MULTI DIMENSIONAL MODELLING OF A HIGH PRESSURE NATURAL GAS FUEL INJECTION PROCESS IN PRECHAMBER OF A SINGLE CYLINDER FOUR-STROKE ENGINE

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Report submitted in fulfillment of the requirements
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SUPERVISOR’S DECLARATION

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DEDICATION

My kindly supervisor, Mr Mohd Fadzil Abdul Rahim

My beloved parent Mohd Yamin bin Ali &Norizam binti Che Harun

My lovely brother and sister

My precious friends

May Allah bless all of you

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ABSTRACT

This thesis deals with the numerical setup of a simulation for compressed natural gas in spark ignition engine with pre-chamber for the full cycle four-stroke engine using Computational Fluid Dynamics (CFD). Single-cylinder Yamaha FZ 150i engine is used as the base line engine design for the simulation. The engine is modified by replacing the pre-chamber at the spark plug hole. This study is focused on the combustion model with different ignition and injection timing. 2000 revolution per minute (rpm) and 500 iterations are set as the tested speed and number of iterations per time step respectively. This simulation is started from intake valve open until exhaust valve close as a completed cycle of engine. Turbulence is captured using k-ε-realizable model. This simulation is divided in 2 cases. Case 1 is set for study the effect of injection timing while Case 2 is set for study the effect of ignition timing. The injection pressure is set 20 bar and 15° crank angle (CA) of injection period. The injection timing that used are 40° before top dead centre (BTDC), 50° BTDC and 60° BTDC while the ignition timing that used are 20° BTDC, 30° BTDC, and 40° BTDC at the 50° CA BTDC. The predicted maximum pressure due to the injection and ignition timing is 27 bar while maximum temperature is 1200 K. For further simulation, more data from experimental work is needed especially pressure and temperature.

ABSTRAK

Tesis ini berkaitan dengan kajian berangka tentang simulasi untuk gas asli termampat di dalam enjin pencucuhan api dengan pra-ruang sebagai bahagian tambahan untuk kitaran penuh enjin empat lejang menggunakan Dinamik Bendalir Komputeran (CFD). Enjin satu silinder Yamaha FZ 150i digunakan sebagai reka bentuk asas enjin bagi model simulasi. Enjin diubahsuai dengan menggantikan pra-ruang di lubang palam pencucuh. Kajian ini tertumpu kepada model pembakaran dengan pencucuhan yang berbeza dan masa suntikan. 2000 revolusi per minit (rpm) dan 500 iterasi ditetapkan sebagai kelajuan diuji dan bilangan itersai satu masa masing-masing. Simulasi ini bermula dari injap pengambilan dibuka sehingga penutupan injap ekzos sebagai kitaran lengkap enjin. Pergolakan ditangkap menggunakan model k-ε-realisasi. Simulasi ini dibahagikan dalam 2 kes. Kes 1 ditetapkan untuk mengkaji kesan masa suntikan manakala Kes 2 ditetapkan untuk kajian kesan masa penyalaan. Tekanan suntikan ditetapkan 20 bar dan 15 ° CA tempoh suntikan. Masa suntikan yang digunakan adalah 40 ° BTDC, 50 ° BTDC dan 60 ° BTDC manakala masa penyalaan yang digunakan ialah 20 ° BTDC, 30 ° BTDC, dan 40 ° BTDC pada 50 ° CA BTDC. Tekanan maksima yang dijangkakan akibat suntikan dan pemasaan penyalaan ialah 27 bar manakala suhu maksimum ialah 1200 K. Untuk simulasi lanjut, lebih banyak data daripada kerja eksperimental diperlukan terutama tekanan dan suhu.

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LIST OF SYMBOLS

κ-ε	k-epsilon
λ	fuel air ratio
μ	micro
$\circ$	degree
%	percentage
ρ	density
rpm	revolution per minute
C_R	compression ratio
V_s	swept volume
V_c	clearance volume
s	second

LIST OF ABBREVIATIONS

2D	Two dimensional
ATDC	After top dead center
BDC	Bottom dead center
BTDC	Before top dead center
CA	Crank angle
CAD	Computational Aided Design
CFD	Computational Fluid Dynamics
CI	Compression ignition
CNG	Compressed natural gas
DI	Direct injection
DISI	Direct injection-spark ignition
ICE	Internal combustion engine
K	Kelvin
N	Newton
PA	Pascal
SI	Spark ignition
SOHC	Single Over-Head Cam
TDC	Top dead center

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

This chapter discuss the information about this project include background of project, problem statement, objectives and scope of project.

1.2 BACKGROUND OF PROJECT

Compressed Natural gas (CNG) is one of the alternative fuel that find that can be used as the vehicle fuel replacing the gasoline (petrol) or diesel fuel. In this project the engine used is gasoline engine with single cylinder that convert to the CNG engine.

To improve the combustion and emission performance of CNG DISI engines, the pre-chamber is designed to help combustion in air fuel-mixing process. This design is a function to make engine efficiency increase with the high temperature and pressure in combustion chamber without knocking. Air-fuel mixing process in the cylinder is affected by many parameters such as the compression, combustion chamber and precombustion chamber geometry, injection pressure, nozzle-hole number and arrangement and swirl intensity. In this case, the model of prechamber is designed in order to improve the engine performance. The importance of this study is investigating the influences of combustion flow in the prechamber in case injection and ignition timing were controlled.

1.3 PROBLEM STATEMENT

The inclusion of pre-chamber to improve combustion in a spark ignited gasoline engine to run on compressed natural gas need to be investigated. This is because the inclusion of pre chamber has reduced the effective compression ration of the engine. As results, the cylinder pressure and temperature have become lower than the original engine configuration. Two methods of investigation have been proposed. One is to vary the injection timing and the other is by ignition timing. The effect of both strategies is evaluated primarily based on cylinder combustion pressure and temperature.

1.4 OBJECTIVES

Based on the problem statement above, this project is conduct to simulate different injection and ignition timing for high pressure fuel injection in precombustion chamber of a single cylinder four-stroke ignition engine.

1.5 SCOPES OF STUDY

The scope of this project are:

1. Structual design of main and prechamber in 2D geometry model based on Yamaha FZ150i engine dimension.
2. Design the precombustion chamber based on the actual main chamber volume where the prechamber volume less than 5% of the main volume.
3. Mesh generation and boundary condition setup
4. Simulation of cold flow and reacting flow process in precombustion chamber using CNG.

1.6 FLOW CHART

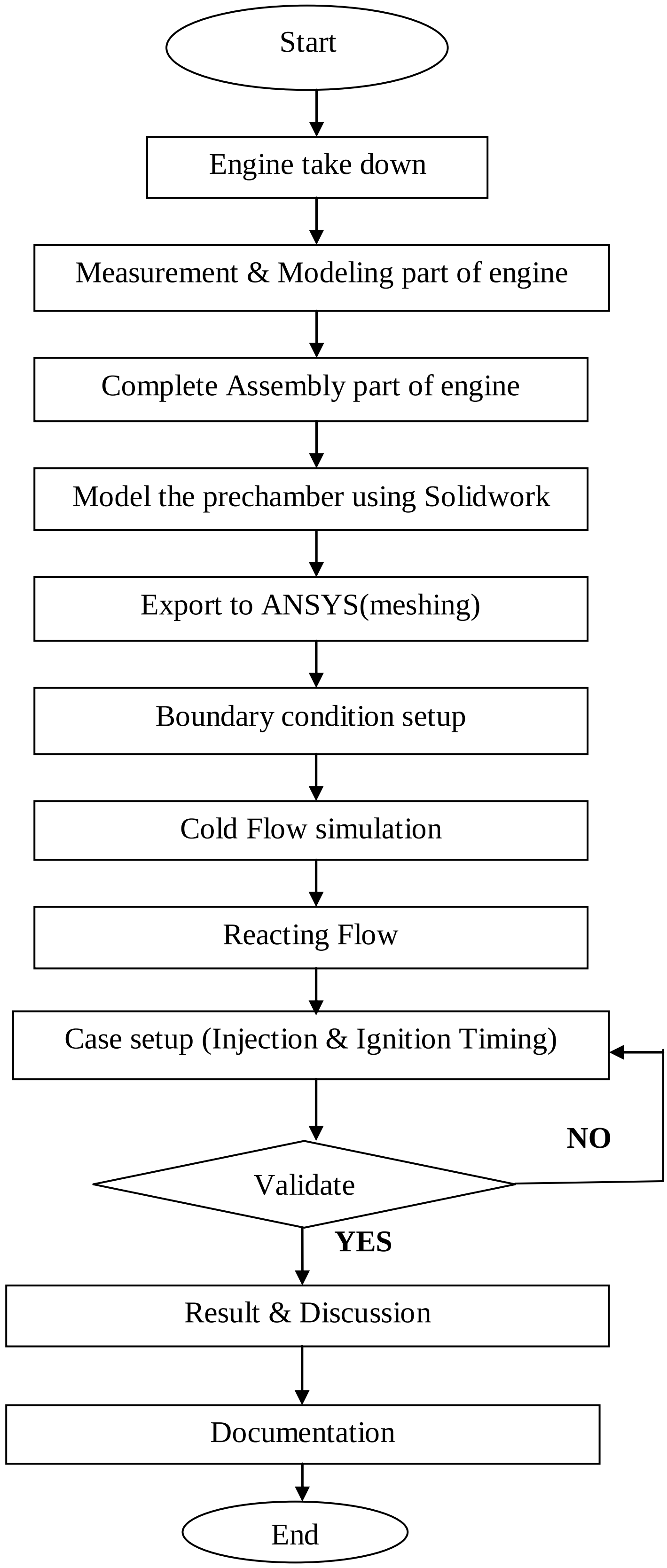


Figure 1.1 : Project flow chart

1.7 SUMMARY

This chapter is generally described the project from the background, problem statement, objectives and scope of the project.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

The purpose of this chapter is to provide a review of past research efforts related to spark ignition four-stroke engine, compressed natural gas as a fuel, internal combustion engine using prechamber model, direct injection control strategy and fluent analysis using ANSYS 12. The review is organized chronologically to offer insight to how past research efforts have laid the groundwork for subsequent studies, including the present research effort. The review is detailed so that the present research effort can be properly tailored to add to the present body of literature as well as to justly the scope and direction of the present research effort.

2.2 INTERNAL COMBUSTION ENGINE

An internal combustion engine means that engine uses the explosive combustion of fuel to move the piston and turns a crankshaft that then turns the car wheels via a chain or a drive shaft. The different types of fuel commonly used for car combustion engines are gasoline (or petrol), diesel, and kerosene.

2.2.1 Four-Stroke and Two-Stroke Engine

Four-stroke and two-stroke engine usually refer to the operating cycles of the engine. Most of the engine refers to reciprocating engines, where the piston moves back and forth in a cylinder and transmits power through a connecting rod and crank mechanism to the drive shaft (Heywood, 1988).

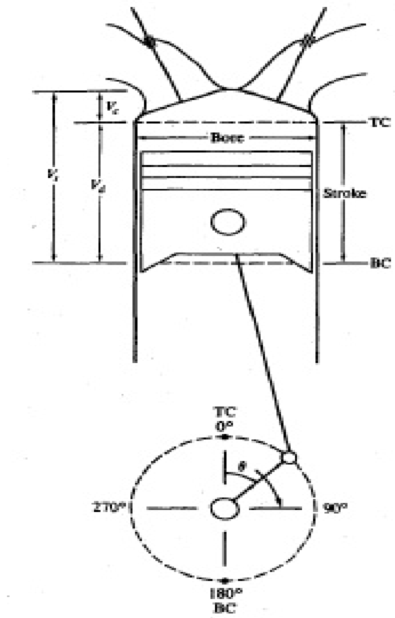


Figure 2.1 : Basic geometry of reciprocating internal combustion engine.

Source: Heywood 1988

The cyclical piston motion is produced by the rotation of the crank. The piston comes to rest at the top dead center (TDC) crank position and bottom dead center (BDC) crank position when the cylinder volume is a minimum or maximum. The minimum cylinder volume is called the clearance volume, V_c. Compression ratio, r_c is the ratio of maximum volume to the minimum volume. For spark ignition engine, typical value of r_c are 8 to 12 while for the compression ignition are 12 to 24.

The cycle reciprocating majority operate based on the four-stroke cycle (Heywood, 1988). In four-stroke engine, it has four cycles before complete the combustion. Each cyclinder is required four strokes of its piston, and need two revolution of the crankshaft to complete the sequence of events which produces one power stroke. Both spark ignition (SI) and compression ignition (CI) using this cycle. This four cycle are intake stroke, compression stroke, expansion stroke and exhaust stroke.

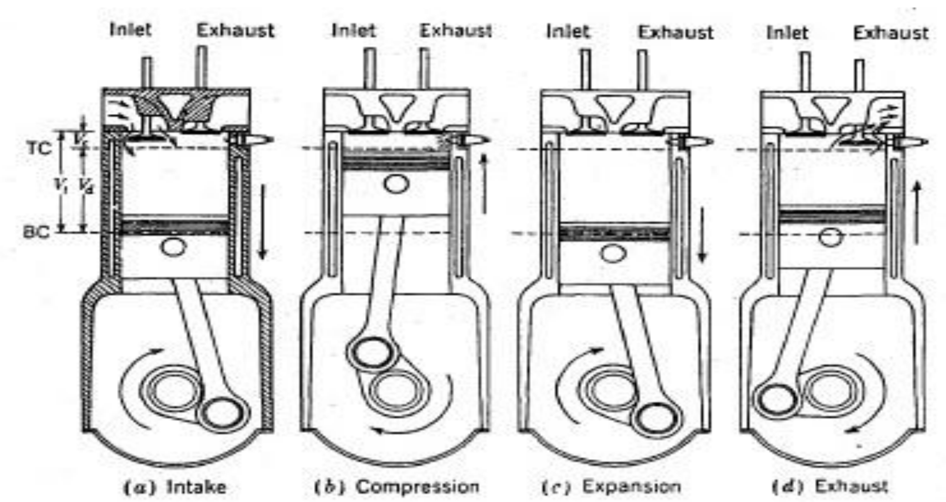


Figure 2.2: The four-stroke operating cycle.

Source: Heywood 1988

The operating cycle to four-stroke start from intake stroke, which the piston at TDC and ends with the piston at BDC. In this stroke, fresh mixture is drawn into the cylinder. The inlet valve opens shortly before the stroke start to increase the mass inducted and closes after the stroke ends.

In the second stage of the cycle, a compression stroke is operated when the inlet and exhaust valves are closed and the mixture in the cylinder is compressed to a small fraction of its initial volume. At the end of this stroke, combustion initiated and the cylinder pressure rises more rapidly.

The piston starts at TDC and ends at BDC in expansion stroke. In this stroke, the high temperature, high pressure, and gases push the piston down and force the crank to rotate. During the expansion stroke, the piston had to do about five times as much work done of compression. As the piston approaches at BDC the exhaust valve is opened to initiate the exhaust process and drop the cylinder pressure to close to the exhaust pressure.

The last process in the four-stroke engine cycle is the exhaust stroke. During this stroke, gases remaining burned exit the cylinder because the cylinder pressure may be substantially higher than the exhaust pressure. Then they swept out by the piston as it moves toward TDC. As the piston approaches TDC the inlet valve opens. After the exhaust valve closed, the cycle will be started again.

To obtain a higher power output from a given engine size, and a simpler valve design, the two-stroke cycle was developed (Heywood, 1988). The two-stroke cycle is applicable to both SI and CI engines.

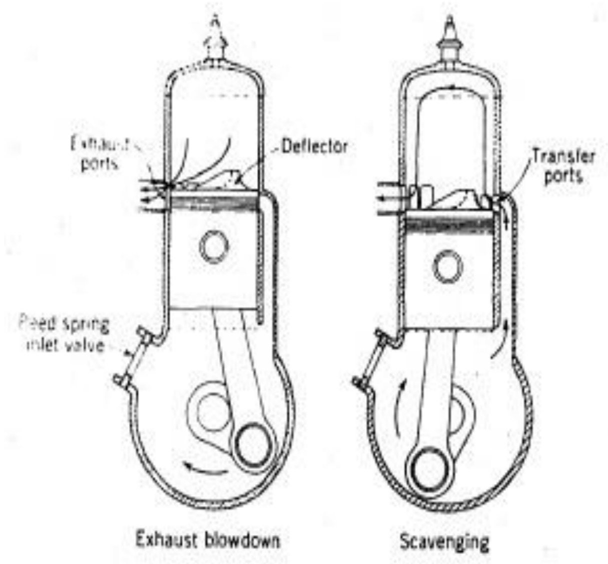


Figure 2.3: The two-stroke operating cycle.

Source: Heywood 1988

For the design of two-stroke cycle in figure 2.3, it has ports in the cylinder liner. This ports operate opened and closed by piston motion, control the exhaust and inlet flows while the piston is close to BDC. In two-stroke cycle, it only have two process compare with four-stroke cycle that have four cycles to complete. This cycle are compression and expansion stroke.

A compression stroke which starts by closing the inlet and exhaust ports, and then compress the cylinder contents and draws a fresh charge into the crankcase. When the piston approaches TDC, combustion is initiated.

After the compression stroke, the piston moves down and approaches the BDC, so this called the expansion stroke when first the exhaust ports and the intake ports are uncovered. Most of the burnt gases exit the cylinder in an exhaust blowdown process. When the inlet ports uncovered, the fresh charge which has been compressed in the crankcase flows into the cylinder. The piston and the ports are generally shaped to deflect the incoming charge from flowing directly into the exhaust ports.

2.2.2 Spark Ignition and Compression Ignition Engine

The internal combustion engine also can be classified by type of ignition on the engine (Pulkrabek, 2004). Two types of ignition that found in internal combustion engine are spark ignition (SI) and compression ignition (CI). An SI engine starts at the combustion process in each cycle by use of a spark plug. The spark plug gives a high voltage electrical discharge between two electrodes which ignites the air-fuel mixture in the combustion chamber surrounding the plug. In CI engine, the proses start when the air-fuel mixture self ignites due to high temperature in the combustion chamber caused by high compression.

Spark ignition usually found in the gasoline or petrol engine because they have higher self-ignition temperature. SI engine works on the Otto cycle (Gupta, 2006). A. Nicolaus Otto in 1876 proposed an ideal air-standard cycle with the constant volume heat addition , which formed the basis for the practical spark-ignition engines (petrol and gas engines). The cycle shown on *p-V* and *T-s* diagram in figure 2.4 (a) and figure 2.4 (b) respectively.

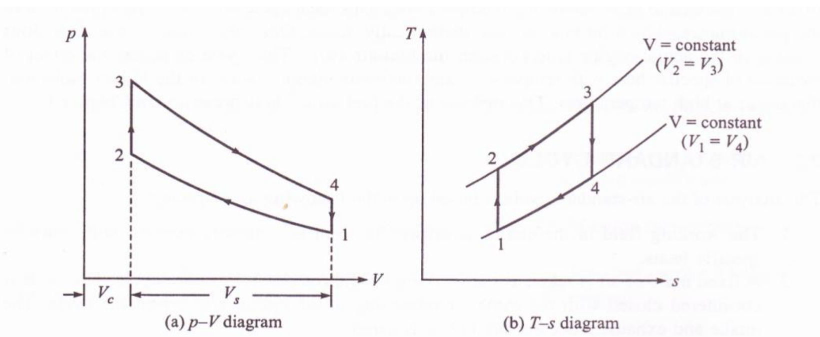


Figure 2.4: Otto cycle.

Source: Gupta 2006.

In SI engine, air and fuel mixture in gaseous form is inducted through the carburettor into the cylinder during the suction stroke. The throttle valve of the carburetor controls the quantity of charges. The quality of the charge remains almost fixed during normal running conditions at variable speed and load. The spark is required to burn the fuel. A compression ratio of this engine is 6 to 10.5. The upper limit is fixed by the anti-knock quality of fuel. The engine tends to knock at higher compression ratios.

Compression ignition engine also known as the diesel engine. A fuel having a lower self ignition temperature is desirable such as diesel oil. So, this engine will operate using the diesel cycle. Rudolf Diesel in 1892 introduced this diesel cycle (Gupta 2006). It is a theoretical cycle for slow speed compression ignition diesel engine. In this cycle, heat is added at constant pressure and rejected at constant volume. The *p-V* diagram and *T-s* diagram are shown in figure 2.5 (a) and figure 2.5 (b).

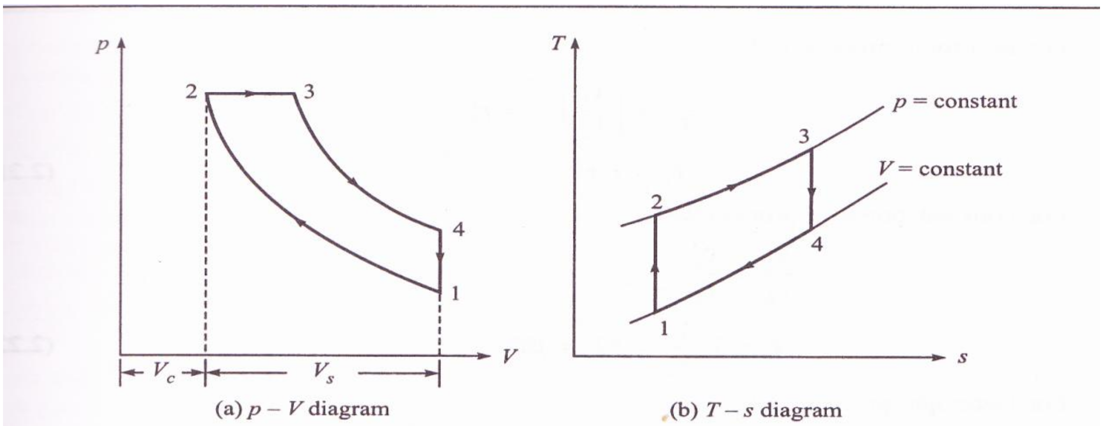


Figure 2.5: Diesel cycle.

Source: Gupta 2006.

For CI engine, only air is introduced into the cylinder during the suction stroke and therefore the carburetor is not required. Fuel is injected at high pressure through the fuel injectors direct into the combustion chamber. The amount of air is fixed but the amount of fuel injected is varied by regulating the quantity of fuel in the pump. The air-fuel ratio is varied at varying load. A compression ratio of CI engine is higher than gasoline engine. This compression ratio is 14 to 20.

2.2.3 Fuel Injection and Carburetors

Fuel injectors are nozzle that injected fuel into the intake air. They are normally controlled electronically, but mechanically controlled injectors, which are camoperated. The amount of fuel injected each cycle is controlled by injector pressure and time duration of injection.

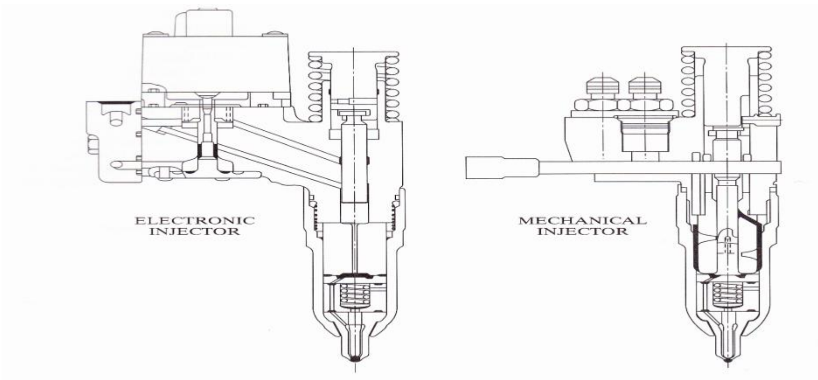


Figure 2.6: Electronic and mechanical injector.

Source: Pulkrabek 2004

An electric fuel injector consists of the following basic components; valve housing, magnetic plunger, solenoid coil, helical spring, fuel manifold, and pintle (needle valve). The fuel exits the injectors at velocities greater than 100m/sec, and flow rates of 3 to 4 gm/sec (Pulkrabek, 2004). In mechanically controlled injectors there is no solenoid coil, and the plunger is moved by the action of a camshaft.

Carburetors were used on most SI engine for several decades as means of adding fuel to the intake air. Carburetors are still found on few automibiles, but the vast majority of car engines use simpler, better controlled, more flexible fuel injection system.

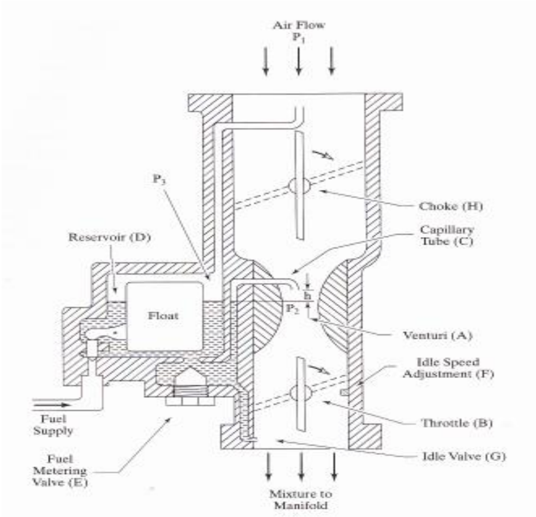


Figure 2.7: Basic carburetor.

Source: Pulkrabek 2004.

Figure 2.7 shows that the basic carburetor is a venturi tube that mounted with a throttle plate and a capillary tube to the input fuel. It is usually mounted on the upstream end of the intake manifold, with all air entering the engine passing first through this venturi tube. Most of the time , there will be an air filter mounted directly on the upstream side of carburetor. Other main parts of the carburetor are the fuel reservoir, main metering needle valve, idle speed adjustment, idle valve, and the choke.

2.3 PRECOMBUSTION CHAMBER

Precombustion chamber or indirect injection have been used to generate vigorous charge motion during the compression stroke. There are categories of combustion chamber that can be classified as:

1) Swirl or turbelent chamber.
2) Precombustion chamber.
3) Air and energy cells.

2.3.1 Prechamber Design

Chambers are divided into two parts, which the main chamber is located between piston and cylinder head and the other one is prechamber that located at the cylinder head. Fuel is injected into prechamber and under full-load condition sufficient air to complete combustion not present in this chamber. These prechambers are usually small in size and are used as an ignition assist for very lean mixtures and to increase the combustion rate of large bore engines. The prechamber was located in the current spark plug hole location, with the spark plug cavity forming the prechamber chamber. The prechamber nozzle was mounted in the original spark plug hole.

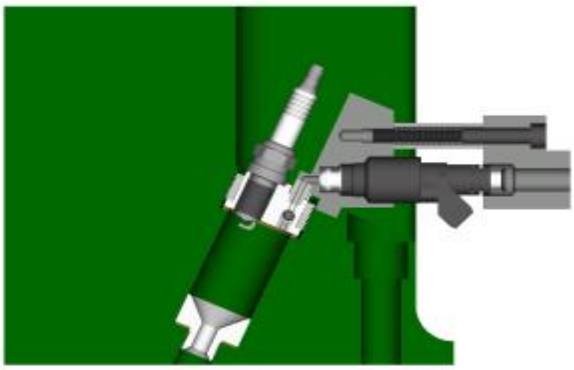


Figure 2.8: Example of cutaway of prechamber design.

Source: John Kubesh 2002

The prechamber volume is equal to 20% of the total combustion chamber clearance volume, with one or more outlets leading to the main chamber. This volume large enough to contain sufficient fuel-air mixture to operate the engine at light loads over the speed range of engine. The prechamber volume also should be 20% or less for good results in terms of emissions and fuel economy (Gruden, 1976).

2.3.2 Advantages and Disadvantages of Precombustion Chamber

There are many advantages of the prechamber combustion such as higher speed and brake mean effective pressure and the power with less smoke are feasible. Since the nozzle is at the side, there is no more room for the larger intake and exhaust valves, so volumetric efficiency will be higher. Using the prechamber, mechanical stress and noise become less because of the lower rate of pressure rise and the lower maximum pressure in the main chamber due to the throttling effect of throat. The engine will become smoother and quieter idling with precombustion chamber. And the most important that using precombustion chamber is less air pollution because the cleaner of exhaust.

Besides the advantage, this process also has disadvantages. The disadvantages of prechamber is higher specific fuel consumption that will resulting in poorer fuel economy. It is because of greater heat losses and pressure losses through the throat which result in lower thermal efficiency and higher pumping losses. The flow of combustion gases through the throat leads to thermal cracks in the cylinder head and creates sealing problems. When precombustion occurs, more thermal energy is lost to the exhaust gases. So, it may decrease the life of the exhaust valve which will run hotter and increase cracking and sealing problems of the exhaust manifold.

2.3.3 Sequence of Precombustion Chamber Process

A precombustion chamber is connected to the main combustion chamber by spark plug hole. There are several steps occur during the combustion process.

During the compression stroke of the engine, air is forced into the precombustion chamber and the air become hot because its was compressed. At the beginning of injection , the precombustion chamber contains a definite volume of air.

As the injection begins, combustion starts in the precombustion chamber. The burning of the fuel, combined with the restricted passage to the main combustion chamber. It creates a large amount of pressure in the precombustion chamber. The pressure and the initial combustion cause a heated fuel charge to enter the main combustion at high velocity.

The mixture enter the hollow at the top of piston and creating turbulance in the chamber to make sure the mixing of fuel charge with the air is completed. The chamber design will provide satisfactory performance with low fuel injector pressures and gross spray patterns because a large amount of vaporization takes place in the combustion chamber.

2.4 DIRECT INJECTION STRATEGY

Direct injection is important to assist the combustion in main chamber or prechamber. In combustion process, fuel injection must control to get perfect combustion in combustion chamber.

2.4.1 Ignition Timing and Control

Ignition timing is produce by the spark timing when the spark is ignited towards the TDC, the compression process at the point of ignition is covered more (Gupta, 2006). From this process, there are increasing in temperature and pressure, and therefore the ignition lag reduces. Advancing the spark timing increases the ignition lag.

From the previous research, to investigate the particle emission and combustion characteristics under diffrent spark timing. The ignition timing was varied in increments of 4º CA from 24º CA BTDC to 40º CA BTDC (Liu et al., 2010). From this research, the initial combustion duration increase when the spark timing advance. As the spark timing slowed, the initial in-cylinder temperature and pressure increase because more compression work. This effects promotes the formation of the flame kernel and thus shortens the initial combustion duration.

2.4.2 Injection Timing and Control

Injection timing is refer to the timing of fuel injector to inject the fuel into combustion chamber. This process will effect the combustion duration such as the initial combustion duration decrease linearly as a fuel injection is advanced. When the fuel injection is slow, the initial combustion duration is expected to be shortened owing to the increased degree of charge stratification and the intensity of turbulance (Huang et al., 2006).

The behaviour of CNG fuel injected is going down and produces certain spray angle i.e. 70 degree from the detection of spray pattern. At the time of 240º CA, which is 10º CA after fuel injection begins, the spray formation creates some swirl structures within cylinder as required to be mix with the air. During the late part of the fuel injection process, the fuel within combustion chamber nearly touches the top surface of piston before it is compressed until the spark ignition starts.

2.5 CFD SIMULATION

Computational fluid dynamics (CFD) is the science of predicting fluid flow, heat and mass transfer, chemical reactions, and related phenomena by solving numerically the set of governing mathematical equations. It is a computer-based mathematical modeling tool that incorporates the solution of the fundamental equations of fluid flow, the Navier-Stokes equations, and other allied equations. CFD incorporates empirical models for modelling turbulence based on experimentation, as well as the solution of heat, mass and other transport and field equations. In order to done the calculations, computers are used to compute such task by using specific software that allows complex calculation for simulation of intended flow process. There are three phases to CFD: pre-processing, or creation of a geometry usually done in a CAD tool; mesh generation of a suitable computational domain to solve the flow equations on; and solving with post processing, or visualization of a CFD code's predictions.

2.5.1 Turbulence Model

The k-ε two-equation turbulence model is widely used in CFD application. The k-ε model was proposed by Launder and Spalding (1974). The entire constant used in the implements of the RNG k-ε turbulence model, the enthalpy equation and the calculation of the effective viscosity are the same as starting in the article turbulance modeling of internal combustion engines using RNG k-ε models (Han et.al, 1995).

CHAPTER 3

METHODOLOGIES

3.1 INTRODUCTION

This chapter presents the main outline of the study which contains the engine baseline specification, important parameters, initial condition, boundary condition, and numerical modelling approach in cold flow and reacting flow.

3.2 FLOW CHART FOR METHODOLOGIES

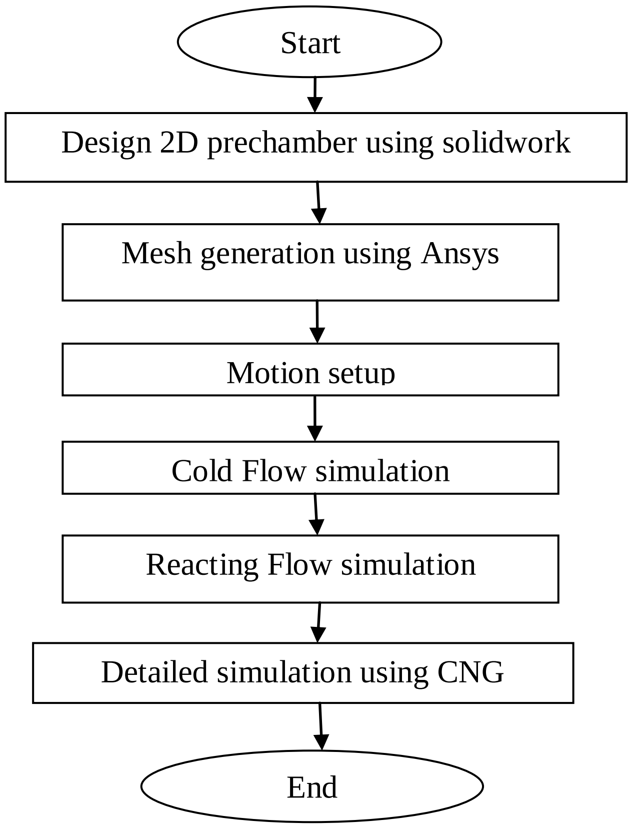


Figure 3.1: Flow chart process for methodology.

3.3 BASELINE ENGINE SPECIFICATION

A single cylinder Yamaha FZ150i 1.5 liter engine is selected for the baseline engine. The engine specification is remained unchanged except the original injector and spark plug position. The position of the spark plug is replaced by prechamber. At the prechamber, fuel injector and spark plug are positioned. Table 3.1 listed the details of the engine specifications. From the table, the other important key to developing the computational domain is the compression ratio. The compression ratio is the most important parameter to be fixed.

Table 3.1 : Engine specification Yamaha FZ150i

Source: Engine Specification Data, (2010)

Parameter	**Size and Feature**
Valve train type	4 strokes, SOHC
Number of cylinder and valve per cylinder	1 cylinder with 4 valves (two intake valve and two exhaust valves)
Combustion chamber type	Pent-roof type
Total displacement (cm^3)	1,498
Cylinder bore (mm)	57
Piston stroke (mm)	58.7
Compression ratio	10.4
Intake valves open/close	29º BTDC/59º ABDC
Exhaust valves open/close	59º BBDC/29º ATDC
Maximum power	11.1 kW@8500 rpm
Maximum Torque	13.1 Nm@7500 rpm
Connecting rod (mm)	117

3.4 TRANSIENT ENGINE MODELING

The engine modelling is started from the design the prechamber model in Solidwork software. This model refers to the engine specification in Table 3.1. The volume of prechamber is considered 2% off clearance volume at main chamber. Below is the calculation of volume prechamber that calculate from clearance volume:

From the table 3.1:

Cylinder bore = 57 mm

Piston stroke = 58.7 mm

Compression ratio, C_R = 10.4

Swept volume, $V_s = \Pi \times bore^2 \times stroke$ (3.1)

$$= \Pi \times (\frac{57}{2})^2 \times 58.7$$

$$= 149,788.23 \ mm^3.$$

$$\text{Compression Ratio, } C_R = \frac{swept\ \ volume\ ,Vs + clearance\ \ \ volume\ ,Vc}{clearance\ \ \ volume\ ,Vc} \quad (3.2)$$

$$\text{Clearance volume, } V_c = \frac{swept\ \ volume\ ,Vs}{Compression\ \ \ ratio\ ,CR - 1} \quad (3.3)$$

$$= \frac{149{,}788.23 \text{ mm } 3}{10.4 - 1}$$

$$= 15934.92 \ mm^3.$$

Volume of Prechamber = 2% x 15934.92 mm^3

= 318.698 mm^3

Prechamber is designed based on the volume that has been calculated. The injector is located at the top of prechamber while spark plug is located at the right side of prechamber. In this case, injector that used is HP injection valve (HDEV 5) manufactured by Bosch.

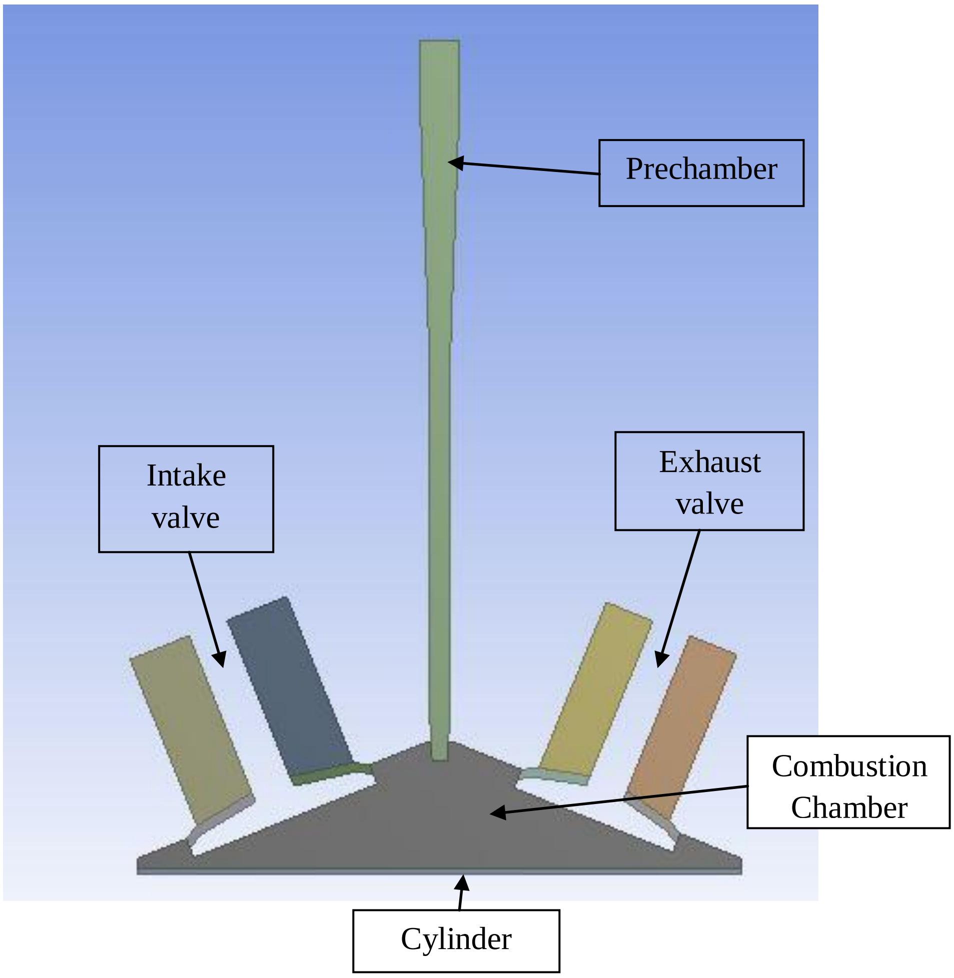


Figure 3.2: 2-D Engine modeling using SOLIDWORK.

3.4.1 Mesh Genaration

The model is the imported to Ansys to generate the mesh and change their condition as a fluid. This model is meshed using triangles and uniform quad mesh method. In Ansys software, each edge is defined in order to setup the dynamic mesh. In dynamic mesh, the model is divided into three groups which are moving rigid body, deforming and stationary.

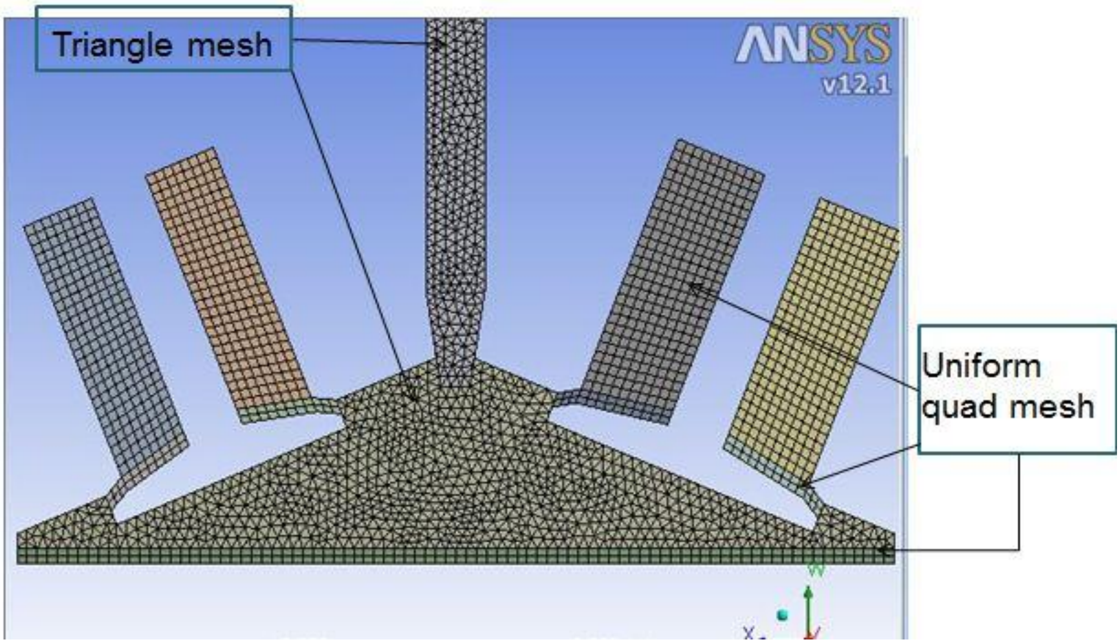


Figure 3.3: Mesh geometry model

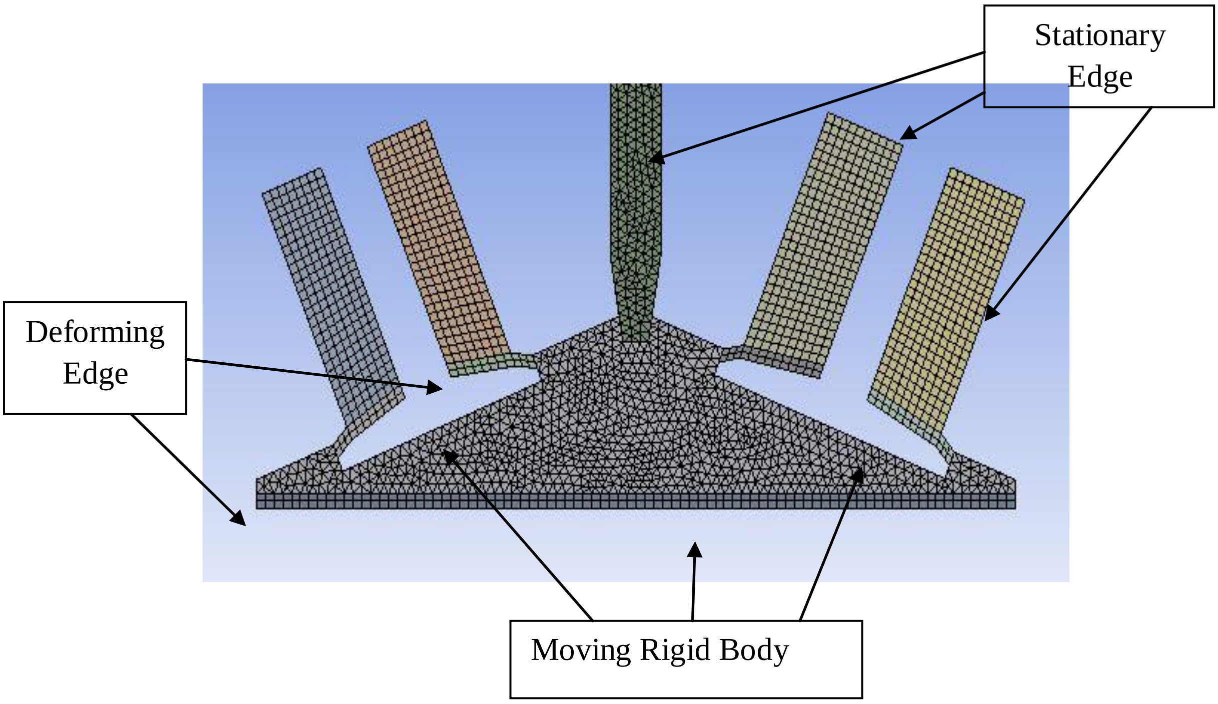


Figure 3.4: Detailed of dynamic mesh

3.5 CFD SIMULATION USING FLUENT

In CFD approach, the model is simuated in two type which are cold flow simulation and reacting flow simulation. In FLUENT simulation, partial differential equations of flow variables is used to calculate and to simulate the kind of analysis concerning the fluid flow. Among of them are energy conservations and quantities of turbulence. FLUENT is also able to predict heat transfer in periodically repeating geometries, thus greatly reducing the required computational effort in certain cases.

3.5.1 Turbulence Model

Turbulance model is unsteady or irregular motion in which transported quantity (mass, momentum, scalar species) fluctuate in time and space. Turbulance model adopts *k*-*ε* turbulance model:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial X_i}(\rho k u_i) = \frac{\partial}{\partial X_j}[(\mu + \frac{\mu_t}{\sigma_k}) \frac{\partial k}{\partial X_j}] + G_k + G_b - \rho\varepsilon - Y_m + S_k \tag{3.4}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial X_i}(\rho\varepsilon u_i) = \frac{\partial}{\partial X_i}[(\mu + \frac{\mu_t}{\sigma_\varepsilon}) \frac{\partial \varepsilon}{\partial X_j}] + C_{1\varepsilon}\frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} + S_\varepsilon \tag{3.5}$$

Where;

G_k = engender item of rapid kinetic energy karosed by average speed grads.

G_b = engender item of rapidkinetic energy,*k* arosed by flotage.

Y_M = the effect ofcompressible onflow pulse expand on the whole dissipationrate.

α_k= 1.0

α_k= 1.3

On flow viscidity degree,

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{3.6}$$

Where, $C_{1\varepsilon}$= 1.44, $C_{2\varepsilon}$= 1.92, $C_{3\varepsilon}$= 1,C_μ= 0.09

3.5.2 Combustion Model

The combustion model was combined with species transport and finite rate chemistry with simplified chemistry reactions to simulate the overall combustion process in a CNG engine.This approach is based on the solution of transport equations for species mass fractions. The reaction rates that emerge as source terms in the species transport equations are computed from well known Arrhenius rate expressions. A chemical kinetic mechanism from the FLUENT database is used for modeling CNG combustion. For chemical species, local mass fraction of each species, Y_i and through the solution of a convection diffusion equation for the i^{th} species can be used to solve conservation equation. This conservation equation takes the following general form;

$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla.(\rho \underset{v}{\rightarrow} Y_i) = \nabla.J_i + R_i + S_i \tag{3.7}$$

where R_i is the rate of production species $_i$ by chemical reaction and S_i is the rate of creation by addition from dispered phase.

3.6 BOUNDARY CONDITION SETUP

The all edges of engine geometry are set as walls. The materials for all walls are aluminium. The boundary condition that will be specified is the wall temperature. At the point of injection is set the pressure of injector and state the gap of spark plug for the ignition. Parameter that will be set include the engine speed, load of engine, pressure, temperature and ignition duration. Initial condition for this simulation are setup in parameter of ignition timing and injection timing. Other than that, the pressure for both intake and exhaust manifold are assumed as same as ambient pressure which 101 kPA and all the temperature is defined 300 K same as the romm tempreature. Table 3.2 is the data for initial condition of simulation.

Table 3.2: Operating condition of simulation

Attribute	Value
RPM	2000
Pressure (kPA)	101
Temperature (K)	300
Load (kN)	0

3.7 NUMERICAL SETUP

3.7.1 Injection Timing Setup

In this study, CNG is injected into the prechamber before going down to the main chamber during the compression stroke. The injector is defined as single hole injector. Based on the manufacturer, the maximum injection pressure is 200 bar while the operating temperature is 70 ℃. This injection process is set in different timing while the duration of injection is set contant for 15° CA. Different injection timing are used 40° CA BTDC as Case 1, 50° CA BTDC as Case 2and 60° CA BTDC as Case 3 in compression process.

Table 3.3 : Injection condition

Injection Condition	Value
Pressure	20 bar
Temperature	300K
Crank angle duration	15°

3.7.2 Ignition Timing Setup

For this setup, ignition timing is set as a variable parameter. This timing refer to the degree of crank angle. For this case, ignition timing are set at 40° CA BTDC as Case 1, 30° CA BTDC as Case 2, and 20° CA BTDC as Case 3. Another parameter for ignition setup is stated at table 3.4.

Table 3.4: Ignition Condition

Ignition Condition	Value
Ignition Point (mm)	X=-0.2564042 Y=116.7132
Radius of ignition (mm)	0.4538065
Ignition duration (s)	0.001

3.7.3 Combustion Setup

The combustion model for this simulation is Eddy-Dissipation as the flow is turbulence. This combustion start during the compression stroke after the intake valve close.

This combustion range start from the injection process and the spark plug ignited until all of fuel in cyinder is burned. The important of this event to investigate the pressure and temperature inside the combustion chamber and cylinder with different injection and ignition timing. The initial value before combustion is set based on the cold flow simulation at the 60° CA BTDC.

3.7.4 Event Definition

Table 3.5: Event of full crank angle in single cycle

Source: Specification Data Yamaha, (2010)

Crank Angle	Events
-360°	Start of intake process (intake and exhaust valves already open at TDC
-331°	Intake process (exhaust valve closed)
-180°	End of intake process / Start compression stroke
-121°	Compression process(intake valve closed)
-50°	Injection of fuel process
-25°	Ignition timing(2000rpm)
0°	End of compression / Start of power stroke
121°	Power stroke (exhaust valve opened)
180°	End of power stroke/ Start of exhaust stroke
331°	Exhaust stroke (intake valve opened)
360°	End of single cycle

Table 3.5 represented the full crank angle event based on the single cylinder engine. The important event that considered in simulation are intake process, compression process, injection process, ignition timing, power stroke and the last event is exhaust stroke. The valve is overlapping which is only 58° between intake and exhaust stroke. The overlapping period for the both valves is defined from -331° CA to 331° CA as in the actual condition.

3.8 Limitation of study

Since the study only based on the simulation using CFD approach, some of the parameters have to be assumed due to lack of exact data. Besides, the model engine has to be simplified because of the limitation of computer processor. The model that simplified will reduce the difficulties during problem setup, meshing process, and computing time for run the simulation. The mesh size of model is considered in average size. The smaller size of mesh will produce the better result but will increase the computing time.

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

This chapter presents the results and discussion on the CFD simulation. The presented results are discussed the contour of intake jet flow in terms of velocity magnitude , the contour of injection process in terms of temperature and species mass fraction, contour of ignition process in terms of temperature and species mass fraction, and focused to the temperature and pressure difference ignition and injection timing.

4.2 CONTOUR OF INTAKE JET FLOW

From the simulation, the contour of intake jet flow in terms of velocity magnitude represented. When the piston at TDC move downward, intake stroke begins. In figure 4.1 (a) the intake valve start to open and the air forced enters into the cylinder due to the pressure between ambient and vacuum condition inside the cylinder. Figure 4.1 (b) until figure 4.1 (f) show that the flow of air at the intake valve enter to the cylinder with the increased speed. Flow speed increased is directly proportional to the movement of the piston. The average velocity during the intake process at 2000 rpm is 14.58 ms^{-1} and the maximum velocity is occured at the opening intake valve region.

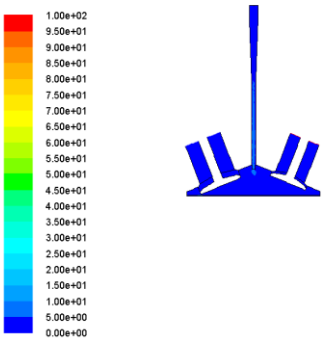


Figure 4.1 (a) : CA = 10° ATDC

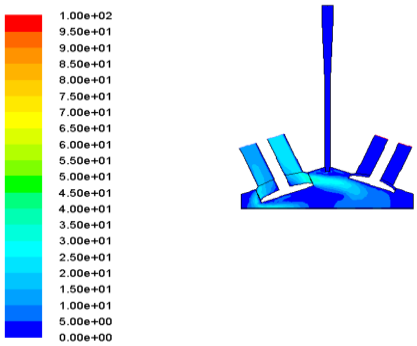


Figure 4.1 (b) : CA = 30° ATDC

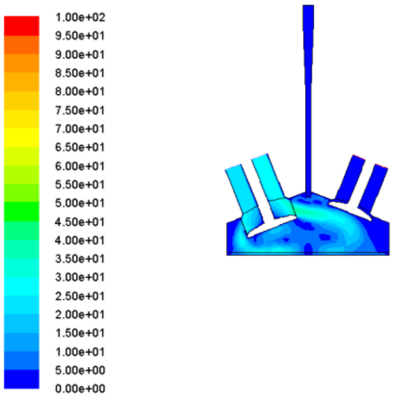


Figure 4.1 (c) : CA = 45° ATDC

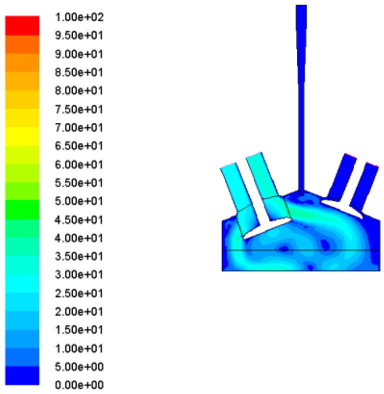


Figure 4.1 (d) : CA = 60° ATDC

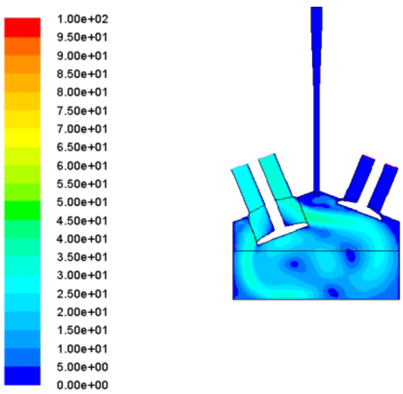


Figure 4.1 (e) : CA = 80° ATDC

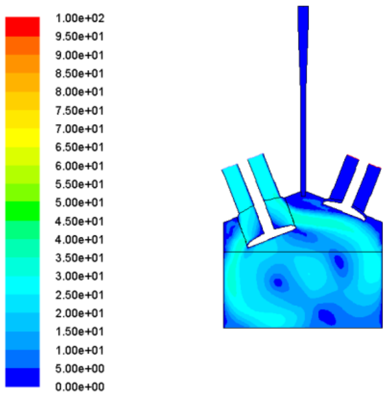


Figure 4.1 (f) : CA = 100° ATDC

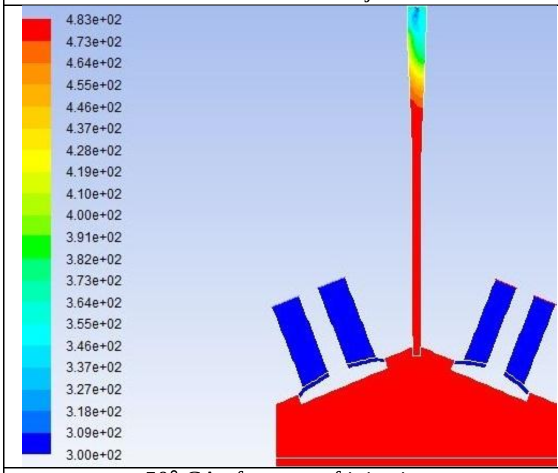
Figure 4.1 (a)-(f) : Contour of velocity magnitude at intake process

4.3 PROGRESSIVE COMBUSTION VISUALIZATION

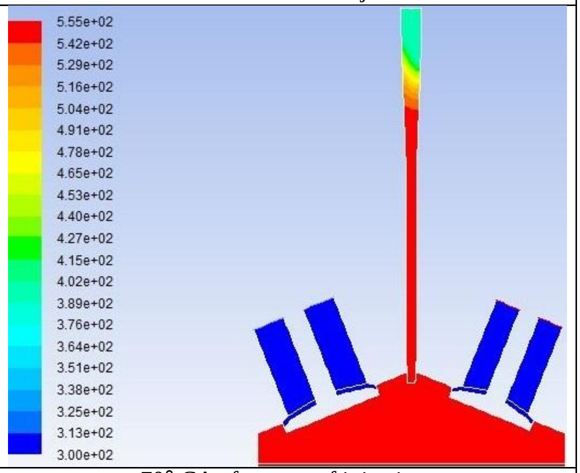
Table 4.1 : Contour of temperature in different injection timing

Case 1 : Injection Timing at 60 ° BTDC

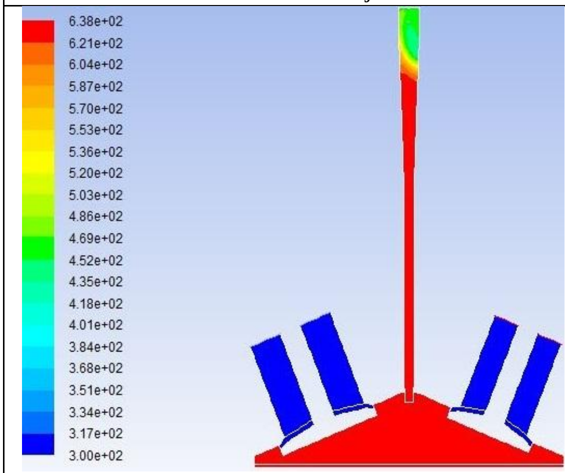
10° CA after start of injection



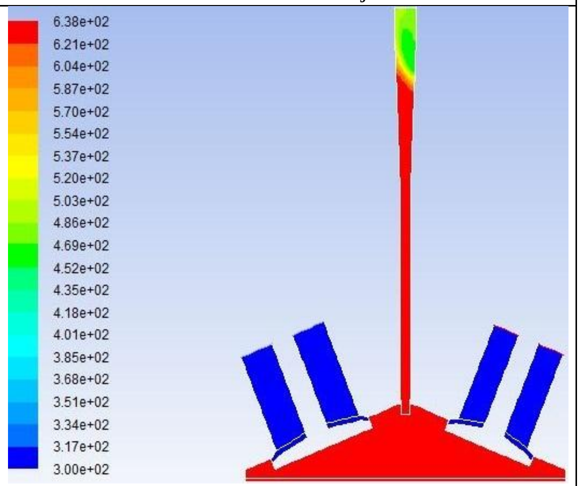
30° CA after start of injection



50° CA after start of injection



70° CA after start of injection



100° CA after start of injection

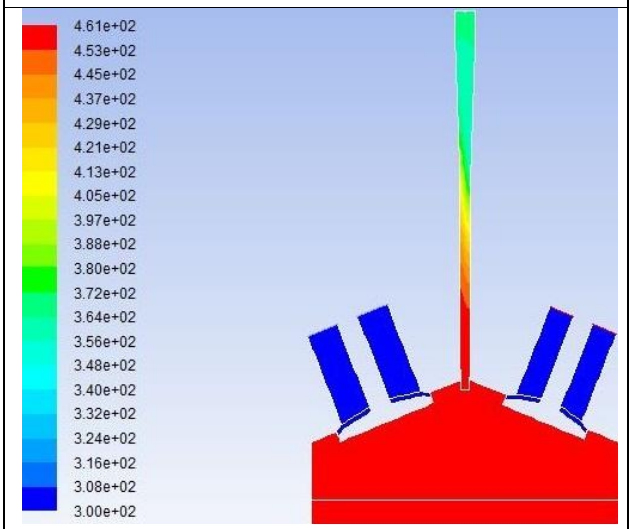
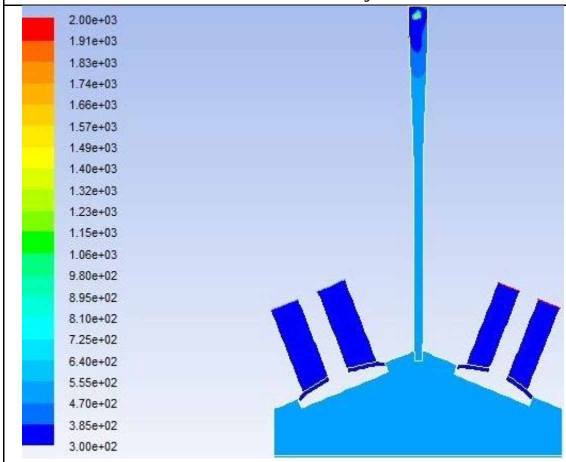


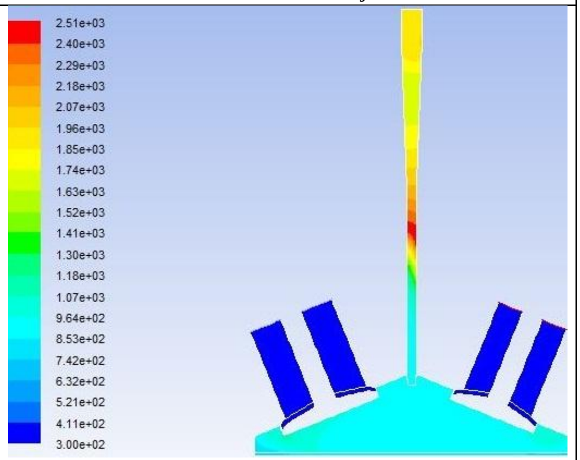
Table 4.1 : Contour of temperature in different injection timing(Cont.)

Case 2 : Injection Timing at 50 ° BTDC

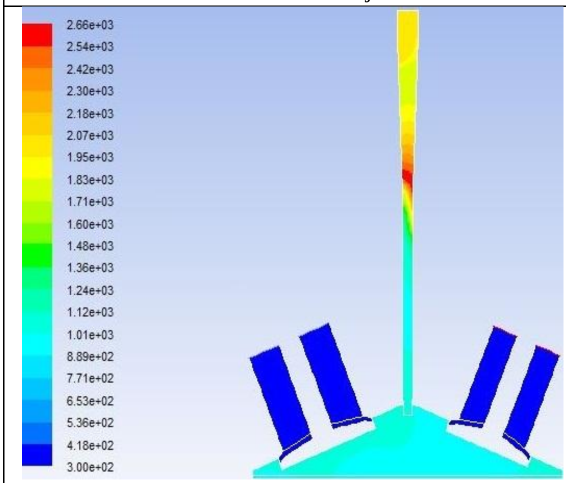
10° CA after start of injection



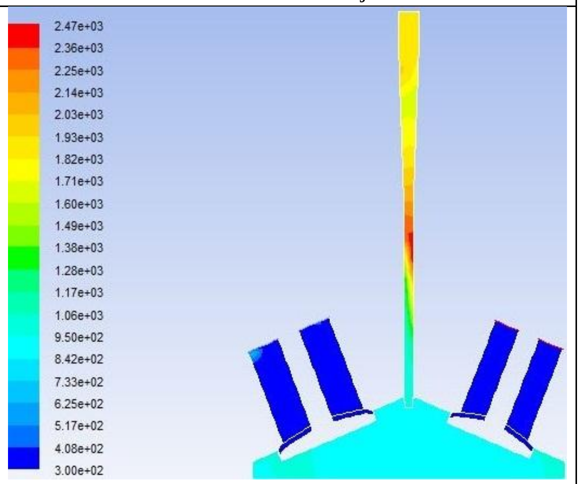
30° CA after start of injection



50° CA after start of injection



70° CA after start of injection



100° CA after start of injection

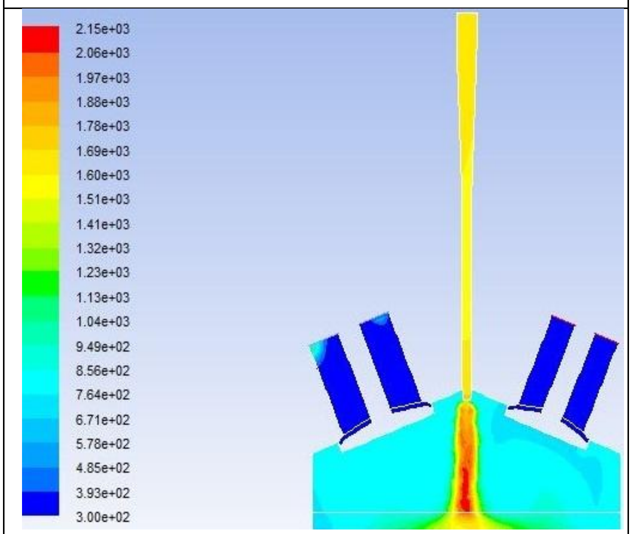
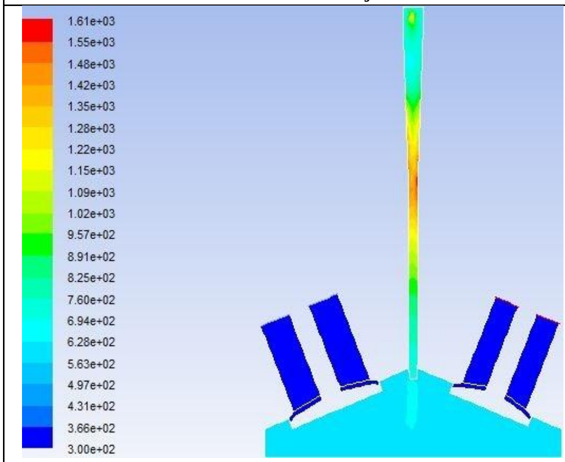
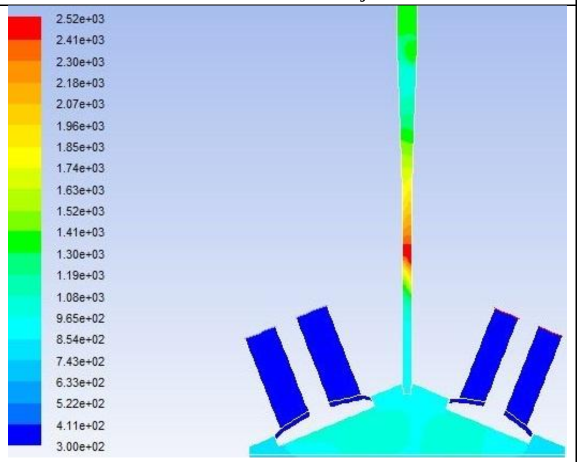
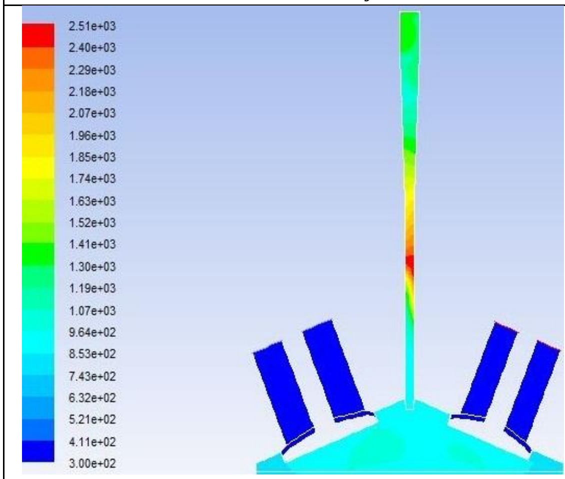
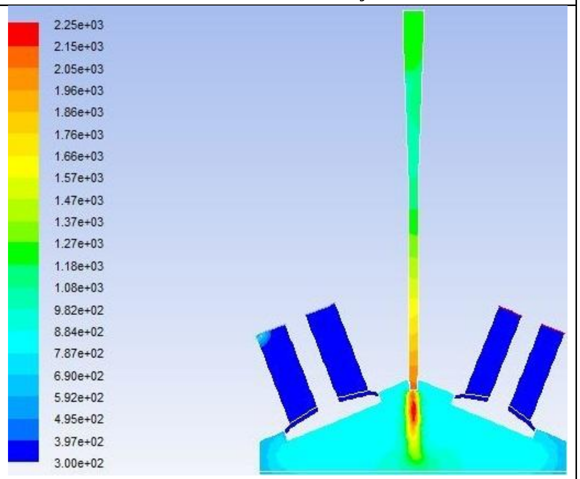
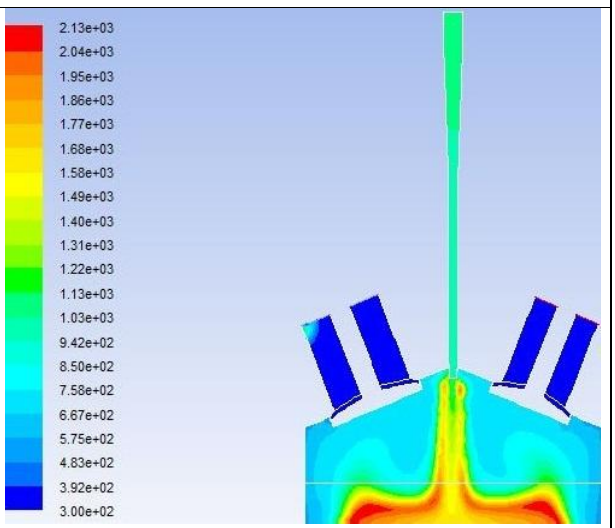


Table 4.1 : Contour of temperature in different injection timing(Cont.)

Case 3 : Injection Timing at 40 ° BTDC	
10° CA after start of injection	30° CA after start of injection
	
50° CA after start of injection	70° CA after start of injection
	

100° CA after start of injection


Based on figure 4.1, the contour of temperature at different injection timing showed that the range of temperature is 300 K to 2470 K. Refer to case 1 which the injection start at 60° CA BTDC, the temperature start to rise from 483 K and the maximum temperature is 638 K. At this temperature, maximum at the main cylinder within the temperature at the prechamber is lower than the main cylinder. Temperature in main cylinder becomes hot when start the compression stroke until the piston reach to TDC.

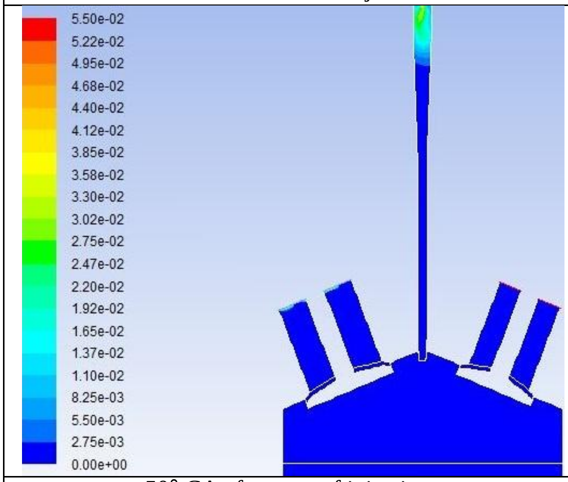
Refer to the case 2, temperature in prechamber is higher than combustion chamber compare with the situation in case 1. The temperature increased to 2000 K during combustion process. It showed that the spark plug properly ignited after 10° CA start of injection. Temperature in this case increased until 2660 K at the TDC. At 100° CA after start of injection the contour showed that the hot temperature is transfered to the combustion chamber.

Case 3, showed that the hottest temperature is 1600 K located at the prechamber. After the cylinder reached at the TDC, temperature increased to the 2520 K. Compare with case 2, temperature in combustion chamber become hot than prechamber at 70° CA after start of injection

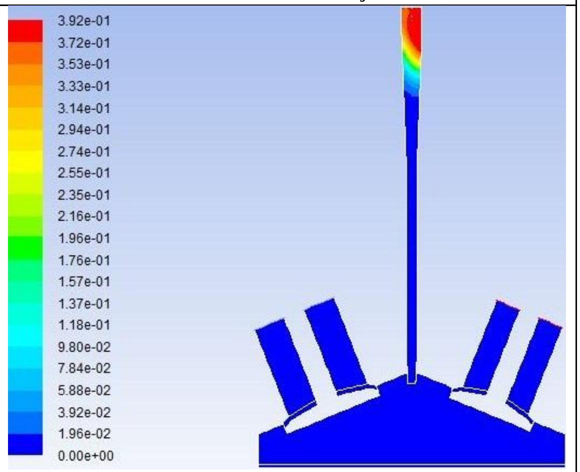
Table 4.2 : Contour of methane mass fraction in different injection timing

Case 1 : Injection Timing at 60 ° BTDC

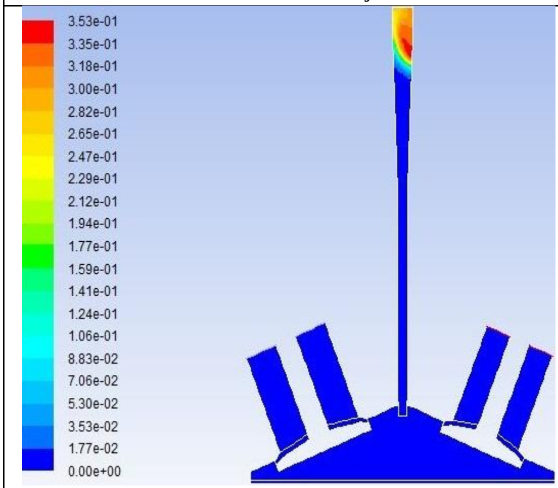
10° CA after start of injection



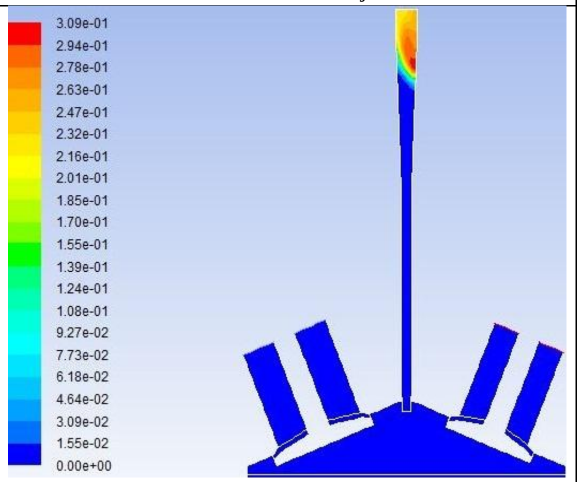
30° CA after start of injection



50° CA after start of injection



70° CA after start of injection



100° CA after start of injection

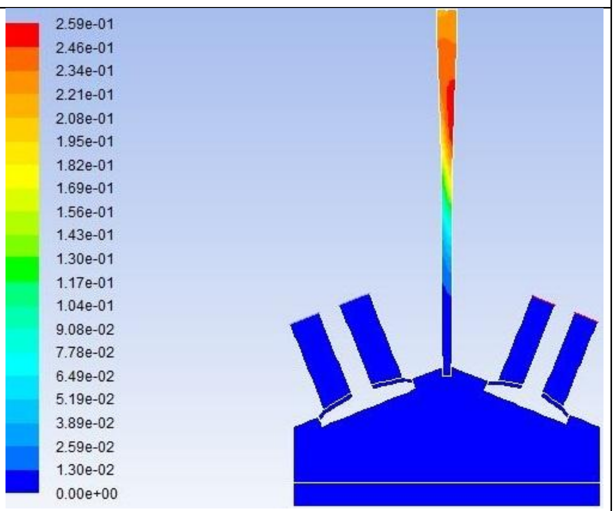
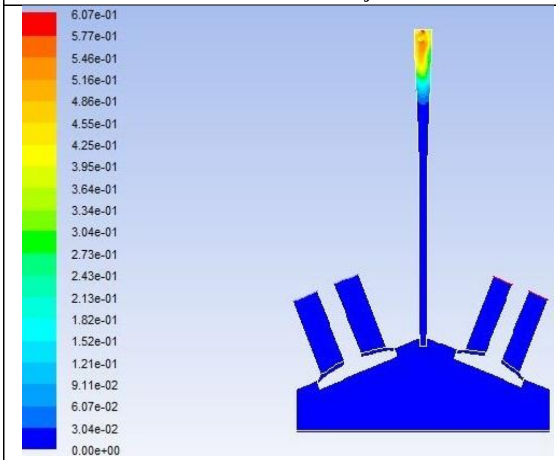
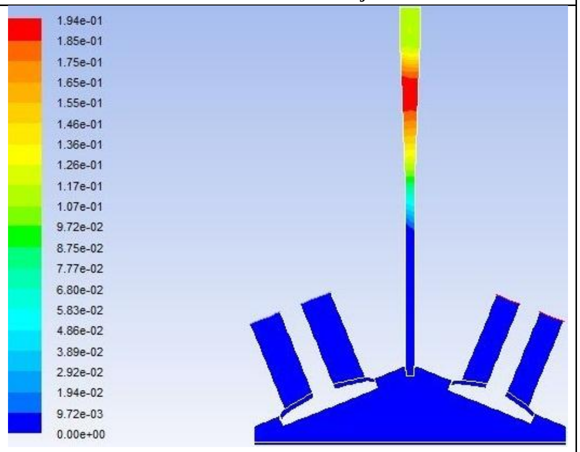


Table 4.2 : Contour of methane mass fraction in different injection timing(Cont.)

<table>
<tr><th colspan="2">Case 2 : Injection Timing at 50 ° BTDC</th></tr>
<tr><td>10° CA after start of injection</td><td>30° CA after start of injection</td></tr>
<tr><td>
</td><td>
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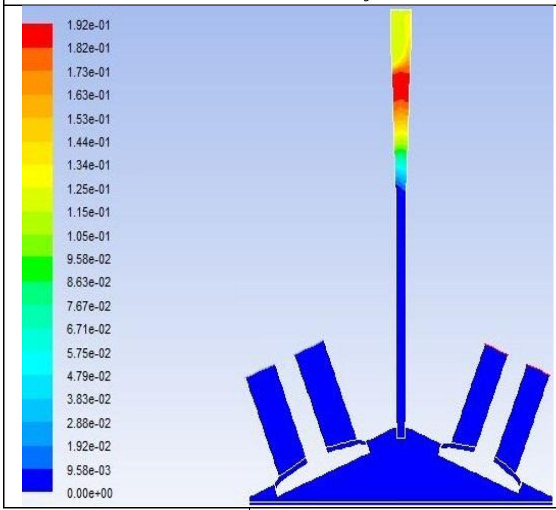
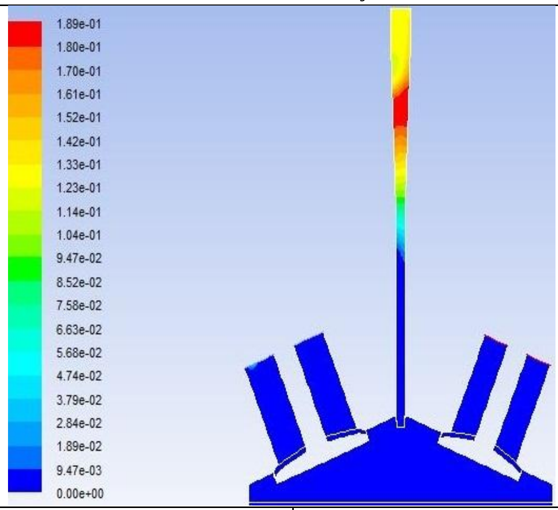
<tr><td>50° CA after start of injection</td><td>70° CA after start of injection</td></tr>
<tr><td>
</td><td>
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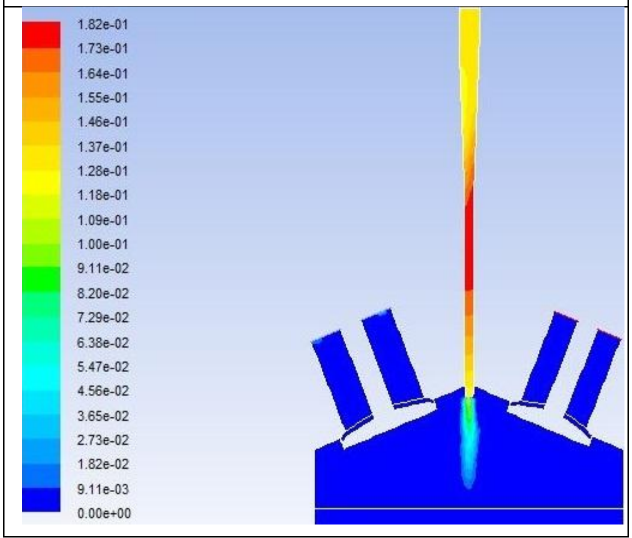
<table>
<tr><td>100° CA after start of injection</td></tr>
<tr><td>
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Table 4.2 : Contour of methane mass fraction in different injection timing(Cont.)

<table>
<tr><th colspan="2">Case 3 : Injection Timing at 40 ° BTDC</th></tr>
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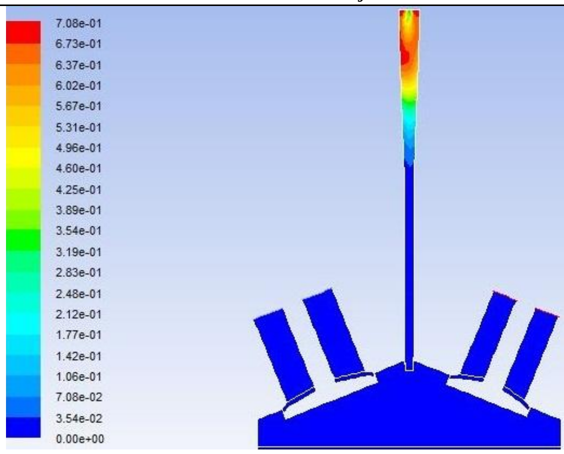
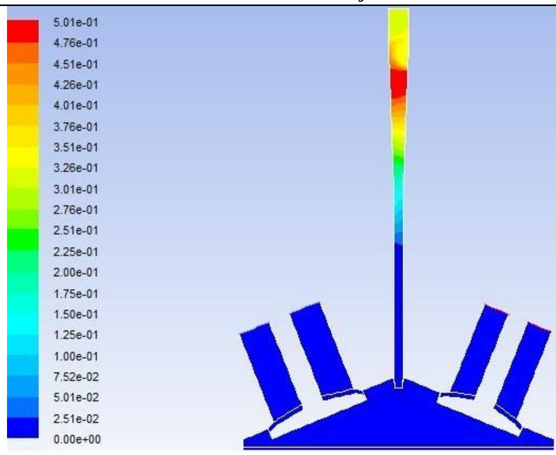
<tr><td>
</td><td>
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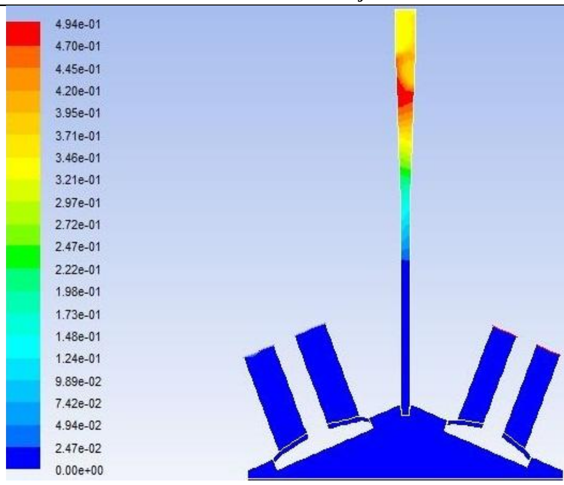
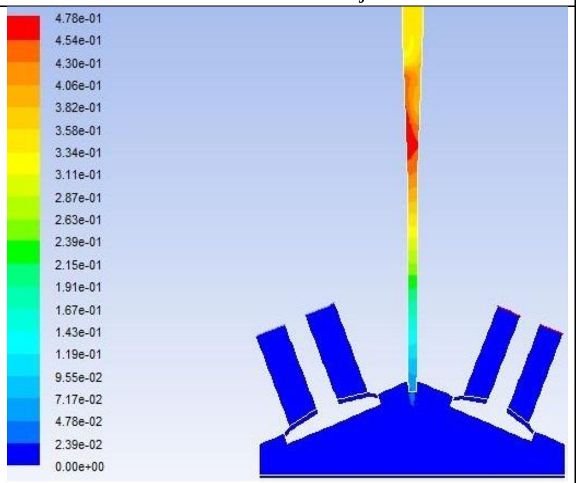
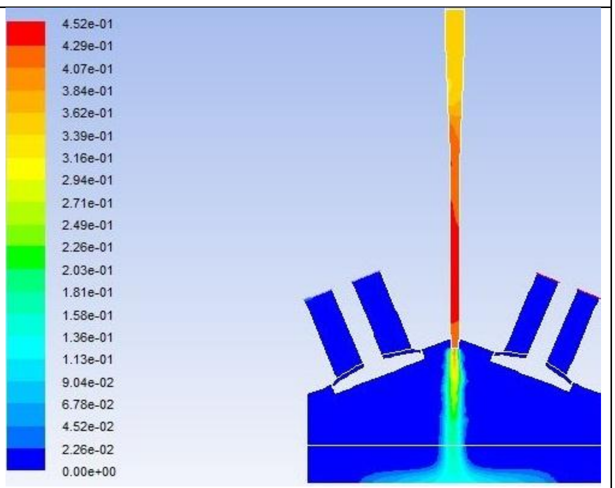
<tr><td>50° CA after start of injection</td><td>70° CA after start of injection</td></tr>
<tr><td>
</td><td>
</td></tr>
<tr><td colspan="2">100° CA after start of injection</td></tr>
<tr><td colspan="2">
</td></tr>
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In table 4.2, contour of the methane mass fraction is shown at different injection timing. The contour is representing the mass fraction of methane that exist in the model. For the case 1, that injection process methane exist in prechamber is 0.55. Mass fraction of methane become decreased after reaction process occur during combustion process. After the combustion process, methane still exist in prechamber based on the amount of methane mass fraction in range 0.38 until 0.295.

Case 2 showed that the amount of methane mass fraction is 0.66. At this time, the injection process is still occured. Case 2 has a smaller amount of methane mass fraction. The amount of methane mass fraction in range 0.185 until 0.18 after 50° CA after start the injection process. The contour showed methane is started going down to the combustion chamber after 100° CA.

In case 3, methane has 0.708 of mass fraction after start the injection. Mass fraction for this methane become smaller to the 0.45 but it still greater compare with case 1 and case 2. It shows an incomplete combustion occured for this case.

Table 4.3 : Contour of temperature in different ignition timing

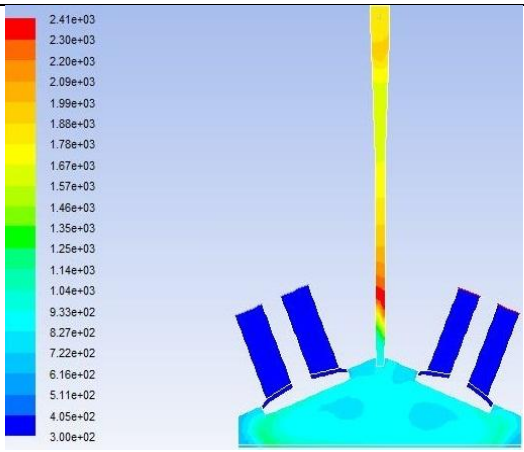
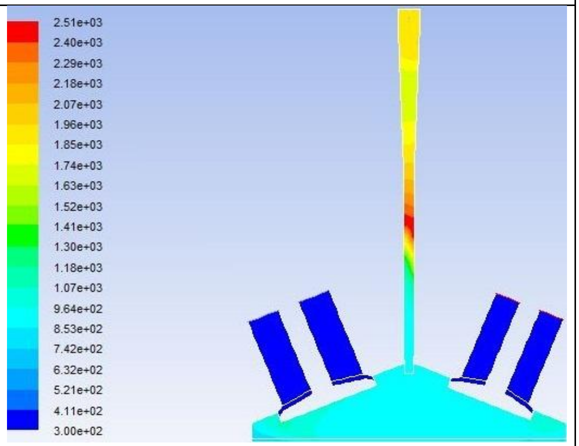
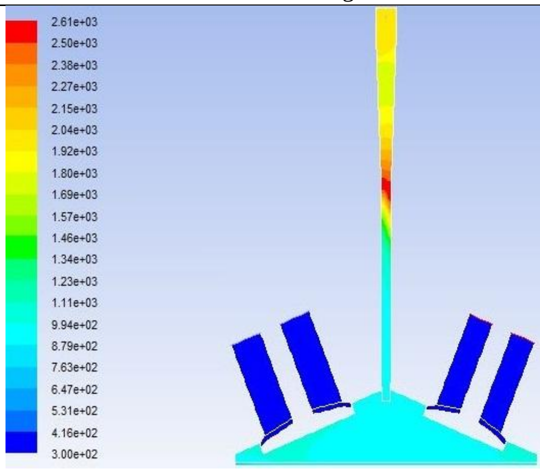
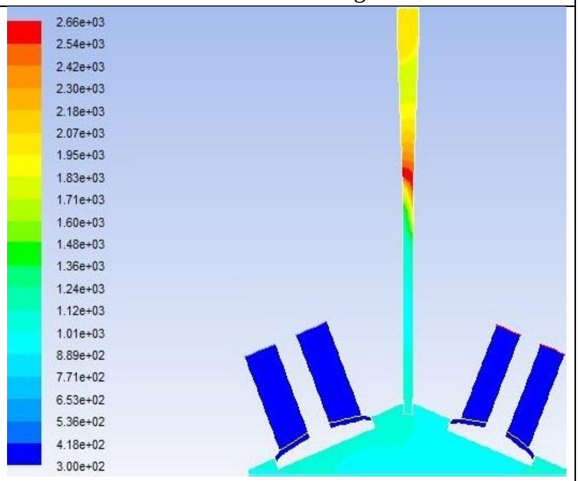
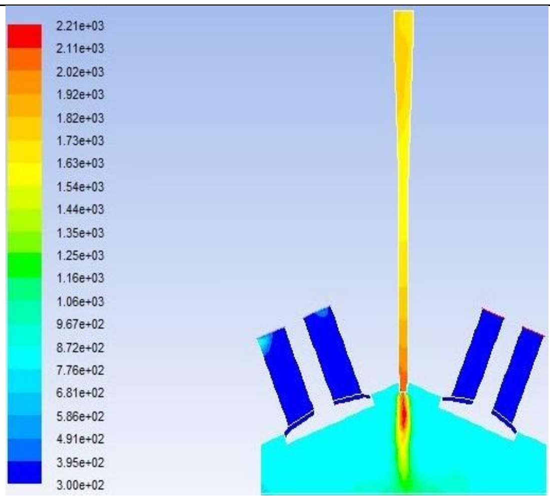
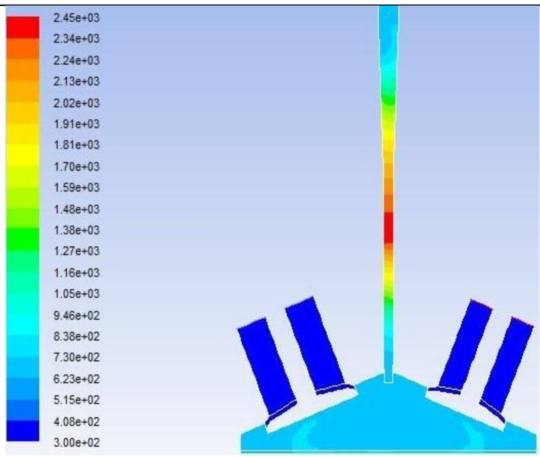
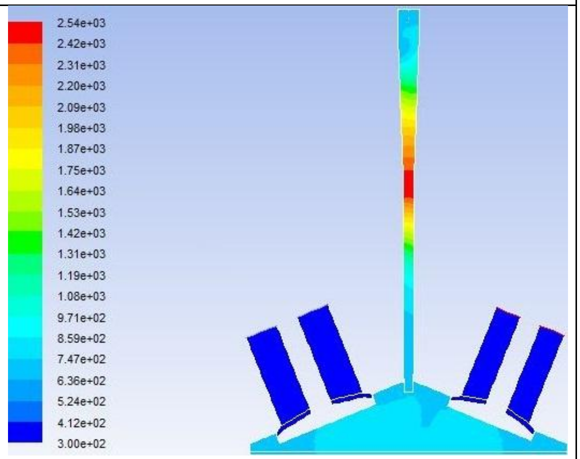
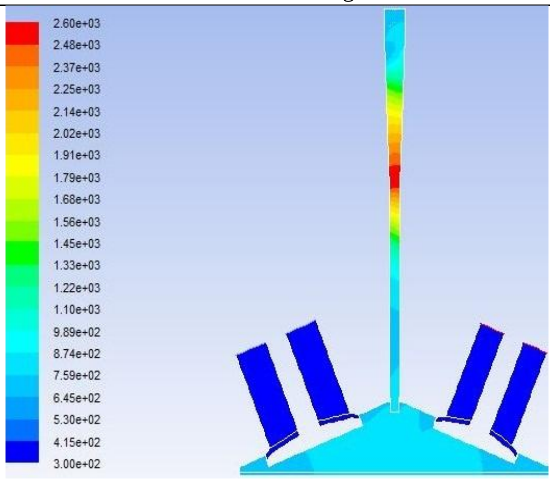
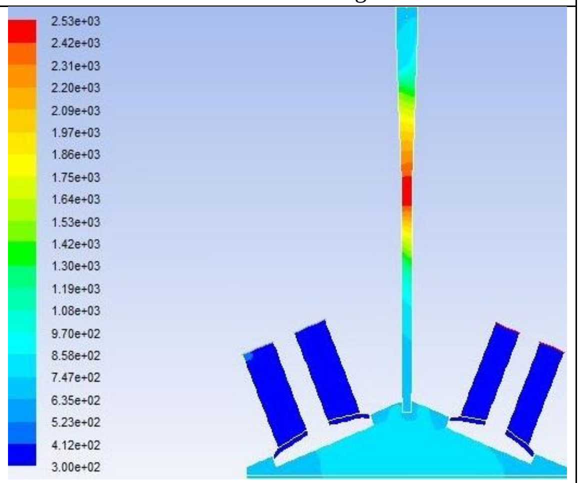
<table>
<tr><th colspan="2">Case 1 : Ignition Timing at 40 ° BTDC</th></tr>
<tr><td>10° CA after start of ignition</td><td>20° CA after start of ignition</td></tr>
<tr><td>
</td><td>
</td></tr>
<tr><td>30° CA after start of ignition</td><td>40° CA after start of ignition</td></tr>
<tr><td>
</td><td>
</td></tr>
<tr><td colspan="2">50° CA after start of ignition</td></tr>
<tr><td colspan="2">
</td></tr>
</table>

Table 4.3 : Contour of temperature in different ignition timing(Cont.)

Case 2 : Ignition Timing at 30 ° BTDC	
10° CA after start of ignition	20° CA after start of ignition
	
30° CA after start of ignition	40° CA after start of ignition
	

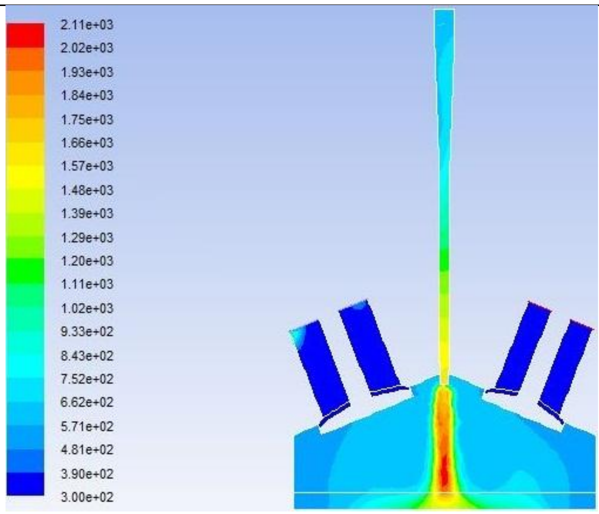
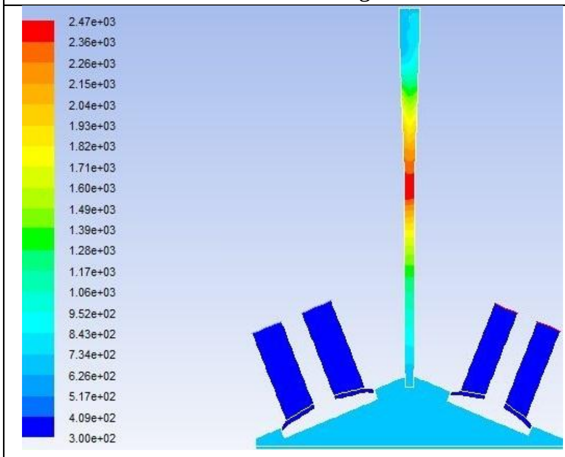
50° CA after start of ignition


Table 4.3 : Contour of temperature in different ignition timing(Cont.)

<table>
<tr><th colspan="2">Case 3 : Ignition Timing at 20 ° BTDC</th></tr>
<tr><td>10° CA after start of ignition</td><td>20° CA after start of ignition</td></tr>
<tr><td>


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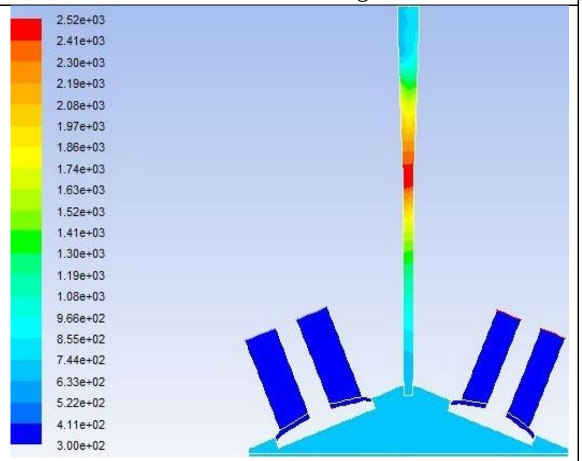
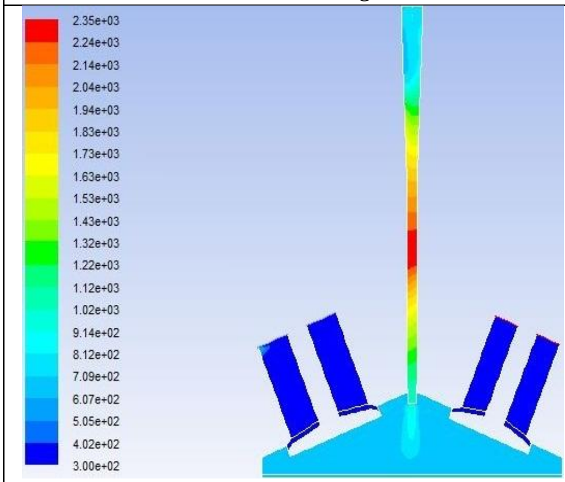
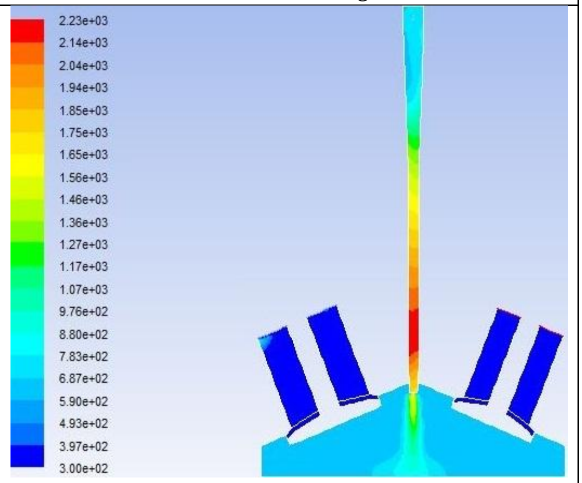
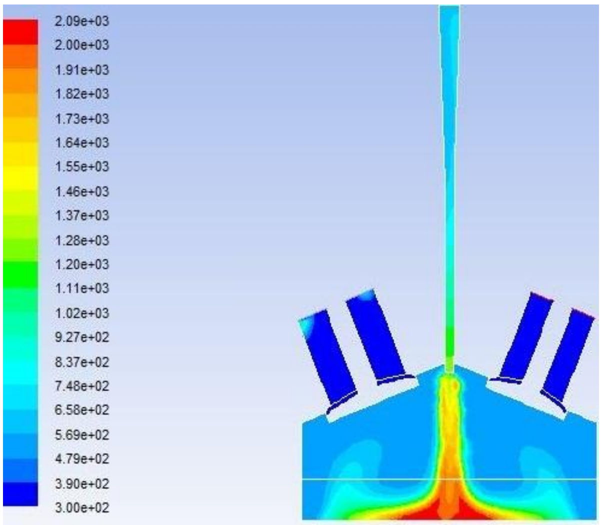


</td></tr>
<tr><td>30° CA after start of ignition</td><td>40° CA after start of ignition</td></tr>
<tr><td>


</td><td>


</td></tr>
<tr><td colspan="2">50° CA after start of ignition</td></tr>
<tr><td colspan="2">


</td></tr>
</table>

Table 4.3 above showed that the temperature at different ignition timing. In case 1 has shown the temperature range from 2610 K to 2210 K. It means that the temperature increased at the compression stroke and decreased after start the expansion stroke. The maximum temperature of case 1 is 2610 K at TDC. The hot temperature is located at the prechamber because the ignition process is occurring in this process.

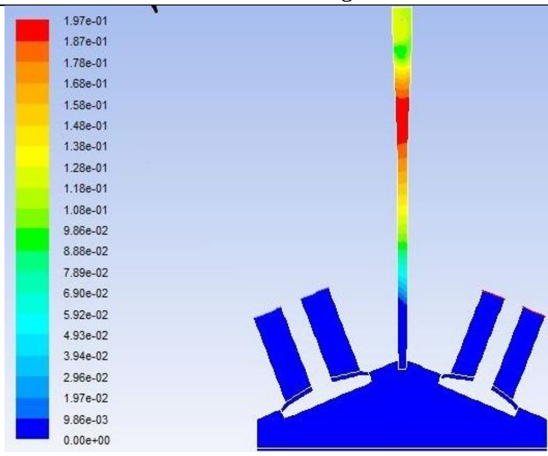
In case of different ignition timing, the temperature is decreased when the timing of ignition is near to TDC. In case 2, temperature after 10° CA of ignition is 2410 K and increased until the maximum value is 2530 K at TDC. This ignition occurred after the injection process stopped.

The maximum value of temperature for case 3 is 2530 K. This contour showed that the hot temperature increased and going up to the top of prechamber until reach at TDC. At 40° CA after the ignition for this case, temperature in combustion chamber become hot compare within the prechamber.

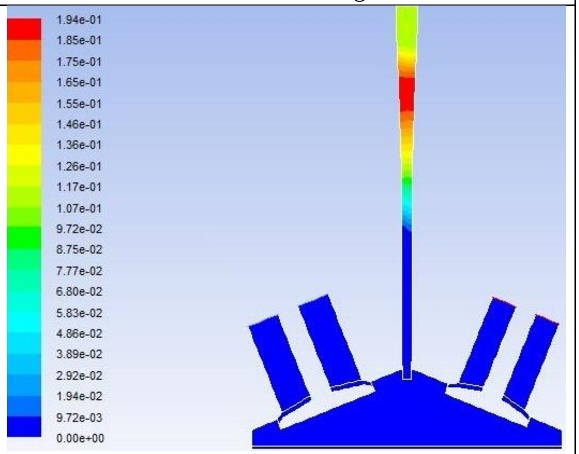
Table 4.4 : Contour of methane mass fraction in different ignition timing

Case 1 : Ignition Timing at 40 ° BTDC

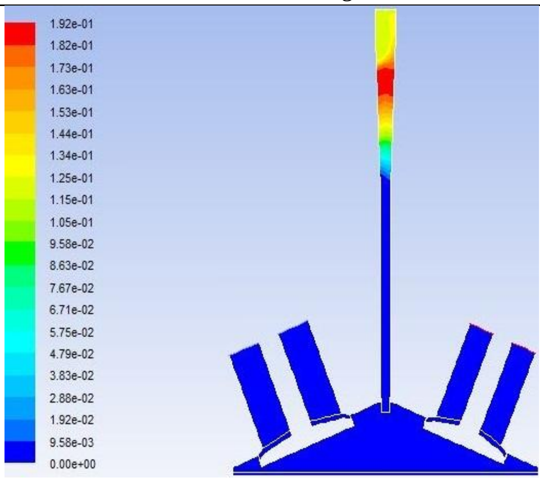
10° CA after start of ignition



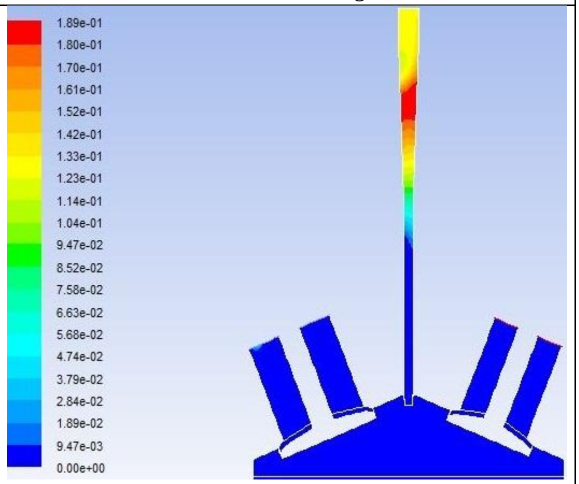
20° CA after start of ignition



30° CA after start of ignition



40° CA after start of ignition



50° CA after start of ignition

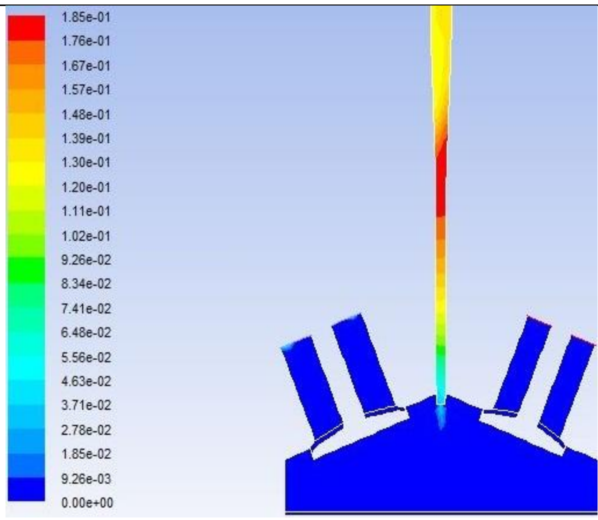
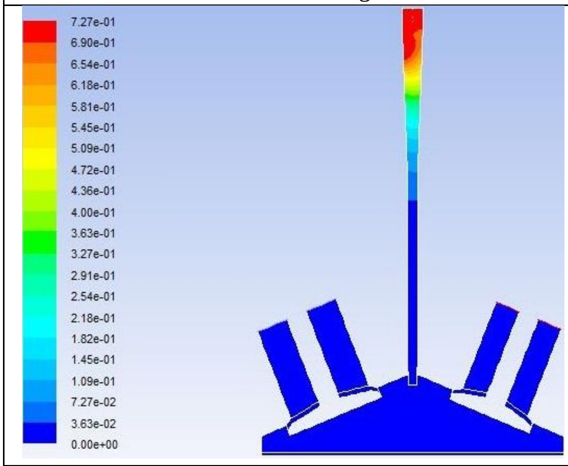


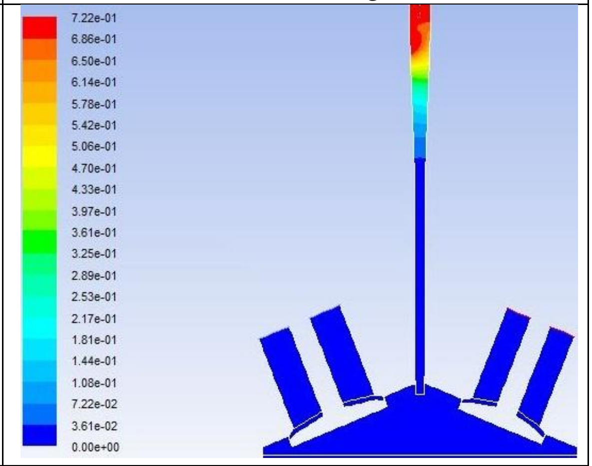
Table 4.4 : Contour of methane mass fraction in different ignition timing(Cont.)

Case 2 : Ignition Timing at 30 ° BTDC

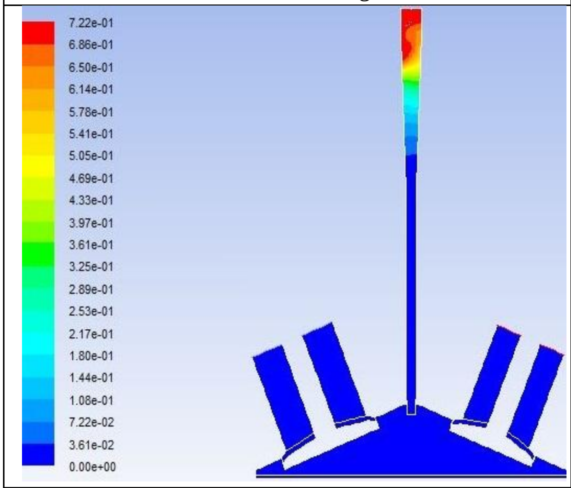
10° CA after start of ignition



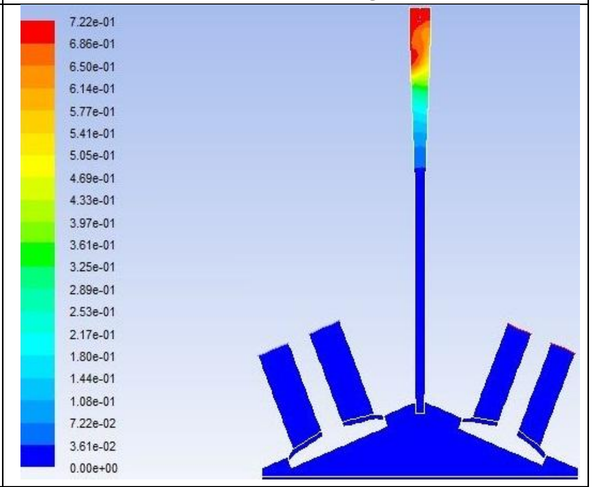
20° CA after start of ignition



30° CA after start of ignition



40° CA after start of ignition



50° CA after start of ignition

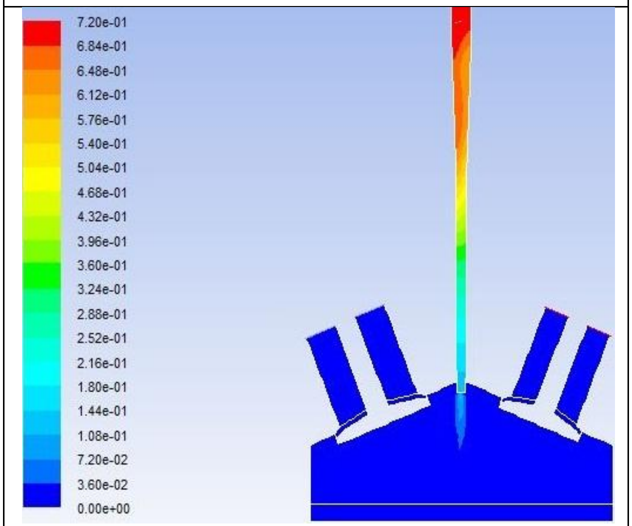
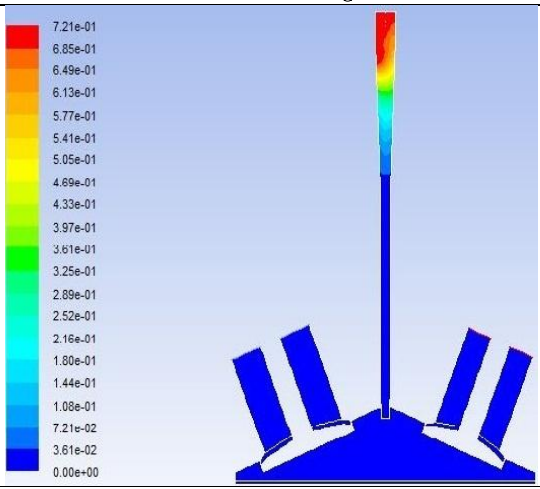
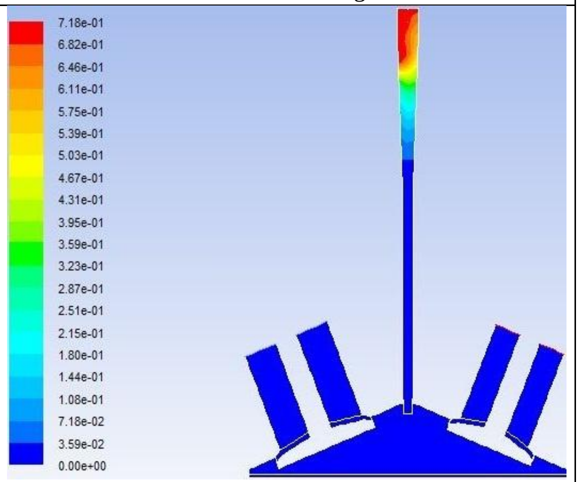
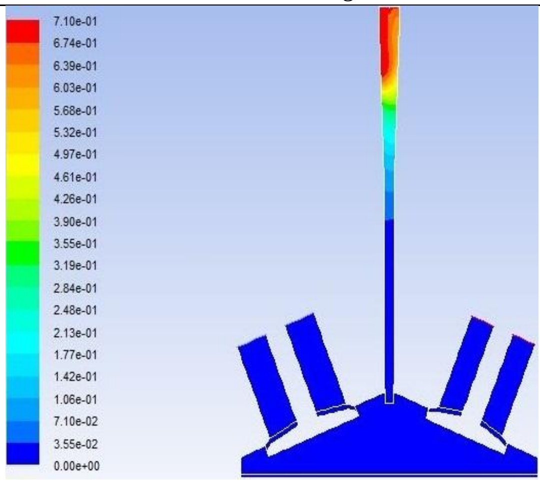


Table 4.4 : Contour of methane mass fraction in different ignition timing(Cont.)

<table>
<tr><th colspan="2">Case 3 : Ignition Timing at 20 ° BTDC</th></tr>
<tr><td>10° CA after start of ignition</td><td>20° CA after start of ignition</td></tr>
<tr><td>
</td>
<td>
</td></tr>
<tr><td>30° CA after start of ignition</td><td>40° CA after start of ignition</td></tr>
<tr><td>
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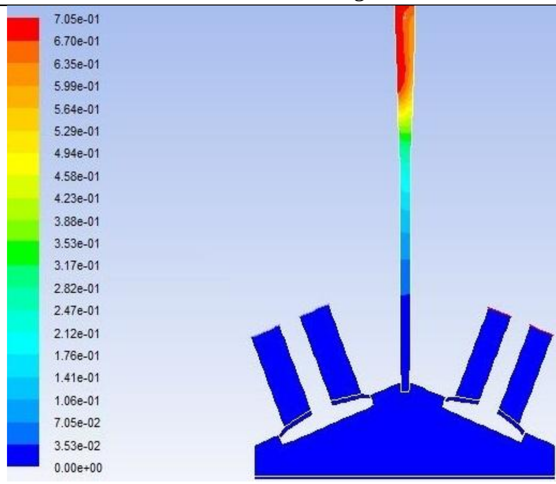
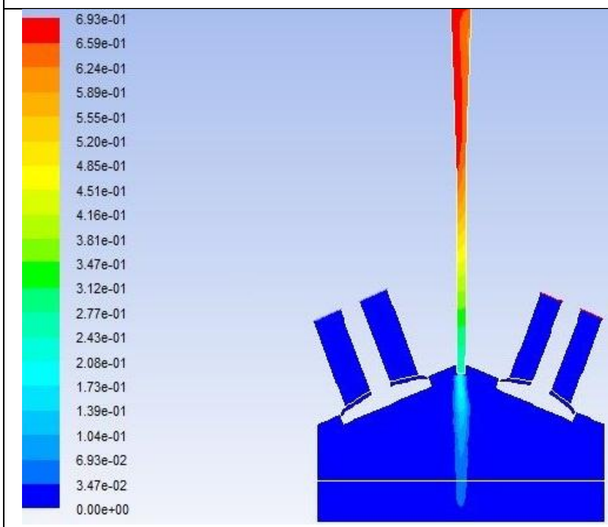
<td>
</td></tr>
<tr><td colspan="2">50° CA after start of ignition</td></tr>
<tr><td colspan="2">
</td></tr>
</table>

Refer to the table 4.4, contour of methane mass fraction is showed at different ignition timing. The contour is representing mass fraction of methane that exist in the model. In the case 1, amount of methane that exist in the prechamber in the range 0.192 until 0.18. After the ignition, mass fraction of methane become decreased because of reaction process occurs during the combustion process.

Case 2 is showing the amount of the methane mass fraction is 0.727. At this time, the injection process is stopped. So, the mass fraction of methane is still greater after the ignition process. Methane mass fraction is not change at prechamber until it transfer to the main cylinder. After 50° CA start of ignition, main cylinder contains 0.108 of methane.

At the combustion process in case 3, methane still has 0.721 of mass fractions. Based on table 4.4, methane only exists at the prechamber after the ignition process. After 50° CA of ignition, 0.708 of methane is located at prechamber and slowly going down to the main cylinder.

4.4 CASE I : VARIABLE INJECTION TIMING

4.4.1 Pressure In Cylinder

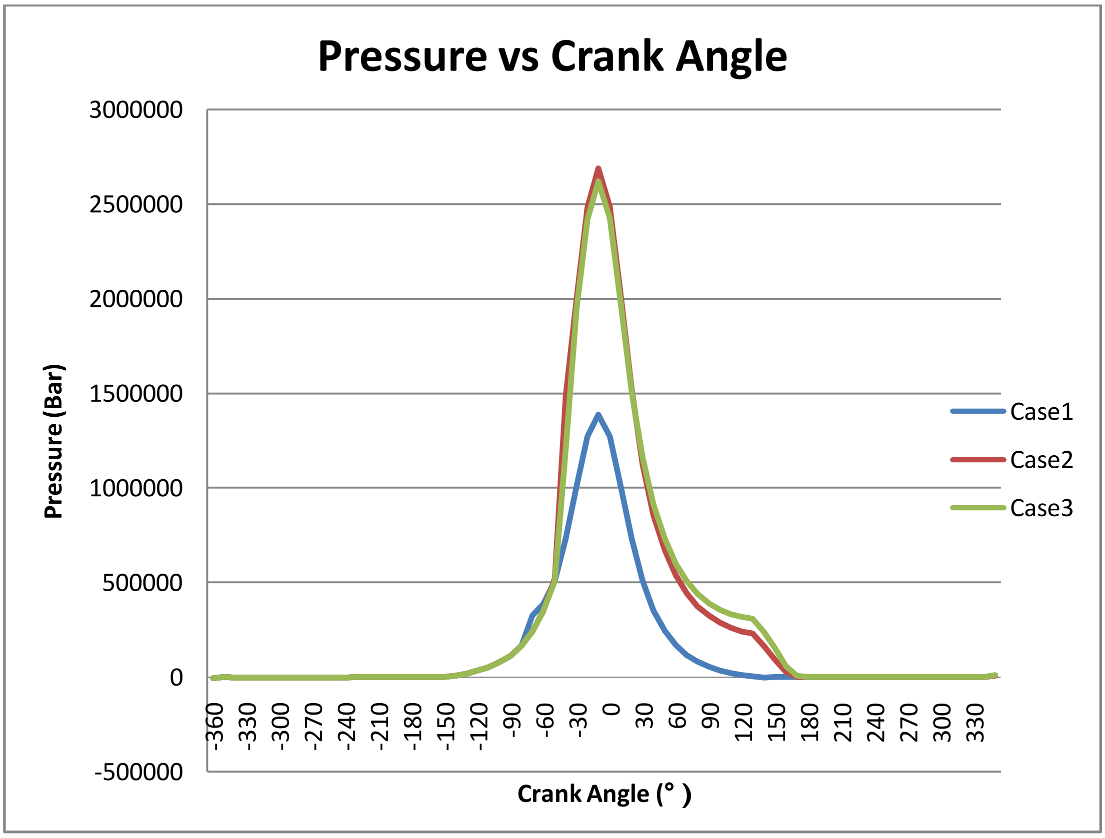


Figure 4.2 : Pressure of different injection timing

Based on the graph that showed in figure 4.2, the average pressure in cylinder have a different value at the different timing. The maximum peak of pressure is 27 bar at the injection case 2. In this case the timing of ignition is constant at the 10°CA after the start of injection. After the ignition point, the average pressure is start increased due to the heat released from the combustion of methane and air mixtures.

Based on the case 1, average pressure is too low compare with another case. This situation occured because the pressure in cylinder is not enough compressed during the compression stroke. So, its can effect the result of the simulation. For case 2 and case 3, value of pressure are not critically different because both of case have a greater pressure in cylinder compare with case 1. When the pressure injection attached to the model, the pressure start to increased.

4.4.2 Temperature In Cylinder

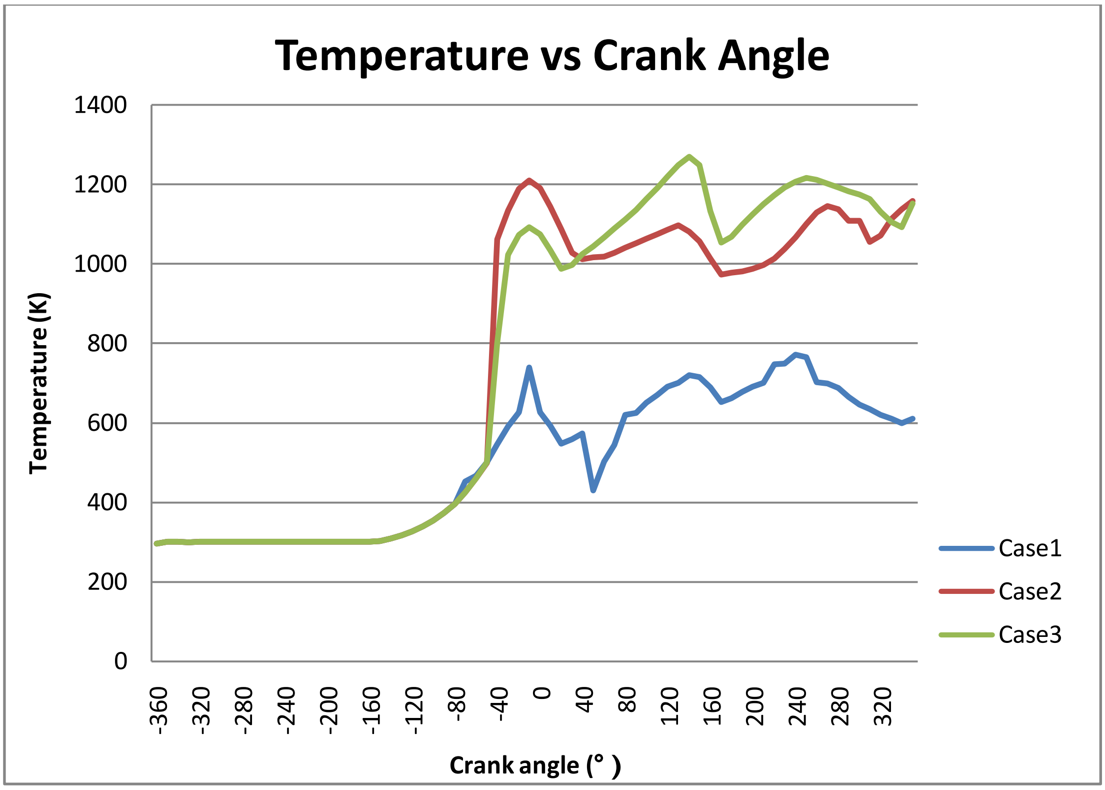


Figure 4.3 : Temperature of different injection timing

Based on the graph that showed in figure 4.3, the average temperature in cylinder have a different value at the different timing. The pattern for the graph at the figure 4.3 is increased when the degree of crank angle almost reach TDC. The maximum average temperature is 1210 K at TDC. At the 10° CA after start the injection process, which is ignition point, the average temperature for combustion simulation is drastically increased. The reason of the increment in average temperature is due to the heat release from the combustion process.

4.5 CASE II : VARIABLE IGNITION TIMING

4.5.1 Pressure In Cylinder

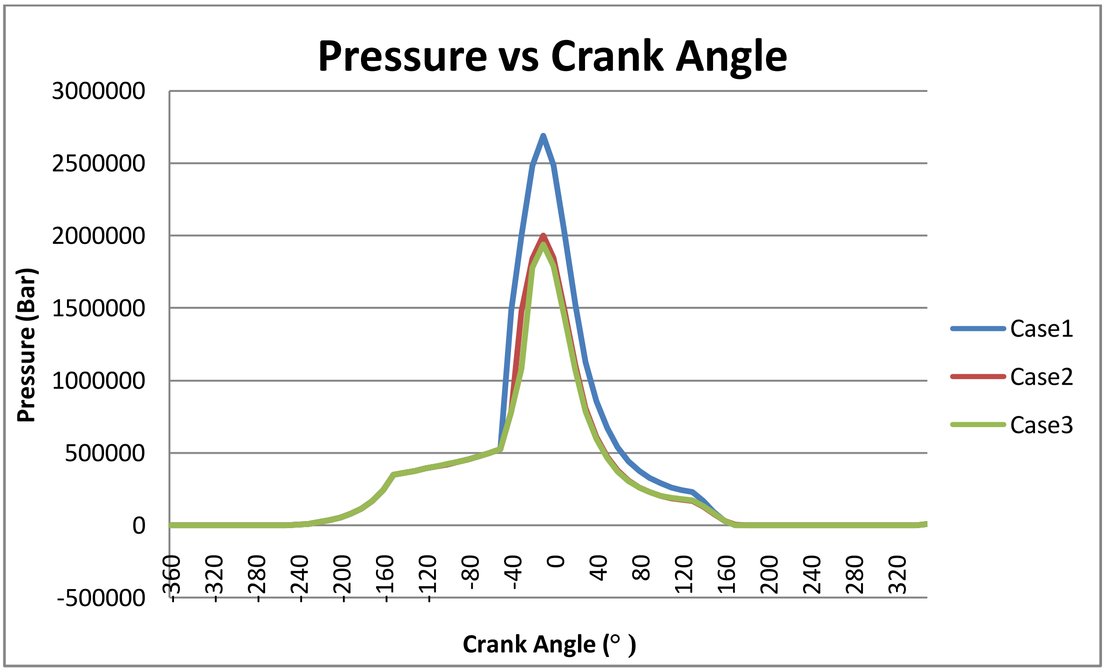


Figure 4.4 : Pressure of different ignition timing

Based on the figure 4.4, the trend of this graph is comparable with the actual profile of pressure. There are three case of ignition timing are done for this simulation. The highest average pressure for this study is 27 bar at the ignition 40° CA BTDC.

For the case 2 and case 3, pressure have a same value that showed in figure 4.4. This average pressure represented lower compared with the case 1. Same as the figure 4.2, this average pressure at the different ignition timing decreased after start the process expansion stroke because the expanding volume in combustion cylinder.

4.5.2 Temperature In Cylinder

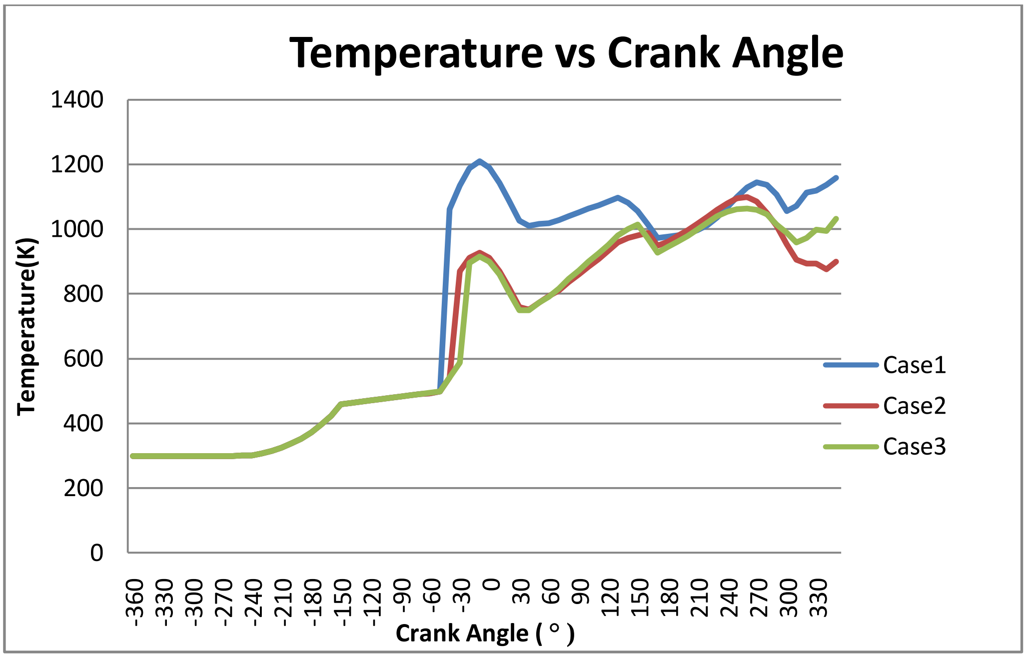


Figure 4.5: Temperature of different ignition timing

Based on the figure 4.4, the graph represented average temperature versus crank angle of the cylinder. There are three cases of ignition timing are done in this simulation. The maximum average temperature based on figure 4.5 is 1210 K at the ignition point at case 1. It shows that the temperature increased by the pressure increased during the compression stroke.

After the ignition point, the temperature on the graph showed that fluctuating pattern. This pattern means the combustion is still occurring until complete the full cycle of the engine.

4.6 SUMMARY

This chapter has presented the simulation data and the experimental result. The simulation data have been compared in case different injection timing and ignition timing. Combustion pressure analysis using CFD prediction is the main interest in this study.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

Simulation of CNG fuel in SI engine has been done with using computational fluid dynamic modelling approach for fluid flow problems. The mesh has been used is 0.7 mm and the thickness of the geometry is 1 mm. The result from the simulation showed that the maximum average pressure for Case 2 and Case 3 is 27 bar while the maximum average temperature is 1210 K.

The main problems in this simulation is the actual pressure of injector. The pressure of the injector in this study is assumed same as the gasoline combustion process. The increasing temperature will cause the pressure increase. For this study main factor that affect the simulation accuracy is heat transfer process. This simulation is not considered the heat transfer along the wall, although in the actual condition the heat transfer along the wall is very important in order to avoid from over heat and engine failure. As the conclusions, factors that effect the model prediction are recognized and critically need refinement in future works.

5.2 RECOMMENDATION

The consideration of input data properties and heat transfer need to be fully aware and take it sseriously in order to achieve the accurate result. More data from the actual condition is needed such as pressure, temperature and injection data to implement in the simulation. In modelling work, temperature and pressure is needed to set as a dependent properties and more data of boundary condition is needed.

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