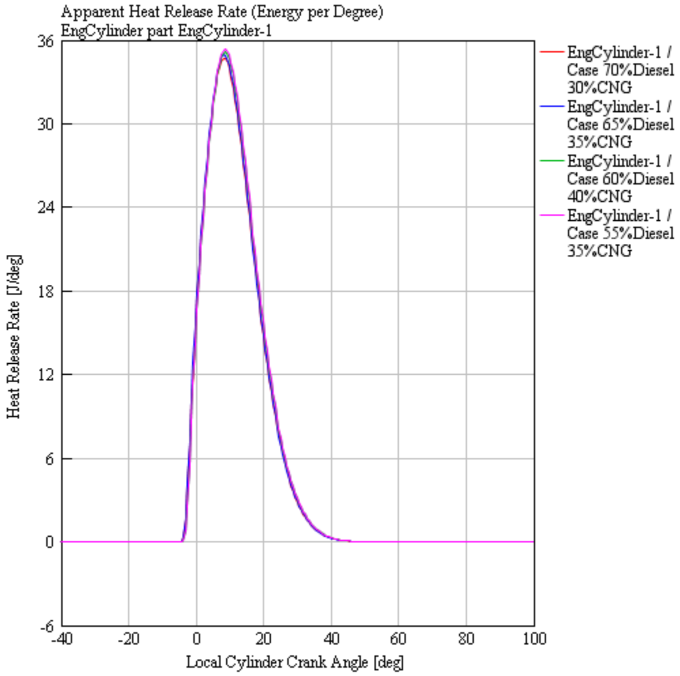
3. Heat Release Rate



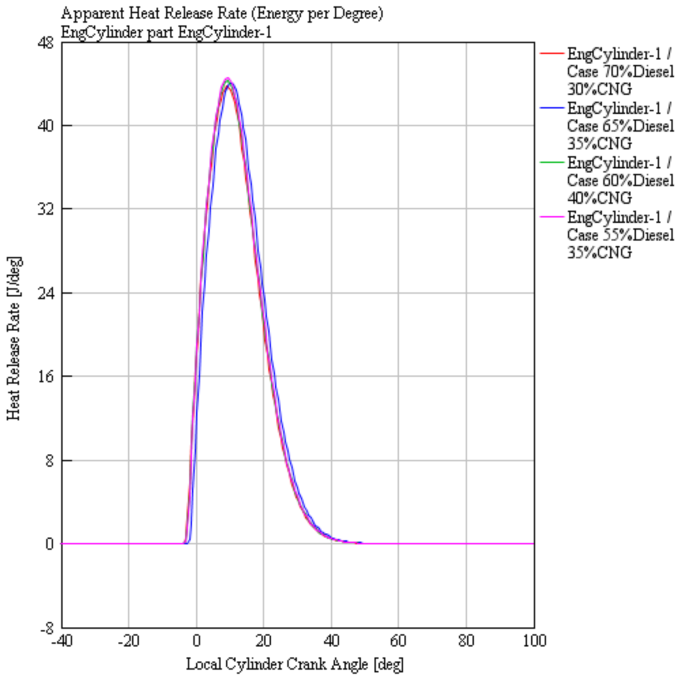
Heat release rate with gas consumption variation at load 1000 watt and speed 2000 rpm



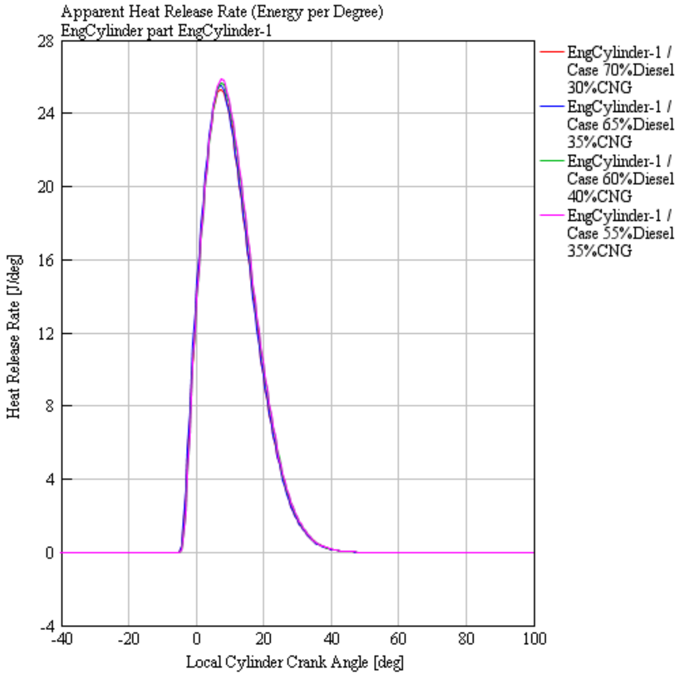
Heat release rate with gas consumption variation at load 2000 watt and speed 2000 rpm



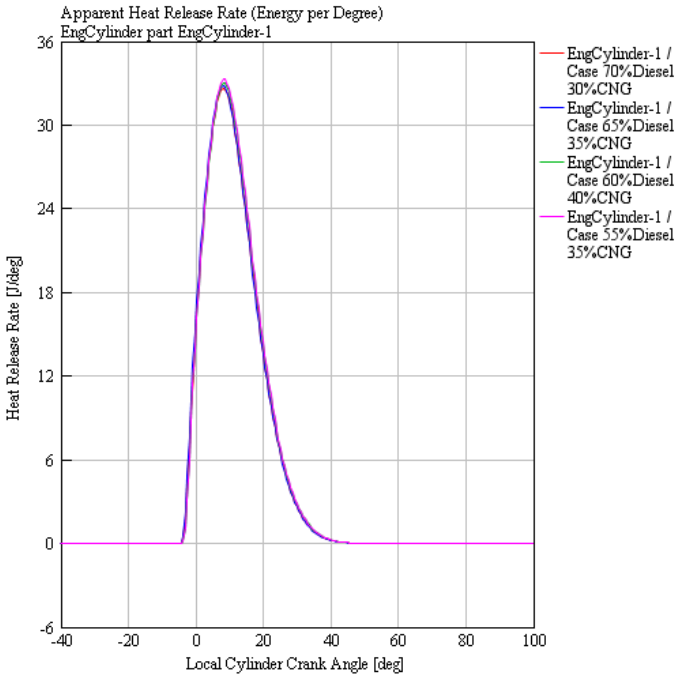
Heat release rate with gas consumption variation at load 3000 watt and speed 2000 rpm



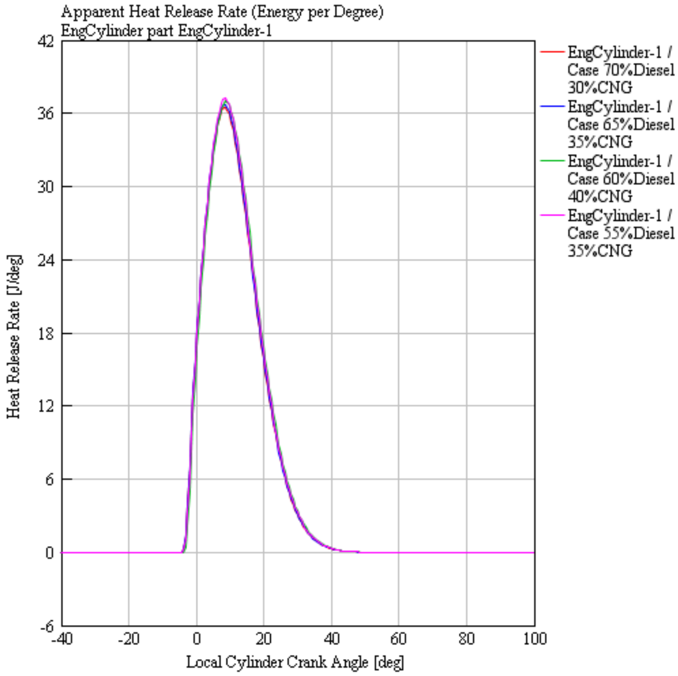
Heat release rate with gas consumption variation at load 4000 watt and speed 2000 rpm



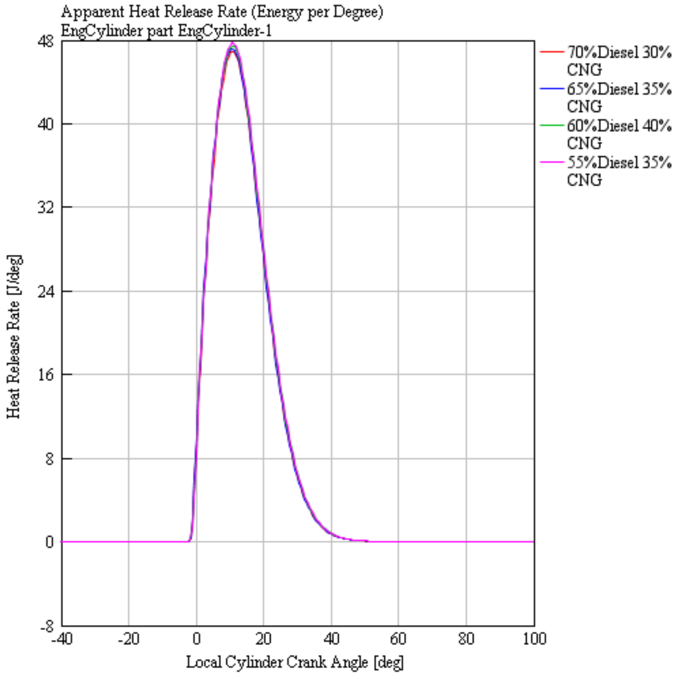
Heat release rate with gas consumption variation at load 1000 watt and speed 2100 rpm



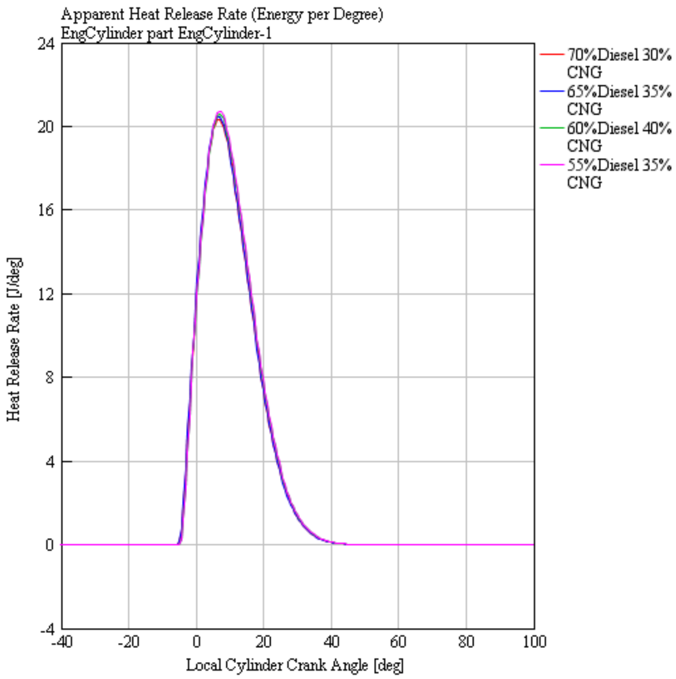
Heat release rate with gas consumption variation at load 2000 watt and speed 2100 rpm



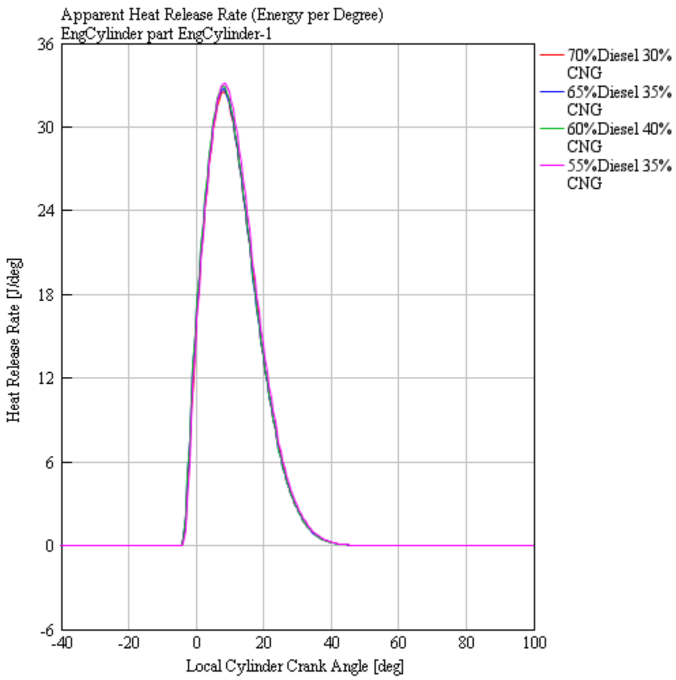
Heat release rate with gas consumption variation at load 3000 watt and speed 2100 rpm



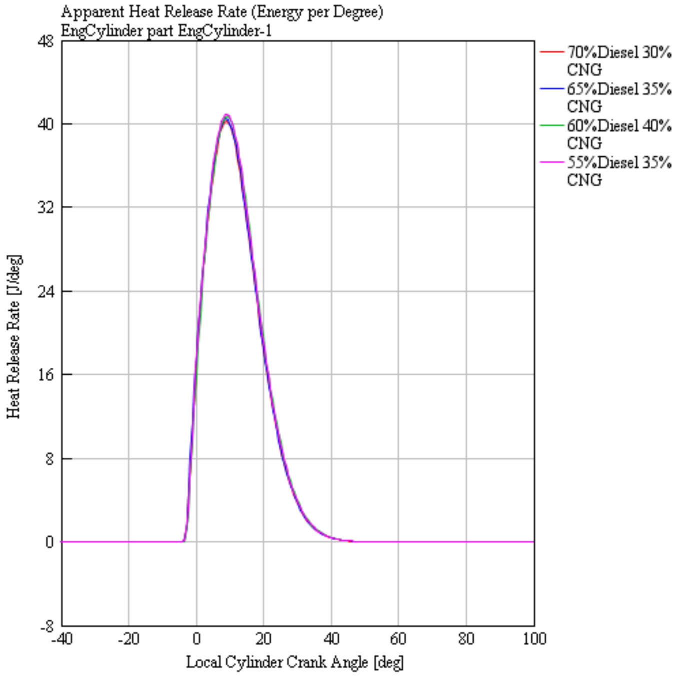
Heat release rate with gas consumption variation at load 4000 watt and speed 2100 rpm



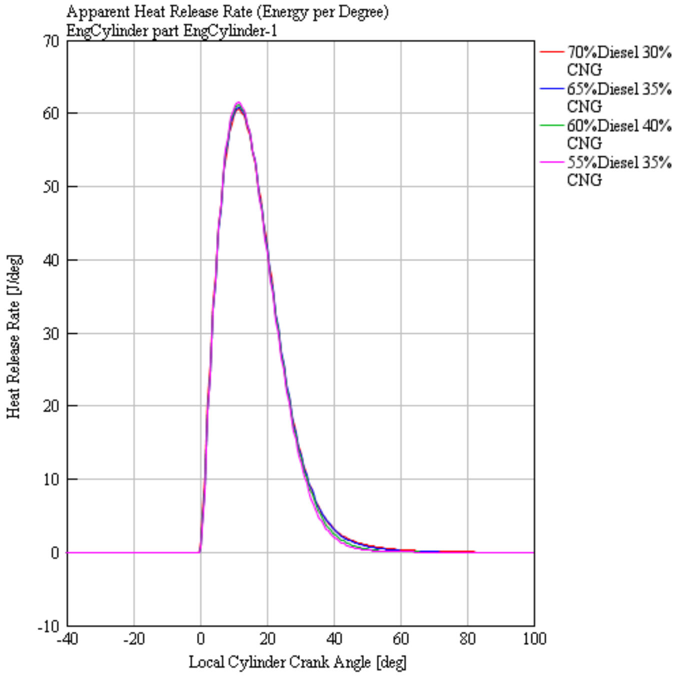
Heat release rate with gas consumption variation at load 1000 watt and speed 2200 rpm



Heat release rate with gas consumption variation at load 2000 watt and speed 2200 rpm



Heat release rate with gas consumption variation at load 3000 watt and speed 2200 rpm



Heat release rate with gas consumption variation at load 4000 watt and speed 2200 rpm

AUTHOR BIOGRAPHY

Theresia Dianita was born in Jakarta on October 20th, 1998. The author grew up in Jakarta with her parent and her brothers. Her father is a mechanical teacher in on of vocational high school. The author was educated and graduated from elementary school in SDS GKPI Rawamngun, junior high school in SMPN 92 Jakarta, and senior high school in SMAN 31 Jakarta. She was beginning her study in Sepuluh Nopember Institut of Technology on year 2016 through SNMPTN program then accepted as a student in double degree program, department of marine engineering, faculty of marine technology.

During the author's college life in ITS, she was active in some organization, he became a staff in big event of HIMASISKAL on her second year. On her third year, she was active in religion organization, Persekutuan Mahasiswa Kristen, as a treasurer in Christmas and Passover division. Also, she was active as assistant laboratory in Marine Machinery System (MMS) Laboratory, department of marine engineering, faculty of marine technology. On the fourth year, she was registered as a member in Marine Power Plant (MPP) laboratory.

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