BACHELOR THESIS & COLLOQUIUM – ME 141502

TECHNICAL ANALYSIS OF FIN EFFECT IN INLET VALVE ON AIR-GAS INTAKE FLOW OF MODIFICATION YANMAR TF 85 MH - DI; BASED ON COMPUTATIONAL MODELLING

Muhamad Azka Asykarullah
NRP 04211441000021

Supervisors:
Prof. Semin, S.T., M.T., Ph.D.
Beny Cahyono, S.T., M.T., Ph.D.

Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Surabaya
2018

BACHELOR THESIS & COLLOQUIUM – ME 141502

TECHNICAL ANALYSIS OF FIN EFFECT IN INLET VALVE ON AIR-GAS INTAKE FLOW OF MODIFICATION YANMAR TF 85 MH - DI; BASED ON COMPUTATIONAL MODELLING

Muhamad Azka Asykarullah
NRP 04211441000021

Supervisors:
Prof. Semin, S.T., M.T., Ph.D.
Beny Cahyono, S.T., M.T., Ph.D.

Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Surabaya
2018

“This page deliberately left blank”

BACHELOR THESIS & COLLOQUIUM – ME 141502

TECHNICAL ANALYSIS OF FIN EFFECT IN INLET VALVE ON AIR-GAS INTAKE FLOW OF MODIFICATION YANMAR TF 85 MH - DI; BASED ON COMPUTATIONAL MODELLING

Muhamad Azka Asykarullah
NRP 04211441000021

Supervisors:
Prof. Semin, S.T., M.T., Ph.D.
Beny Cahyono, S.T., M.T., Ph.D.

Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Surabaya
2018

“This page deliberately left blank”

APROVAL SHEET

TECHNICAL ANALYSIS OF FIN EFFECT IN INLET VALVE ON AIR-GAS INTAKE FLOW OF MODIFICATION YANMAR TF 85 MH - DI; BASED ON COMPUTATIONAL MODELLING

BACHELOR THESIS

Submitted to Comply One of The Requirements to Acquire a Bachelor of Engineering Degree in Double Degree of Marine Engineering Program
Department of Marine Engineering - Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Department of Marine Studies
Hochschule Wismar, University of Applied Sciences

Submitted by:
Muhamad Azka Asykarullah
NRP. 04211441000021

Approved by Thesis Supervisors:

Prof. Semin, S.T., M.T., Ph.D. ( )
NIP. 1971 0110 1997 02 1001

Beny Cahyono, S.T., M.T., Ph.D. ( )
NIP. 1979 0319 2008 01 1008

SURABAYA
JULI, 2018

"This page deliberately left blank"

APROVAL SHEET

TECHNICAL ANALYSIS OF FIN EFFECT IN INLET VALVE ON AIR-GAS INTAKE FLOW OF MODIFICATION YANMAR TF 85 MH - DI; BASED ON COMPUTATIONAL MODELLING

BACHELOR THESIS

Submitted to Comply One of The Requirements to Acquire a Bachelor of Engineering Degree in Double Degree of Marine Engineering Program
Department of Marine Engineering - Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Department of Marine Studies
Hochschule Wismar, University of Applied Sciences

Submitted by:
Muhamad Azka Asykarullah
NRP. 04211441000021

Approved by
Head Department of Marine Engineering:

KEMENTERIAN RISET, TEKNOLOGI DAN PENDIDIKAN TINGGI
INSTITUT TEKNOLOGI SEPULUH NOPEMBER
FAKULTAS TEKNOLOGI KELAUTAN
DEPARTEMEN TEKNIK SISTEM PERKAPALAN

Dr. Eng. M. Badrus Zaman., ST., MT
NIP. 197708022008011007

“This page deliberately left blank”

DECLARATION OF HONOUR

With signed below, I declare that:

This thesis has been written and developed independently without any plagiarism act. All contents and ideas drawn directly from internal and external sources are indicated such as cited sources, literatures, and other professional sources.

Student Name	: Muhamad Azka Asykarullah
Student ID Number	: 0421144100021
Thesis Title	: Technical Analysis of Fin Effect In Inlet Valve on Air-Gas Intake Flow of Modification Yanmar Tf 85 Mh - Di; Based on Computational Modelling
Department	: Double Degree Marine Engineering Program Faculty of Marine Engineering Institut Teknologi Sepuluh Nopember

Surabaya, July 3, 2018

Muhamad Azka Asykarullah

“This page deliberately left blank”

Technical Analysis of Fin Effect in Inlet Valve on Air-Gas Intake Flow of Modification Yanmar Tf 85 Mh - Di; Based on Computational Modelling

Student Name	**: Muhamad Azka Asykarullah**
NRP	**: 04211441000021**
Departement	**: Double Degree of Marine Engineering**
Supervisors	**: Prof. Semin, S.T., M.T., Ph.D.**
	Beny Cahyono, S.T., M.T., Ph.D.

Abstract

Diesel engine commonly used because of their good reliability, high combustion efficiency, and high cost effectiveness. However, diesel engine is one of any machinery that causes environmental pollutions, which these elements are the main constituents of acid rain and human health disorders Dual fuel engine is one solution to reduce the amount of air pollution expenditure caused by the use of diesel fuel. the purpose of fin additing on air intake valve in modification engine YANMAR TF 85 MH-DI is produce more turbulence intensity especialy of swirl flow. this research simulating air flow through intake valve used CFD simulation with Ansys Fluent. there are three compared valve on this research which are, conventional valve, 3 finned intake valve, and 5 finned intake valve on two different situation, that are 372° and 402° of crank angle. The result of simulation can be concluded that the addition of fin produced more the turbulence intensity. the best produced turbulence intensity was by 5 finned valve with 0.9651 followed by 3 finned valve with 0.9040 and by conventional valve with 0.8707.

Keywords: CFD Simulation, Dual Fuel Engine, Fin Addition, Intake Valve, Turbulence Intensity

“This page deliberately left blank”

Analisa Teknis Terhadap Pengaruh Penambahan Fin Pada Katup Hisap pada Aliran Udara-Gas Masuk Pada Mesin Modifikasi Yanmar TF 85 MH - DI; Berbasis Pemodelan Komputasi

Nama Mahasiswa : Muhamad Azka Asykarullah
NRP : 04211441000021
Departemen : Double Degree of Marine Engineering
Pembimbing : Prof. Semin, S.T., M.T., Ph.D.
Beny Cahyono, S.T., M.T., Ph.D.

Abstrak

Mesin diesel umumnya digunakan karena keandalannya yang baik, pembakaran efisiensi tinggi, dan efektivitas biaya yang tinggi. Namun, mesin diesel adalah salah satu permesinan yang menjadi penyebab utama pencemaran lingkungan, yang mana pencemaran lingkungan dari bahan bakar diesel tersebut merupakan penyusun utama hujan asam dan permasalahan kesehatan. Mesin berbahan bakar ganda merupakan salah satu solusi untuk mengurangi jumlah polusi udara yang disebebkan oleh bahan bakar mesin diesel. Tujuan penambahan sirip pada air intake valve pada mesin modifikasi YANMAR TF 85 MH-DI adalah menghasilkan intensitas turbulensi yang lebih besar terutama dari aliran pusaran. Penelitian ini mensimulasikan aliran udara melalui intake valve menggunakan simulasi CFD dengan Ansys Fluent. ada tiga katup berbedakatup hisap konfensional, katup hisap bersirip 3, dan katup hisap bersirip 5 pada dua situasi yang berbeda, yaitu 372 ° dan 402 ° dari sudut engkol. Hasil simulasi dapat disimpulkan bahwa penambahan sirip menghasilkan intensitas turbulensi yang lebih besar. intensitas turbulensi terbaik dihasilkan oleh 5 katup bersirip dengan 0,9651 diikuti oleh 3 katup bersirip dengan 0,9040 dan oleh katup konvensional dengan 0,8707.

Kata Kunci: CFD Simulasi, Dual Fuel Engine, Fin Addition, Intake Valve, Turbulence Intensity

“This page deliberately left blank”

PREFACE

Praise be to Allah Almighty Azza Wa Jalla, who has given His grace and grace, so that the writer is able to finish the Bachelor Thesis with the title of **Technical Analysis of Fin Effect in Inlet Valve on Air-Gas Intake Flow of Modification Yanmar TF 85 MH - DI; Based on Computational Modelling** as well and on time. This Thesis is submitted to Comply One of The Requirements to acquire a Bachelor of Engineering Degree in Double Degree of Marine Engineering Program in Department of Marine Engineering, Faculty of Marine Technology, Institut Teknologi Sepuluh Nopember Surabaya.

In the process of completion of this Bachelor Thesis and the success of undergraduate study program, can not be separated from the help and support from various parties. Therefore, the authors express gratitude to the parties below, namely:

1. Both parents of the author, Mrs. Aini Hilmiyah Qurata'ayun and Mr. Abdullah Azzam who always support and give spirit to the author of every activity and activity to date and reminded to obey the worship.
2. Mr. Ir. Aguk Zuhdi Muhammad Fatahallah, M.Eng., Ph.D. as a lecturer advisor of the author during collage on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi SepuluhNopember.
3. Mr. Prof. Semin, S.T., M.T., Ph.D as a supervisor of the author and lecturer of MPP Laboratory who has guided the author in completing this Bachelor Thesis and providing motivation to continue develop themselves.
4. Mr. Beny Cahyono, S.T., M.T., Ph.D as a supervisor of the author and lecturer of MPP Laboratory who has guided the author in completing this Bachelor Thesis and providing motivation to continue develop themselves.
5. Mr. Dr. Eng. Muhammad Badrus Zaman, ST., MT as a Head of Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
6. To all my brother and sister, Sibghotulloh Aliy Asyauqi, Khanza Mujahidah Biizatillah, Izzudin Hikmatyar Al Madani, and Sarah Huriyah Qurata'ayun who always give their prayers and support for any author chooses.
7. Hana Rafidah Hasan who always be author's loyal supporter in every conditions.
8. To all member of ITS Marine Solar Boat Team Family who always be author's personal support during collage on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
9. To all member of Marine Power Plant Laboratory on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
10. To all my High School mate of SMAN 1 Surabaya who always entertaining in any situation and be a great discussion partner.

11. Mercusuar ’14 and Double Degree Marine Engineering ’14 for assistance and support.
12. Bismark ’12 and Double Degree Marine Engineering ’12 as a great senior mate during collage on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
13. Barakuda ’13 and Double Degree Marine Engineering ’13 as a great senior mate during collage on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
14. Salvage ’15 and Double Degree Marine Engineering ’15 as a great junior mate during collage on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
15. Voyage ’16 and Double Degree Marine Engineering ’16 as a great junior mate during collage on Marine Engineering Department, Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember.
16. Great amounts of friends that can’t be mentioned, for their support, help and ideas for this thesis.

The author realizes that the research done in this Bachelor Thesis is far from being perfection, so criticism and suggestions are very open to making better work and providing usefulness.

The author hopes that this final work can be useful for the author and for all readers in the future.

Surabaya, July 2018

Author

TABLE OF CONTENTS

TABLE OF FIGURES

TABLE OF TABLES

"This page deliberately left blank"

CHAPTER I
INTRODUCTION

1.1 Background

Many of the modern-day industries use diesel engines such as power plant industry, oil and gas industry, mining vehicles, and marine transportation. Diesel engines are typical engine with internal combustion. This engine commonly used because of their good reliability, high combustion efficiency, and high cost effectiveness (Bayraktar et al., 2008). However, diesel engine is one of any machinery that causes environmental pollutions such as particular matter (PM), sulfur oksida (SOx) and nitrogen oksida (NOx). which these elements are the main constituents of acid rain and human health disorders (Wei et al., 2015).

Natural gas is an alternative fuel with the main constituent components are methane gas (CH_4) (Semin et al., 2008; Wei et al., 2016). Natural gas is an environmentally friendly fuel because it contains less carbon per unit of energy than fossil fuels. The availability of natural gas reserves in nature are abundant at a relatively cheaper compared to gasoline and diesel fuel, but there is no method has been found to optimally utilize it (Semin, 2008).

Table 1. Natural Gas Composision

Component	*Typical Analysis (vol%)*	*Range (vol%)*
Methane	94.9	87-96
Ethane	2.5	1.8-5.1
Propane	0.2	0.1-1.5
Isobutane	0.03	0.01-0.3
n-Butane	0.03	0.01-0.3
Isopentane	0.01	Trace to 0.14
n-Pentane	0.01	Trace to 0.14
Hexane	0.01	Trace to 0.06
Nitrogen	1.6	1.3-5.6
Carbondioxide	0.7	0.1-1.0
Oxygen	0.02	0.01-0.1
Hydrogen	Trace	Trace to 0.02

(Wei et al., 2016)

Dual fuel engine is an effective solution to reduce the environmental pollutions that produced by conventional diesel engine as an emission (Ohashi et al, 2015). Dual fuel engine utilized the Compressed Natural Gas (CNG) as fuel aid. Methode of dual fuel engine is natural gas injected into the intake manifold to mixed with the air. Then, introduced them to the cylinder and ignited by direct injected high cetane number fuel. However, heat value of gas-air mixing of dual fuel engine is lower compared to conventional diesel engine and caused the combustion duration of dual fuel engine is longer. Therefore, the performance is lower than diesel engine (Saha et al., 2008).

In order to mend the duration of combustion duration and the performance of dual fuel engine, it needs to raise the kinetic energy inside the cylinder. Kinetic energy can be raised up by producing more turbulent intensity and high velocity flow (Yuesheng et al., 2007). Turbulent flow intensity also one of the factor to increase engine efficiency. Turbulent flow can be made by improving a fin object in inlet valve. In this thesis, author will analyze of fin effect on air-gas intake flow of dual fuel engine; based on computational modelling.

1.2 Research Problems

Based on background mentioned above, it can be concluded some problems of this thesis are:

1. Analyze how the effect of intake valve fin application in intake air/gas motion inside the combustion chamber on dual fuel engine?
2. Determine how the best shape, size, and numbers of fin in intake valve based on turbulent intensity?

1.3 Research Objectives

Based on problems mention above, the objectives of this thesis are:

1. To know the effect of intake valve fin application in intake air/gas motion inside the combustion chamber on dual fuel engine.
2. To know the best shape, numbers and size of fin in intake valve based on turbulent intencity.The object of this thesis is designed as Yanmar TF85Di Diesel engine.

1.4 Research Limitations

This thesis can be focused and organized, with limitations on problem which are:

1. This thesis use data input from engine guide, manual measurement, and engine simulation software.
2. Design modelling and flow simulating use 3D modelling and CFD Software Student License.
3. This thesis only analyzes the air/gas flow that caused by the application of modified inlet valve.
4. This thesis only analyzes the turbulent intensity and swirl ratio of air/gas flow.
5. This thesis the simulation ignores disturbance of environmental factors (Adiabatic).

1.5 Research Benefits

This thesis is expected to give benefits for the various kind of parties. The benefits that can be obtained are:

1. Provides information about the effect of intake valve fin application in intake air/gas motion inside the combustion chamber on dual fuel engine.
2. Provides information about the best shape, numbers and size of fin in intake valve based on turbulent intencity.
3. as a reference for other likely research of studying about the turbulent flow effect that caused by the application of finned inlet valve in dual fuel engine.

“This page deliberately left blank”

CHAPTER II
LITERATURE STUDY

2.1. Dual Fuel Engine

2.1.1. Four-Stroke Dual Fuel Engine

Gas fuel is supplied during the intake stroke of a diesel engine, and the exhaust emissions vary according to the proportions of diesel fuel and gas fuel in the total fuel see **Figure 2.1.** Because the concentration of CO2 in the exhaust gas depends on the composition of the fuel, it can be reduced by up to about 25% according to the proportions of diesel fuel and gas fuel. On the other hand, because the concentration of NOx depends on the combustion temperature, reductions in NOx can only be achieved by reducing the proportion of diesel fuel.

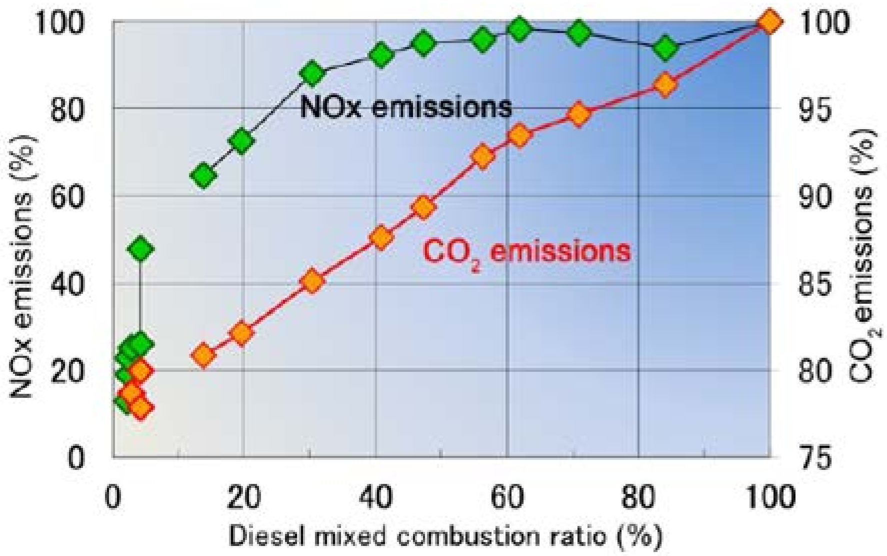


Figure 2.1. Emissions in Mixed Diesel/Gas Mode

While achieving the IMO Tier 3 regulations (80% reduction in NOx) requires that the proportion of diesel fuel be reduced to 2%, no currently available nozzles are able to inject the full range of fuel injection quantities, from the amount required under rated load down to this very small quantity, in a reliable manner (without variation in the quantity injected for each cycle).To overcome it, adding a small dedicated nozzle (micro-pilot fuel injector) that supplies diesel fuel for the ignition when operating on gas fuel is necessary.

Figure 2.2. shows a cross-section of a four-stroke dual-fuel engine. Gas fuel is supplied from a gas valve in the air intake manifold where it flows into the cylinder as a mixture with air. Two injectors, one is the main fuel injector used for diesel mode and another is the micro-pilot fuel injector used for gas mode, are equipped in the cylinder

head, the engine can switch between diesel mode and gas mode freely with certain operating condition restrictions.

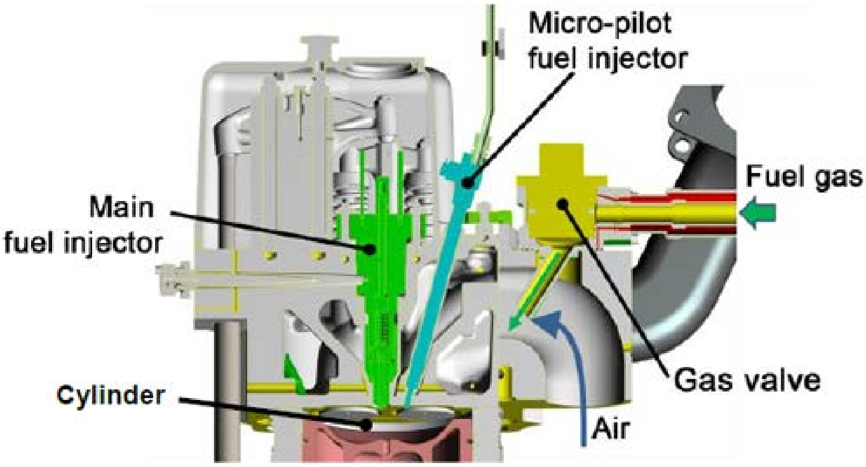


Figure 2.2. Cross-Section of Four-Stroke Dual-Fuel Engine While

2.1.2. Two-Stroke Dual Fuel Engine

Because intake air is used for scavenging in a two-stroke engine, it is not allowed to mix the gas fuel with the intake air. Instead, the gas fuel is injected into the compressed air in the same way as diesel fuel, and then ignition is achieved by injecting fuel via the micro-pilot fuel injector see **Figure 2.3.** Because this results in diffusion combustion, as with diesel fuel, it can reduce CO2 emissions by 20% or more, with a low level of unburned gas and CO emission and without knocking. However, the level of NOx emissions is high due to the same reason.

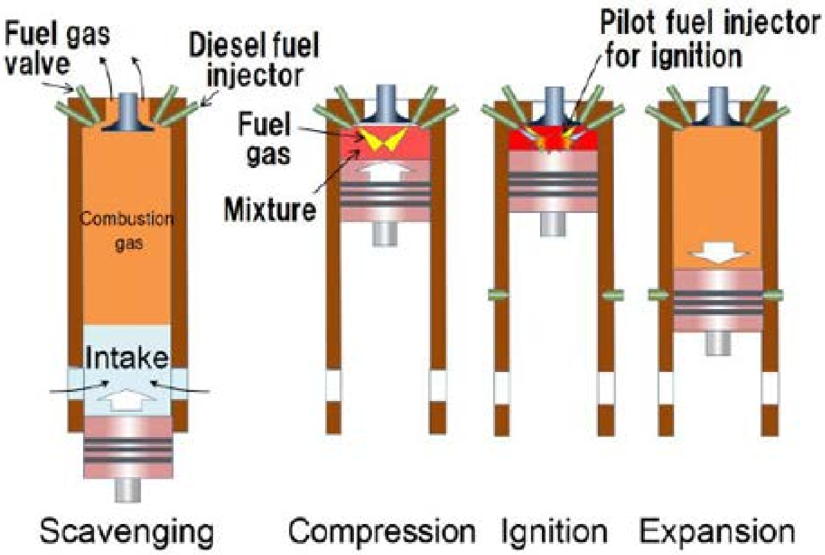


Figure 2.3. Combustion Cycle for Two-Stroke Dual-Fuel Engine

Figure 2.4. shows the cylinder cover design for a two-stroke dual-fuel engine. In order to pressurize gas fuel injected into the compressed air up to 30 MPa, natural gas that has been pressurized in liquid form is vaporized and then injected into the cylinder. And because there is no risk of knocking, the switch between diesel fuel and gas fuel can be performed comparatively easily and quickly.

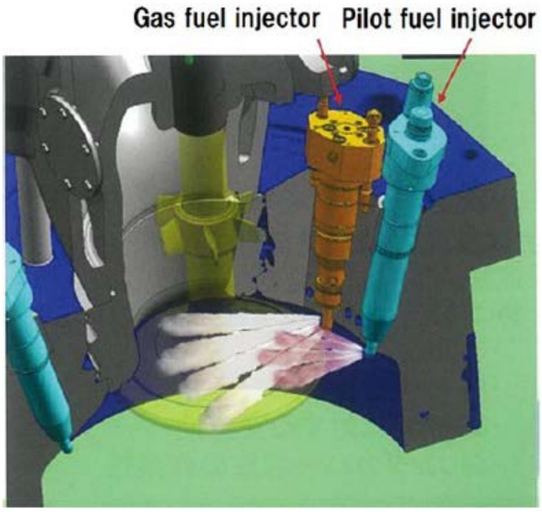


Figure 2.4. Cylinder Cover Design for Two-Stroke Dual-Fuel Engine

One proposal for improving the level of NOx emissions for two-stroke dual-fuel engines is to use a low-pressure gas fuel injection engine that can operate with a lean pre-mixed fuel mixture despite being a two-stroke engine. This involves using a fuel injection timing control method that enables the time delay of pre-mixing of gas fuel and air during the intake stroke described above that prevents the fuel-air mixture from coming into direct contact with the exhaust gas see **Figure 2.5.** Such an engine is currently under development with the aim of achieving low-NOx emissions similar to a four-stroke engine.

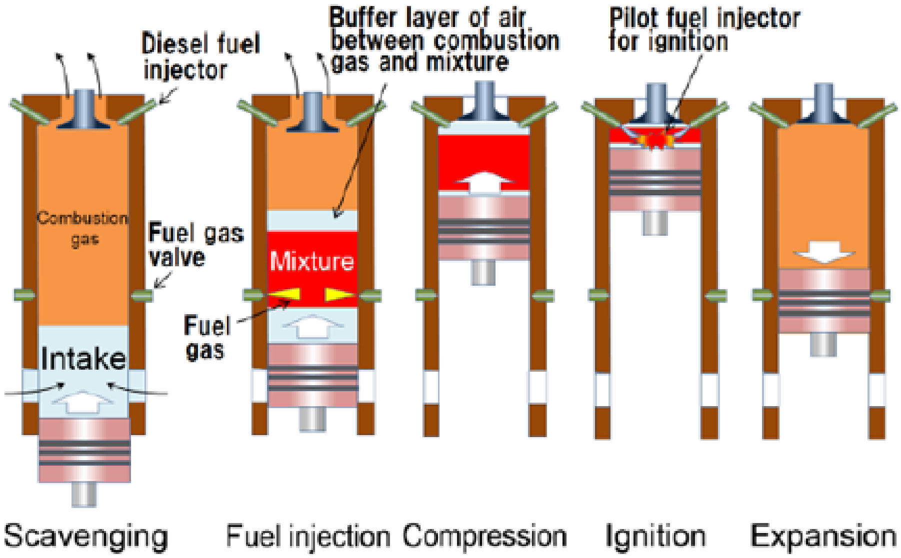


Figure 2.5. Combustion Cycle of Two-Stroke Dual-Fuel Engine

2.1.3. Control of Dual Fuel Engine

Dual-fuel engines use an electronic controller for fuel flow control to enable switching between diesel mode and gas mode as required. In diesel mode, an actuator mechanically operates the fuel pump's flow control lever, whereas in gas mode, the timing for opening the gas valve is controlled electrically with the energizing duration of time for an electromagnetic solenoid.

While it is possible to switch from gas mode to diesel mode by instantaneously shutting off the gas fuel and starting diesel fuel injection, switching from diesel mode to gas mode involves the engine speed control working through a procedure for transitioning from diesel mode to gas mode as the gas valve is slowly opened. This is because of restrictions on the range of the fuel-air ratios that can be used when operating in gas mode, which are dependent on operation condition such as the load or fuel characteristics, and the switchover is controlled automatically while monitoring for signs of knocking or misfiring.

The three main error modes in gas mode are knocking, misfiring, and gas leaks. Knocking can be detected by using a vibration sensor to find changes in combustion vibration or detected directly from the internal cylinder pressure waveform. If knocking is detected, the engine automatically adjusts its operation to prevent it. Although the usual practice is to reduce the maximum pressure in the cylinder by using the micro-pilot injector to delay the ignition timing, if this fails to halt the knocking, the operation automatically switches to diesel mode.

Misfiring can be detected from factors such as the exhaust gas temperature, internal cylinder pressure waveform, or engine speed fluctuation. If misfiring continues for a number of cycles without recovery, the operation automatically switches to diesel mode. Gas leaks are detected with a gas sensor. If a gas leak is detected, the gas valve is immediately shut off and the operation automatically switches to diesel.

2.1.4. Advantages of Dual Fuel Engine

The advantages of using dual fuel engine is when there is a failure of gas fuel air mixing, the engine can work properly by switching dual fuel operational mode into a conventional diesel engine operating mode that used only diesel fuel (Sahoo et al, 2009).

Another advantages of using dual fuel engine are :

- It requires a minimum of modification to the engine, since primary fuel injector is placed at the take air manifold. Also, low control of the fuel can be managed by a simplified device and fuel supply system.
- The primary fuel system is separate from the diesel system. This flexibility enables diesel engines, equipped with the fumigation system, to be operated with diesel fuel only. The engine can switch from dual fuel to diesel fuel operation and vice-versa by disconnection and connection of the alcohol source to the injector.
- Full original power capacity
- Diesel cam timing keeps exhaust cooler and provides better scavenging, contributes to higher power density and longer valve life.
- Higher compression ratio, better efficiency, nearly all dual fuel engines have better efficiency than spark gas.
- Diesel ignition, this is a huge one. You will have very long service intervals with the ignition system (same as normal diesel injector service). Lean burn combustion capacity, far beyond any spark ignition system, contributes to reduced misfire, better efficiency, higher power density, reduced NOX emissions. Diesel pilot fuel provides lubrication to valves and rings, when combined with clean gas, maintenance service intervals are longer than strait diesel not shorter like spark gas.
- Exhaust emissions, specifically Nitrogen oxides, CO_2 and particulates are significantly reduced.
- Fail safe operation, if a problem exists with the gas system full diesel backup is instantly provided.
- No changes should be made to your standard engine warranty.

Should be noted that certain aspects of a lean burn spark gas and dual fuel are similar.

2.1.5. Disadvantages of Dual Fuel Engine

The disadvantages of using dual fuel engine is the combustion duration is longer than using conventional diesel engine. Therefore, the performance is lower than diesel engine (Saha et al., 2008).

The disadvantages of using dual fuel engine are :

- Dual fuel engines require diesel for ignition. Run out of diesel and you are not running.

- Higher emissions of CO compared to strait diesel. Similar emissions of CO as spark gas (without catalytic converter).

- Two fuel systems to maintain. Keep in mind however that diesel fuel ignition is less maintenance than spark systems and that the gases control system is mostly maintenance free.

- Less oil contaminates leads to longer lasting engine, possibly by a factor of two.

Various ways to improve the effectiveness of combustion process on dual fuel engines and have been studied previously, that is by regulating some operating parameters of the machine, increasing the temperature and pressure of diesel and gas fuel injection and adjusting the injection time fuel (Semin, et al., 2009) (Murthy, et al., 2012). Another way to improve the effectiveness of combustion process by adding a fin in intake valve (Bakar, et al., 2007). However, this method is currently applied in diesel engine.

2.2. Intake Valve Dual Fuel Engine

Intake Valve is a valve that located in the cylinder head of an internal combustion engine. This valve opens at the certain time in every cycle of the combustion process to allow the fresh air to be drawn into the cylinder.

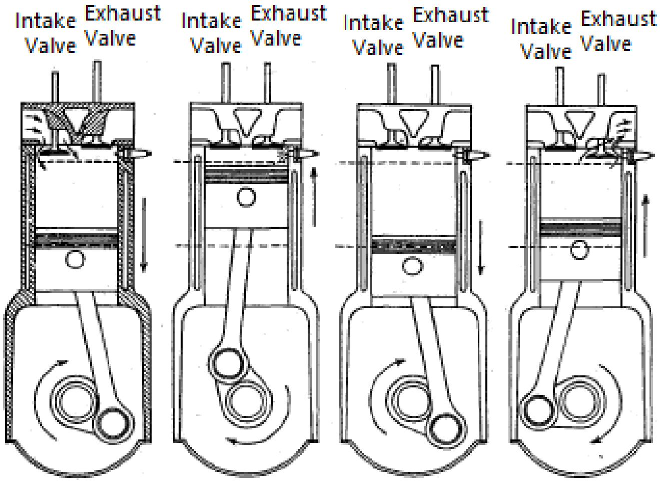


Figure 2.6. Intake Valve Location in Engine

Among the commonly used sleeve, sliding, rotary, and poppet type valves, the intake valve is most common because this offers reasonable weight, good strength and good heat transfer characteristics. The most popular shape of the intake valve uses a small cup at one end of the stem. The valve stem is placed in a guide hole made centrally in a circular passage in the cylinder head. The valve disc head opens and closes the ported passage leading to the cylinder during in and out movement of the stem. the structural componen shown below:

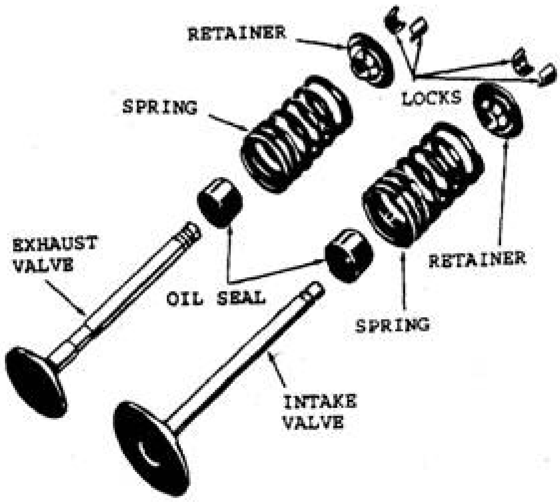


Figure 2.7. Valve Component

2.2.1. Function Intake Valve

The function of intake valve is take control the amount of air as working fluid entering the combustion chamber of an engine (Heywood, 1998). The amount of air is an important factor in the combustion process. The intake valve mechanism takes responsibility for the proprietary and efficient delivery of fresh air to the engine. For that, proper valve timing should be designed and computed (Zbierski, 2007).

2.2.2. Mechanism of Intake Valve

The valve mechanism opens or closes the intake valve at the proper timing in order to draw the air-fuel mixture into the cylinder. The rotation of the crankshaft is transferred to the camshaft via the timing chain (timing belt), rotating the cam. The number of teeth on the camshaft sprocket (pulley) is double that of the crankshaft so that the camshaft rotates once for every two rotations of the crankshaft. As the camshaft rotates, the cam forces the valve to open or close.

The valve seat is press-fitted into the cylinder head. When the valve closes, the valve face and valve seat fit closely together to make the combustion chamber airtight. The valve seat also transfers heat from the valve to the cylinder head, serving to cool the valve as well. Since the valve seat is exposed to high temperature combustion gas, and repeated contact with the valve, it is constructed of a metal that excels in resistance to heat and wear. When the valve seat wears, it can be ground by the carbide cutter or replaced. In recent years, lasers have been used to weld a wear-resistant alloy valve seat layer directly to the cylinder head making the valve seat and the cylinder head one unit on some engines. With this type of laser clad valve seat, replacement is impossible.

Because of a variation range of the engine rotational speed under it work, the valve timing has to be a compromise between economical work with low fuel consumption by the engine and good performance for the user. Because of work with the constant rotational speed of the industrial engine, valve timing is fixed to the engine maximum volumetric efficiency. To increase the efficiency and decrease in exhaust gases emission, the over-expanded cycle can be applied to the engine (Miller, 1947).

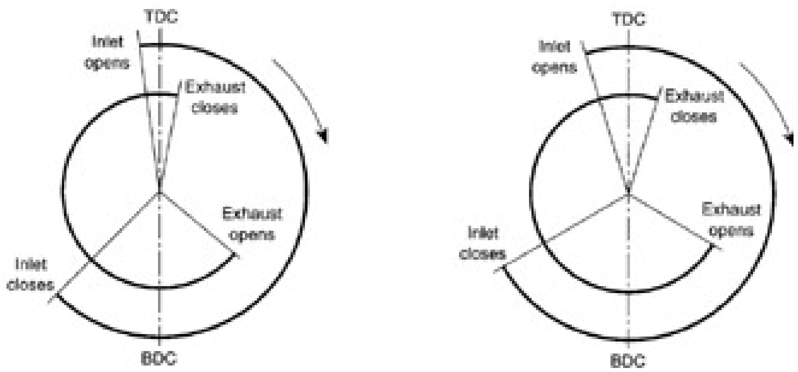


Figure 2.8. Valve Timing in a Cycle

There are several methods to modify the valve train and change the valve timing. The most popular from it, the modification of intake valve cam slope in that way that it provides early intake closure before bottom dead center (BBDC) or late intake valve closure after bottom dead center (ABDC). However, with very early or very late intake valve closure relative to BDC, decrease in indicated mean effective pressure and indicated efficiency might occur. One of the methods to reduce this drawback is to apply supercharging the engine what in effect contributes to increase in both IMEP and indicated efficiency.

2.2.3. Intake Valve Material

Materials that are commonly used for performance valve applications include carbon steel alloys, stainless steels, high-strength nickel-chromium-iron alloys and titanium. The alloys that are most commonly used for performance engines include various high chromium stainless alloys for intake valves

There are essentially two basic types of steel used to make valves. One is "martensitic" steel and the other is "austenitic" steel. The difference is in the microstructure of the steel and how the various ingredients in the alloy interact when the molten steel is cast and cooled. This affects not only the hardness and strength of the steel, but also its corrosion resistance and magnetic properties. As a rule, martensitic steels are magnetic while austenitic steels are non-magnetic.

The bottom line here is that intake valves and exhaust valves both require different types of alloys. The same alloy can be used for both intake and exhaust valves (say 21-2N or 21-4N, for example), but the best results are usually obtained when different alloys are selected for the intake and exhaust valves. Because an exhaust alloy that has good high temperature strength and corrosion resistance really isn't needed on the intake side, and it may not have the hardness and wear resistance of an intake alloy at lower temperatures. Even so, some companies sell the same alloy for both intake and exhaust valves while others offer different alloys for intake and exhaust valves.

The intake flow is throttled to below atmospheric pressure by reducing the flow area when the power required (at any engine speed) is below the maximum which is obtained when the throttle is wide open. The intake manifold is usually heated to promote faster evaporation of the liquid fuel and obtain more uniform fuel distribution between cylinders. To maintain high mixture flows at high engine speeds (and hence high power outputs) the inlet valve, which opens before TC, closes substantially after BC. During intake, the inducted fuel and air mix in the cylinder with the residual burned gases remaining from the previous cycle. After the intake valve closes, the cylinder contents are compressed to above atmospheric pressure and temperature as the cylinder volume is reduced. Some heat transfer to the piston, cylinder head, and cylinder walls occurs but the effect on unburned gas properties is modest.

Volumetric efficiency is affected by the following fuel, engine design, and engine operating variables:

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

Flow effects on volumetric efficiency depend on the velocity of the fresh mixture in the intake manifold, port, and valve. Local velocities for quasi-steady flow are equal to the volume flow rate divided by the local cross-sectional area. Since the intake system and valve dimensions scale approximately with the cylinder bore, mixture velocities in the intake system will scale with piston speed. Hence, volumetric efficiencies as a function of speed, for different engines, should be compared at the same mean piston speed. The valve, or valve and port together, is usually the most important flow restriction in the intake and the exhaust system of four-stroke cycle engines

2.3. Fin Addition on The Intake Valve

Nowadays, modifying an intake valve is a common topic to discuss. There are many methods to modify an intake valve, for example: Fin addition, mask addition, mask with hole addition. These modifications done with purpose to increase the engine efficiency.

2.3.1. Existing research of Fin Addition in Diesel Engine

Yerrennagoudaru in 2014 doing an experiment about fin addition in the intake valve on a single cylinder diesel engine with variation design, with detail below:

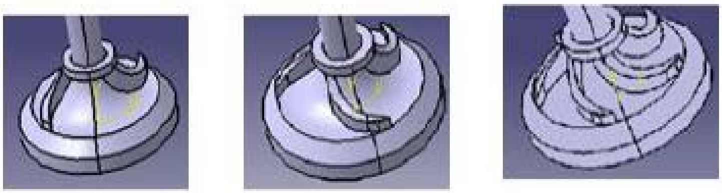


Figure 2.9. Intake Valve Modeling

The fins are equi- spaced on a circular orbit with varying numbers say two, three and five. The dimensions are:

- inner ring dia-7.8mm,
- Thickness of ring: 2mm,
- Width of ring: 2mm,
- Length of vane: 12mm,
- Width and thickness of vane: 2mm,
- Angle of vane: 450 bent down to adjust on valve head.

Here valve lift is considered as major criterion for simulation. Total valve lift = 12mm

This 12mm is divided into 3 parts and called as:

- Low Lift (valve at 4mm downward movement)
- Medium lift (valve at 8mm downward movement)
- High lift (valve at12mm downward movement)

Hence CFD simulations were carried out for analyzing swirl, turbulence, velocity of inlet air and also pressure distribution inside the cylinder during suction stroke and are analyzed at different inlet valve lift positions in comparison with base model.

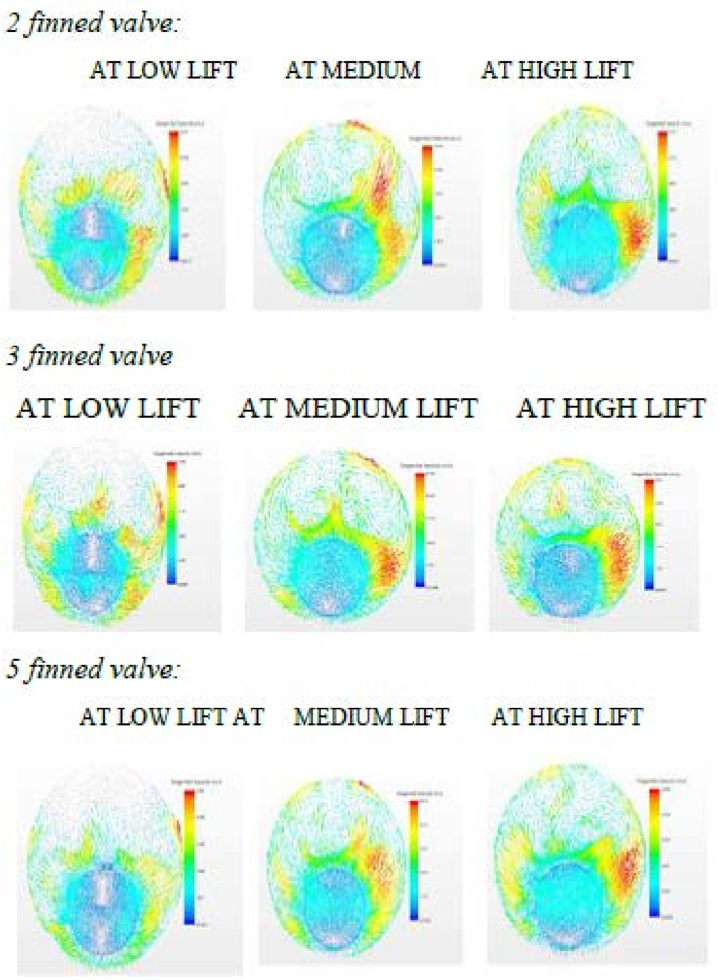


Figure 2.10. Intake Valve Flow CFD Simulation

Type of Valve	Parameter	Lift		
		Low(4.3mm)	Medium	High
Base	Swirl Ratio	2.2500	0.9371	0.4840
2 Fins		2.7022	1.3334	0.6206
3 Fins		2.5523	0.9832	0.6496
5 Fins		2.5216	1.1215	0.7035

the result, intake valve with two fin produce turbulent flow ratio up to 0.6206, intake valve with three fin produce turbulent flow ratio up to 0.6496, intake valve with five fin produce turbulent flow ratio up to 0.7035.

Yerrennagoudaru in 2014 doing an experiment about mask addition in the intake valve on a single cylinder diesel engine with variation design, with detail below:

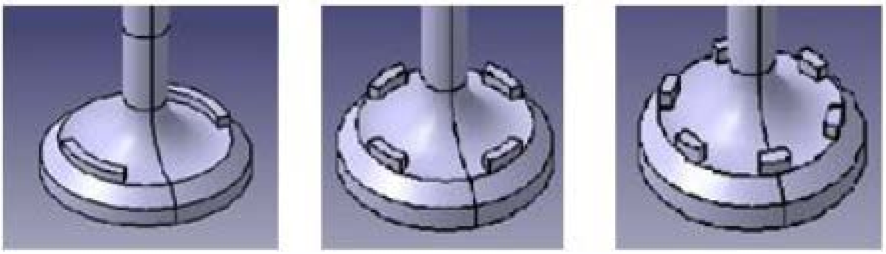


Figure 2.11. Intake Valve Modeling

The mask are blocking spaced on a outer orbit with varying numbers say two, four and six. The dimensions are:

- Thickness of mask: 2mm,
- Width of mask: 2mm,
- Angle of mask: 120 degree .

Here valve lift is considered as major criterion for simulation. Total valve lift = 12mm

This 12mm is divided into 3 parts and called as:

- Low Lift (valve at 4mm downward movement)
- Medium lift (valve at 8mm downward movement)
- High lift (valve at12mm downward movement)

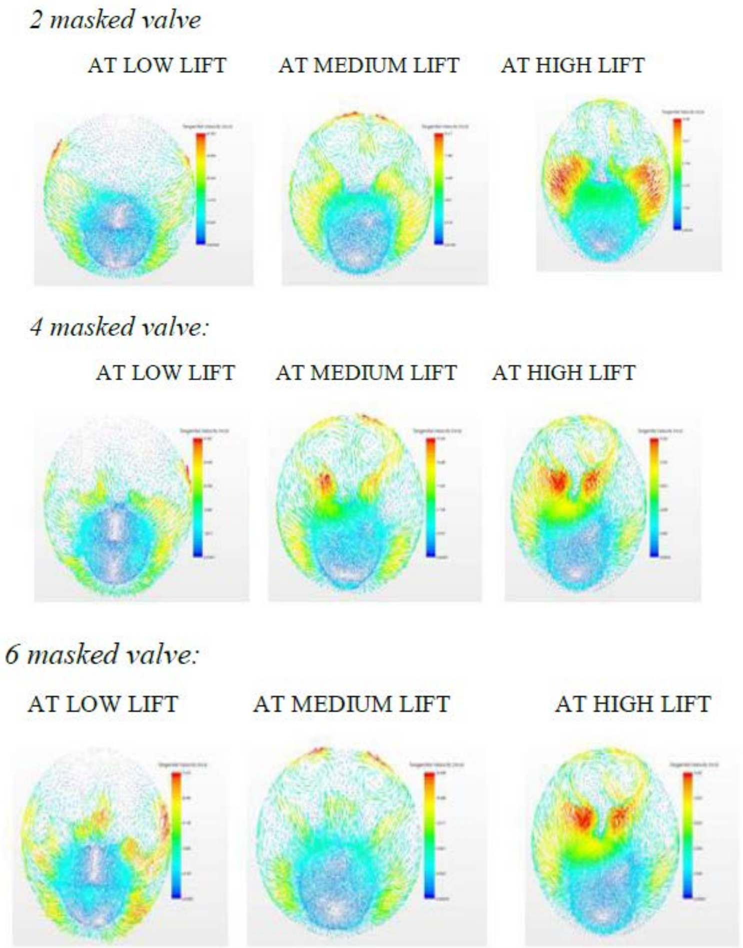


Figure 2.12. Intake Valve Flow CFD Simulation

Type of Valve	Parameter	Lift		
		Low(4.3mm)	Medium	High
Base	Swirl Ratio	2.2500	0.9371	0.4840
2 Mask		2.1600	1.0600	1.0300
4 Mask		2.1300	1.0954	1.0445
6 Mask		2.2945	1.0027	1.1858

the result, intake valve with two mask produce turbulent flow ratio up to 1.0300, intake valve with four mask produce turbulent flow ratio up to 1.0445, intake valve with five mask produce turbulent flow ratio up to 1.1858.

Yerrennagoudaru, 2015, also done an experiment discuss about fin addition on the intake valve in diesel engine. in this experiment, Yerrennagoudaru modeling 3 type of intake valve, with detail below:

Model-1 valve (Intake valve with 5 straight grooves on its poppet head):

Figure 2.13. Intake Valve with 5 Straight Grooves Modeling

Grooving is the process of removing a small piece of metal on the valve head without disturbing the valve seating; the small cavities are called grooves. Here for analysis we have used 2-grooved valve, 3- grooved valve and 5- grooved valve, which are having the following dimensions:

- Width of groove: 3mm
- Depth of groove: 3mm
- Outer dia of grooves: 22mm
- Inner dia of grooves: 8mm

Model-2 valve (Intake valve with 3 masks and 3 straight grooves on its back)

Figure 2.14. Intake Valve with 3 Masks and 3 Straight Grooves Modeling

In this type a valve consisting a grooves three in numbers with small pieces build on head as shown in fig-2. This type of design utilizes a combination of small built-up pieces called MASKS and a CAVITIES called grooves. For analysis 2groove-2mask, 3Groove-3masks are simulated through CFD, which are having the following dimensions:

- Width of groove: 3mm
- Depth of groove: 3mm
- Outer dia of grooves: 22mm
- Inner dia of grooves: 8mm
- Angle of each mask: 45degrees.
- Thickness of mask: 4mm
- Width of mask: 4mm

Model-3 valve (Intake valve with 5 straight grooves with 3 fins on its poppet head)

Figure 2.15. Intake Valve with 5 Straight Grooves with 3 Fins Modeling

In this type a small ring with blades are attached to valve as shown in figure-3. It uses a combination of grooves three in numbers and a freely rotatable ring with 3 blades. For CFD analysis the 3Groove-5bladed ring and 5groove-3bladed ring are simulated. which are having the following dimensions:

- Width of groove: 3mm
- Depth of groove: 3mm
- Outer dia of grooves: 22mm
- Inner dia of grooves: 8mm
- Dia of ring: 7.9mm
- Length of blade: 6.5mm
- Width: 4mm

Here valve lift is considered as major criterion for simulation. Total valve lift = 12mm This 12mm is divided into 3 parts and called as Low Lift (valve at 4mm downward movement), Medium lift (valve at 8mm downward movement), High lift (valve at12mm downward movement), Hence CFD simulations were carried out for analyzing swirl, turbulence, velocity of intake air and also pressure distribution inside the cylinder during suction stroke and are analyzed at different intake valve lift positions in comparison with base model.

Intake valve with 5 straight grooves

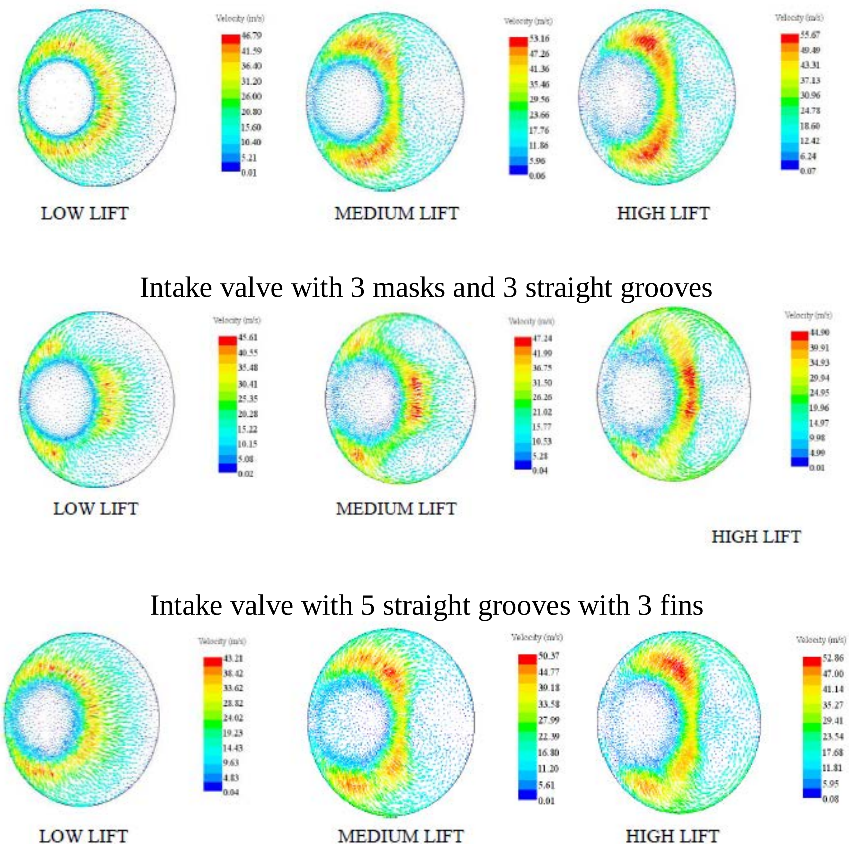


Figure 2.16. Intake Valve Modification Flow CFD Simulation

Intake valve with 5 straight grooves

		Valve movement (full 24mm)					
		Valve lift in (mm) Forward			Backward		
Five Straight		4.00	8.00	12.00	16.00	20.00	24.00
	Swirl Ratio	1.72	1.62	1.45	1.49	1.36	0.83
	Tumble Ratio	14.64	15.96	16.57	17.50	30.21	1.68
	TKE(j/kg)	8.68	53.25	90.09	107.13	54.92	11.87
	Mass Flow Rate(kg/s)	0.0279					

Intake valve with 3 masks and 3 straight grooves

		Valve movement (full 24mm)					
		Valve lift in (mm) Forward			Backward		
3Grooves3Mask		4.0000	8.0000	12.0000	16	20	24
	Swirl Ratio	1.43	1.38	1.29	1.35	1.11	0.87
	Tumble ratio	15.28	16.09	16.29	17.59	14.23	0.49
	TKE(j/kg)	8.26	23.55	71.18	82.40	56.32	1.48
	Mass Flow Rate(kg/s)	0.02351					

Intake valve with 5 straight grooves with 3 fins

		Valve movement (full 24mm)					
		Valve lift in (mm) Forward			Backward		
5Grooves3Fin		4.0000	8.0000	12.0000	16	20	24
	Swirl Ratio	1.86	1.62	1.60	1.52	1.33	0.83
	Thumble Ratio	14.63	14.89	14.68	16.71	28.45	1.24
	TKE(j/kg)	9.25	50.39	88.58	97.18	54.07	7.07
	Mass Flow Rate(kg/s)	0.02729					

2.3.2. Effect of Fin Addition on the Intake Valve in Diesel Engine

Turbulent flow inside the combustion chamber have an effect to combustion process that also take an effect to engine performance (Semin et al, 2007). Turbulent flow intensity can mend the combustion duration. Moreover, turbulent flow also contributed in gas-air mixing process. There are two types of turbulent flow in engine, swirl flow and tumble flow.

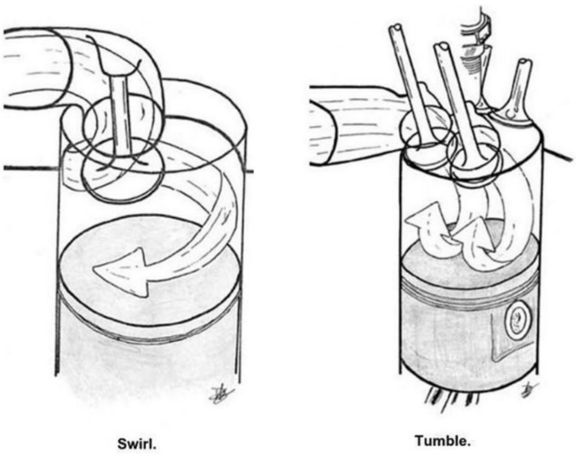


Figure 2.17. Swirl and Tumble Flow in Combustion Chamber

Fluid flow in the combustion chamber can be adjusted by optimizing engine components and developing machine design. Intake manifold, intake valve, and piston head are components that take effect of the fluid flow formation in the combustion chamber. Adding a fin componen in Intake valve can also produce a swirl flow in the combustion chamber (Raghavan et al, 2014).

Swirl flow that introduced into the combustion chamber have an effect to increasing the torque output of the engine. This is caused by gas-air mixing and fuel flow rate process occur better. The combustion surface area also take an effect to reduce the combustion duration.

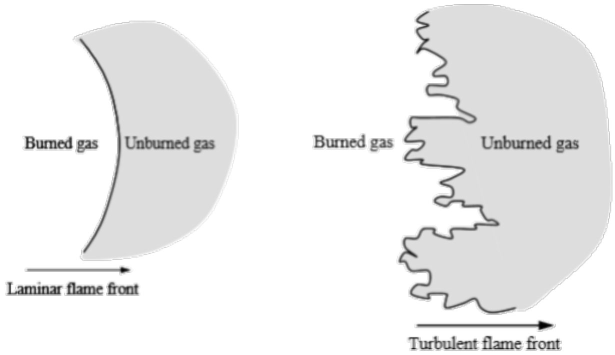


Figure 2.18. Laminar and Turbulent Flame Front

The increased combustion surface area raises heat transfer to nearby unburned gas. Since the temperature of the unburned gas increases more quickly due to the increased heat transfer, it can reach ignition temperature and initiate ombustion much faster than without a wrinkled flame front.

Gas motion within the engine cylinder is one of the major factors that controls the combustion process in spark-ignition engines and the fuel-air mixing and combustion processes in diesel engines. It also has a significant impact on heat transfer. Both the bulk gas motion and the turbulence characteristics of the flow are important. The initial in-cylinder flow pattern is set up by the intake process. It may then be substantially modified during compression. if the Intake valve is closed too early, volumetric efficiency will decrease gradually with increasing.

.

2.4. Design Intake Valve

The Intake port is generally circular, or nearly so, and the cross-sectional area is no larger than is required to achieve the desired power output. For the exhaust port, the importance of good valve seat and guide cooling, with the shortest length of exposed valve stem, leads to a different design. Although a circular cross section is still desirable, a rectangular or oval shape is often essential around the guide boss area.

Larger valve sizes (or four valves compared with two) allow higher maximum air flows for a given cylinder displacement. The instantaneous valve flow area depends on valve lift and the geometric details of the valve head, seat, and stem. There are three separate stages to the flow area development as valve lift increases. The maximum valve lift is normally about 12 percent of the cylinder bore (Heywood, 1988).

The effect of valve geometry and timing on air flow can be illustrated conceptually by dividing the rate of change of cylinder volume by the instantaneous minimum valve flow area to obtain a pseudojlow velocity for each valve

In addition to valve lift, the performance of the Intake valve assembly is influenced by the following factors: valve seat width, valve seat angle, rounding of the scat corners, port design, cylinder head shape. In many engine designs the port and valve assembly are used to generate a rotational motion (swirl) inside the engine cylinder during the induction process, or the cylinder head can be shaped to restrict the flow through one side of the valve open area to generate swirl.

2.5 Computational Fluid Dynamics

Computational fluid dynamics (CFD) is a subsidiary method of fluid mechanics used numerical analysis and data structures to solve and analyze any problems that involve fluid flows. Computers are used to perform the calculations required to simulate the interaction of liquids and gases with surfaces defined by boundary conditions. The ultimate goal of the field of computational fluid dynamics is to understand the physical event that occur in the flow of fluids around and within designated objects. These events are related to the action and interaction of phenomena such as dissipation, diffusion, convection, shock waves, slip surface, boundary layers, and turbulence.

In the simulation process, there are three steps that must be done, there are pre-processing, solving and post-processing.

1. Pre-Processor

Pre-processor is the initial stage in Computational Fluid Dynamic (CFD) which is the stage of data input that includes the determination of domain and boundary condition. At this stage, meshing is also done, where the analyzed object is divided into the number of specific grids.

2. Processor

The next step is the processor stage. At this stage, is done the process of calculating data that has been entered using iterative related equations until the results obtained can reach the smallest error value.

3. Post Processor

The last step is the post processor stage, the results of the calculations at the processor stage will be displayed in pictures, graphs and animations.

Regardless of what the numerical errors are called, if their effect are not thoroughly understood and controlled, they can lead to serious difficulties, producing answers that represent little, if any, physical reality.

CHAPTER III
METHODOLOGY

Methodology represents of the stages from start to finish the thesis. The methodology of this thesis cover all the activity that supports the completion of this thesis. The stages of this methodology are as follows as flowchart in **Figure 3.1.**

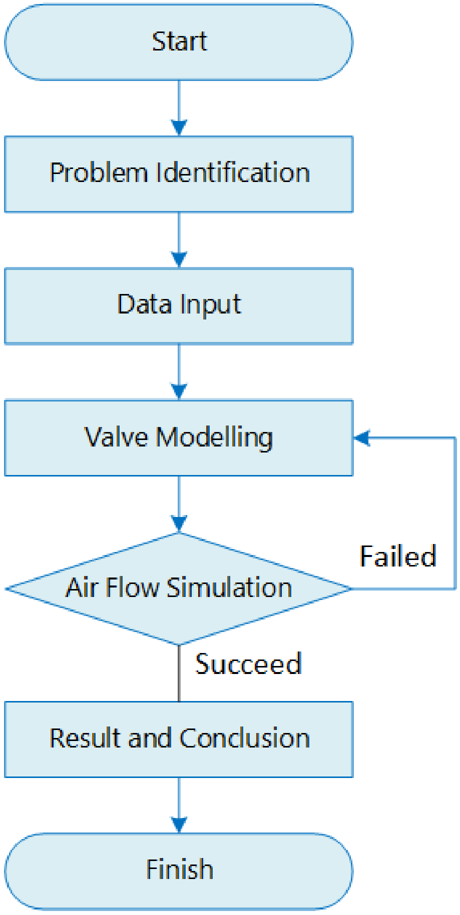


Figure 3.1.Methodology Flowchart

Explanation of the stages of this methodology flowchart is as follows:

3.1 Problem Identification

This stage, problem identification become the first stage of this thesis. Problem identification is identified the problems in dual fuel engine compared with diesel engine conventional. The object of research is the application of Intake valve fin in intake manifold and turbulent flow effect inside the cylinder that caused. Turbulent flow intensity is one of the factor to increase engine efficiency.

3.2 Data Input

This stage data input needed for simulation in this thesis is the dimensional of diesel engine such as detail valve size, cylinder head type, intake runner design, combustion chamber volume and internal boundary condition. All the dimensional obtained from engine guide and manual measurement. The boundary condition obtained from engine simulation software (GT Suites Software).

Model	Unit	TF50	TF50-H	TF60	TF60-H	TF70	TF70-H	TF80	TF80-H	TF90	TF90-H	TF110	TF110-H	TF120	TF120-H	TF140	TF140-H	TF160	TF160-H
Type		Horizontal, water-cooled 4-cycle diesel																	
Combustion system		Direct injection																	
No. of cylinders		1																	
Borex stroke	mm	74 x 72		75 x 80		78 x 80		80 x 87		85 x 87		88 x 96		92 x 96		96 x 105		102 x 105	
Displacement	ℓ	0.309		0.353		0.382		0.437		0.493		0.583		0.638		0.760		0.857	
Rated continuous output	HP/rpm	4.5/2400		5.0/2400		6.0/2400		7.5/2400		8.5/2400		10.0/2400		10.5/2400		12.5/2400		14.0/2400	
	kW/rpm	3.4/2400		3.7/2400		4.5/2400		5.6/2400		6.3/2400		7.5/2400		7.8/2400		9.3/2400		10.4/2400	
At 1-hr. rated output	HP/rpm	5.0/2400		6.0/2400		7.0/2400		8.5/2400		9.5/2400		11.0/2400		12.0/2400		14.0/2400		16.0/2400	
	kW/rpm	3.7/2400		4.5/2400		5.2/2400		6.3/2400		7.1/2400		8.2/2400		9.0/2400		10.4/2400		11.9/2400	
Specific fuel consumption	g/HP-hr	175						174		170				169		155			
Compression ratio		18.4		17.9		18.1		18.0				17.9		17.7		17.8			
Position of PTO		Flywheel side																	
Direction of crankshaft rotation		Counterclockwise viewed from flywheel																	
Fuel oil applicable		Gas-oil or Light oil (UK BS 2869 A1 or Equivalent)																	
Fuel injection pump		Bosch type																	
Injection timing	deg.	bTDC 12.5		bTD 17.0				bTDC18.0				bTDC 17.0							
Injection pressure	kg/cm²	200																	
F.O. tank capacity	ℓ (US gal)	5.6 (1.48)		7.1 (1.88)				10.5 (2.77)				11.0 (2.91)				14.3 (3.78)			
Lubrication system		Complete enclosed forced lubricating system with hydraulic pressure regulating valve																	
Lubricating oil applicable		API grade CB or CC																	
Lubricating oil capacity (Oil pan) Effective/Total	ℓ (US gal)	0.4/1.2 (0.11/0.32)		0.6/1.8 (0.16/0.48)				0.8/2.2 (0.21/0.58)				1.0/2.8 (0.26/0.74)				1.5/3.0 (0.40/0.79)			
Cooling system		Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper
Cooling water capacity	ℓ (US gal)	1.20 (0.32)	5.0 (1.32)	1.25 (0.33)	8.00 (2.11)	1.25 (0.33)	8.00 (2.11)	1.65 (0.44)	8.9 (2.35)	1.65 (0.44)	8.9 (2.35)	2.3 (0.61)	12.0 (3.17)	2.3 (0.61)	12.0 (3.17)	3.00 (0.79)	13.00 (3.43)	3.00 (0.79)	13.00 (3.43)
Starting system		Manual or Electric																	
Engine dimensions Overall Length	mm	523		607				675				696				776			
Engine dimensions Overall Width	mm	311		311				329				349				380			
Engine dimensions Overall Height	mm	463		469				496				530				621			
Engine dry weight	kg (lb)	47.5 (104.7)	46 (101.4)	68 (149.9)	65 (143.3)	68 (149.9)	65 (143.3)	86.5 (190.7)	85 (187.4)	86.5 (190.7)	85 (187.4)	101 (220.7)	99 (218.3)	101 (220.7)	99 (218.3)	140 (308.6)	136 (299.6)	140 (308.6)	136 (299.8)

Figure 3.2. Engine Specification

From engine guide and manual measurement, dimensional presented as follows:

- Cylinder bore : 85 mm
- Cylinder length : 87 mm
- Intake valve diameter : 32.5 mm
- Intake valve length : 84.1 mm
- Exhaust valve diameter : 27.5mm
- Exhaust valve length : 84.1 mm
- Compression ratio : 1:16

Another input data is taken from engine simulation software, GT Suites. The data consist internal boundary condition as follows:

Table 2 Boundary Condition

No.	Bounday Condition	value
1	Air Flow Rate	68.26 m/s
2	Pressure Inside Cylinder	68.232 Pa
3	Valve Overlap	1.5 mm

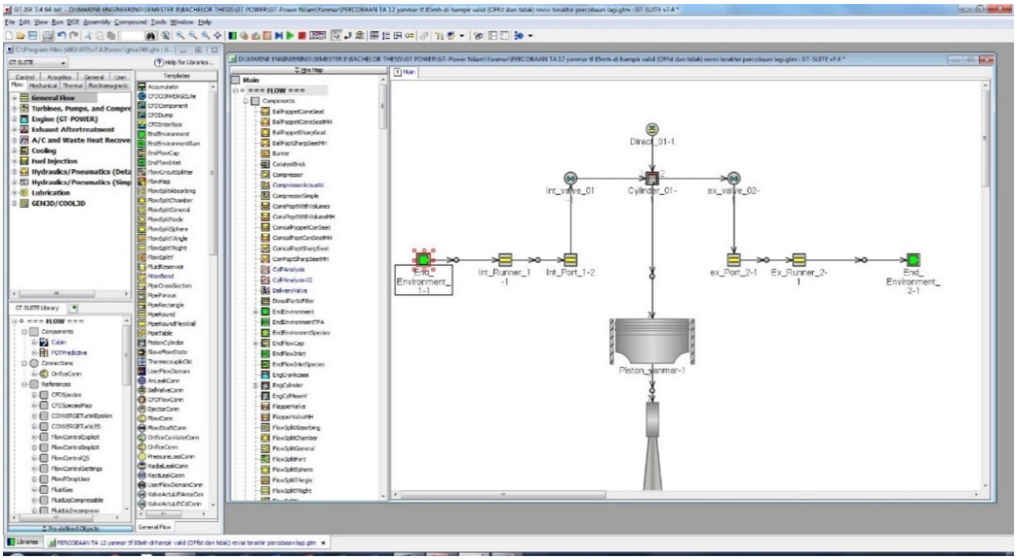


Figure 3.3. Engine Simulation Software

3.3 Valve Modeling

Modelling stage is where the real dimensional data from engine guide drawn using 3D modelling software. In this case, used Autodesk Inventor 2019 Student License. Intake Valve Modeling based on engine specification of diesel engine YANMAR TF85 MH and manual measurement on engine.

Figure 3.4. Conventional intake valve

From diesel engine YANMAR TF85 MH can be measured the dimension of intake valve that shown in figure 4.1.

In case to adding fin in intake valve, fin designed precisely as previous research by Yerrennagoudaru in 2015. with detail dimension drawing below:

- Length of fin : 12 mm,
- Thickness of fin : 2 mm.

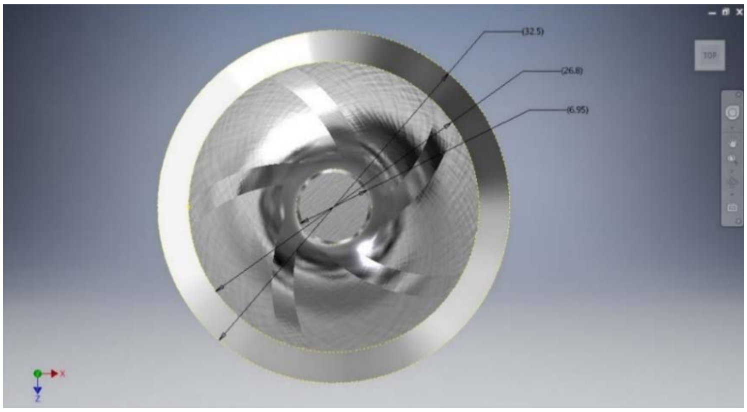


Figure 3.5. Dimensional Design of Intake Valve Fin Top View

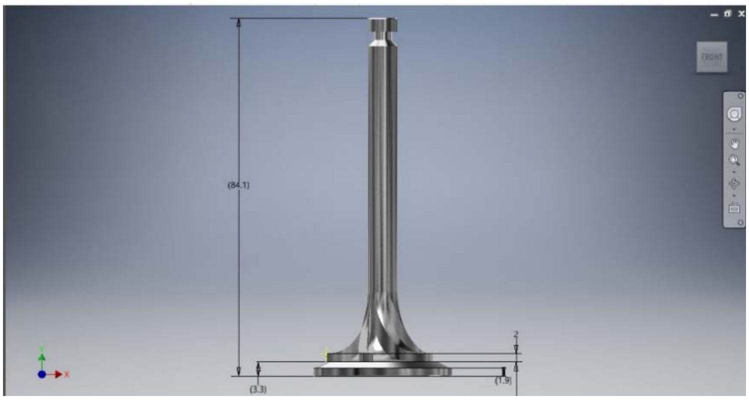


Figure 3.6. Dimensional Design of Intake Valve Fin Side View

Modeling intake valve with two variation, there are three fins and five fins with the same shape and size. model of these valve can be seen below:

Figure 3.7. Intake Valve with Three Fin Addition

Figure 3.8. Intake Valve with Five Fin Addition

3.3.1. Modelling Support Components

Inorder to doing CFD software simulation also needed other supporting model design, here is some model that has been designed to complete the simulation process.

Figure 3.9. Air-Gas Flow Area

Figure 3.10. Model of Exhaust Valve

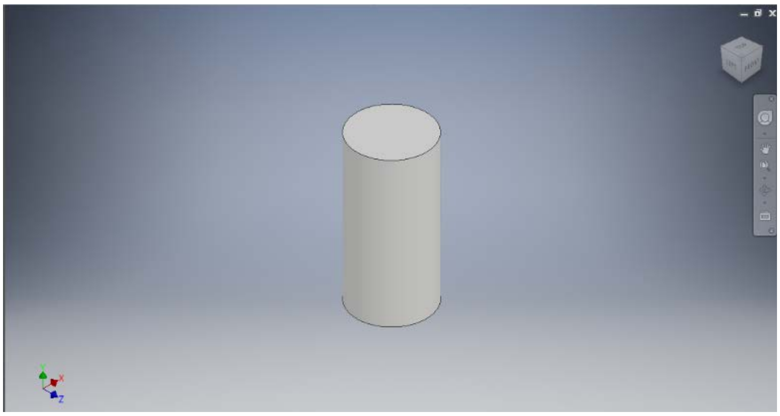


Figure 3.11. Model of Connector

3.3.2. Assembly Modeling

After all part model designed, Assemble these all into a complete arrangement. In this thesis, valve lift condition based on engine simulation software. It taken at RPM 2200 and no load of diesel engine modified by gas addition as follows:

At crank angle 372°, Intake Valve lift 2.083 and Exhaust Valve lift 1.344

At crank angle 402°, Intake Valve lift 5.022 and Exhaust Valve lift 0.295

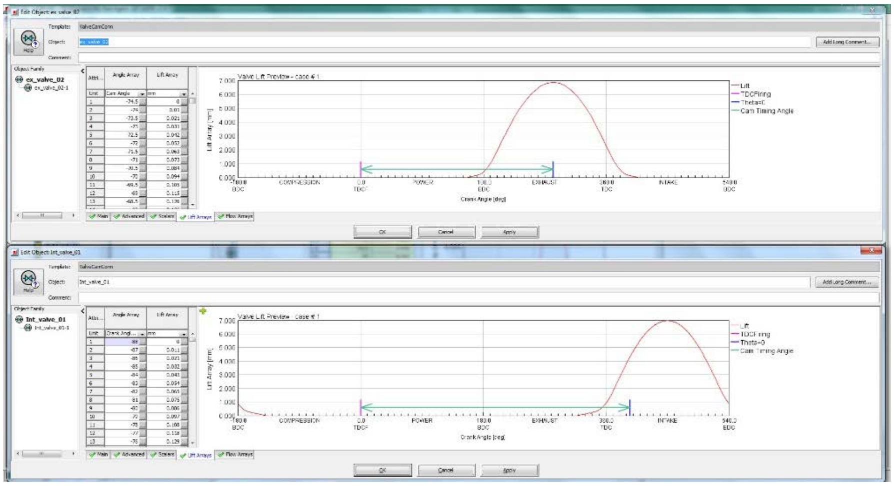


Figure 3.12. Valve lift Condition based on GT Suites

With valve lift condition above, here the picture of assembled parts into a complete arrangement system.

Figure 3.13. Assembly of Each Part Model Isometric View

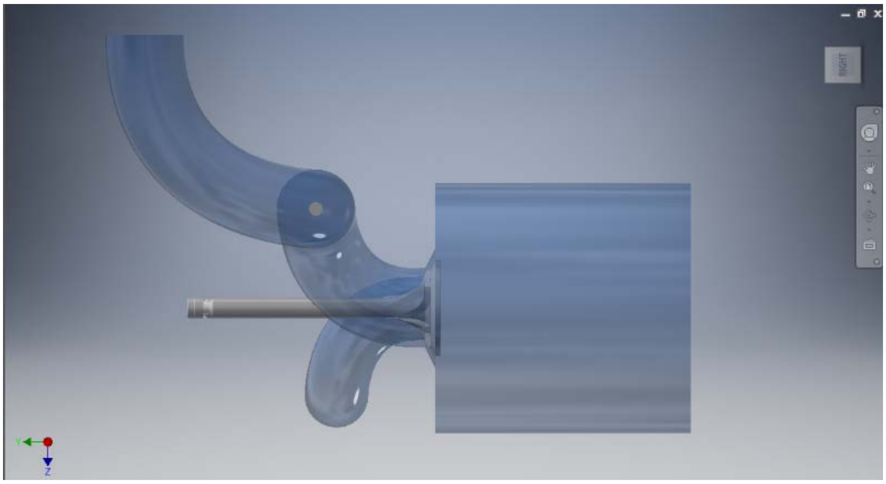


Figure 3.14. Assembly of Each Part Model Side View

3.4. Air Flow Simulation

Air flow simulation using Ansys 16.2 Student License. Same as the modelling stage, in this stage author simuling the air flow inside cylinder with the application of two type of valves; plain valve, and finned valve.

First thing in computational fluid dynamics simulation is choose the analysis system of it software. Here choosed fluent analysis for complete the simulation.

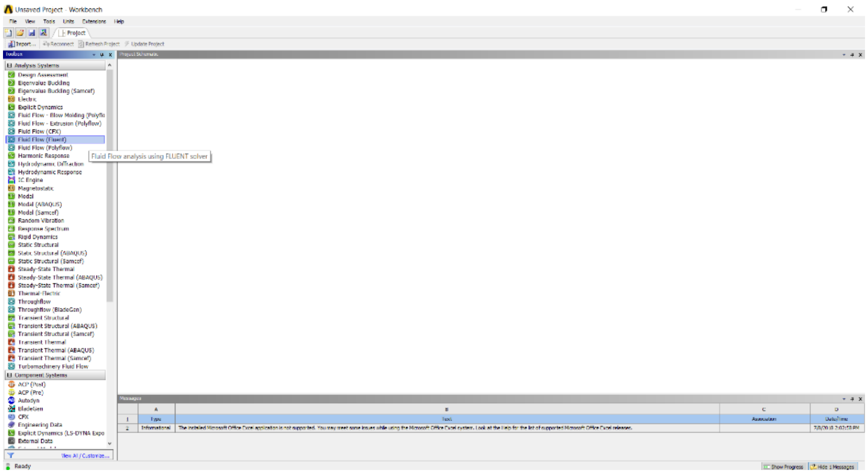


Figure 3.15. Main panel of Ansys Workbench R16.2

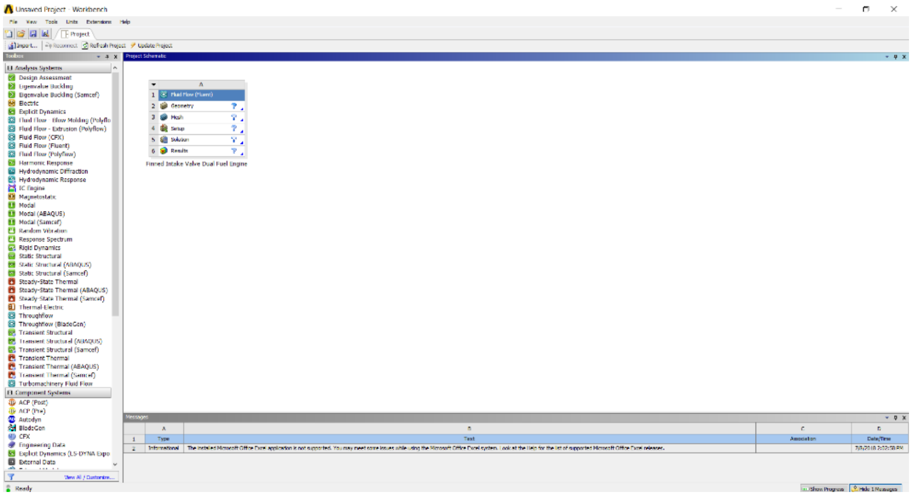


Figure 3.16. Work Window of Fluent Analysis in Ansys Workbench R16.2

3.4.1. Import Geomerty

Import geometry to workbench and named all componen for supporting simulation process in set up stage. Picture of geometry below:

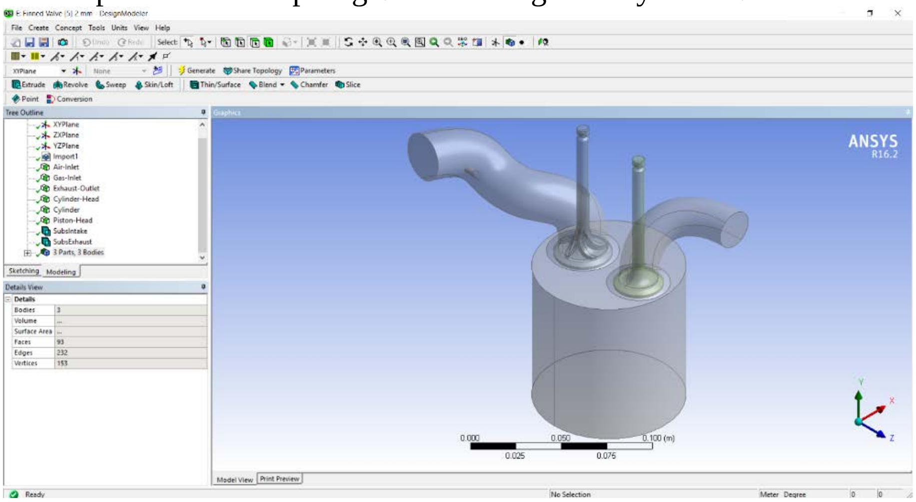


Figure 3.17. Import Geometry into work window in Ansys Workbench R16.2

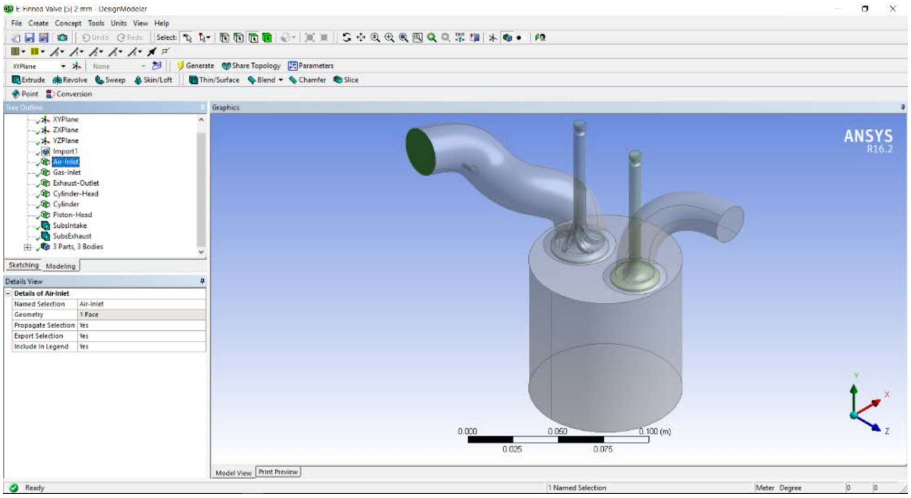


Figure 3.18. Named all Component and Surface of Geometry

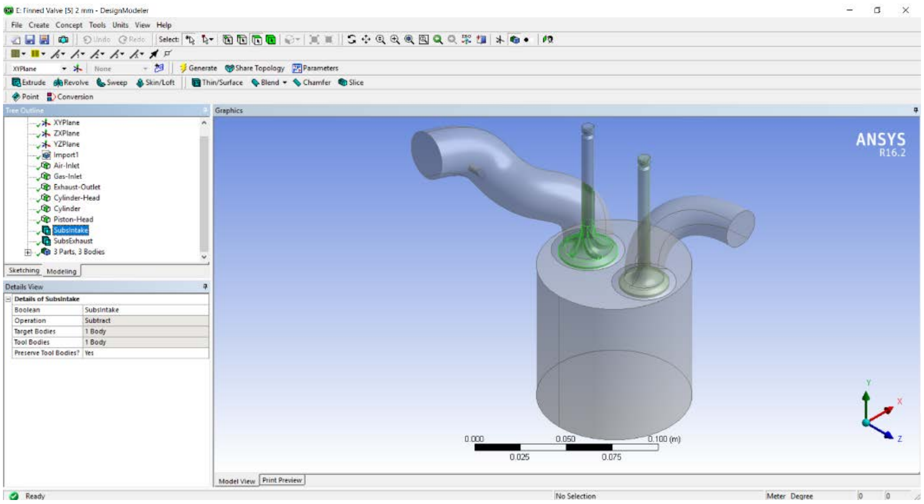


Figure 3.19. Defined the interface body of Geometry

3.4.2. Meshing Process

After import all model to workbench, it needs to build a good enough mesh process. Here picture of meshing process in Ansys R16.2:

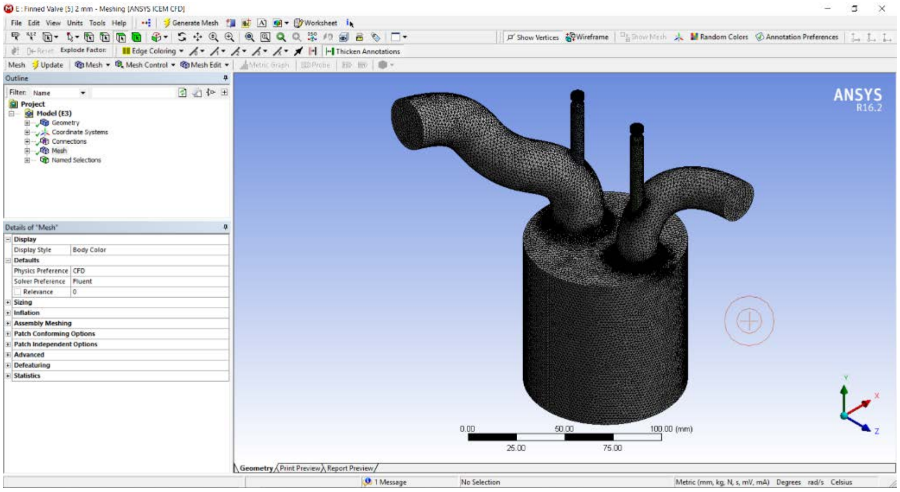


Figure 3.20. Generating Mesh in Isometric View

Meshing more detail to get more accurate result, here used refinement and face sizing in several surface in model during mesh processing.

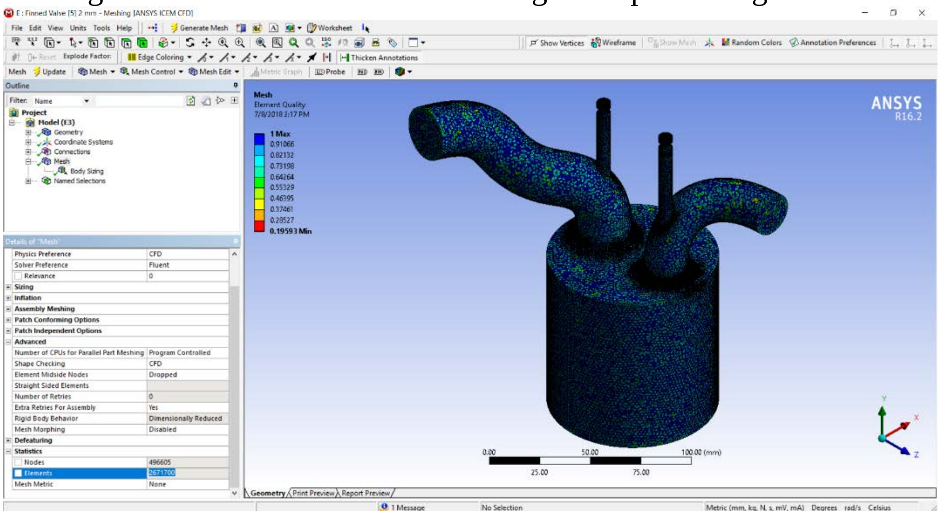


Figure 3.21. Refinement Meshing in Isometric View

3.4.3. Setup and Solution

After meshing process done, set up the fluent analysis and fill in any internal boundary condition-based Engine Simulation Software (GT Suites) as follows:

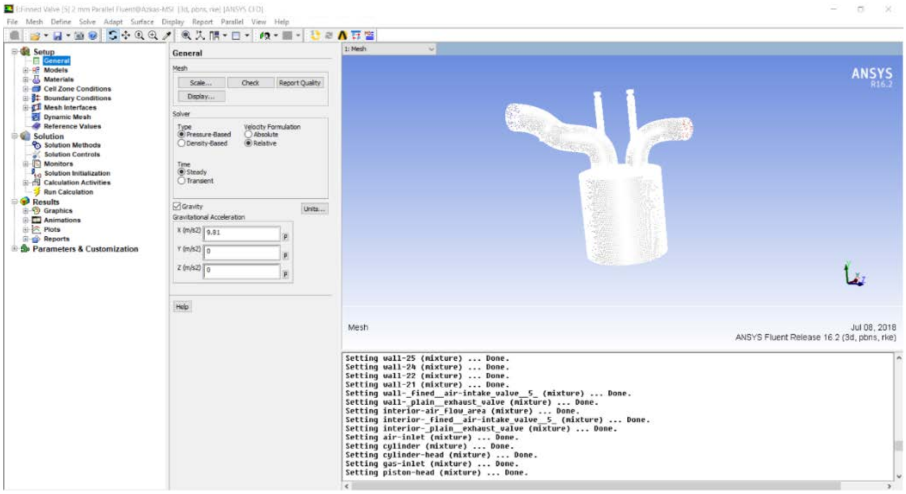


Figure 3.22. General Setup for Fluent Analysis

In general setup, it defined the mesh scaling and gravity direction at x-axis of geometry.

In fluent analysis must be defined viscous model as initial reference of simulation. This thesis defined geometry as K-epsilon with realizable model.

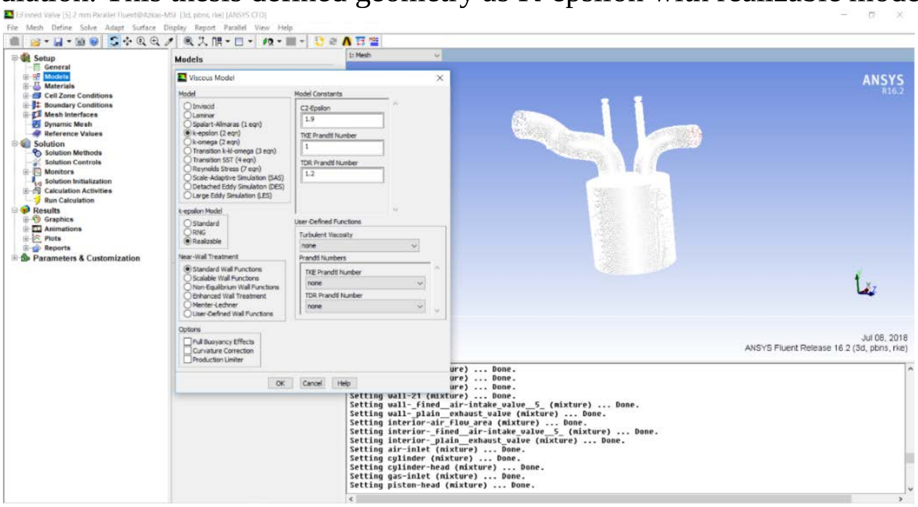


Figure 3.23. Model Setup for Fluent Analysis

Before input any boundary condition, there must be inputed the material of each body of model with detail properties, such an air as fluid or steel and aluminium as solid body.

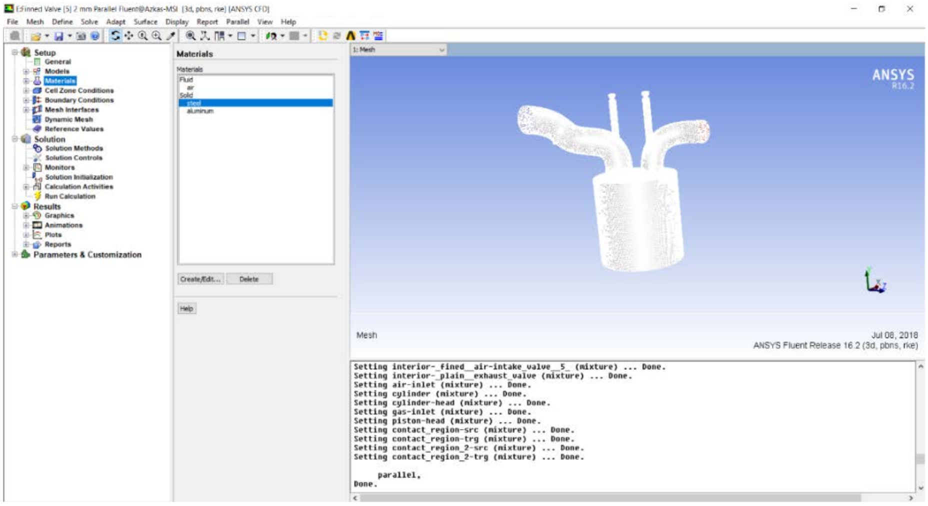


Figure 3.24. Material Selection Setup for Fluent Analysis

Defined any body as a solid or fluids based on it properties. In this case, intake valve and exhaust valve defined as solid body and flow area as fluid body.

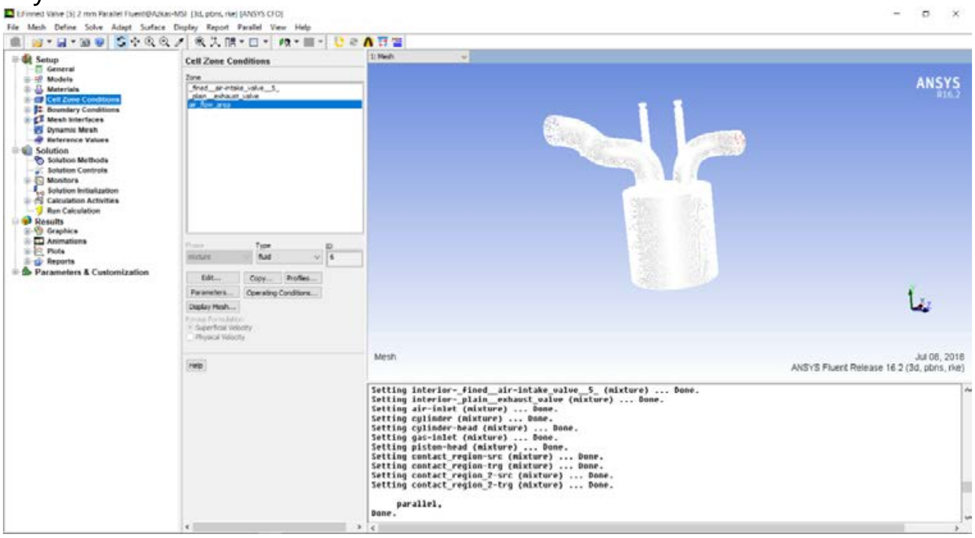


Figure 3.25. Cell Zone Setup for Fluent Analysis

In this setup, input all boundary condition to build a realistic simulation. Boundary condition can be input to named section in geometry model.

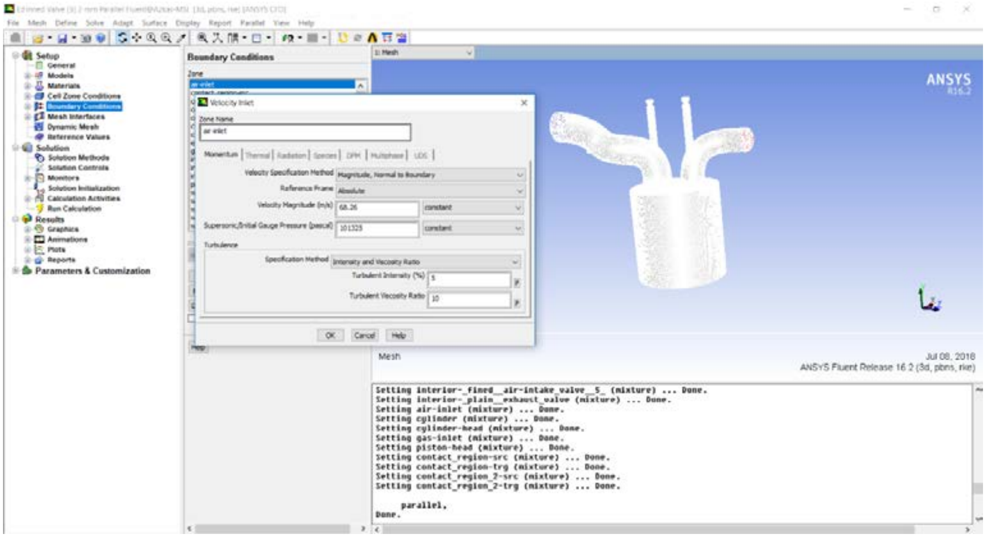


Figure 3.26. Boundary Condition Setup for Fluent Analysis

Interfaces body must be defined as well to build a valid boundary. In this case, there are two interface in geometry that is intake valve and flow area, and exhaust valve and flow area.

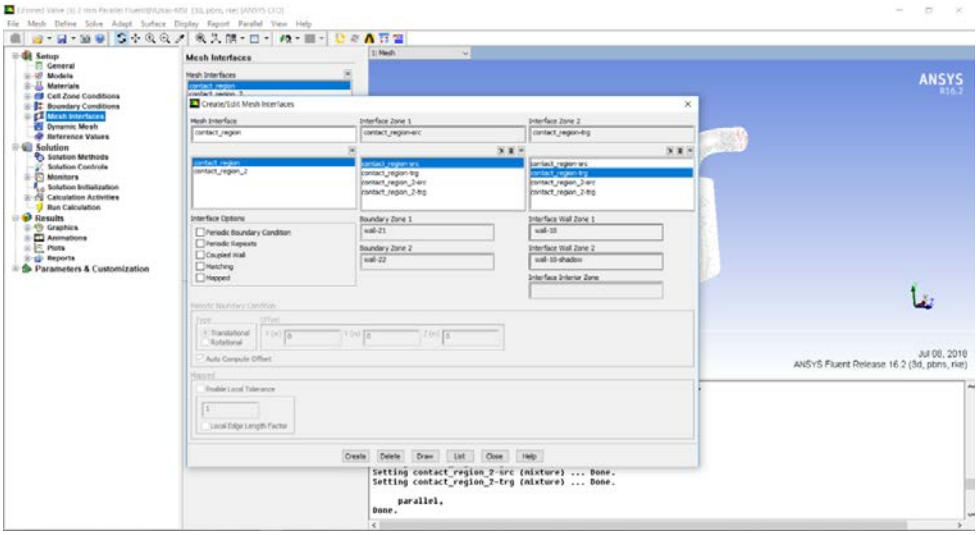


Figure 3.27. Interfaces Setup for Fluent Analysis

Input the reference value to complete fluent setup. This value includes detail numbers of density, entalphy, pressure, velocity, and viscosity.

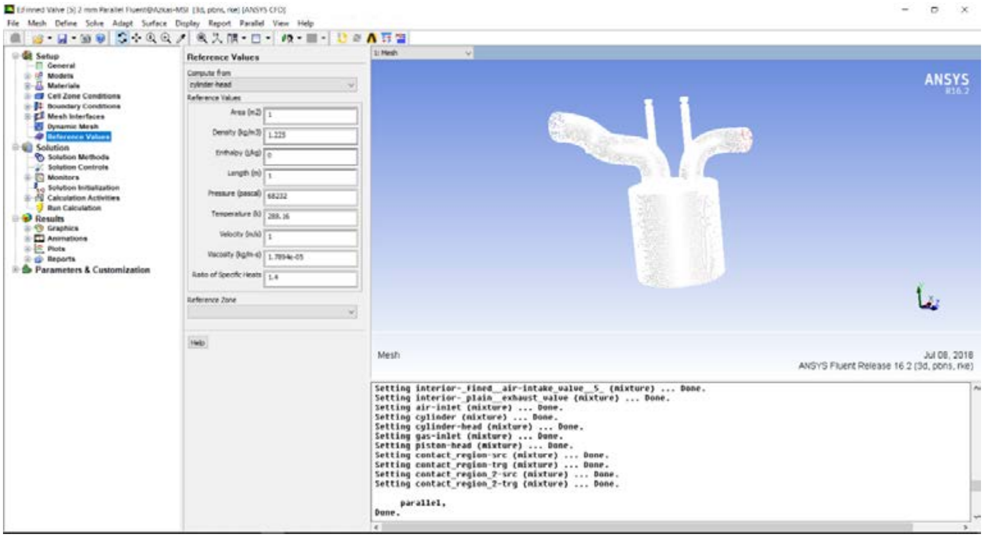


Figure 3.28. Reference Setup for Fluent Analysis

After set all the things up, it need to choose the solution methods that applied in simulation process. This method includes scheme of pressure-velocity coupling, gradient of spatial discretization and more.

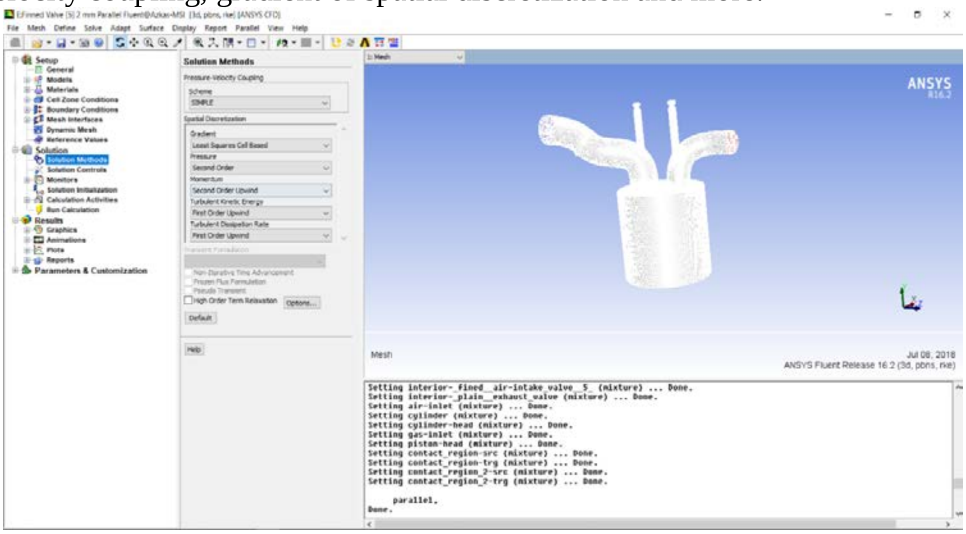


Figure 3.29. Solution Methods for Fluent Analysis

Decide which component and surface of geometry model to monitor in simulation process. This monitor will follow the iteration of calculation.

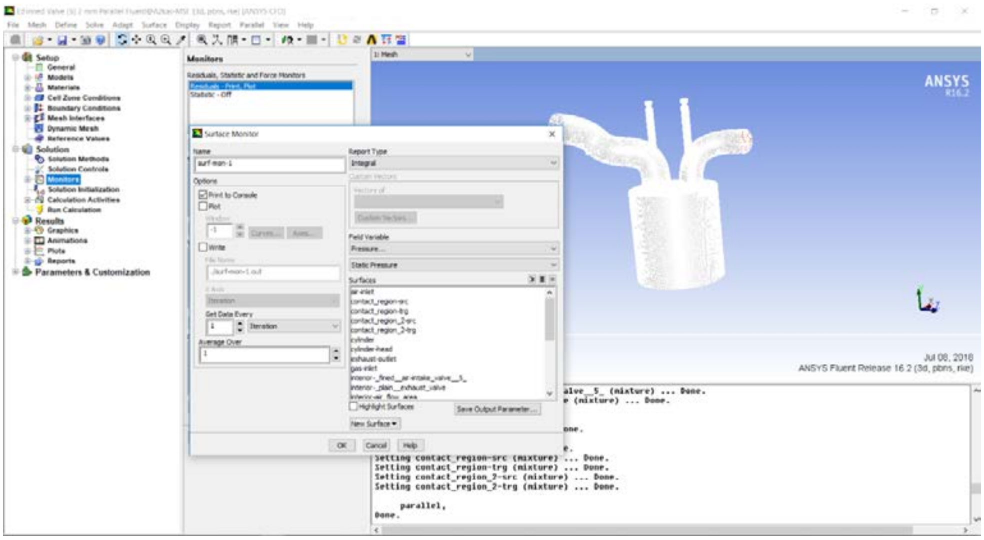


Figure 3.30. Monitors Control for Fluent Analysis

Right before calculating the simulation, it needs to initialize the setup and solution. There are two options in initialize the solution, standart initialization and hybrid initialization.

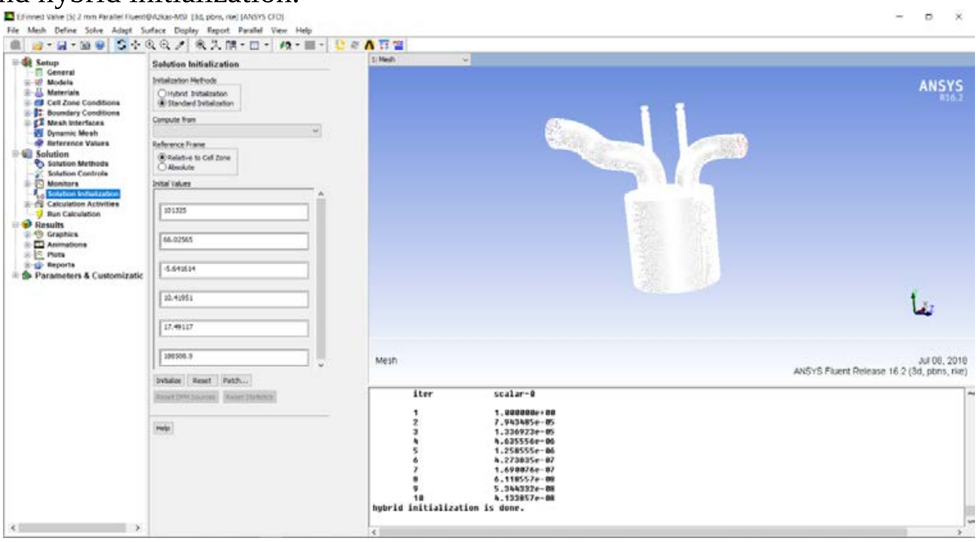


Figure 3.31. Solution Initialization for Fluent Analysis

Run the calculation to analyze setup and solution by fluent analysis. Then input how many iterations to complete calculation.

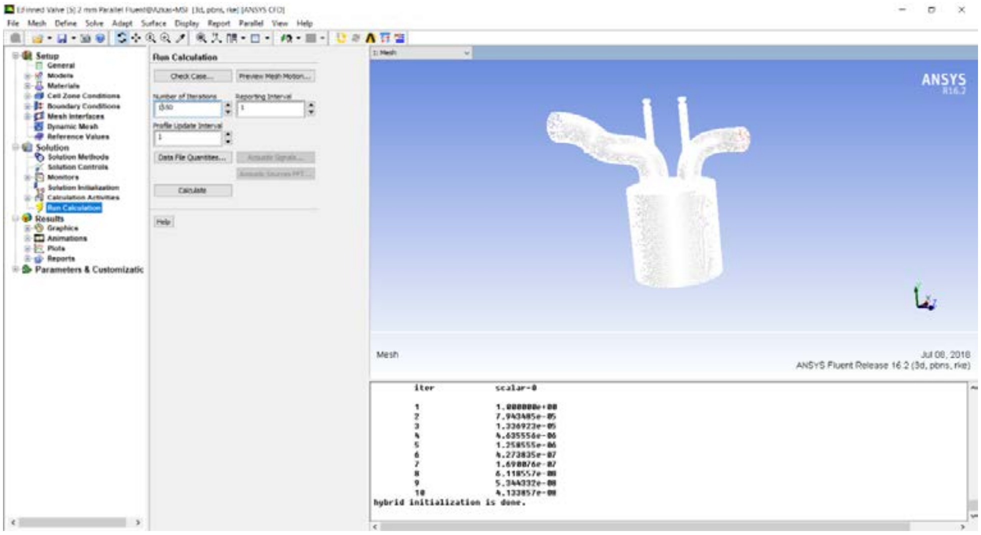


Figure 3.32. Running Calculation for Fluent Analysis

3.5. Result and Conclusion

The last stage is look at the result and identify the conclusion. the conclusions that expected in this thesis is able to answer the problem. Aim of this thesis is to know the turbulent flow effect inside the cylinder that caused by the application of finned Intake valve on modified engine. And To know the number of fin quantity on Intake valve that cause the best turbulent flow into cylinder in engine modification.

\

“This page deliberately left blank”

CHAPTER IV
DISCUSION AND RESULT

After run all solution with enough iteration or reach convergent calculation. The result can be shown in work window of Ansys Workbench main panel. Here the result of simulation.

4.1. Flow Simulation Result

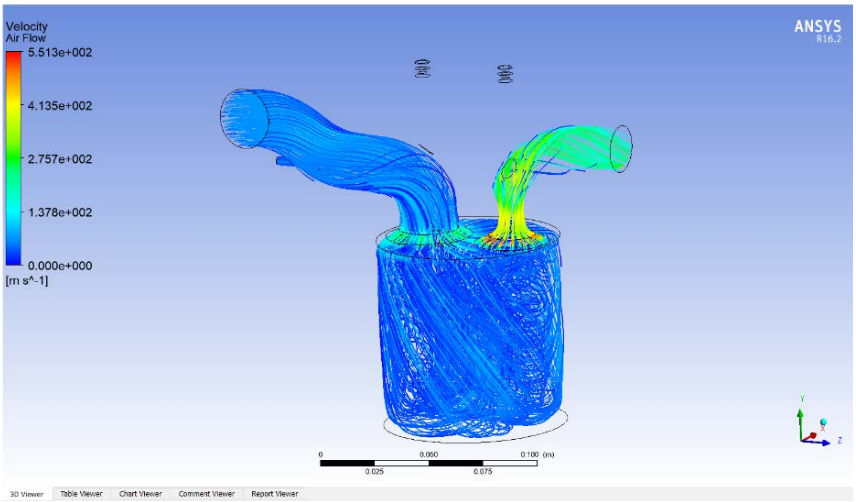


Figure 4.1. Simulation Result of Conventional Valve lift 2 mm

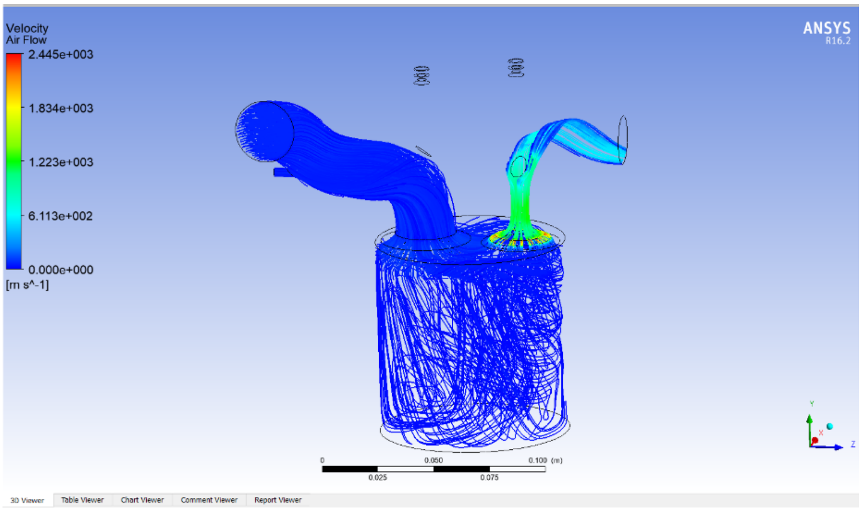


Figure 4.2. Simulation Result of Conventional Valve lift 5 mm

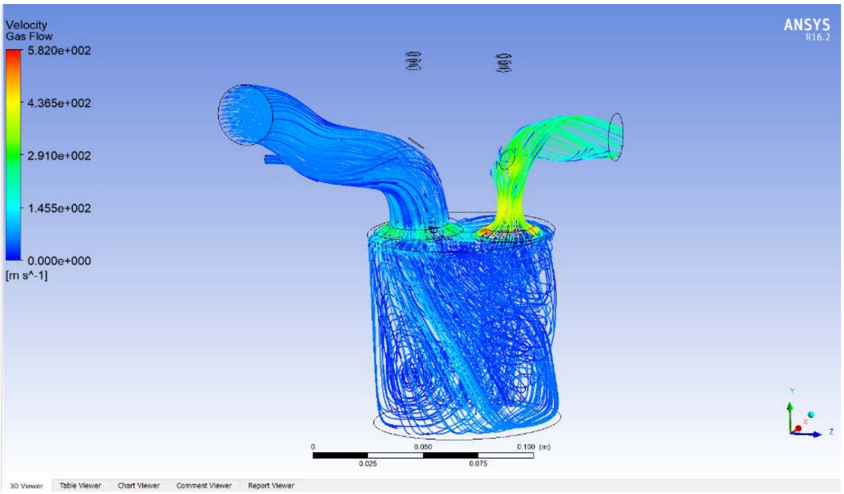


Figure 4.3. Simulation Result of 3 Fined Valve lift 2 mm

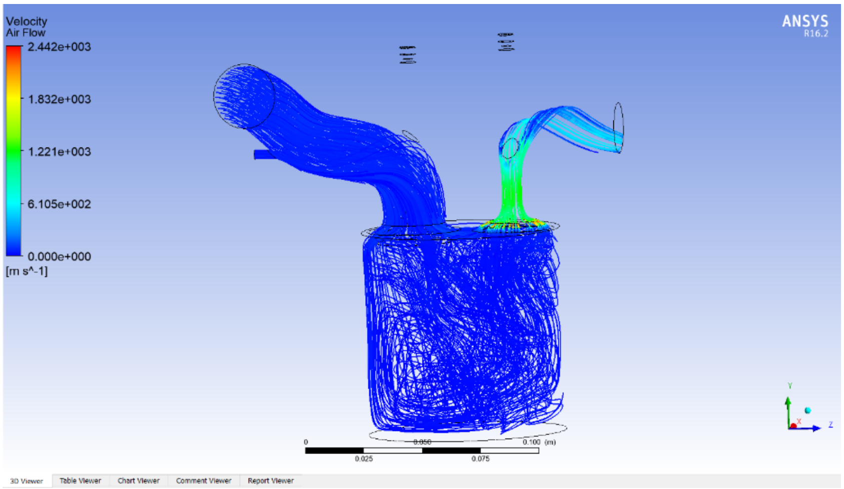


Figure 4.4. Simulation Result of 3 Fined Valve lift 5 mm

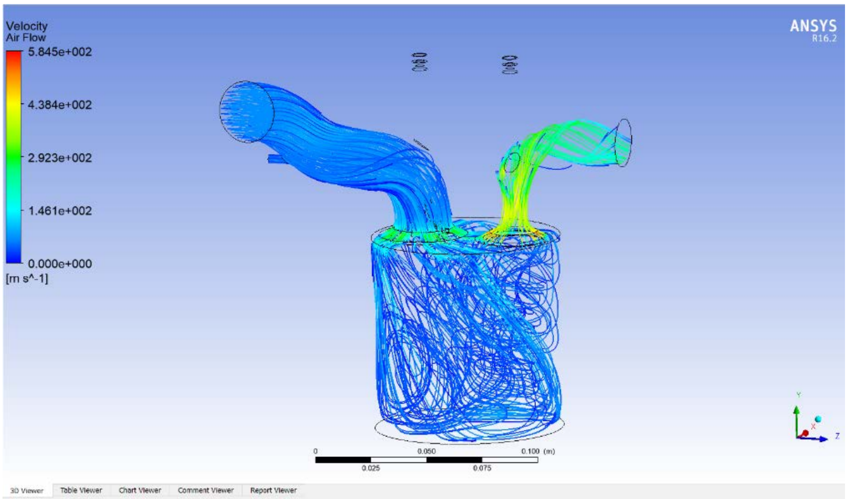


Figure 4.5. Simulation Result of 5 Fined Valve lift 2 mm

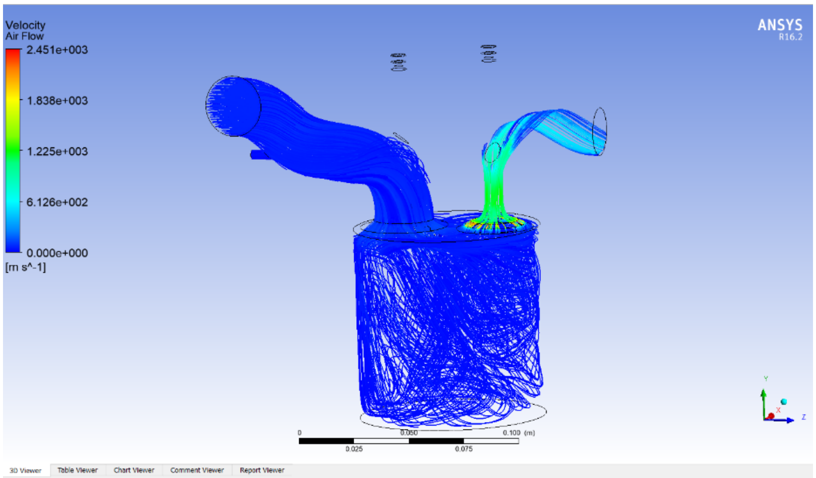


Figure 4.6. Simulation Result of 5 Fined Valve lift 5 mm

4.2. Average Air Flow Velocity

On post processing simulation, can be obtained calculation results of simulation for average air flow velocity. This value measured in internal flow inside the cylinder.

Table 3 Average Air Flow Velocity

Average Air Flow Velocity			
Valve Lift	Conventional Valve	3 Finned Valve	5 Finned Valve
Intake Valve 2 mm	84.67 m/s	89.49 m/s	96.64 m/s
Intake Valve 5 mm	167.59 m/s	180.91 m/s	199.31 m/s

The data provides information of average air flow velocity by two groups. These groups determined by two different valve lifts.

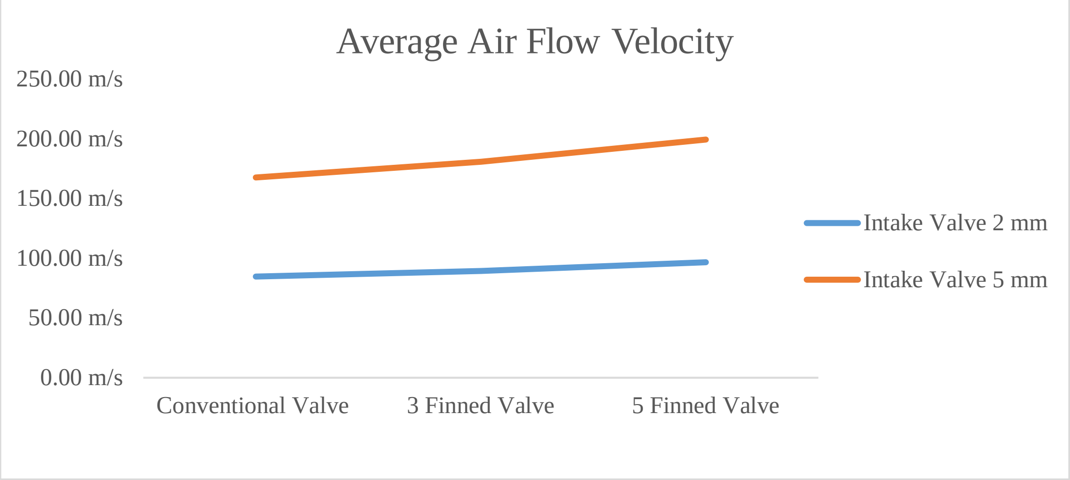


Figure 4.7. Average Air Flow Velocity

From the chart, it can be clearly seen that the average air flow velocity of simulation result slightly increased by the number of fin addition. Which are highest average velocity on 5 finned intake valve with 199.31 m/s in 5 mm valve lift and 96.64 m/s in 2 mm valve lift. By contrast, the lowest average velocity is conventional valve with 167.59 m/s in 5 mm and 84.67 m/s in 2 mm. By comparison, average velocity on 3 finned intake valve is 180.92 m/s in 5mm and 89.49 m/s in 2 mm.

4.3. Average Turbulence Kinetic Energy

From the result, can be obtained calculation results of simulation for average turbulence kinetic energy.

Table 4 Average Turbulence Kinetic Energy

Average Turbulence Kinetic Energy			
Valve Lift	Conventional Valve	3 Finned Valve	5 Finned Valve
Intake Valve 2 mm	567.95 J/kg	573.58 J/kg	577.43 J/kg
Intake Valve 5 mm	5,299.57 J/kg	5,720.80 J/kg	6,511.21 J/kg

The table above present an information of average turbulence kinetic energy by two groups. These groups determined by two different valve lifts.

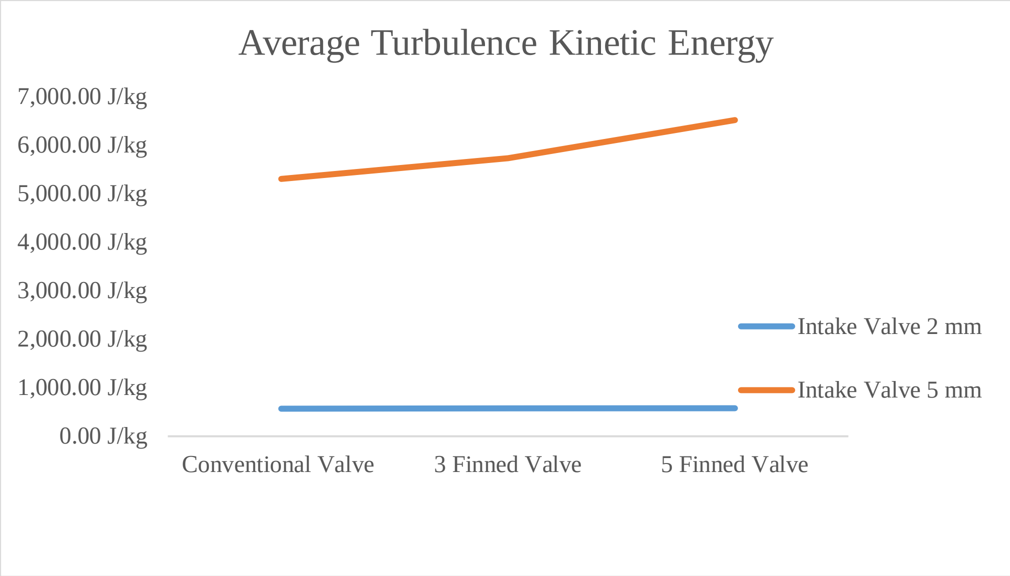


Figure 4.8. Average Turbulence Kinetic Energy

The graphic clearly shown that the average turbulence kinetic energy of simulation result quite increased by the number of fin addition. Which are highest average kinetic energy on 5 finned intake valve with 6511.21 J/kg in 5 mm valve lift and 577.43 J/kg in 2 mm valve lift. By contrast, the lowest average velocity is conventional valve with 5299.57 J/kg in 5 mm and 567.95 J/kg in 2 mm. By comparison, average velocity on 3 finned intake valve is 5720.82 J/kg in 5mm and 573.58 J/kg in 2 mm.

4.4. Turbulence Intensity

From the result, can be obtained calculation results of simulation for turbulence intensity.

Table 5 Turbulence Intensity

Turbulence Intensity			
Valve Lift	Conventional Valve	3 Finned Valve	5 Finned Valve
Intake Valve 2 mm	0.2850	0.2864	0.2874
Intake Valve 5 mm	0.8707	0.9040	0.9651

The table above present an information of turbulence intensity by two groups. These groups determined by two condition valve lifts.

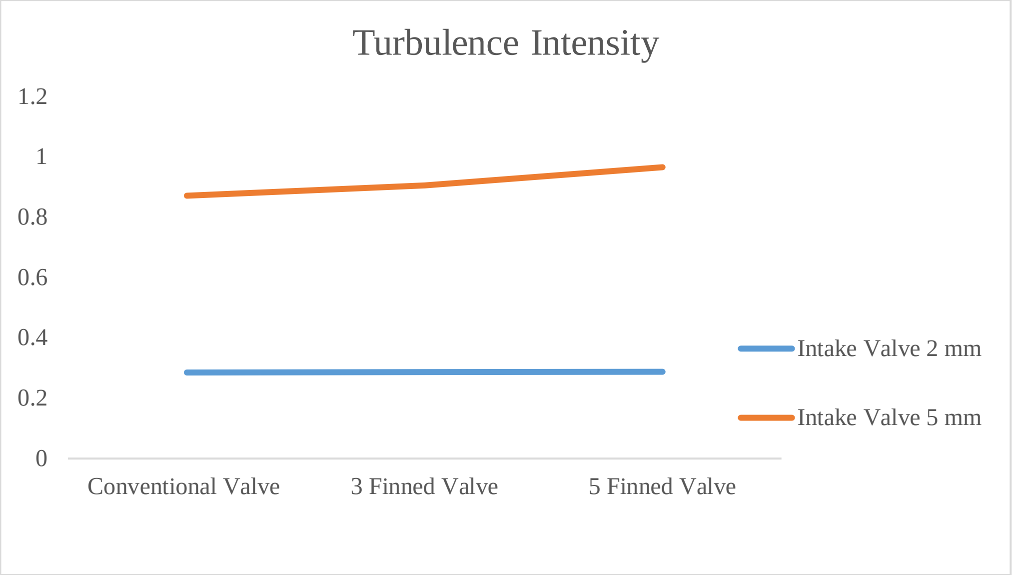


Figure 4.9. Turbulence Intensity

The graphic clearly shown that the average turbulence intensity result quite increased by the number of fin addition. Which are highest turbulence intensity on 5 finned intake valve with 0.9651 in 5 mm valve lift and 0.2874 in 2 mm valve lift. By contrast, the lowest average velocity is conventional valve with 0.9040 in 5 mm and 0.2864 in 2 mm. By comparison, average velocity on 3 finned intake valve is 0.8707 in 5 mm and 0.2850 in 2 mm.

CHAPTER V
CONCLUSION AND SUGGESTION

5.1. Research Conclusion

Based on the results of calculations that have been done by the author related to the effect analysis of additional fin in intake valve on air-gas intake flow of modification YANMAR TF 85MH-DI, it can be concluded that:

1. From the result, it obtained that fin addition on intake valve will increase turbulence intensity, which is raised from 0.8707 in valve lift 5 mm on conventional valve to 0.9040 in valve lift 5 mm on finned 3 intake valve and 0.9651 in valve lift 5 mm on finned 5 intake valve.
2. The best number of fin based on this simulation is intake valve with 5 fin addition. With turbulence intensity 0.9651 in valve lift 5 mm and 0.2874 in valve lift 2 mm.

5.2 Suggestion

Based on the analysis and whole process that has been done by the author in conducting the analysis of fin effect in inlet valve on air-gas intake flow of modification Yanmar Tf 85 Mh-Di, there are some things that need to be considered in conducting flow analysis that is: for obtaining more subsidiary result, it need to simulate in other condition. Such as another number of fin, another engine load and speed. It can be simulated by another modification engine.

“This page deliberately left blank”

BIBLIOGRAPHY

Bayraktar H., 2008, An Experimental Study on the Performance Parameters of an Experimental CI Engine Fueled with Diesel-Methanol-Dodecanol Blends, Fuel, Volume 87, pp. 158-164.

Semin and Bakar R.A., 2008, A technical review of compressed natural gas as an alternative fuel for internal combustion engines, American Journal of Engineering and Applied Sciences 1, pp 302-311.

Semin, Bakar R.A., Ismail A.R., 2008, Steady state and transient simulation of gas pressure in intake port engine, Journal of Engineering and Applied Sciences 3 (1), 47-54.

Wei L., P. G., 2016, A Review on Natural Gas/Diesel Dual Fuel Combustion, Emissions and Performance. Fuel Processing, Volume 142, pp. 264-278.

Ohashi I., 2015, Dual fuel marine engine (highly reliable environmentally friendly engine), YANMAR Technical Review.

Bakar R.A., Semin, Ismail A.R., Ali I., 2007, Computational modeling of compressed natural gas as an alternative fuel for diesel engines, Proceeding od the 2nd ANGVA Conference, Bangkong, Thailand.

Imran S., Embrson D.R., Ihracska B., Wen D.S., Crookes R.J., Korakianitis T., 2014, Effect of pilot fuel quantity and type on performance and emissions of natural gas and hydrogen based combustion in compression ignition engine, International Journal Hydrogen Energy 39 (10), pp 5163-5175.

Seddiek I. S. ; Elgohary M. M., 2014, Eco-friendly selection of ship emissions reduction strategies ith emphasis on SOx and NOx emissions, International Journal of Naval Architecture and Ocean Engineering, Volume 6, pp 737-748.

Taritas I., Sremec M., Kozarac D., Blazic M., Lulic Z., 2017, The effect of operating parameters on dual fuel engine performance and emissions - An overview, Transactions of famena XLI-1.

Candra Pavan P.V., Chaitanya Siva A., 2015, Design and analysis of the effect of a modified valve with helical guideways on combustion performance of an IC engine.

Liy Jie, Yang F., Wang H., Ouyang M., Hao S., Effect of pilot fuel quantity on the emission characteristics of a CNG diesel dual fuel engine with optimozed pilot injection timing, Applied Engergy 110, pp 201-206.

Sahoo B.B. ; Sahoo N. ; Saha U.K., 2009, Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engine – A critical riview, Renewable and Sustainable Energy Riviews, Volume 13, pp 1151-1184.

Murthy K., Madhwesh N., Shrinivasarao B.R., 2012, Influence of injection timing on the performance of dual fuel compression ignition engine with EGR, International Journal of Engineering Research and Development, Vol 1 (11), pp 36-42.

Zbierski, K., Regulacja napełnienia silnika o zapłonie iskrowym za pomocą rozrządu magnetoelektrycznego, Monografia, Wydawnictwo Politechniki Łódzkej, 2007.

Miller, R. H., Supercharging and internal cooling cycle for high output, ASME Transaction 69, pp. 453-464, 1947.

ATTACHMENTS

ATTACHMENT A INPUT DATA

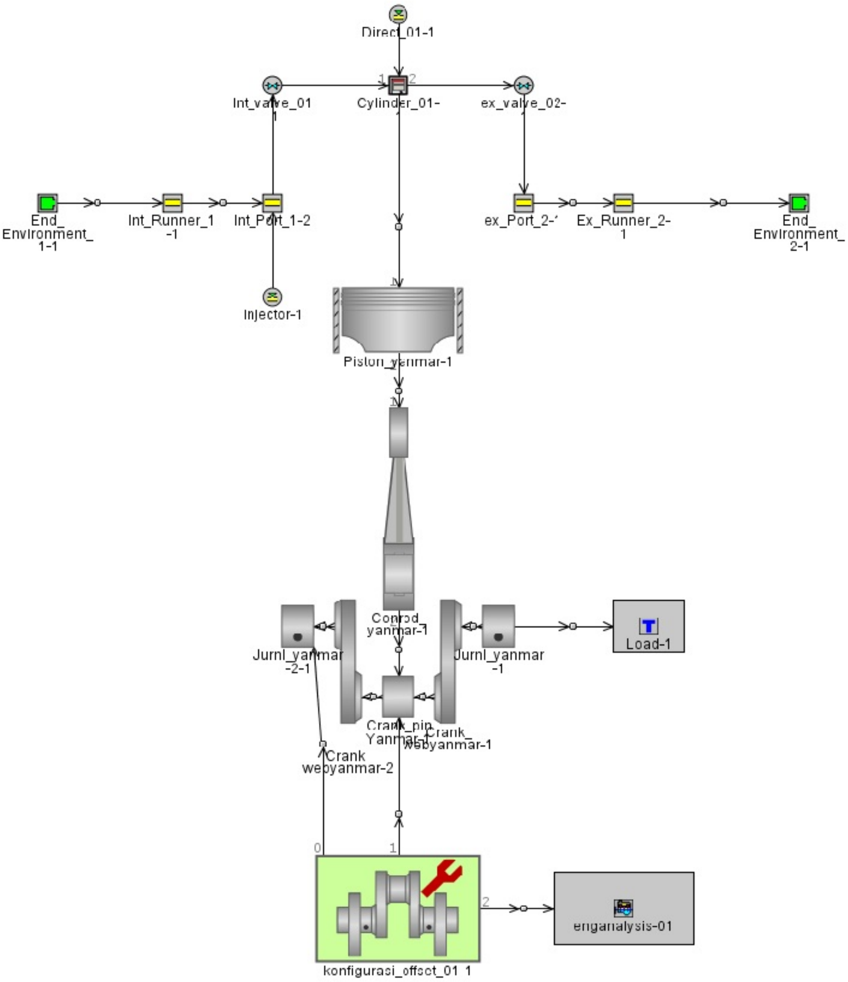


Figure A. 1. GT Power Engine System

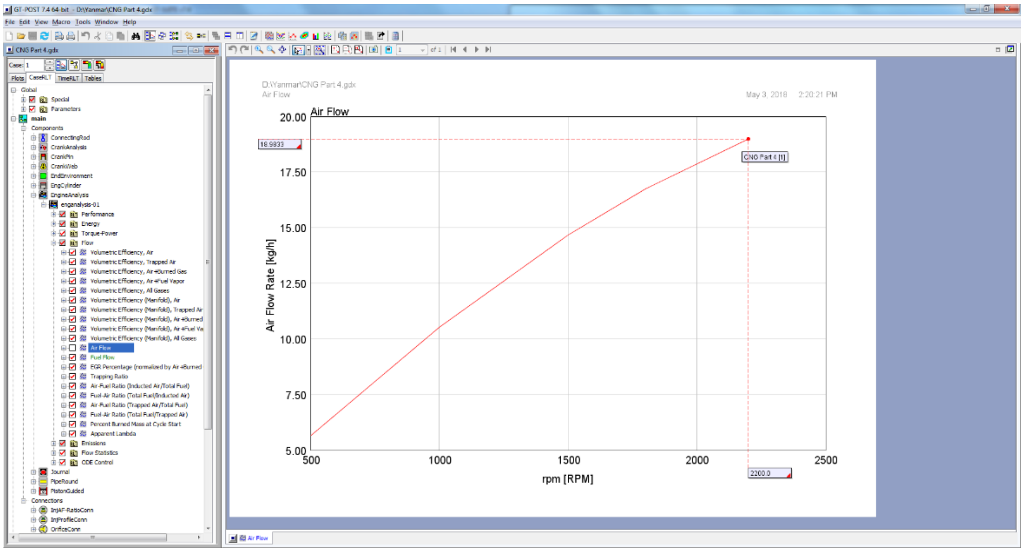
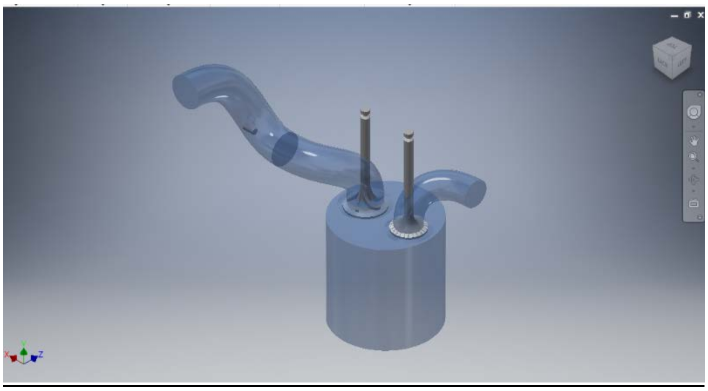


Figure A. 2. Air Flow in GT Power

Model	Unit	TF50	TF50-H	TF60	TF60-H	TF70	TF70-H	TF80	TF80-H	TF90	TF90-H	TF110	TF110-H	TF120	TF120-H	TF140	TF140-H	TF160	TF160-H
Type		Horizontal, water-cooled 4-cycle diesel																	
Combustion system		Direct injection																	
No. of cylinders		1																	
Borex stroke	mm	74 x 72		75 x 80		78 x 80		80 x 87		85 x 87		88 x 96		92 x 96		96 x 105		102 x 105	
Displacement	ℓ	0.309		0.353		0.382		0.437		0.493		0.583		0.638		0.760		0.857	
Rated continuous output	HP/rpm	4.5/2400		5.0/2400		6.0/2400		7.5/2400		8.5/2400		10.0/2400		10.5/2400		12.5/2400		14.0/2400	
	kW/rpm	3.4/2400		3.7/2400		4.5/2400		5.6/2400		6.3/2400		7.5/2400		7.8/2400		9.3/2400		10.4/2400	
At 1-hr. rated output	HP/rpm	5.0/2400		6.0/2400		7.0/2400		8.5/2400		9.5/2400		11.0/2400		12.0/2400		14.0/2400		16.0/2400	
	kW/rpm	3.7/2400		4.5/2400		5.2/2400		6.3/2400		7.1/2400		8.2/2400		9.0/2400		10.4/2400		11.9/2400	
Specific fuel consumption	g/HP-hr	175						174		170				169		155			
Compression ratio		18.4		17.9		18.1		18.0				17.9		17.7		17.8			
Position of PTO		Flywheel side																	
Direction of crankshaft rotation		Counterclockwise viewed from flywheel																	
Fuel oil applicable		Gas-oil or Light oil (UK BS 2869 A1 or Equivalent)																	
Fuel injection pump		Bosch type																	
Injection timing	deg.	bTDC 12.5		bTD 17.0				bTDC18.0				bTDC 17.0							
Injection pressure	kg/cm^2	200																	
F.O. tank capacity	ℓ (US gal)	5.6 (1.48)		7.1 (1.88)				10.5 (2.77)				11.0 (2.91)				14.3 (3.78)			
Lubrication system		Complete enclosed forced lubricating system with hydraulic pressure regulating valve																	
Lubricating oil applicable		API grade CB or CC																	
Lubricating oil capacity (Oil pan) Effective/Total	ℓ (US gal)	0.4/1.2 (0.11/0.32)		0.6/1.8 (0.16/0.48)				0.8/2.2 (0.21/0.58)				1.0/2.8 (0.26/0.74)				1.5/3.0 (0.40/0.79)			
Cooling system		Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper	Radiator	Hopper
Cooling water capacity	ℓ (US gal)	1.20 (0.32)	5.0 (1.32)	1.25 (0.33)	8.00 (2.11)	1.25 (0.33)	8.00 (2.11)	1.65 (0.44)	8.9 (2.35)	1.65 (0.44)	8.9 (2.35)	2.3 (0.61)	12.0 (3.17)	2.3 (0.61)	12.0 (3.17)	3.00 (0.79)	13.00 (3.43)	3.00 (0.79)	13.00 (3.43)
Starting system		Manual or Electric																	
Engine dimensions: Overall Length	mm	523		607				675				696				776			
Engine dimensions: Overall Width	mm	311		311				329				349				380			
Engine dimensions: Overall Height	mm	463		469				496				530				621			
Engine dry weight	kg (lb)	47.5 (104.7)	46 (101.4)	68 (149.9)	65 (143.3)	68 (149.9)	65 (143.3)	86.5 (190.7)	85 (187.4)	86.5 (190.7)	85 (187.4)	101 (220.7)	99 (218.3)	101 (220.7)	99 (218.3)	140 (308.6)	136 (299.6)	140 (308.6)	136 (299.8)

Figure A. 3. Engine Specification

ATTACHMENT B 3D MODELLING



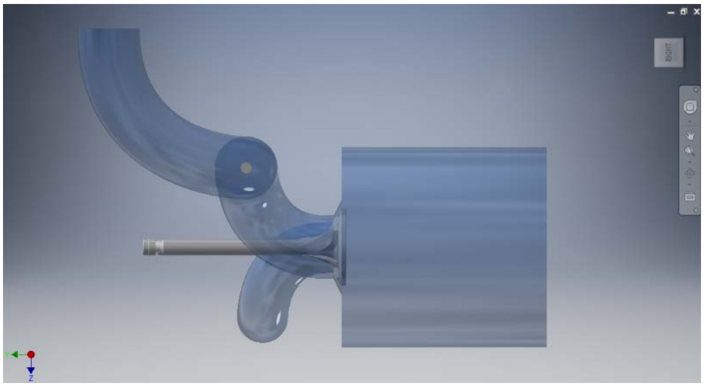


Figure B. 1. 3D Modelling in Autodesk Inventor

"This page deliberately left blank"

AUTHOR BIOGRAPHY

The Author's name is Muhamad Azka Asykarullah, was born in Kupang, November 3, 1996 as a third son of five brothers. He started his formal education in Muhammadiyah 3 Bandung elementary school, Luqman Al Hakim Islamic junior high school, and SMAN 1 Surabaya as high school. In 2014, author proceed to pursue bachelor degree at Department of Marine Engineering (Double Degree Program with Hochschule Wismar, German), Faculty of Marine Engineering, Institut Teknologi Sepuluh Nopember Surabaya specializes in marine power plant. During collage, author did many activities in campus organizations such as: Karate-Do ITS as a staff, Basketball ITS as a member, and ITS Marine Solar Boat Team as a Team Manager. The Author also joined in several On the Job Training, such as: PT Batamec Shipyard in Batam and PT Chevron Pasific Indonesia, Rumbai.

Muhamad.azka@outlook.com
(+62) 87854023335

“This page deliberately left blank”