CRANKCASE ANALYSIS FOR TWO-STROKE SPARK IGNITION ENGINE

TING SWEE KEONG

UNIVERSITI MALAYSIA PAHANG

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A report submitted in partial requirements
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SUPERVISOR'S DECLARATION

We hereby declare that we have checked this project and in our opinion this project is satisfactory in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive.

Signature

Name of Supervisor : NIK MOHD IZUAL BIN HJ NIK IBRAHIM
Position : LECTURER
Date :

Signature
Name of Panel : DEVARAJAN A/L RAMASAMY
Position : LECTURER
Date :

STUDENT'S DECLARATION

I hereby declare that the work in this thesis is my own except for quotations and summaries which have been duly acknowledged. The thesis has not been accepted for any degree and is not concurrently submitted for award of other degree.

Signature

Name : TING SWEE KEONG

ID Number : MH06071

Date : 10^{th} NOVEMBER 2008

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ABSTRACT

Crankcase of two-stroke spark ignition engine is an important part since it compresses the air fuel mixture before going into cylinder. Parameter inside crankcase such as pressure largely affects the performance of this engine. In this project, the simulation of flow inside of two-stroke engine had been carried out. The objective of this project is to simulate the visualization of crankcase flow process. The advantages of using CFD in this research are low cost and easy to apply compared to the laser. Before the main concept had been applied, COSMOS application is the one of the software that introduces to the basic of flow pattern in this research. After complete the entire tutorial this software, will proceed with the CFD application. CFD application was started by modeling the crankcase in three-dimensional in SOLIDWORK. After that, GAMBIT was use to generate grids before export to FLUENT for flow analysis. Crankcase model was simulated in motoring condition, which means no combustion or firing. As a result, pressure and contours inside the crankcase flow was observed. And found that the simulation results are slightly different from calculation.

ABSTRAK

Kotak engkol merupakan satu bahagian yang penting dalam engine pembakaran dalam dua lejang. Ia berfungsi untuk memampatkan campuran udara dan bahan api sebelum memasuki silinder. Parameter dalam kotak engkol umpamanya tekanan memberi kesan yang besar terhadap prestasi enjin ini. Dalam projek ini, simulasi terhadap aliran dalam kotak engkol bagi enjin dua lejang telah dilakukan. Objektif projek ini ialah untuk simulasi dan memerhatikan proses aliran dalam kotak engkol Kelebihan teknik analisis pengaliran dinamik dalam kajian ini merupakan analisis yang murah dan senang digunakan berbanding penggunaan laser. Sebelum mengaplikasikan konsep yang utama, aplikasi COSMOS merupakan perisian yang memperkenalkan asas proses aliran dalam kajian ini. Selepas tamat latihan tutorial dalam aplikasi ini, teknik analisis aliran dinamik akan diteruskan. Teknik analisis aliran dinamik dimulakan dengan memodelkan kotak engkol dalam tiga dimensi dalam perisian SOLIDWORK. Selepas itu, GAMBIT pula digunakan untuk menjana grid sebelum dieksport ke FLUENT untuk analisis aliran. Model kotak engkol disimulasikan dalam keadaan bermotor yakni tiada pembakaran. Hasilnya tekanan beserta kontur kelajuan aliran dalam kotak engkol dapat diperhatikan. Didapati tekanan dari simulasi mempunyai ralat berbanding nilai tekanan daripada kiraan.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF SYMBOLS

Roman Symbols

A	_	Stroke
CR_{cc}	_	Crankcase compression ratio
d_b	_	Diameter of cylinder
$H(\theta)$	_	Piston position
L	_	Connecting rod length
P_s	_	Piston position
SE_v	_	Scavenging efficiency on volumetrically
SR_v	_	Scavenged ratio on volumetrically
TE_v	_	Trapping efficiency on volumetrically
V_{am}	_	Volume of air in the mixing zone
V_{as}	_	Air supplied /volume fresh air
V_{cc}	_	Crankcase volume
$V_p(\theta)$	_	Piston velocity
V_s	_	Swept volume
V_{ta}	_	Volume of air trapped
V_{ts}	_	Trapped swept volume

Greek Symbols

$k-\varepsilon$	_	Standard k -epsilon model
λ_d	_	Delivery ratio (or scavenge ratio)
η_{se}	_	Scavenging efficiency
η_{te}	_	Trapping efficiency
η_{ce}	_	Charging efficiency

LIST OF APPENDICES

LIST OF ABBREVIATIONS

ATDC	After Top Dead Center
BC	Boundary Condition
BDC	Bottom Dead Center
CA	Crank Angle
CFD	Computational Fluid Dynamics
DAQ	Data Acquisition System
FLUENT	Computational Fluid Dynamics code software
IC	Internal Combustion
SAE	Society of Automotive Engineers
TDC	Top Dead Center

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

The first two-stroke engine was invented by Sir Dugald Clerk (1900) in England at the end of 19^{th} century. The form of engine using crankcase compression for intake process, including the control of the timing and the area of the exhaust port, transfer and the inlet ports by the piston, was patented by Joseph Day (1891) in England. Same as today conventional two stroke cycle engine, the engine patented by Joseph Day was a three ports engine, where the intake ports used to suck air or mixture from the carburetor or air cleaner and been transfer to the combustion by the transfer port, while the third port is used as a exhaust port to allow burned gases exhausted to the atmospheric [10].

In a conventional two-stroke internal combustion engine, the vacuum caused by a piston moving away from the crankcase draws a mixture of fuel, air and oil into the crankcase through a one-way valve of timed induction mechanism such as a piston port or rotary valve. Crankcase of two-stroke spark ignition engine is an important part since it acts as a compressor to compress the air-fuel mixture before going into cylinder [8][16].

1.2 PROBLEM STATEMENT

Crankcase scavenged two-stroke gasoline engines suffer from fresh charge losses leading to poor fuel economy and it is a reason for large increases of hydro-carbon in the exhaust. In recent years, two-stroke engines have been re-evaluated in terms of improvements to the drawbacks, involving the in-cylinder fuel injection, exhaust catalysts, and control of the optimum port area [1][2].

Previously, there are only a few sources regarding the study of the crankcase for two-stroke spark ignition engine [3][4][5][6]. In addition, some other sources use formulas and calculations to determine the internal flow process of the crankcase. The accuracy of determined parameters are different compare to the actual engine running process. This project is to focus more detail on simulation of two-stroke spark ignition engine by using computational fluid dynamics (CFD) with motoring method and dynamic mesh approach to the crankcase. CFD codes are developed for the simulation of a wide variety of fluid flow process. They can be analyzed steady or unsteady flow, laminar or turbulence flow, flow in two or three dimensions and single or two phase flow [7]. The model was based on a single cylinder research engine and included features to simulate the motion of air-fuel and lubricant mixture inside the crankcase.

1.3 PROJECT OBJECTIVE

The objective of this project is to simulate the visualization of crankcase flow process for a single 30.5 cc two-stroke spark ignition (SI) engine.

1.4 PROJECT SCOPES

i. Literature review regarding the two-stroke spark ignition engine.
ii. Three dimensional CAD modeling and drawing.
iii. Familiar with usage of COSMOS FLOWORKS and FLUENT.
iv. Simulation with COSMOS FLOWORKS and FLUENT.

1.5 THESIS ORGANIZATION

This thesis consists of five chapter summarized as follows:

Chapter 2 is focus about the literature review of two-stroke spark ignition engine as general. Then, the detail of crankcase was mentioned in term of construction and function to the engine.
Chapter 3 consist of methodology of this research, flow chat and simulation set up processes. It is concentrate on the three-dimensional analysis by FLUENT in term of modeling, mesh generation, and define boundary conditions and simulation set up.

Chapter 4 includes the detail results and discussion on this research. The main focus is on the pressure distribution inside the crankcase. Besides, results from COSMOS also contain in this chapter for the initial flow trend observation. It also provided the visualization results for contours and vectors of velocity magnitude and static temperature.

Chapter 5 concludes the results provide some recommendations for future work regarding this research.

1.6 AUTHOR’S CONTRIBUTION

Determine the pressure distribution inside crankcase by build up the three-dimensional model and simulate using the dynamic mesh approaches. Visualization of crankcase velocity characteristics contributes to the study in this research at different crank angle.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

The purpose of this chapter is to provide a review of past research effort related to crankcase analysis of two-stroke spark ignition engine. This chapter will include all the important information regarding the internal combustion engine, two-stroke engine and finally the crankcase of two-stroke engine. Generally, the crankcase is one of the important part in two-stroke spark ignition engine, not only mounted the crankshaft assembly, but also provide the vacuum for intake stroke purpose.

2.2 INTERNAL COMBUSTION ENGINE

The internal combustion (IC) engine is a heat engine that converts chemical energy in a fuel into mechanical energy, usually made available on a rotating output shaft. Chemical energy of the fuel is first converted to thermal energy by means of combustion of oxidation with air inside the engine. This thermal energy raises the temperature and pressure of the gases within the engine, and the high-pressure gas then expands against the mechanical mechanisms of the engine [6].

An internal combustion engine can work on any one of the following cycles [7][8]:

a. Constant volume or Otto cycle.
b. Constant pressure or Diesel cycle.
c. Dual combustion cycle.

Constant volume or Otto cycle – heat is supplied at constant volume. Petrol, gas, light oil engine works on this cycle. In the case of petrol engine, the proper mixing of petrol and air takes place in the carburetor, which is situated outside the engine cylinder. The proportionate mixture is drawn into the cylinder during the suction stroke [8][9].

Constant pressure or Diesel cycle – only air is drawn into the engine cylinder during suction stroke, this air gets compressed during the compression stroke and its pressure and temperature increase by a considerable amount. Just before the end of the stroke a metered quantity of fuel under pressure adequately more than that developed in the engine cylinder is injected in the fine sprays by injector. Heavy oil engines make use of this cycle [9][8].

Dual combustion cycle – also called the semi-diesel cycle. Heat is added partly at constant volume and partly at constant pressure. In this cycle only air is drawn into the engine cylinder during suction stroke. The air is then compressed in hot chamber at the end of the cylinder during the compression stroke to a pressure of about 26 bar. The heat compressed air together with heat of combustion chamber ignites the fuel. The fuel is injected into the cylinder just before the end of compression stroke when it ignites immediately. The application of this cycle is heavy oil engines [8][9].

2.2.1 Background of Two-Stroke Cycle Engine

Some sources claim that two-Stroke engines were produced by Butler and Roots (1887) based on a crankcase scavenged, piston controlled intake, transfer, and exhaust port patent ultimately issued to F.W.C. Cock in England in 1892, and owned by Joseph Day. From 1928 to 1948, General Motors Research Laboratories under CF Kettering made a serious effort to develop a two-stroke powered vehicle. The engine configuration was also an inverted U-type patterned [11].

Some of the early applications of the two-stroke engine include the first engine that produced by Edward Butler in 1887 and by J.D.Roots, in the form of the Day crankcase compression type, in 1892. Usually this kind of engine was used in motorcycle, scooters, small chainsaw engine, light aircraft etc. Besides the small power output usage, the two-stroke cycle engines also being applied in the marine diesel engine due to its simplicity of constructions, low weight and higher power output [10].

2.3 TWO-STROKE ENGINE

The two-stroke engine employs the crankcase as well as the cylinder to achieve all the elements of the Otto cycle in only two strokes of the piston [7][9][10].

In a conventional two-stroke internal combustion engine, the vacuum caused by a piston moving away from the crankcase draws a mixture of fuel, air and oil into the crankcase through a one-way valve of timed induction mechanism such as a piston port or rotary valve. Increased pressure produced by the piston moving toward crankcase forces the mixture of fuel, air and oil into piston cylinder on the side of the piston away from crankcase and, therefore, into combustion chamber, which is at the portion of the piston cylinder that is most distant from the crankcase, because such carbureted fuel cannot escape through the one-way valve or a now closed induction mechanism [9][10].

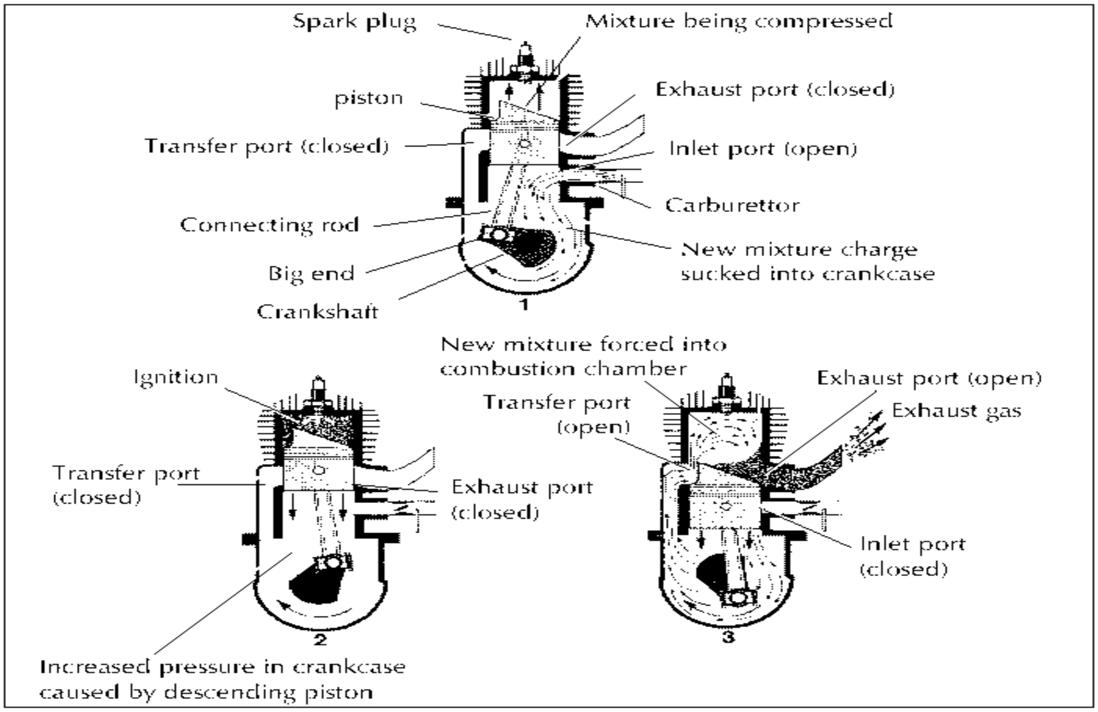

Figure 2.1: Two-stroke cycle engine [12]

In the single cylinder two-stroke spark ignition engine, there are two main of movement of piston that differentiates from other engine that is compression stroke and power stroke. For the intake process, the air-fuel mixture is first drawn into the crankcase by the vacuum created during the upward stroke of the piston. The illustrated engine features a poppet intake valve, however many engines use a rotary value incorporated into the crankshaft [7][9][13].

During the downward stroke, the poppet valve is closed by the increased crankcase pressure. The fuel mixture is then compressed in the crankcase during the remainder of the stroke [7].

Toward the end of the stroke, the piston exposes the intake port, allowing the compressed fuel/air mixture in the crankcase to escape around the piston into the main cylinder. This expels the exhaust gasses out the exhaust out the exhaust port, usually located on the opposite side of the cylinder [7].

The piston then rises, driven by flywheel momentum, and compresses the fuel mixture. (at the same time, another intake stroke is happening beneath the piston). At the top of the stroke, the spark plug ignites the fuel mixture. The burning fuel expands, driving the piston downward to complete the cycle [7][10].

Inherent in the two-stroke cycle is the process of scavenging the burned gases from the engine cylinder with fresh charge. This gas exchange process has several consequences. First charging losses are inevitable. Under normal operation conditions in a typical two-stroke engine, about 20% of the fresh charge that enters the cylinder is lost due to short circuiting to the exhaust and results in very high hydrocarbon emissions and poor fuel economy [12].

2.3.1 Scavenging process

In two-stroke cycle engines, each outward stroke of the piston is a power stroke. In order to achieve this two-stroke, the fresh charge must be supplied to the cylinder at a high-enough pressure to displace the burned gases from previous cycle [14]. The operation of clearing the cylinder of burned gases and filling it with fresh mixture combination between intake and exhaust process is known as scavenging [1][14][12]. The design of port system either the intake or the exhaust ports of two-stroke spark ignition engine is very influential in the short-circuiting of fresh charge [14]. The short-circuiting phenomena are responsible for the lower fuel efficiency and high hydrocarbons emission.

Today, three main categories of scavenging process generally accepted:

- Cross scavenging
- Loop scavenging
- Uniflow scavenging

2.3.1.1 Cross scavenging

Cross scavenging is the simplest way of loading the cylinder with fresh charge. First, the system was designed the way that the piston head towards the cylinder head deflected the incoming gas jet. Thus the leading to the thermal problems with the piston, newer designs are consisting angular intake port areas. The burnt gases are pushed by the fresh charge towards the exhaust port. Since the exhaust port area is right in front of the intake port, there is the risk of direct charge loss in form of short-circuiting.

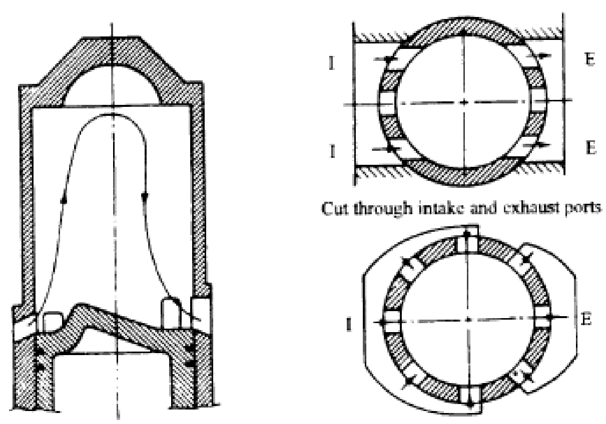

Figure 2.2 : Cross scavenging [15]

2.3.1.2 Loop scavenging

In loop scavenging system, the burnt gases leave the cylinder in reverse flow direction to the incoming fresh charge. There are two sub categories, which is MAN - loop scavenging and SCHNURLE – loop scavenging [10][15].

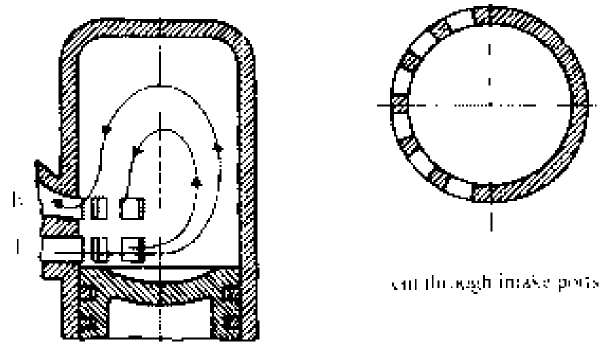

Figure 2.3 : MAN – loop scavenging [15]

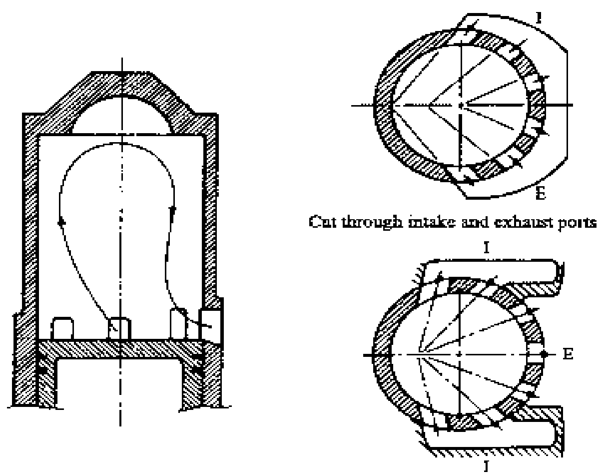

Figure 2.4 : SCHNURLE – loop scavenging [15]

The MAN design has the exhaust area on the top of the intake ports. The incoming charge is supposed to cool the piston. While the SCHNURLE design avoids exhaust on top of intake. Here, the intake ports are found on the both sides of the exhaust port at the whole cylinder diameter. The advantage of this design is a higher effective compression volume [15].

2.3.1.3 Uniflow Scavenging

In uniflow scavenging system, the fresh charge is introduced into the cylinder through intake ports, usually at the bottom of the cylinder. The burnt gases are pushed towards the cylinder head and leave through one or more exhaust valve. In order to optimize the charging effect, the intake area is a angular producing by a swirl flow [10][15].

Uniflow scavenging system is usually used with large slow speed engines. The risk of direct charge loss is very small, thus leading to very high scavenging efficiency. This disadvantage of such a system is the requirement of valves, valves controlling unit etc [10][15].

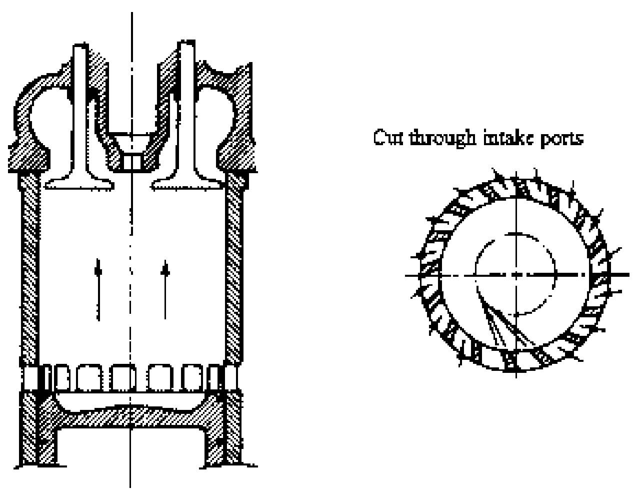

Figure 2.5 : Uniflow scavenging [15].

2.3.2 Advantages and disadvantages of two-stroke cycle engine

2.3.2.1 Advantages:

- Simple engine and construction.
- Light weight and small size.
- Produce one power stroke in every one-engine revolution.
- Piston function as a valve.
- Low cost of manufacturing due to the engine simplicity.
- Applicable to both SI and CI engine.

2.3.2.2 Disadvantages:

- Higher residual gases inside cylinder after scavenging process.
- Higher fuel consumption rate.
- Low combustion efficiency due to incomplete combustion.
- High hydrocarbon pollution due to mixing of fresh charge and burned gases during scavenging process.
- The fresh charge is use to push the burned gases to atmospheric.
- Lower volumetric efficiency.

2.4 CRANKCASE

Crankcase of two-stroke spark ignition engine act as a compressor, this requires the crankcase to have relatively close tolerances between the crank and the crankcase, itself. It is also required that the crankcase be sealed [8][16]. These factors isolate the crankcase from any lubrication that may be in other parts of the engine. Therefore, a secondary lubrication system is necessary. However, any oil in the crankcase would readily be pushed into the combustion chamber. Therefore, to minimize the oil that is pushed into the combustion chamber, oil is continuously added to the crankcase, but only in small quantities. In conventional two-stroke engines this is accomplished either by oil injection or by utilizing fuel, which has been pre-mixed with a suitable quantity of oil [17]. However, no matter how the lubrication is achieved, oil will be introduced into the combustion chamber and combusted. In addition, during the combustion process, such oil creates considerable smoke and other pollution [9].

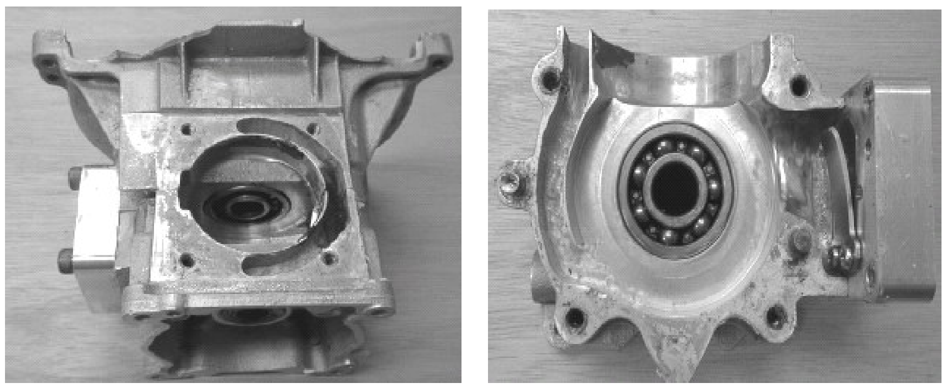

Figure 2.6 : View of two-stroke engine crankcase [18]

In conventional two-stroke engines, crankcase compression largely affects the engine performance since the process provides useful work to compress the fresh air before entering cylinder. The pressurized air is needed to expel the burned gases inside cylinder volume. The higher the fresh air being compressed, the higher momentum they have when entering the cylinder and better scavenging will be [19].

2.4.1 Exposure type of crankcase:

The crankcase is exposing to the vibration while engine is running. There are four main causes vibration in an engine, that is [9][10]:

i. Rotating part – centrifugal force acts on all parts that rotate. Part such as the crankshaft, flywheel and clutch must be balance.

ii. Power impulse – the pistons deliver power to the crankshaft as impulses and this causes a type of rotary vibration in the crankshaft.

iii. Reciprocating part – the piston, in particular, produce an inertia force at the top and bottom of their strokes. This causes up and down vibrations in the engine.

iv. Resonance – vibrations can be transmitted between parts and amplified, even though the parts may not be directly connected.

Crankcase pressure – every time combustion occurs, a certain amount of blow by (from combustion) escape past the piston ring. This blow-by produces a small crankcase pressure. The gases from blow-by are very acidic. If they are allowed to stay in crankcase area, the acids attack oil and metal within the engine [7].

The transfers have extremely short time in which to recharge the cylinder with fuel/fresh air/lubricant mixture, the flow pattern of the charge also must be control to prevent mixture loss out of the exhaust, and drive exhaust gases from the rear of the cylinder towards the exhaust port [7]. Some engine had massive spaces in the crankcase and tuners reasoned, rightly enough, that filling the crankcase with a variety of stuffers would reduce crankcase pressure volume and hence increase crankcase compression when the piston descended to BDC. Increasing crankcase compression naturally enough result in higher crankcase pressure which, all else being equal, raises transfer flow and improves maximum hp output [17]. So effective was this method of cylinder scavenging that the fuel/fresh air 'wedge' was actually being partly lost out of the exhaust before the port closed. Because of more fuel charge being contained within the cylinder, power increased. To ensure, efficient pumping of the/fresh air mixture from the crankcase into the cylinder, one of the ways is by increasing crankcase compression [7][10][15][17]. Also, the density of the charge transferred from the crankcase into the cylinder is dependent on the temperature of the inlet tract and the crankcase surfaces [20].

All compression ratio value is the ratio of the maximum volume in any chamber of an engine to the minimum volume in that chamber. In the crankcase that ratio is known as the crankcase compression ratio CRcc and it's defined by [10][19];

$$CRcc = \frac{Vcc+Vsv}{Vcc} \tag{2.1}$$

where,
Vcc = crankcase clearance volume or crankcase volume at BDC
V_{sv} = swept volume

$$V_{sv} = n\,\frac{\pi}{d}\;d_{bo}{}^{2}\,L_{st} \tag{2.2}$$

where,
d_{bo} = diameter of bore
L_{st} = length of stroke

While it is true that the highest this value becomes the stronger is the crankcase pumping action, the actual numerical value is greatly fixed by the engine geometry of bore, stroke, con-rod length, and the interconnected value of flywheel diameter [11].

2.4.2 Crankcase Scavenging

In crankcase-compression engine, the fresh charge is compressed in the crankcase by the underside of the working piston, prior to its admission to the cylinder through the scavenge ducts. The closing and opening of the inlet, scavenge, and exhaust ports are controlled by the piston itself [12].

The crankcase scavenge pump commonly used in small two-stoke engine is generally reputed to have a very low volumetric efficiency due to the large dead spaces in crankcase [17]. Crankcase scavenged two-stroke gasoline engines suffer from fresh charge losses leading to poor fuel economy and it is a reason for large increases of HC in the exhaust. In recent years, two-stroke engines have been re-evaluated in terms of improvements to the drawbacks, involving the in-cylinder fuel injection, exhaust catalysts, and control of the optimum port area [5][15][17][21].

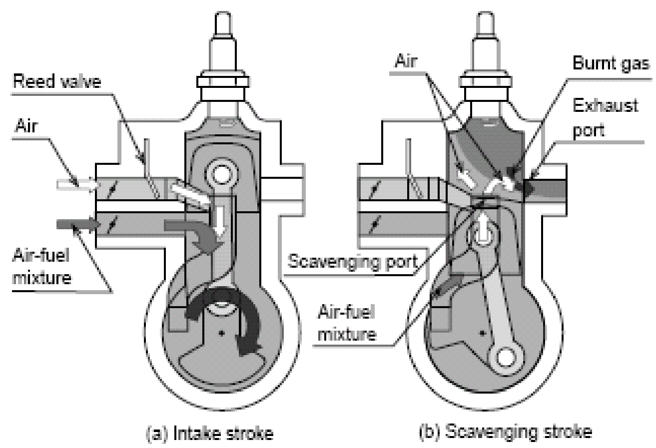

Figure 2.7 : Scavenging flow of conventional two-stroke engine [21]

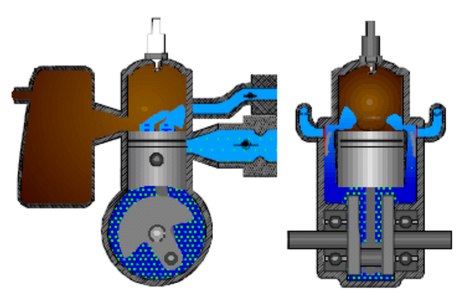

Figure 2.8 : Crankcase scavenging of two-stroke engine [22].

Crankcase scavenging is accompanied by a dry sump lubrication system. A pair of injectors at each cylinder's reed pack spray a metered amount of oil at measured intervals into the intake air. Intake air carries this oil into the crankcase, providing lubrication to the roller bearings of the crankshaft and connecting rods. With no sump, the oil is consumed in the combustion process. [1]

Crankcase scavenge pump is the special feature of two stroke engines whereby the power cylinder is charged with a slightly pressurized mixture and help is rendered in scavenging the exhaust gases [4]. Although such engines have poor fuel economy, they make a strong appeal to the user for its simplicity and compactness. The absence of cylinder head valves and separate scavenging pump, permits low production and maintenance cost. To improve the performance and durability of such an engine, crankcase pressure, volume and temperature are some of the important parameters to optimise [2].

Below is some formula use to define scavenge ratio, scavenge efficiency and trapping efficiency [23] :

Scavenge Ratio,

$$SR = \frac{\text{Vol. of fresh change delivered at cylinder pressure and intake temperature}}{\text{Cylindrical reference volume}} \tag{2.3}$$

Scavenge Efficiency,

$$SE = \frac{\text{Volume of fresh change retained in cylinder}}{\text{Cylinder reference volume}} \tag{2.4}$$

Trapping Efficiency,

$$TE = \frac{\text{Vol. of fresh change retained in cylinder}}{\text{Vol. of fresh change delivered at cylinder pressure and intake temperature}} \quad (2.5)$$

$$= \frac{SE}{SR}$$

The total scavenge ratio is the sum of the scavenge ratios for each plenum [16] :

$$SRvt = SRva + SRvf \quad (2.6)$$

Where, $SRvt$ = Total Volumetric Scavenge Ratio

$SRva$ = Scavenge Ratio from air (Ar) plenum

$SRvf$ = Scavenge Ratio from fuel (SF6/CO2) plenum

The crankcase is modeled as a plenum with variable time dependent volume. The volume function has to be linked to the corresponding cylinder [24]. A volume positioned where the crankcase is on the firing engine is described as a plenum as it volume is significantly larger than the actual engine crankcase [16]. The scavenge ratio for each plenum is determined from the volume outflow which is assumed to expand isothermally in the cylinder, at atmospheric pressure. Therefore the volume that exits, at the plenum pressure is [16]:

$$Vo = A \ x \ L \quad (2.7)$$

where, A = C.S.A of measuring piston

L = Piston displacement

The maximum volume change in the crankcase of a crankcase-scavenged engine is equal to the swept volume of the engine. Due to pumping losses, the volume of air/fuel mixture used for scavenging is less than the swept volume of the cylinder (i.e., the delivery ratio is less than unity) [25].

$$\lambda_d = \frac{\text{mass of delivered air (or mixture) per cycle}}{\text{reference mass}} \tag{2.8}$$

where, λ_d = delivery ratio

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter concludes about all the processes in simulating and analyzing the crankcase. The beginning stage start with the actual size three-dimensional modeling with the simplifications to avoid skewness error and critical geometry of crankcase. COSMOS Flowork initially use to determine the flow pattern inside crankcase at steady state. It is useful to predict the flow trend before running on CFD Simulation. While FLUENT was use in this project to determine the specific parameter inside the crankcase. By using GAMBIT, it generate hexahedral and tetrahedral mesh for simulation in FLUENT. The dynamic approaches of FLUENT able to simulate the internal flow with considering the moving parts such as piston and crankshaft.

3.2 OVERALL OF RESEARCH METHODOLOGY

After understanding and collect the information from literature review. The flowchart of research methodology in this study is shown in Figure 3.1.

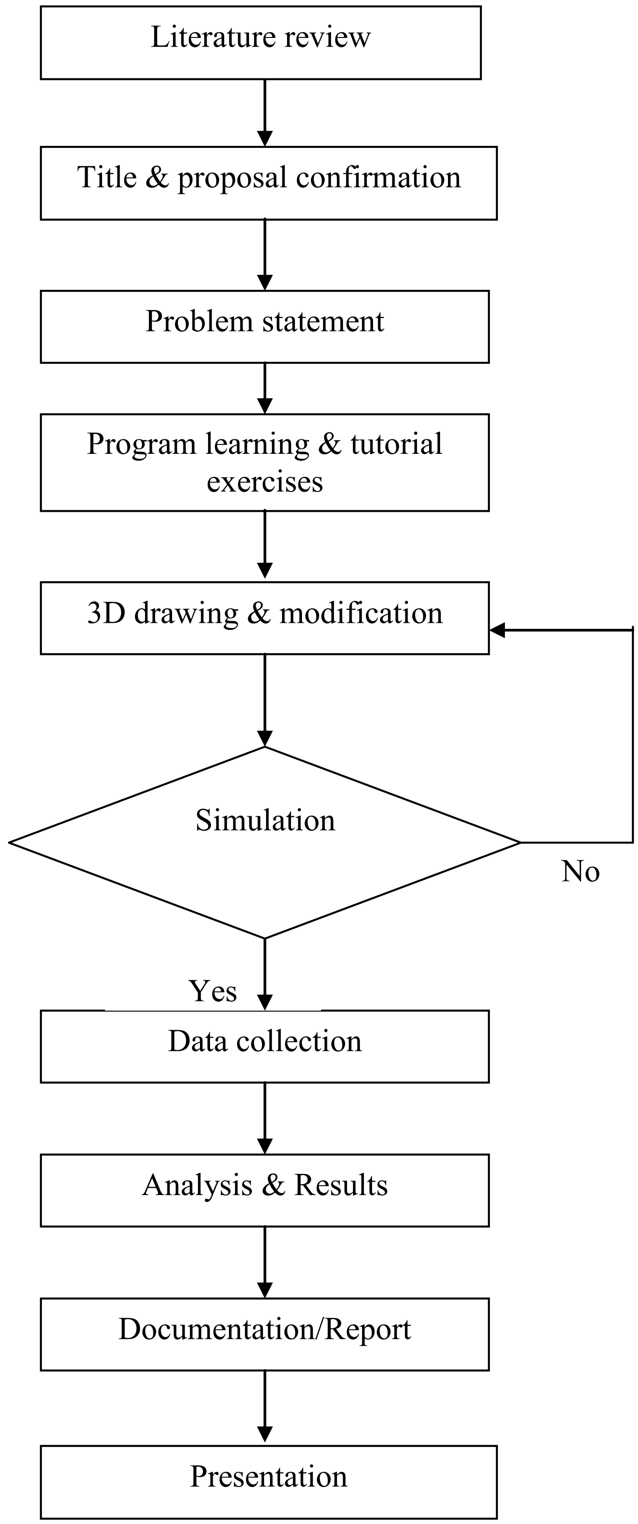

Figure 3.1 : Flow chart of crankcase analysis

3.3 TITLE AND PROPOSAL CONFIRMATION

The project start with the title and proposal confirmation by supervisor, which is regarding the crankcase analysis for two-stroke spark ignition engine. Literature review research and investigation has is carry out to gain information at the beginning level.

3.4 PROBLEM STATEMENT

In this process, the objective and scopes will clearly defined. The current problem face by conventional two-stroke spark ignition engine is stated. Instead of the statement, this stage also expose to the investigation and the information from the journal published by researcher before.

3.5 PROGRAM LEARNING AND TUTORIAL EXERCISE

Some useful tutorials in COSMOS FLOWORKS were is use to expose to the usage of flow analysis whether internal or external. The main objective is to gain basic skills and information before run certain test.

Besides COSMOS FLOWORKS, the same objective can be achieved by practicing the GAMBIT and FLUENT tutorials. For FLUENT, the interface and parameters using are more similar to actual case before perform the analysis using CFD. There are about twenty-four tutorials in this program to be learned.

3.6 THREE-DIMENSIONAL DRAWING AND MODELING

There are some very useful software was use in order to modify the three-dimensional drawing to suite the simulation.

3.6.1 SOLIDWORK

By using SOLIDWORK software, the actual dimension of the engine is draw into three dimensional for future application. All of the parts are separated from engine and measured before drawing process.

All the parts of this engine are shown as below:

Figure 3.2 : Crankcase 1

Figure 3.3 : Crankcase 2

Figure 3.4 : Crankshaft assembly

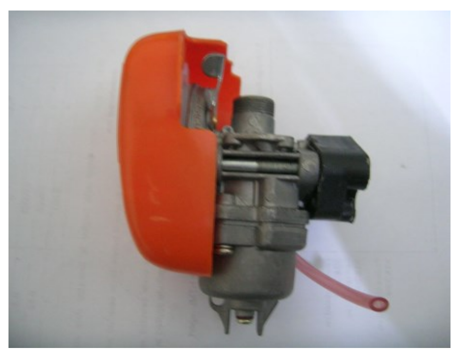

Figure 3.5 : Carburetor

Figure 3.6 : Cylinder block Figure 3.7 : Exhaust

During this process, some of the drawings have will be modified for the primary testing that perform by COSMOS FLOWORKS. The purpose of these modifications is to simplify the complexity of the inner geometry of engine for simulation process.

There was some drawing modifications has been made on the original drawing to simplify the complex part of the component for the simulation purpose. The modifications are includes:

i. Cylinder block - To suite the initial simulation by using COSMOS FLOWORKS, the cylinder block will be separated from head with consider the piston is at BDC position.

ii. Piston - The original piston drawing is slightly smaller than the cylinder diameter. In this case, the diameter of the piston must be enlarged about 1mm to cover the hollow space inside the cylinder block.

iii. Crankcase – End of the crankcase, which the crankshaft bearing located, should be cover. The inner geometry of is without considering the outer part of the crankshaft.

iv. Inlet port – This portion is cover for the purpose of applying the boundary condition.

v. Exhaust port – At the BDC position, the exhaust port is fully open. The modification will fill this portion become solid.

vi. Scavenging or transfer port – The transfer port is open when the reach BDC condition. Modification will cover the expose area and this portion will be applied for the outlet boundary condition.

After all the open and expose portion are covered, the next step in this stage is to generate the inner geometry of the hollow cavity. This is the feature provided by COSMOS FLOWORKS use to determine the volume inside a rigid hollow cavity. This feature also include interface to select which component should be consider for the inner geometry. If the inner geometry is successfully generated, it can be proceed with simulation purposes.

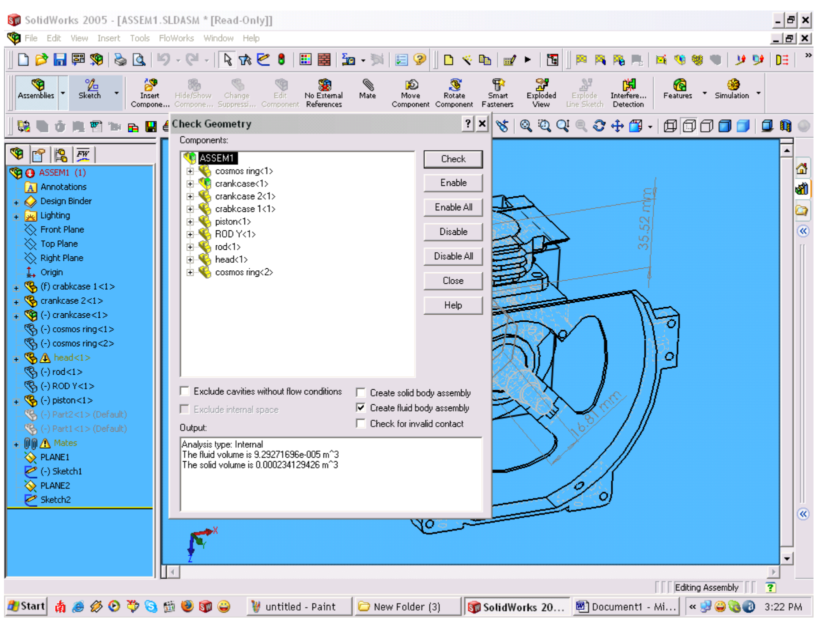

Figure 3.8 : Inner geometry determination

3.6.2 GAMBIT

GAMBIT 2.16 is a useful tool to setup three-dimensional model before export is into FLUENT. Basically, it use to generate meshes grids and imported to FLUENT for modeling analysis. Volume grid generation was established using GAMBIT 2.16 pre-processor and TGrid. The steps of setup a model in GAMBIT are shown as below:

Geometry Setting ⟶ **Mesh Setting** ⟶ **Boundary Condition**

↓

Export

Figure 3.9 : Modeling steps in GAMBIT

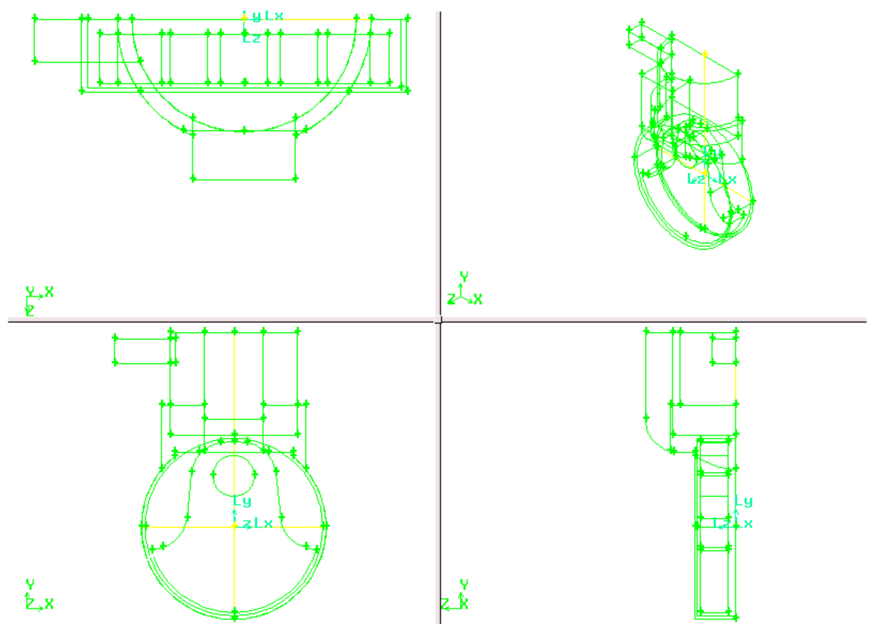

Figure 3.10: Symmetry crankcase modeling

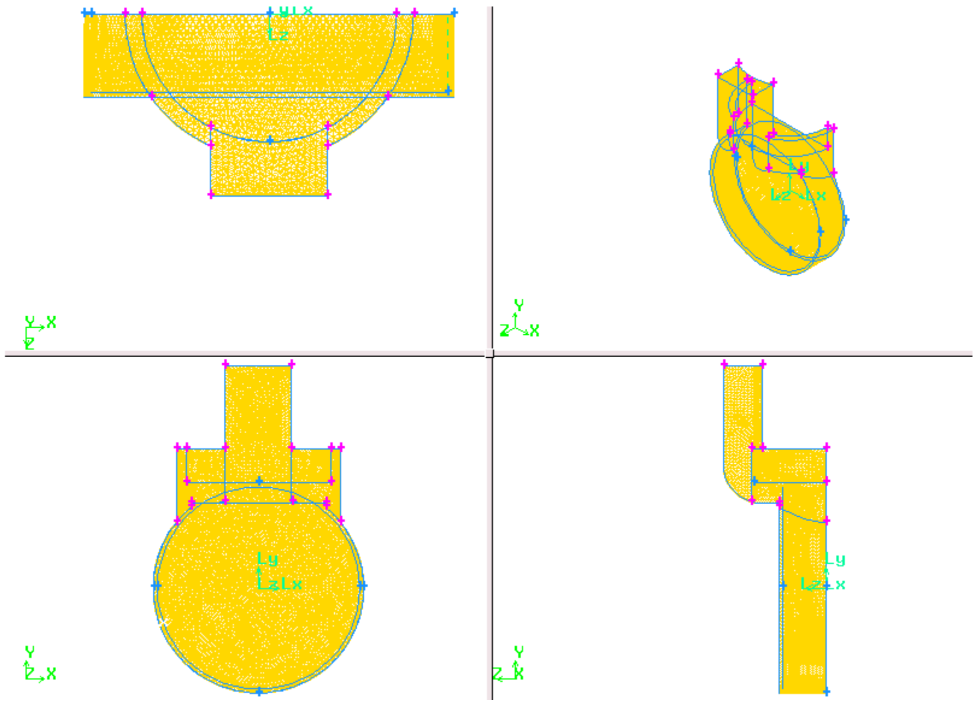

Figure 3.11: Crankcase mesh generation

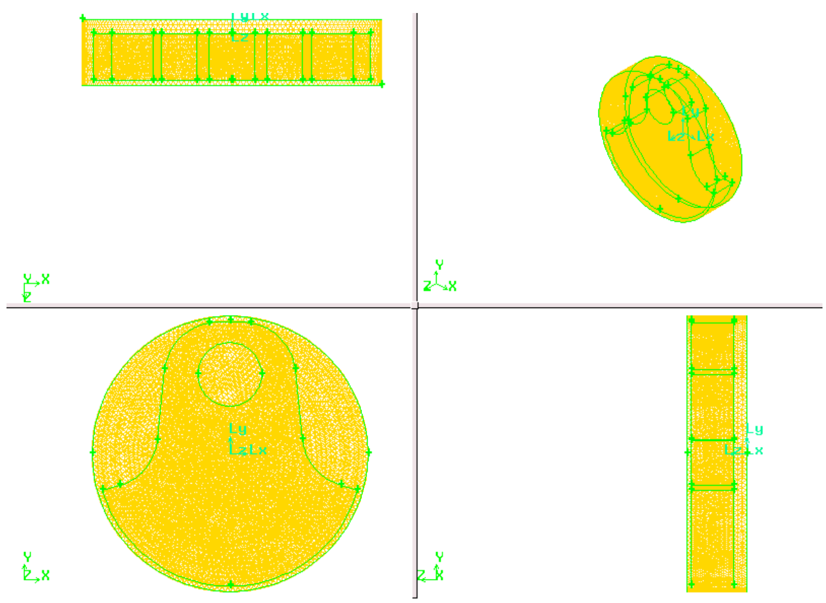

Figure 3.12: Crankshaft mesh generation

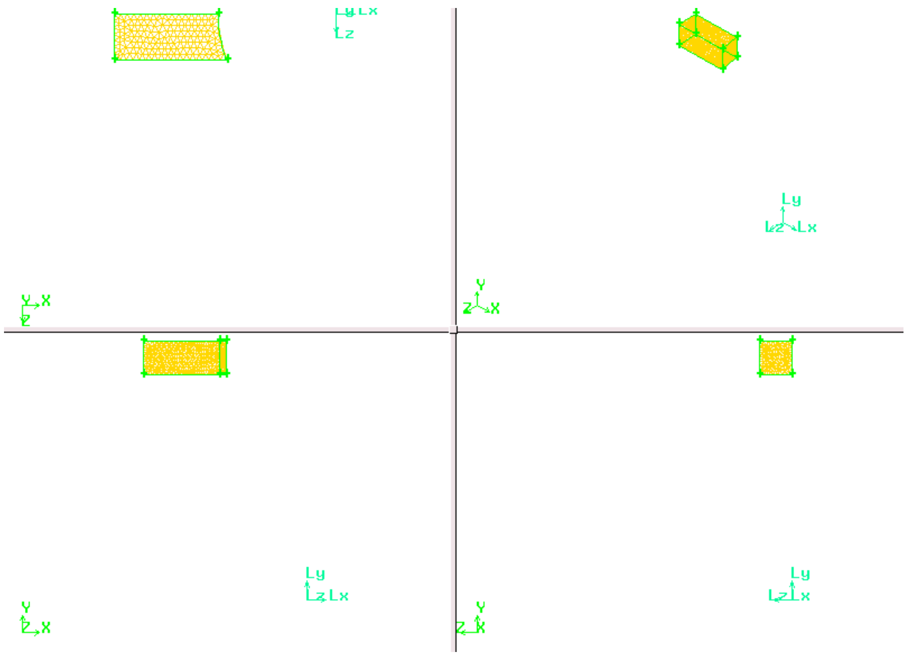

Figure 3.13: Intake port mesh generation

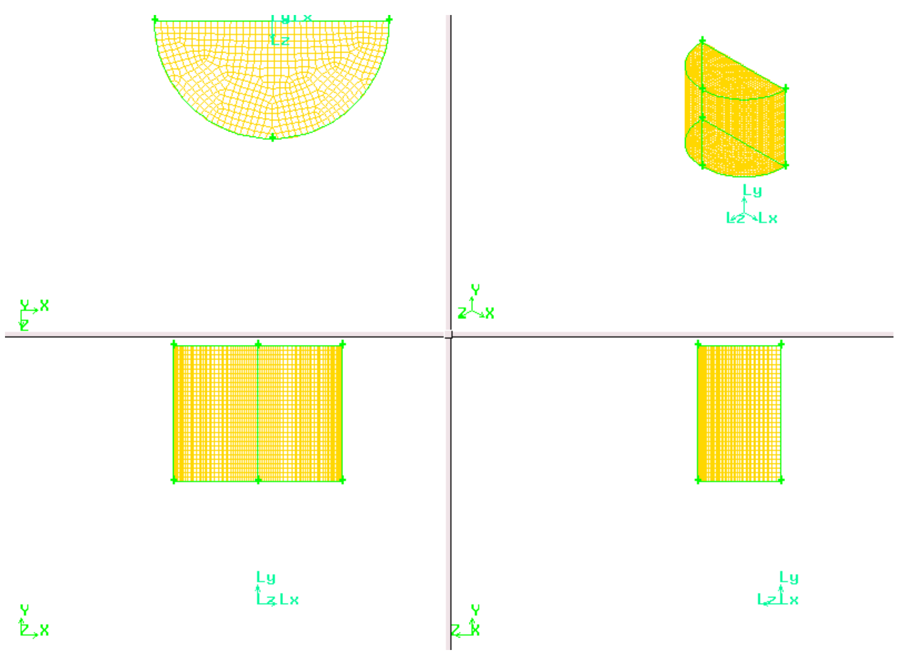

Figure 3.14: Deforming piston mesh generation

3.6.2.1 Geometry Setting

The inner geometry of crankcase imported as ACIS format file from SOLIDWORK into GAMBIT for mesh generation. The geometry of the actual engine was model as half model due to the symmetry of the crankcase layout [14]. The main component includes piston, crankcase, crankshaft and intake port.

Simplification for geometry in order to minimize the skew edges was carried out at this stage. This purpose is to reduce the critical area of the geometry that might affect the process of meshing and simulation period.

3.6.2.2 Mesh Setting

This analysis is using the transient (unsteady) analysis on dynamic mesh approaches using unstructured tetrahedral and layered hexahedral elements [14]. Layered hexahedral meshed is specified for moving part and unstructured tetrahedral element was for stationary region. The volume is divided and the mesh to fulfill the requirement of the dynamic mesh on transient analysis [14][30].

The mesh was constructed using four separate blocks representing the components. Their cell faces at the interface between each other matched exactly setting in the Tgrid stage. The four blocks are piston displacement, intake port, crankcase and crankshaft. The piston displacement was meshed using hexahedral mesh and the other blocks by tetrahedral mesh with the interval size of one.

3.6.2.3 Boundary Condition

The three main boundary types uses in the model were pressure, wall and interface, although the symmetry plane had also to specified as a symmetry boundary to ensure it has no interaction with the flow.

Wall boundaries were specified at the crankcase, piston displacement and crankshaft. The wall velocity simulated the action of rotating crankshaft on the fluid motion. The remaining unspecified surfaces were assigned to the default wall boundary.

Specific boundary type for component defined as interface, wall, symmetry, inlet pressure and pressure outlet accordingly. Interface allow flow go through. There are four interfaces defined in this model involve of sliding wall for crankcase and crankshaft, deforming wall and inlet port wall. Specify continuum type of all the volumes are defined as fluid.

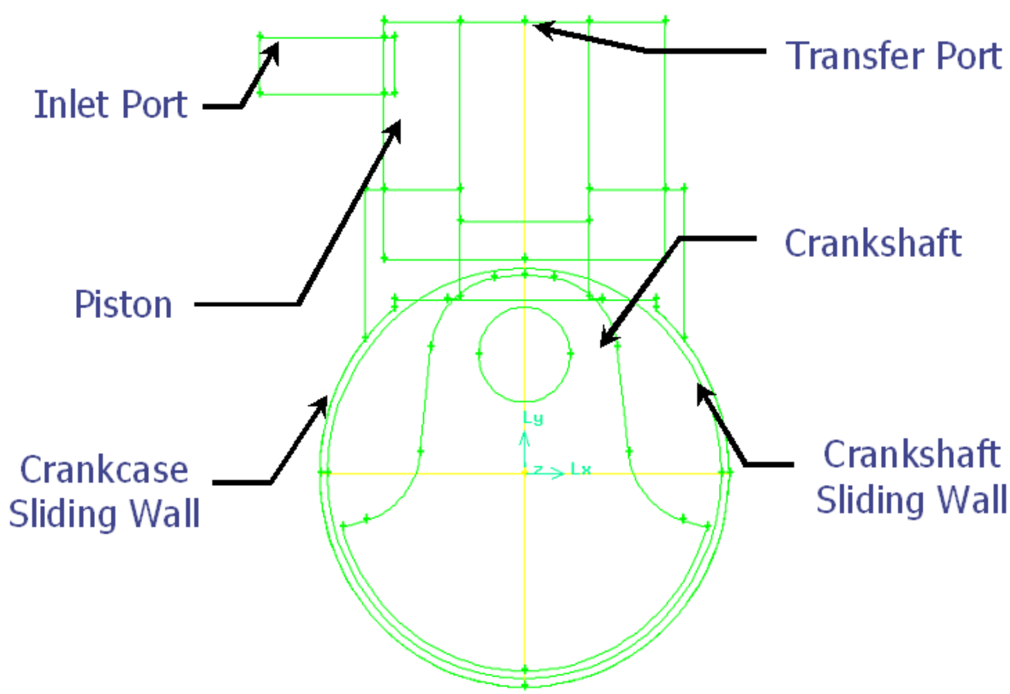

Figure 3.15: Three-dimensional model labeling

3.7 SIMULATION

3.7.1 Simulation by COSMOS FLOWORKS

In this process, the simulation of this engine will perform. For the initial study, analysis performed using COSMOS Flowork on the modified drawing. There are some parameters has to be consider when doing this analysis. The mass flow rate and pressure can be use as parameters to apply for the boundary condition [14][26]. Due to this software only capable to analyze the steady state condition, it is useful to determine the flow pattern inside this engine, which will use as guidance for the real simulation using FLUENT.

3.7.2 Simulation by FLUENT

Computational Fluid Dynamic (CFD) is a major tool in internal combustion engine analysis [30]. FLUENT as one of the CFD code was use in this simulation due to it ability to deal with multi-block solutions domains with moving wall [14]. The computational domain was produced based on the actual and run for half model. The finite volume enclosed the domain started from the intake port and ended at the transfer port locations. The analysis is limited foe motored condition based on three different engine speed (rpm). Below are some steps for the simulation using FLUENT:

Computational Model Setup → **Boundary Condition** → **Mesh Motion** → **Define Parameter** → **Run**

Figure 3.16 : Simulation steps by FLUENT

3.7.2.1 Computational Model Setup

This stage determines the exact geometry to be simulated by using FLUENT code. Meshed model from GAMBIT and Tgrid is imported into FLUENT interface. The function of grid check is to ensure all the meshes are in good condition in the aspect of skewness and critical angle before proceeding with the next process.

Solver of this model defined as unsteady with the first order of implicit since considering the dynamic mesh. $k-\varepsilon$ viscous model with two equations was use. The $k-\varepsilon$ model is one of the most common turbulence models. It is a two-equation model. It includes two extra transport equations to represent the turbulent properties of the flow.

The fluid passing through the engine was set to be air and was allowed to be compressible by the inclusion of the solution of enthalpy; the density was calculated through the use of the ideal gas law [14]. Besides, the boundary condition of sliding and deforming mesh are clearly defined at this stage. The surfaces that represented the interface of the flow domain crankshaft were assumed to have a temperature of 300 K and a wall velocity of 115.19 rad/s. Then, top of deforming cylinder were set to be a rigid body and wall of deforming cylinder assume to be moving mesh with the origin at TDC with coordinate.

In addition, by using FLUENT, it is provided a built-in function to calculate the piston location as a function of crank angle. Using this function, it is needed to specify the piston stroke as 30mm and connecting road length as 57.1mm. The piston location is calculated using the following expansion [14]:

$$p_s = L + \frac{A}{2}\left(1-\cos(\theta_c)\right) - \sqrt{L^2 - \frac{A^2}{4}\sin^2(\theta_c)} \tag{3.1}$$

where p_s is piston position (0 at TDC) and A at BDC, L is connection rod length, A is piston stroke, and θ_c is current crank angle position.

3.7.2.2 Motion Check

Some of the boundary condition unable to define using GAMBIT, therefore sliding and deforming mesh was set-up in FLUENT before going into motion check. Crankshaft and piston assembly as the sliding and deforming mesh was set to speed.

Piston and crankshaft motion within an actual engine dictate that the computational mesh must move if is so to accurately reproduce the system that it is modeling [36]. Therefore, the simulation was set up as a transient problem, start at TDC and running for one complete engine revolution at 360º. Each revolution of the engine was broken down into 360 time steps.

Mesh motion show the exact movement of internal engine part for simulation. Visualization of TDC and BDC of deforming wall is the critical point to ensure the boundary condition defined is correct.

3.7.2.3 Define Parameter

This stage use to define the focus parameters that will display as result. Crankcase pressure and the velocity are the main concern in this simulation. The outcome of this simulation was set to display the pressure value for one complete cycle.

Discretisation of the momentum, turbulence kinetic energy, turbulence dissipation rate and energy are set to be Second Order Upwind. It is useful for reduce numerical diffusion since the model mesh using different meshes.

Each time step was divided into 20 iterations for calculation. It consumed about 24 hours and only can be run at high memory computer to finish one complete cycle iteration.

CHAPTER 4

RESULTS AND DISCSUSSIONS

4.1 INTRODUCTION

This chapter will show the prediction of simulation results from FLUENT. Besides, the simulation by COSMOS FLOWORKS as the initial study also showed. In the other hand, visualization of the vectors of velocity magnitude was included together with the static pressure contours inside crankcase.

4.2 INITIAL STUDY RESULTS

Flow pattern inside crankcase of this engine shows the turbulence and swirl condition from intake port until the crankcase. According to the theory of intake stroke, air-fuel mixture is first drawn into the crankcase by the vacuum created during the upward stroke of the piston [10].

4.2.1 Pressure Distributions

Crankcase pressure introduce to the stage of the characteristics of in-crankcase behavior. Pressure at the intake port is higher than crankcase because the movement of piston creates the vacuum inside the crankcase as shown in the Figure 4.1 from COSMOS.

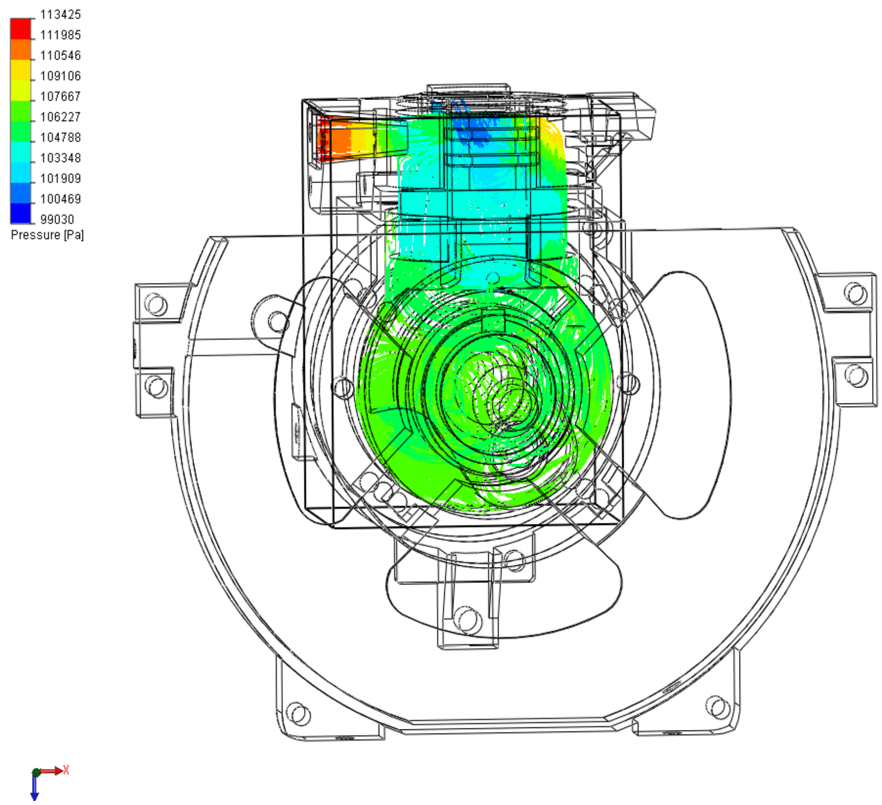

Figure 4.1 : Pressure distributions between intake port and crankcase

4.3 FLUENT Simulation Results

FLUENT version 6.1 was used as the simulation tool for this project due to it capability to simulate the unsteady state of flow condition. The set-up processes are clearly described in Chapter 3. FLUENT provide more satisfactory results to explore more on this study.

4.3.1 Prediction of Crankcase Pressure

Figure 4.2 shows the crankcase pressure results as a function of crank angle degree 1100 rpm. The highest value of crankcase pressure shows 2.1 bar at 178 crank angle degree (CA). Intake port pressure assumed to be at atmospheric pressure with the value of 1.1 bar. During the simulation, the movement of piston from top dead centre (TDC) to bottom dead centre (BDC) compress the air inside the crankcase as the graph showing the increment of pressure. The intake of the fuel/air/oil mixture occurs during the low-pressure phase [10]. After the transfer port open at 140 CA after TDC, the pressure continues to increase slowly until the maximum. The crankcase pressure started to drop after BDC and started to create vacuum inside the crankcase. Thus, the pressure keeps on decrease after the transfer port close until the crankshaft complete one rotation at 360 CA.

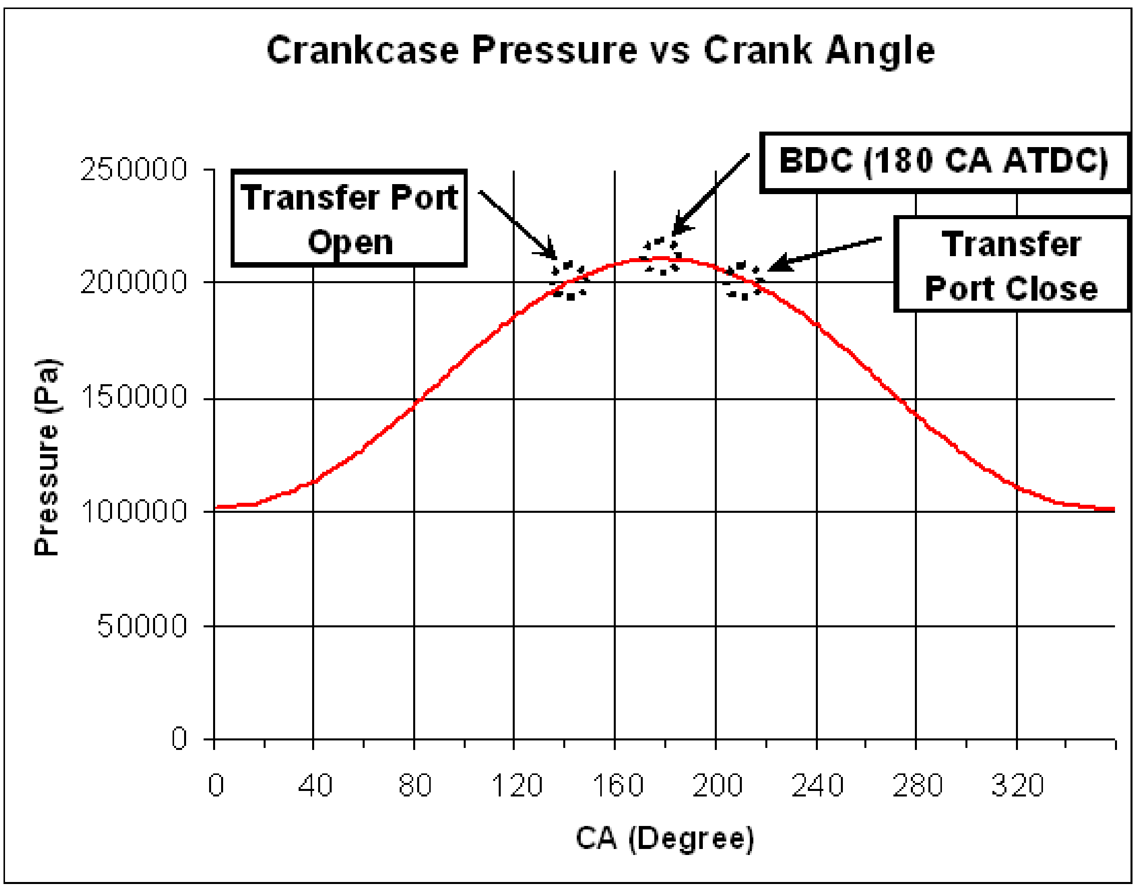

Figure 4.2 : Crankcase pressure versus crank angle at 1100 rpm

As an additional, the crankcase pressure of two-stroke spark ignition engine can be predicted by using the isentropic compression formula as below:

$$P_2 = P_1 \left(\frac{1}{r_c}\right)^k \tag{4.1}$$

Assume k = 1.35 and P_1 as the intake pressure (101325 Pa);

$$r_c = \frac{V_{TDC}}{V_{BDC}} \tag{4.2}$$

$$r_c = \frac{8.7 \times 10^{-5}}{7.1 \times 10^{-5}} = 0.82$$

$$\text{So, } P_2 = 101325 \left(\frac{1}{0.82}\right)^{1.35}$$

$$P_2 = 132454 \text{ Pa} \ @ \ 1.3 \text{ bar}$$

Compare between the simulation and the calculation result, the difference between the peak values of crankcase pressure is about 0.8 bar. The reason is the simulation result may give higher values because refer to ideal cycle and system assumed to be no heat loss. Besides, the model geometry was simplified at the critical angle and region in order to avoid the skewness error.

In addition, the choice of turbulence model can have a considerable effect on the accuracy of the flow predictions. The simulation utilized the k-e turbulence model that is based on the eddy-viscosity concept, which assumes that the turbulence is isotropic [50]. Isotropic turbulence models tend to generate spuriously high turbulence levels in regions of significant streamline curvature and consequently an overestimation of the 'turbulent viscosity' and reproduce the effects of streamline pressure gradients.

4.3.2 Displays the vectors and contours of velocity

By displaying the contours of velocity inside the crankcase, the actual flow process can be visualized for one complete cycle. There is no other method can be used to determine the flow inside crankcase except using CFD simulation or laser capture. FLUENT is the useful code of CFD in generates the high quality picture of crankcase flow.

Vectors and contours of velocity show that high velocity after transfer port open at 120 CA as shown in Figure 4.3 and 4.7. After the transfer port close at 240 CA, the velocity become slowly due to the air trapped inside the crankcase. Refer to Figure 4.6 and 4.6, the high tumble velocity occurs at the curve shape area of crankshaft. Due to the crankshaft and crankcase area are too small, the tumble velocity vector is unable to view clearly. The vector of velocity is large affected by the shape of crankshaft since it designed like a propeller. For example in the Figure 4.4 and 4.8, the crankshaft position influences the velocity of flow due to the propeller shape. There is deflection of airflow from crankshaft wall to crankcase wall once hit the rotating crankshaft.

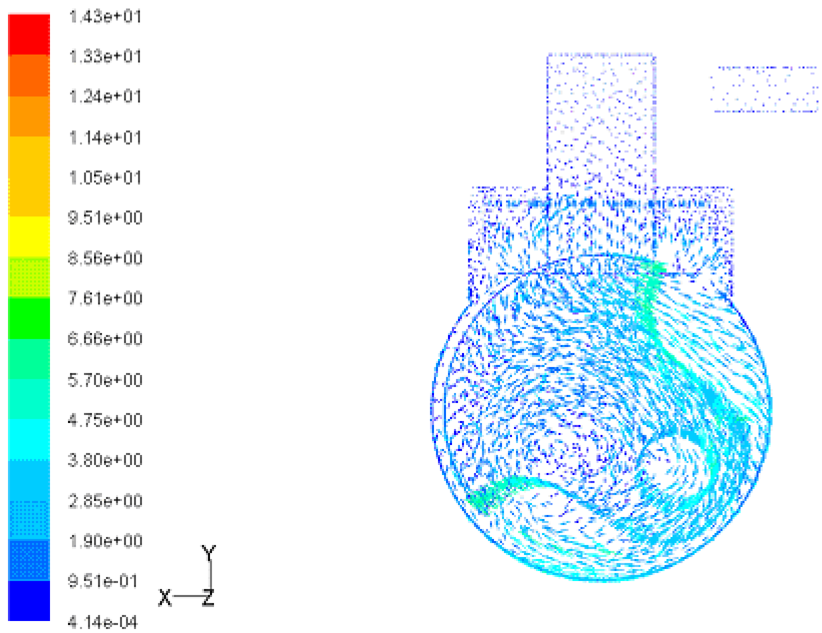

Figure 4.3 : Velocity vector at 120 CA

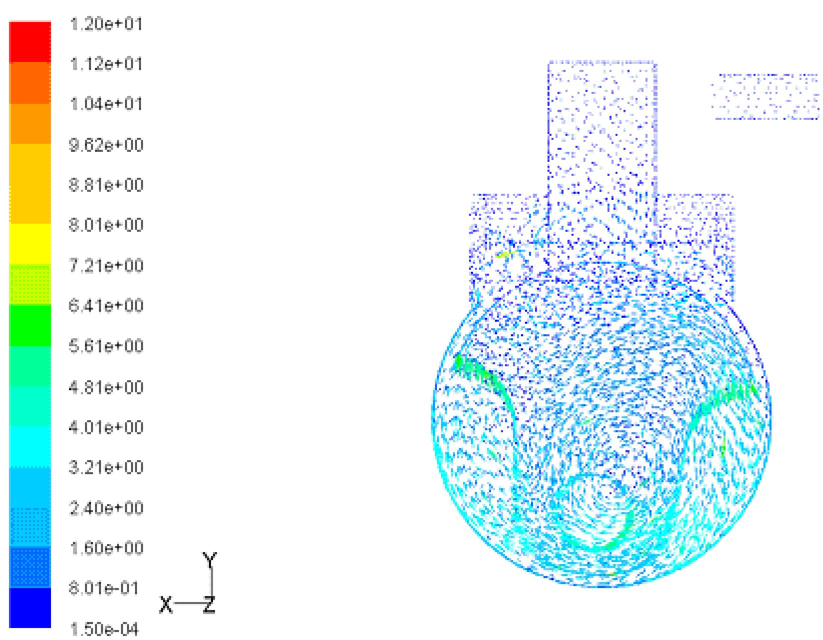

Figure 4.4 : Velocity vector at 180 CA

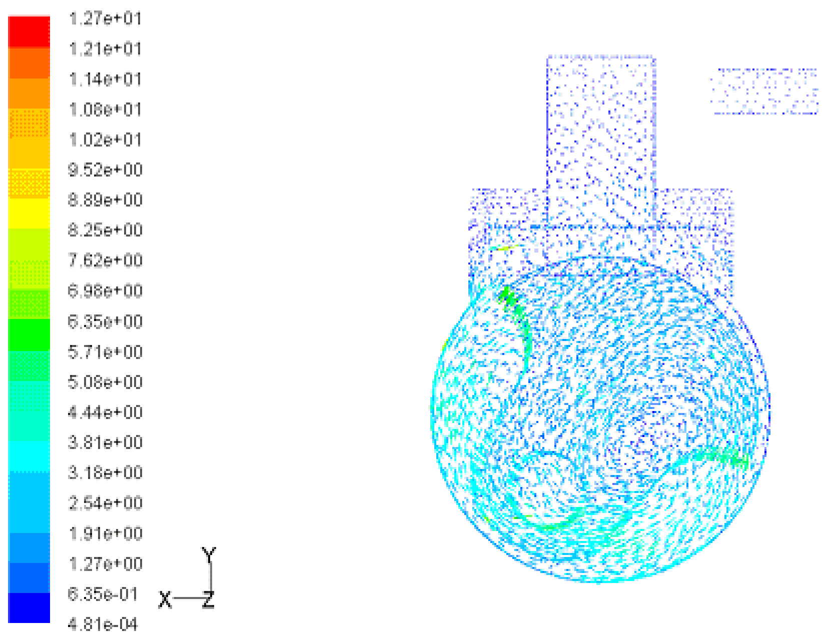

Figure 4.5 : Velocity vector at 210 CA

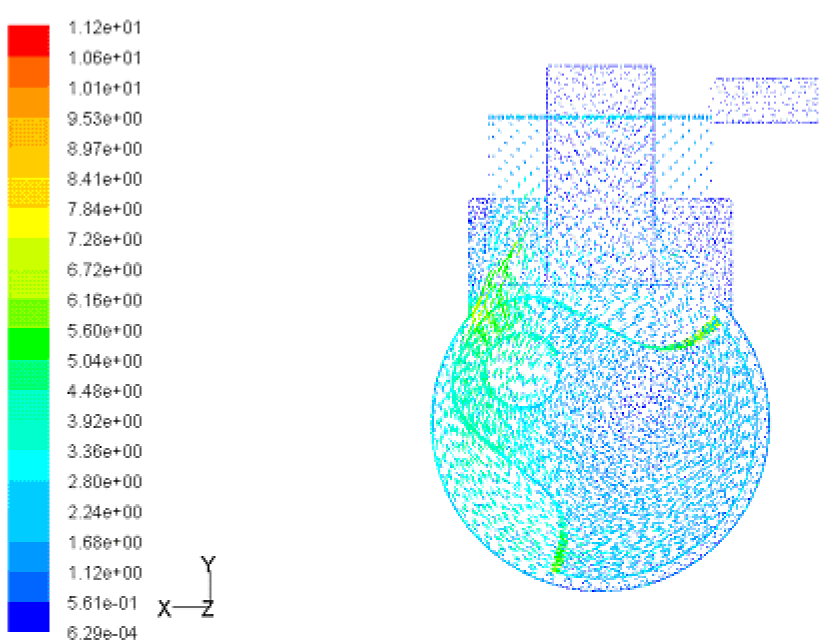

Figure 4.6 : Velocity vector at 300 CA

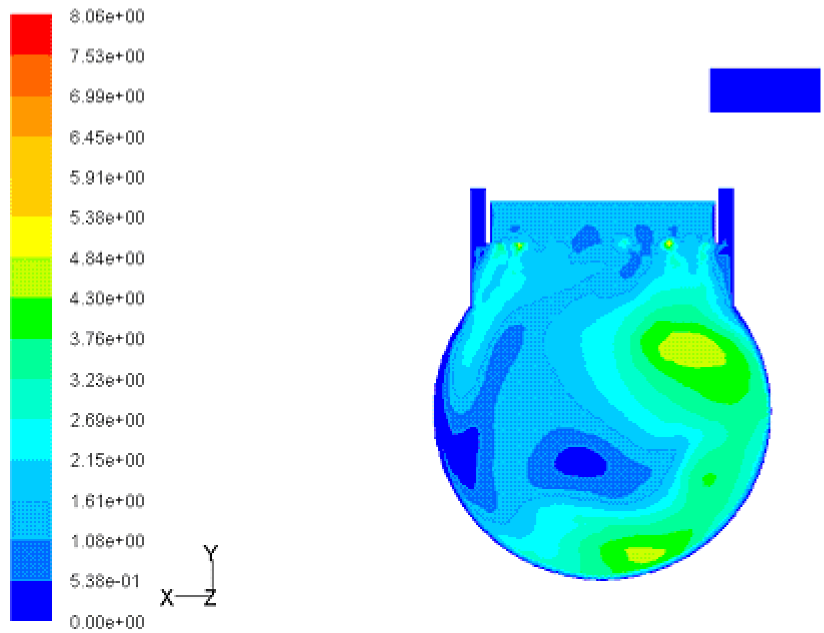

Figure 4.7 : Velocity contour at 120 CA

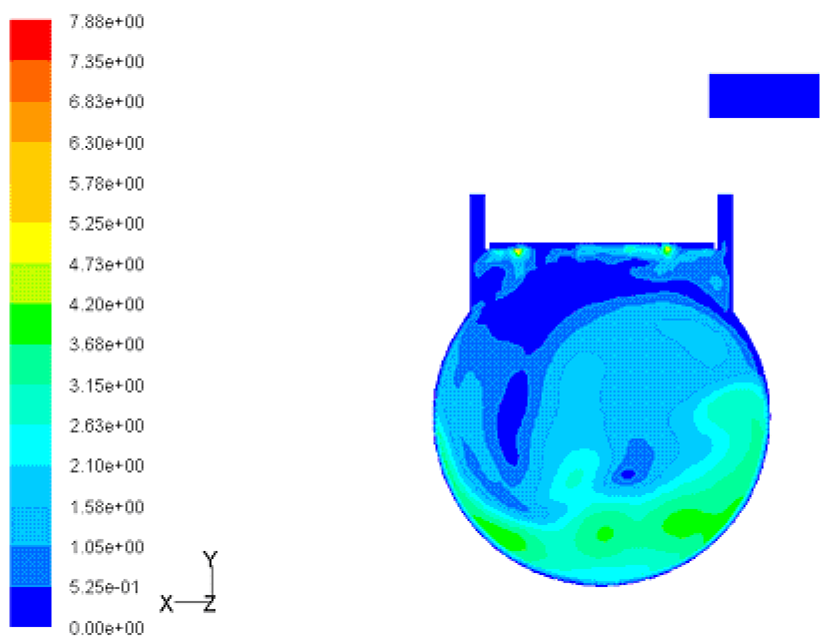

Figure 4.8 : Velocity contour at 180 CA

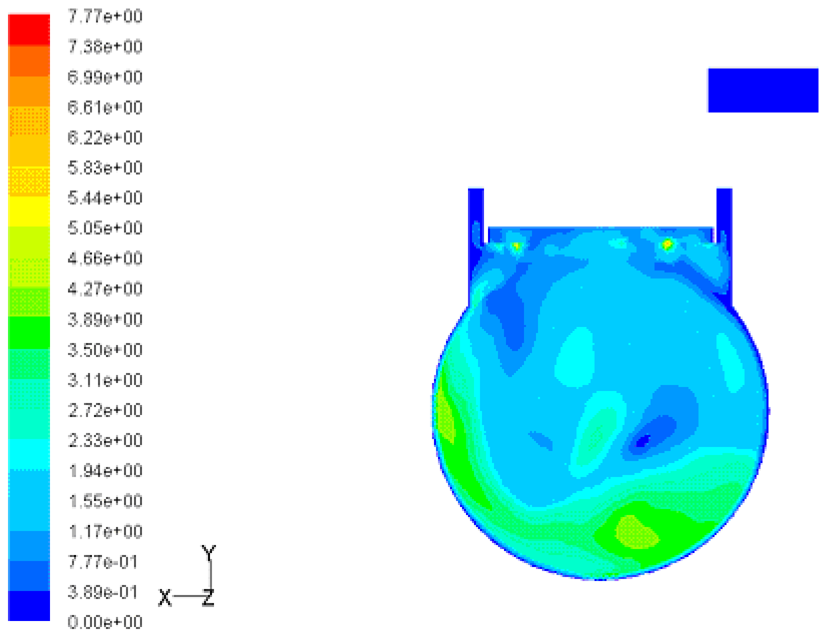

Figure 4.9 : Velocity contour at 210 CA

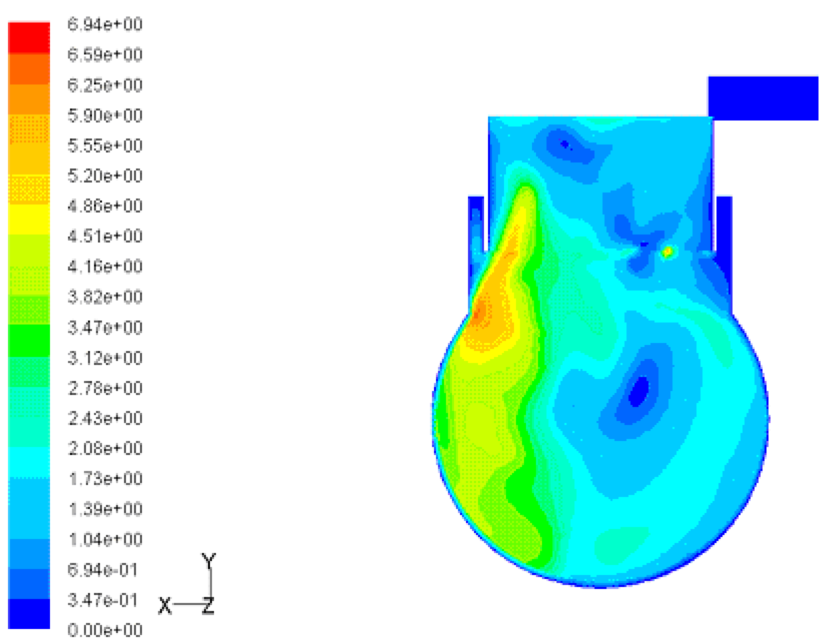

Figure 4.10 : Velocity contour at 300 CA

4.3.3 Displays the contours of temperature

The contours of temperature shows that before transfer port open at 140 CA ATDC, the highest value is about 319 K in side crankcase as shown is Figure 4.11. After the transfer port opening, the average temperature increase to 326 K. The characteristics of crankcase temperature generally effect by piston deforming, location of piston and the movement of crankshaft. According to the relationships between pressure *P* and temperature *T*, if the pressure increase, the temperature also increase. Refer to Figure 4.14, after the transfer port close at 240 CA ATDC, the temperature value of crankcase decreases to 307 K. Movement of piston from BDC to TDC creates vacuum and the pressure drop consequence with the temperature.

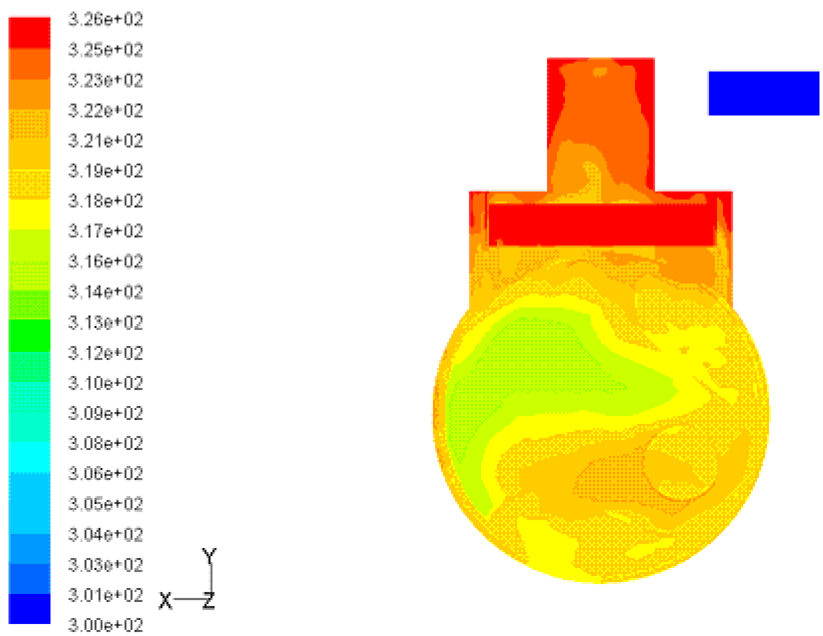

Figure 4.11: Contour of static temperature at 120 CA

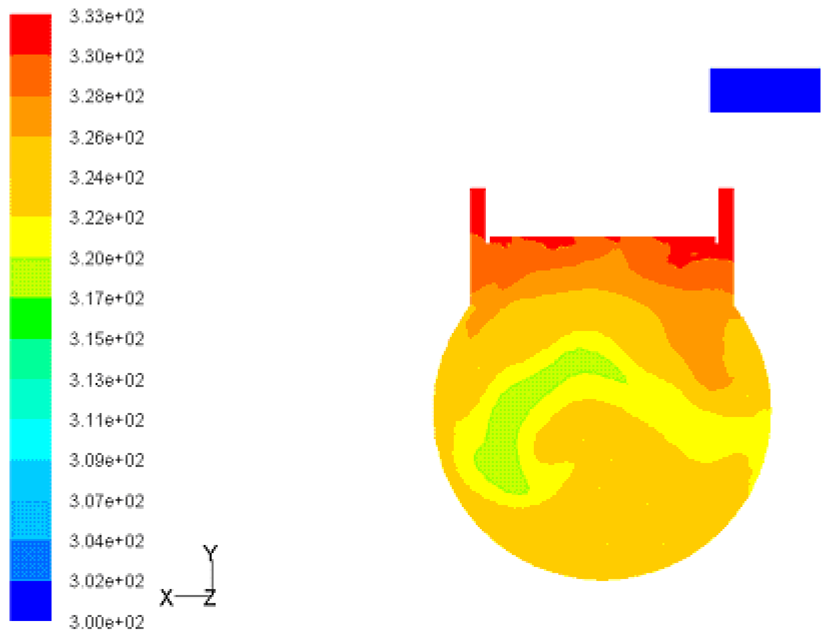

Figure 4.12: Contour of static temperature at 180 CA

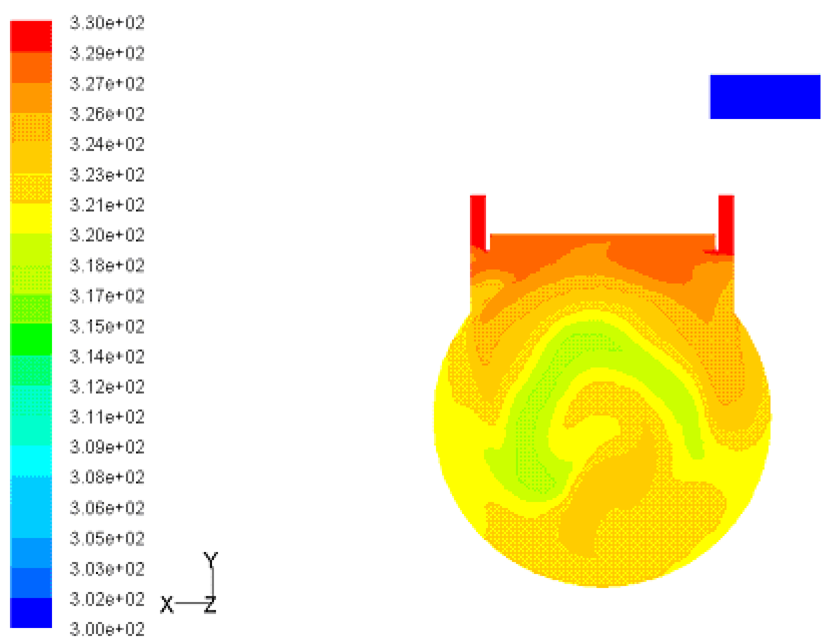

Figure 4.13: Contour of static temperature at 210 CA

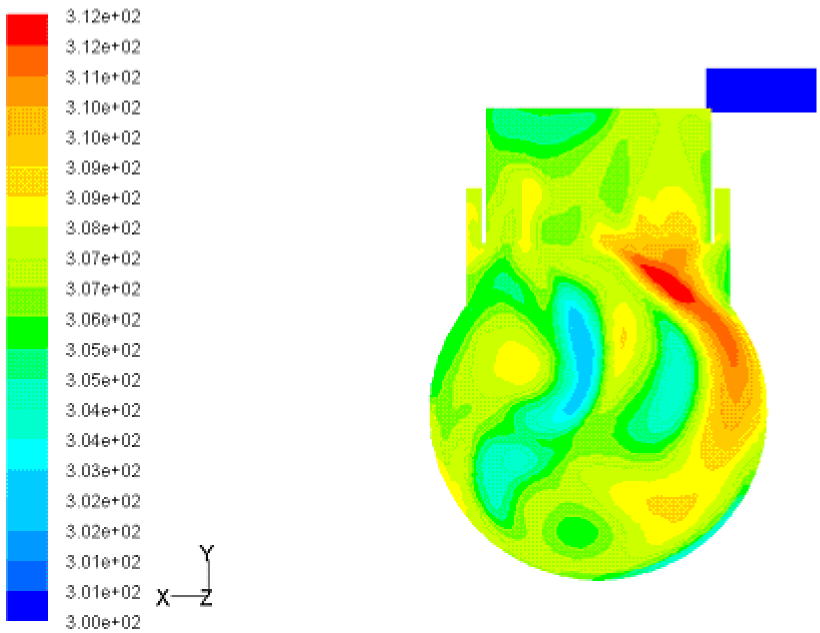

Figure 4.14: Contour of static temperature at 300 CA

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 INTRODUCTION

This chapter summarizes the important findings from the work that carried out from this project. Simulation or numerical approaches is the main concern in this project. In addition, this chapter will include some recommendations and suggestions for future work for this research.

5.2 CONCLUSION

The aim of this project is to analyze the crankcase flow of two-stroke spark ignition engine. By using pressure and temperature as the boundary condition, the characteristic of velocity and pressure inside crankcase at 1100 rpm was determined. Besides, modeling process by using CFD tool was successfully done. By generate this model, future researcher can use as a guide for further analysis.

5.3 RECOMMENDATIONS FOR THE FUTURE RESEARCH

This research still has to carry forward to improve the accuracy of the result. To converge the results, further simulation should be carried out. The recommendations are mentioned as below:

- Use boundary condition of actual running engine for more accurate simulation such as the intake temperature and transfer port pressure.

- Prediction of crankcase pressure by engine test rig to compare the actual engine running result with simulation.

- Velocity visualization results should be comparing with other method result such as laser application.

- Simulation should be run with high-speed devices such as super computer to minimize the calculating time and improve accuracy of results.

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