# SIMULATION OF ELECTROMECHANICAL VALVE DRIVE ON SINGLE CYLINDER FOUR STROKE FREE PISTON DIESEL ENGINE

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UNIVERSITI MALAYSIA PAHANG

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# BORANG PENGESAHAN STATUS TESIS ◆

**JUDUL: SIMULATION OF ELECTROMECHANICAL VALVE DRIVE ON SINGLE CYLINDER FOUR STROKE FREE PISTON DIESEL ENGINE**

**SESI PENGAJIAN: 2008/2009**

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FYROZ ZIFFA BIN MOHD MUSTAFA KAMAL

A report is submitted in partial fulfillment
of the requirements for the award of the degree of
Bachelor of Mechanical Engineering with Automotive Engineering

Faculty of Mechanical Engineering
University Malaysia Pahang

NOVEMBER 2008

# SUPERVISOR DECLARATION

I hereby declare that I have read this thesis and in my opinion this thesis is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering.

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Date : 28 October 2008

## DEDICATION

First thing first, I would like to thanks and gratitude to the god, Allah Subhanahu wa Ta'aalaa whose guidance, help was instrumental in making this project become a reality. To my beloved mother Pn.Hjh.Norzi Binti Hj.Ibrahim and to my brother Fyriz Ziffa. Also to all staff in Faculty of Mechanical Engineering from University Malaysia Pahang especially to my supervisor Prof Madya Dr. Rosli bin Abu Bakar and my co-supervisor Indra Ranu Kusuma.

## ACKNOWLEDGEMENTS

First thing first, I would like to thanks and gratitude to the god, Allah Subhanahu wa Ta'aalaa whose guidance, help was instrumental in making this project become a reality. To my beloved mother Pn.Hjh.Norzi Binti Hj.Ibrahim and to my brother Fyriz Ziffa that always support and encourage me to success.

Also to all staff in Faculty of Mechanical Engineering from University Malaysia Pahang especially to my supervisor Prof Madya Dr. Rosli bin Abu Bakar and my co-supervisor Indra Ranu Kusuma for encouragement, guidance, advices and motivation. I would like acknowledge to all mechanical laboratory staff for their guidance, help and permission for me to use all the necessary mechanical laboratory equipment and for my project. I also want to give thanks to other lecturer and technician who had directly or indirectly contributed in completing my project. Without their continued support and interest, this thesis would not have been the same as presented here.

And not to forget, all my precious friends because of your help, I can get all the things done. I appreciate very much to them because of the idea, information given and for those spending their time in giving me a motivation, moral support and help whenever I need during the process for my project. Last but not least, my sincere gratitude to my partner mate project Hafnizar bin Muhamad Shaharuddin.

Thank you very much.

## ABSTRACT

In conventional internal combustion engines, the camshaft is an apparatus often used in piston engines to operate the valves. The camshaft is connected to the crankshaft. The relationship between the rotation of the camshaft and the rotation of the crankshaft is of critically importance. Since the valves control the flow of air and fuel mixture intake and exhaust gases, they must be opened and closed at the appropriate time during the stroke of the piston. In the free piston engine, there is no crankshaft. So, the electromechanical valve drive (EMVD) will take the part in make the valve moving as usual. In this project, the model of EMVD will be developed in 3D CAD software.

## ABSTRAK

Dalam enjin pembakaran dalam konvensional, aci sesondol merupakan bahagian yang selalu digunakan enjin beromboh untuk mengendalikan injap masuk dan injap ekzos. Aci sesondol berkait dengan aci engkol. Hubungan antara putaran aci sesondol dan aci engkol adalah bahagian yang sangat penting dan kritikal .Injap mengawal aliran campuran udara dan minyak, pada masukan dan ekzos, injap mesti dibuka dan ditutup pada waktu yang tepat ketika lejang omboh. Dalam enjin omboh bebas, tiada aci engkolyang hadir. Jadi sistem seperti panduan injap elektromekanikal akan mengambil bahagian untuk membuat injap bergerak seperti biasa. Di dalam projek ini, model untuk panduan injap elektromekanikal akan dibangunkan di dalam perisian 3 dimensi lakaran bantuan komputer.

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| | |
|---|---|
| $\tau_{\theta}$ | transformer torque in the Θ domain |
| $F_z$ | transformer force in the z domain |
| $J_{\theta}$ | inertia in the Θ domain |
| $m_z$ | mass in the z domain |
| $B_{\theta}$ | friction in the Θ domain |
| $B_z$ | friction in the z domain |
| $K_T$ | motor torque constant |
| $K_z$ | effective spring constant for two spring |
| $i$ | motor current |
| Θ | displacement in the rotational domain |
| z | displacement in the vertical domain |
| g (θ, t) | time-varying gas force acting on the valve reflected to the θ domain. |
| $\frac{d\theta}{dt}$ | the rotational speed in the $\theta$ domain |
| $\frac{dz}{dt}$ | velocity in z domain |
| $\frac{d^2\theta}{dt^2}$ | rotational acceleration |
| $\frac{d^2z}{d^2t}$ | acceleration in the z domain |
| $\frac{dz}{d\theta}$ | slope of the NMT characteristic |

# LIST OF ABBREVIATION

| | |
|---|---|
| BDC | Bottom dead center |
| TDC | Top dead center |
| EMVD | Electromechanical valve drive |
| NMT | Nonlinear mechanical transformer |
| NTF | Nonlinear mechanical transformer |
| VVT | Variable valve timing |

# CHAPTER 1

# INTRODUCTION

## 1.1 PROJECT BACKGROUND

In conventional internal combustion engine (ICE), engine valves are actuated by cams that are located at camshaft. Each cam at the camshaft has its degree and shape by considering a tradeoff between engine speed, power and torque requirements. These cams will determine the duration that means how long each valve will open and closed. Cams also will determine the phase that is when each valve is opened or closed and it also will determine the distance of the valve that will control the fuel into the combustion chamber. This is we called it variable valve timing (VVT).

The camshaft is connected to the crankshaft by a chain or belt, and rotates when the crankshaft does. As the camshaft rotates, it interfaces with the valves though a mechanical linkage and forces the valves open. The springs in the valve train will close the valve again once the cam stops forcing it open. Because the camshaft has a fixed profile, the valve displacement is fixed relative to the position of the crankshaft no matter what speed the engine is running at. In conventional valve drive, the valve motion is very smooth. It means that the valve moving upward and downward easily without any lagging.

The conventional valve drive also are highly regenerative, reliable and economically. But it also has its limitation that is the valve timing is fixed with respect to crankshaft position and it optimized performance at certain speeds and certain load conditions.

In many days, the internal combustion engine continues to evolve. Engines become more efficient because it produces more power per unit mass or volume and it also become cleaner. Many parts in engine have been modified including the valve actuation. After a few researches, the electromechanical valve drive (EMVD) has been found.

In EMVD system, we remove camshaft and we replaced it with motor to make valve moving as usual and we need to design a new cam to connect the valve and the motor. The motor supplies torque and control to open close and hold valve. Mechanical transformer converts rotary motion to linear motion and relates torque to applied force.

An IC engine valve's kinematics profiles for example valve position versus time or valve speed versus time are of fixed shape and are timed relative to the engine crankshaft position. With an electromagnetically driven Variable valve timing (VVT) system, one can independently control the phase and duration of the engine valve profiles, as well as carry out variable engine displacement, where certain cylinders in the engine may be deactivated. In these systems, the valves can be held in the open or closed positions for a variable time called the holding time.

They can also be commanded to transition from fully open to fully closed positions. They make this transition at a time that is well suited to the engine's instantaneous operation. In effect, VVT systems would replace the current camshaft driven valve actuation systems. To this end, VVT systems must have some of the same features as conventional valve actuation systems, while allowing the additional flexibility mentioned previously to be feasible.

## 1.2 PROBLEM STATEMENT

This project is concerns about the simulation of EMVD on single cylinder four stroke free piston diesel engine (FSFPDE).

## 1.3 OBJECTIVE

Firstly, the objective of this final year project is to build a model of EMVD by using 3D CAD software. Secondly, after modeling the model, the next step is to analyze the motion of valve at intake and exhaust.

## 1.4 SCOPE

The scope for these projects are, first is the literature study on EMVD. Then another is to design the valve on FSFPDE and lastly is to make analysis on the motion of valve for intake and exhaust.

**CHAPTER 2**

# LITERATURE REVIEW

## 2.1 CONCEPT OF EMVD

In this electromechanical valve drive (EMVD), the motor offers control over the whole stroke in both directions, providing the desired variable valve timing (VVT) behavior. The springs store and regenerate the inertial energy (from potential energy in the springs to kinetic energy in the valve) as the valve moves between the two ends of the stroke, sharply reducing the power requirement of the motor for producing a fast transition time. Most importantly, the characteristic of the nonlinear mechanical transformer (NMT) can be designed such that low seating velocity, zero holding power and low drive current can be achieved.

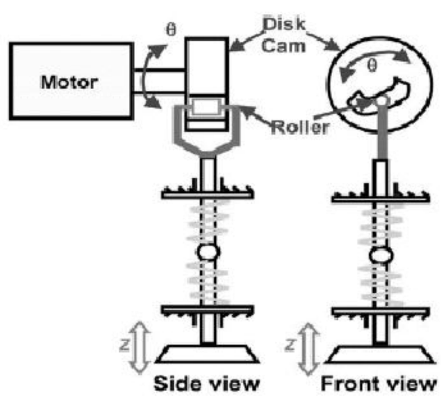



**Figure 2.1** Schematic of the proposed EMVD

The nonlinear mechanical transformer is implemented as a disk cam with a nonlinear relation prescribed in its slot.

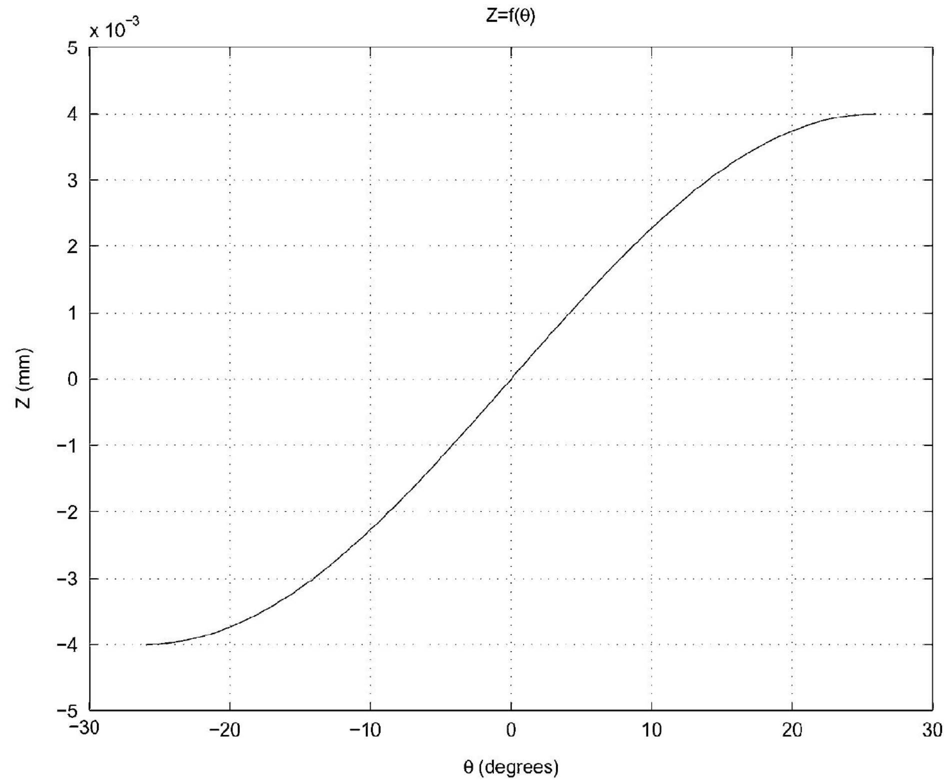



**Figure 2.2** Desirable transfer characteristic for the nonlinear mechanical transformer.

The slope of this profile is the gear ratio of the EMVD when viewed from the $\theta$ domain.

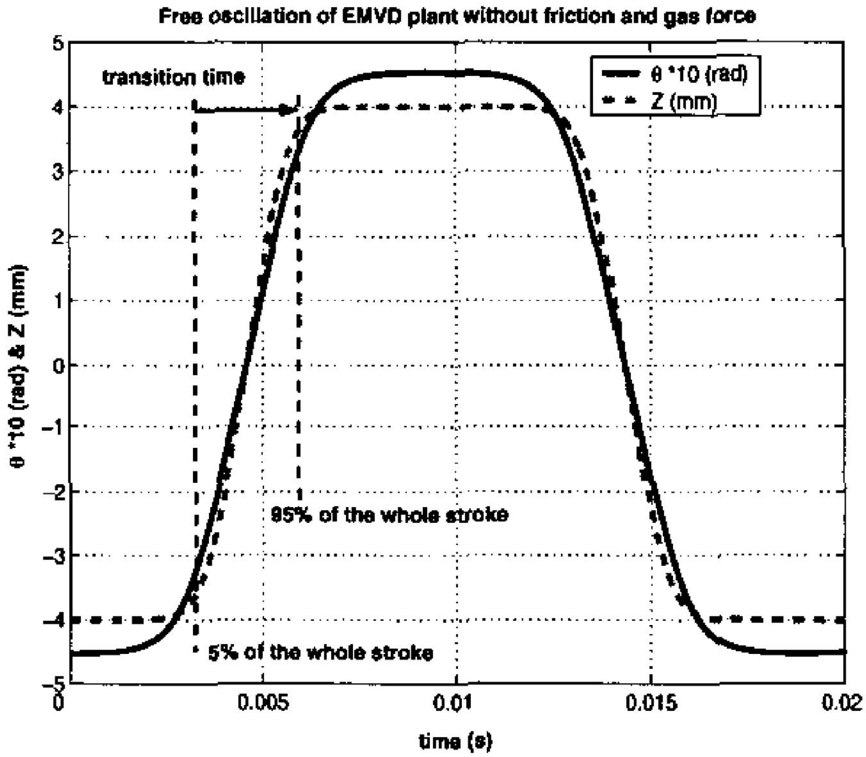



**Figure 2.3** Trajectory of free oscillation of the EMVD without friction or gas force

The benefit of using the resonant valve-spring system in the EMVD is the springs that are it allows lossless transitions of the valve from one end of the stroke to the other. For the valve-spring system, the forces are largest at the ends of the stroke because the spring forces increase linearly with valve displacement from the middle of the stroke. If the relationship between valve motion and motor shaft motion were linear, these large holding forces would make it difficult to hold the valve in the open or closed position without using large motor torques, and thus a lot of electrical power. Precise control of valve seating velocity would require precise control of motor velocity.

The equations of motion for the proposed EMVD shown in are as follow:

$$F_z = m_z \frac{d^2 z}{dt^2} + B_z \frac{dz}{dt} + K_z z$$

$$J_\theta \frac{d^2 \theta}{dt^2} + B_\theta \frac{d\theta}{dt} + \tau_\theta = K_T i$$

Where,

$\tau_\theta$ = transformer torque in the Θ domain

$F_z$ = transformer force in the z domain

$J_\theta$ = inertia in the Θ domain

$m_z$ = mass in the z domain

$B_\theta$ = friction in the Θ domain

$B_z$ = friction in the z domain

$K_T$ = motor torque constant

$K_z$ = effective spring constant for two spring

$i$ = motor current

Θ = displacement in the rotational domain

Z = displacement in the vertical domain

Two equations above can be combined using the NTF characteristic relations. In this manner, the equation can obtain a single equation of motion in either the z or the θ domains. In the θ domain, if we are to assume a time-varying gas force disturbance also acts on the valve (typical of exhaust valves in an internal combustion engine), the resulting equation of motion is

$$\left(J_\theta + m_z\left(\frac{dz}{d\theta}\right)^2\right)\frac{d^2\theta}{dt^2} + \left(B_\theta + B_z\left(\frac{dz}{d\theta}\right)^2\right)\frac{d\theta}{dt} + \left(m_z\,\frac{dz}{d\theta}\frac{d^2z}{d\theta^2}\frac{d\theta}{dt}\right)\frac{d\theta}{dt} + K_z f(\theta)\frac{dz}{d\theta}$$

$$= K_T i + g(\theta, t)$$

Where is g(θ, t) is the time-varying gas force acting on the valve reflected to the θ domain.

A block diagram of the feedback-controlled EMVD apparatus is shown in figure below. The reference input is the desired valve position, and the system output is the actual valve position. The difference between the two is passed into a controller which provides an appropriate current input to a motor drive. This motor drive supplies the desired current to the motor. Note that, for simplicity, this model assumes a perfectly responding motor drive which supplies as much current to the motor as desired, and assumes nothing about the dynamics of the motor drive.

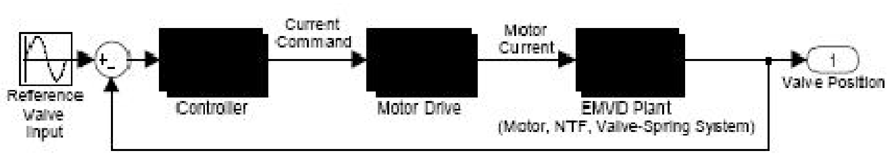



**Figure 2.4** The control block diagram

## 2.2 CHARACTERISTICS OF THE EMVD

a. Bi-positional motion achieved via a rotary motor and springs:
   i. The rotary motor provides the timing control and the power to overcome friction and gas force acting on the valve
   ii. The springs provide the large inertial power for valve transitions

b. The disk cam forces a nonlinear gear ratio between the rotary and translational domains that allows for:
   i. Inherently smooth transitions
   ii. Small valve seating velocity
   iii. Zero power to hold the valve at the open or closed positions
   iv. Low peak power/torque requirement in transitions
   v. Overshoot in the rotary domain

## 2.3 VARIABLE VALVE ACTUATION (VVA)

a. A variable engine valve actuation system offers:
   i. Duration Control
   ii. Phase Control
   iii. Lift Control

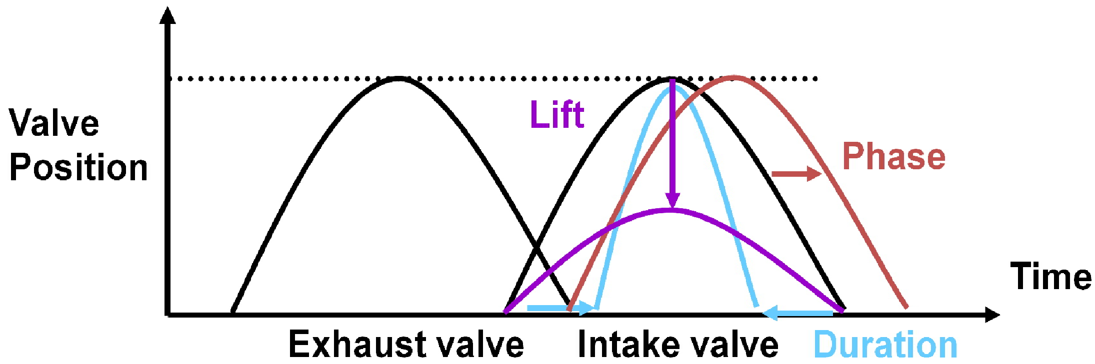



**Figure 2.5** VVA graph

## 2.4 BENEFITS OF VVA

a. Fuel efficiency improvement up to 20% per drive cycle
b. Torque improvement of 5-13%
c. Emission reduction of 5-10% in HC, 40-60% in NOx
d. Other possible benefits
    i. Smaller starter/battery
    ii. Combined starter/alternator

The disk cam is the most important part in this EMVD apparatus. The disk cam is the nonlinear mechanical transformer element that provides the unique features of this valve drive. The cam was press-fit onto the motor shaft. The motor obtained a CFS- 5-V roller follower (IKO International, Parsippany, NJ, USA) to connect the valve holder (at the top of the valve stem) to the disk cam. For the smooth profile of the center of the roller follower in the disk cam slot, we decided to implement the continuous function.

In the EMVD apparatus, the valve-spring system is carried in a larger subassembly called the valve assembly. In addition to the valve and springs, the valve assembly contains the plates, which constrain the stationary ends of the springs (the top and bottom plates), valve guides, and a plate with a valve seat, guide pins, and a linear position sensor. The model assembled two valve assemblies, the first with stiff springs and the second with soft springs.

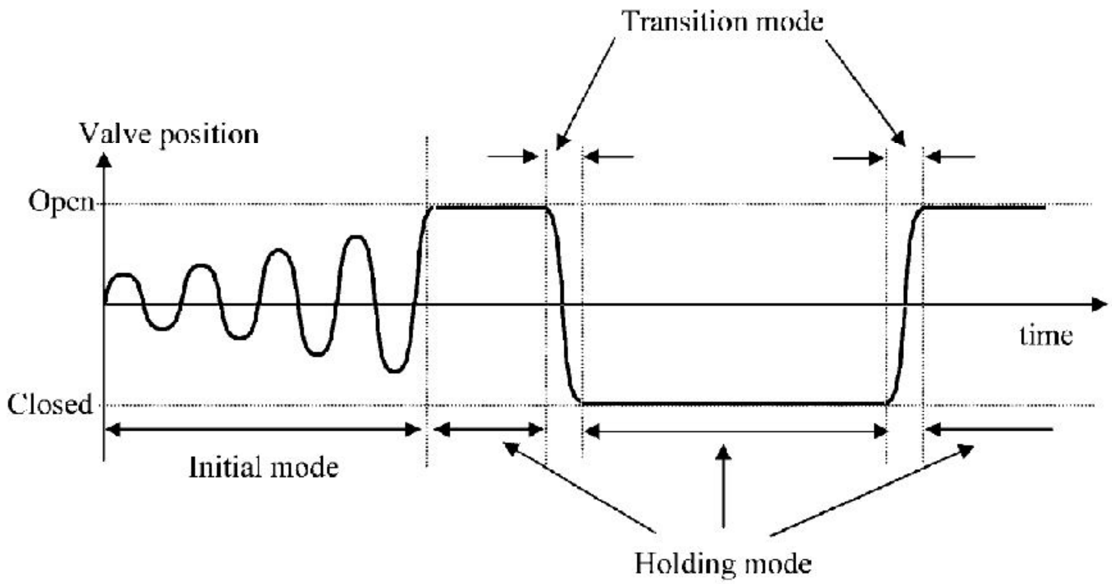



**Figure 2.6** Three modes for the EMVD: initial, holding, and transition modes

**Figure 2.7** The disk cam

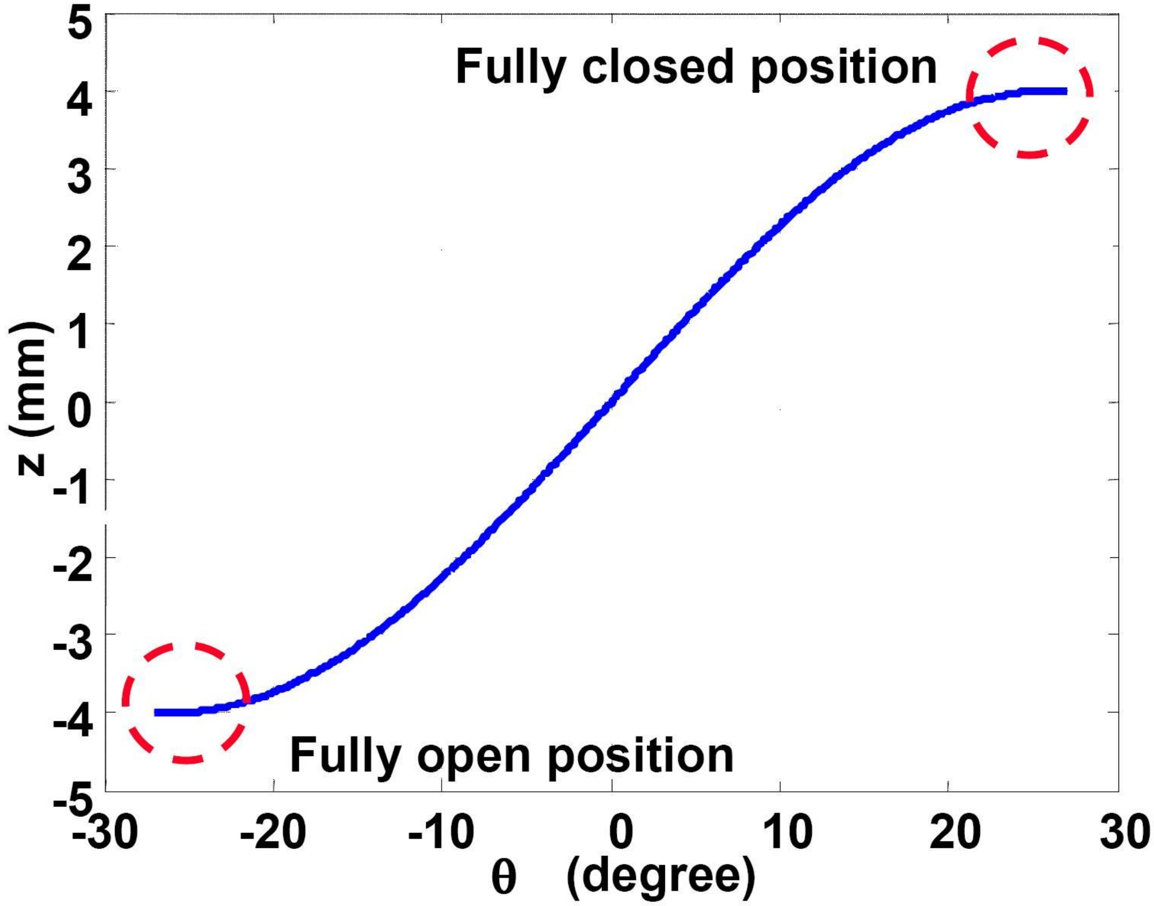



**Figure 2.8** Characteristic of valve motion

There are several issues that must be considered when designing controllers for the proposed EMVD. First, it is important to note that the dynamic characteristics of the proposed EMVD change along the valve stroke. At the ends of the stroke, the effective inertia in the *z* domain is large, while at the midpoint of the stroke, this effective inertia is small. Thus, in the *z* domain, the effective system gain of the valve-spring system decreases at the ends of the stroke and increases at the midpoint of the stroke.

First, analyze the relationship created by the nonlinear mechanical transformer (NMT). Assume the relation between displacements in the Θ domain and the z domain is, z = f (Θ). Then it is easy to show that the following relations of velocity and acceleration hold between Θ and z,

$$\frac{dz}{dt} = \frac{dz}{d\theta} \cdot \frac{d\theta}{dt}$$

$$\frac{d^2z}{dt^2} = \frac{d^2z}{dt^2} \cdot (\frac{d\theta}{dt})^2 + \frac{dz}{d\theta} \cdot \frac{d^2\theta}{dt^2}$$

Where,

$\frac{d\theta}{dt}$ is the rotational speed in the $\theta$ domain,

$\frac{dz}{dt}$ is the velocity in z domain,

$\frac{d^2\theta}{dt^2}$ is the rotational acceleration,

$\frac{d^2z}{dt^2}$ is the acceleration in the z domain

$\frac{dz}{d\theta}$ is the slope of the NMT characteristic.

# CHAPTER 3

# METHODOLOGY

## 3.1 LITERATURE STUDY

Before start with the project, all of the information must be gather to get familiar with the electromechanical valve drive (EMVD) system. All of the information for the project is available in established journals, internet and books. After going through the materials, the related information and data are collected before started the project. The concept of EMVD is important in this project. It is because the elimination of crankshaft in free piston four stroke diesel engines makes there is no torque that can make the movement of the camshaft. The camshaft movement is for to open and closed the valve. So, if there is no camshaft movement, then the valves cannot be opened or closed. Thus, a system like the EMVD system is an alternative for the free piston engine.

## 3.2 OBJECTIVE, SCOPE AND PROBLEM STATEMENT

In determining these three subjects that is objective, scope and problem statement, it is important because these three subjects must be achieved to make the project successful. First, the problem statement must be decided because the problem statement will show what must be done for this project. Many things should be consider before decide the objective. It is because the objective will make the limit for the project. The scope of the project is for what is related to the project.

## 3.3 MEASUREMENT AND MODELING.

Before designing the model and simulate it, it is important to take the measurement for the model from the actual engine. First of all, the cylinders block. Figure at the left below is the actual engine. After get all the data that required the cylinder block is design in SolidWorks software. All of the parts that consist in the engine have been designed in Solidworks such as cylinder head, intake and exhaust valves, rocker arms and many more.

### 3.3.1 Configuration of free piston engine

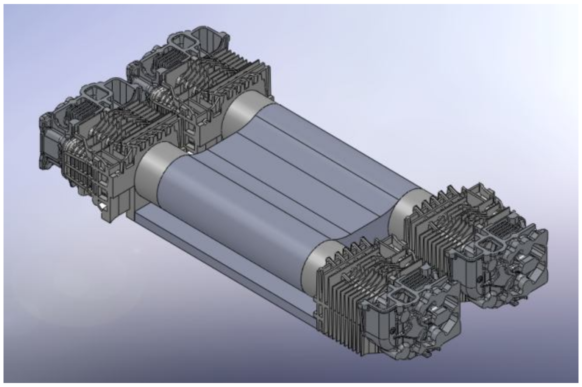


**Figure 3.1** Four stroke free piston diesel engine

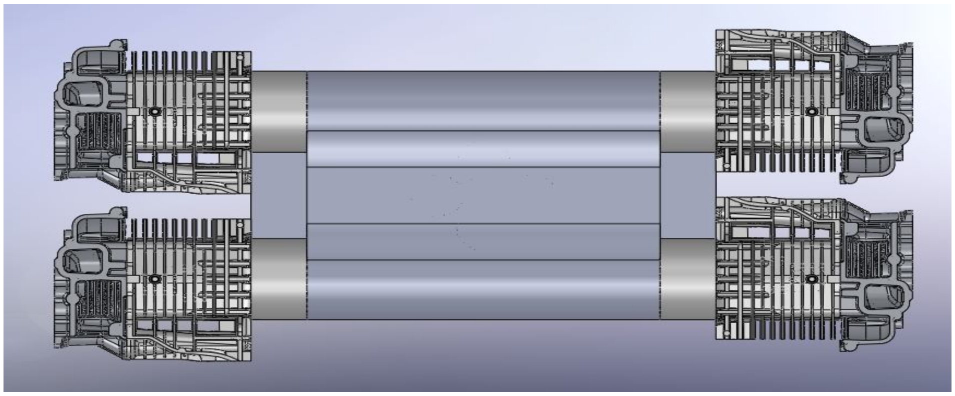


**Figure 3.2** Four stroke free piston diesel engine upper view

### 3.3.2 The cylinder block

**Figure 3.3** Actual cylinder blocks.

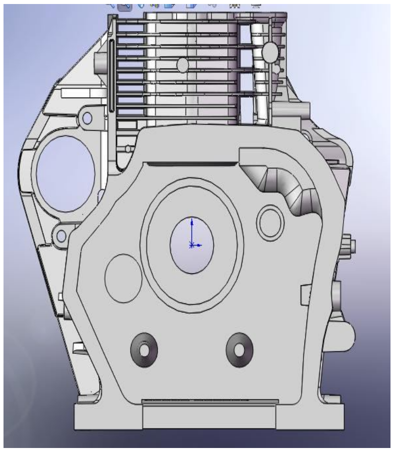


**Figure 3.4** Cylinder block in Solidworks model

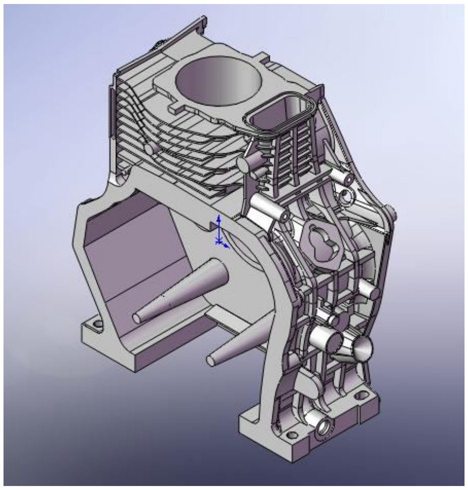


**Figure 3.5** 3D view of cylinder block

### 3.3.3 The cylinder head

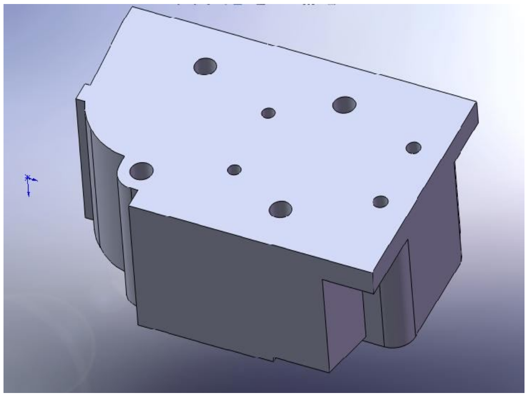


**Figure 3.6** Actual Cylinder Head

**Figure 3.7** Cylinder head design in Solidworks

### 3.3.4 The intake valve

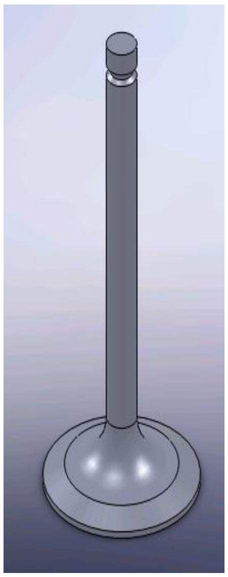


**Figure 3.8** Intake valve

**Figure 3.9** Intake valve in Solidworks

### 3.3.5 The Exhaust valve

**Figure 3.10** Exhaust valve

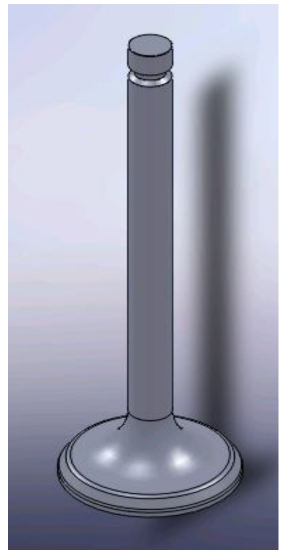


**Figure 3.11** Exhaust valve in Solidwork

### 3.3.6 The rocker arm

**Figure 3.12** Actual rocker arm

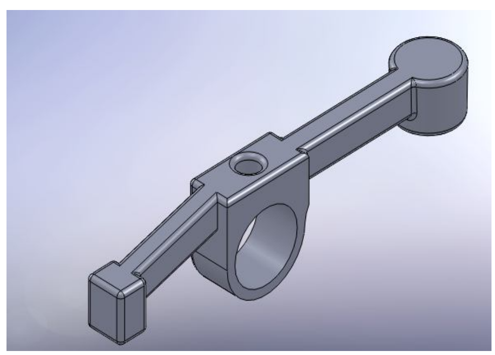


**Figure 3.13** Rocker arm in Solidworks model

### 3.3.7 Spring

**Figure 3.14** Actual spring and seat

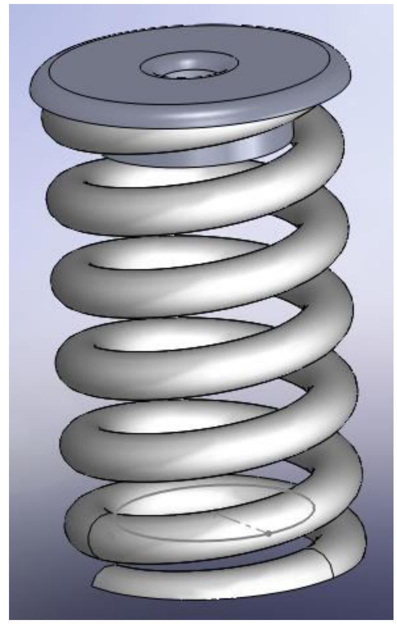


**Figure 3.15** spring and seat in solidworks model

### 3.3.8 Push rod

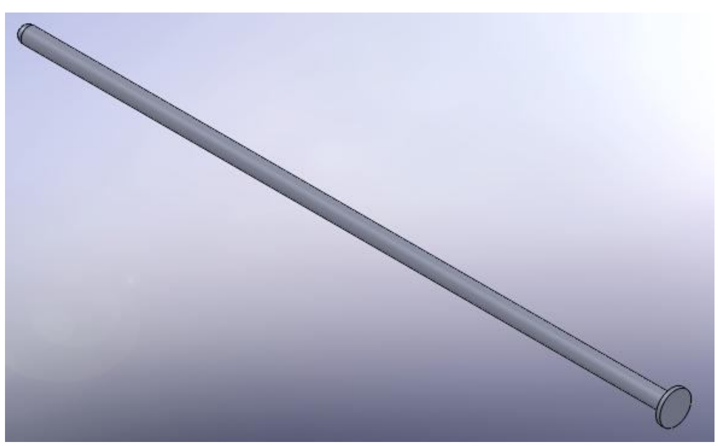


**Figure 3.16** Pushrod in Solidworks model

### 3.3.9 Camshaft

**Figure 3.17** The camshaft

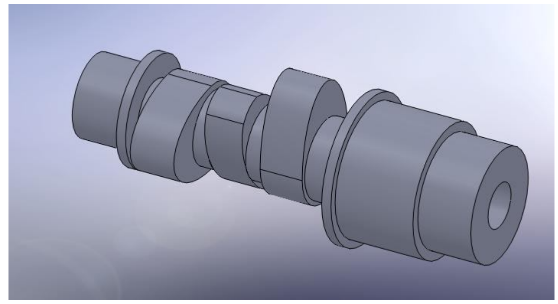


**Figure 3.18** Camshaft model in Solidworks

### 3.3.10 Electric motor

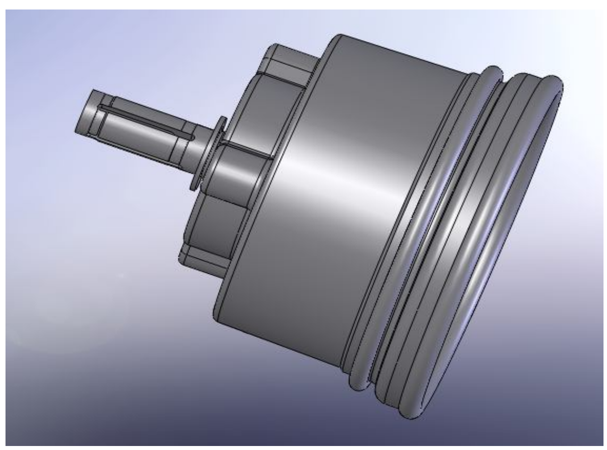


**Figure 3.19** Electric motor in Solidworks model

### 3.3.11 Connecting rocker arm

**Figure 3.20** Actual connecting rocker arm

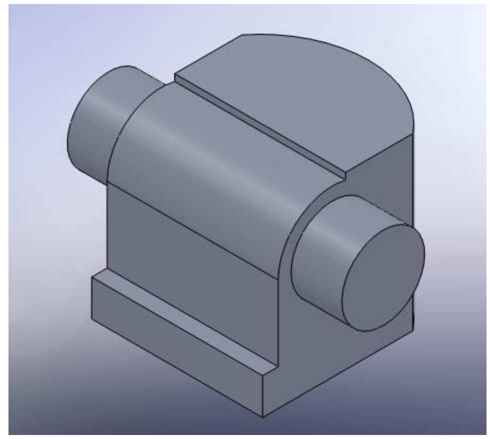


**Figure 3.21** Model of connecting rocker arm in Solidworks

## 3.4 DESIGN

Concept of designing the EMVD system is follows the journal that has established. According to the journal, the electric motor will replace the function of the crankshaft. In the easy way, it means that the electric motor will act just like crankshaft, which is to supplies torque and to control the valves whether to open, close or to hold the valve.

The mechanical transformer will converts rotary motion to linear motion and relates torque to applied force. On initial consideration, there are a number of important issues in the controller design.

First, it has established that the plant is fundamentally nonlinear. Optimal performance will almost certainly require nonlinear control. Secondly, because of the importance of soft landing, it seems essential that the control be precise near the end of the valve closing stroke.

The controller will control valve position, with opening and closing transitions occurring when the controller is given a reference position that describes an idealized free flight trajectory. In this respect, the controller detects the effects of friction and gas loads as position error.

The controller should be able to keep position error small in the presence of these variable loads and also in the presence of parameter variation. Lastly, to function in a diesel engine, a transition time must be achieved.

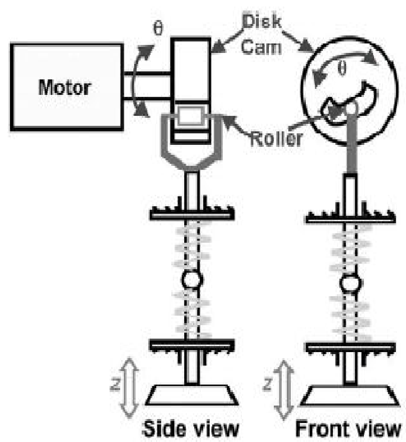



**Figure 3.22** The proposed EMVD

In the proposed EMVD, as the motor rotates, the roller moves within the slotted cam, allowing the valve to move vertically. As the valve moves, it compresses one spring and extends the other. If the relationship between valve motion and motor shaft motion were linear (as in the MAEMVD), these large holding forces would make it difficult to hold the valve in the open or closed position without using large motor torques, and thus a lot of electrical power.

## 3.5 MODELING IN GT-POWER

First of all, this software is chosen because the software is the first product that is a true virtual engine or powertrain or vehicle simulation code. It integrates all GTI products, in one environment: that is GT-POWER, GT-DRIVE, GT-VTRAIN, GT-COOL, GT-FUEL, and lastly GT-CRANK.

All products are contained in a single executable, and have single identical pre-and post-processing. GT-SUITE allows multi-disciplinary "Integrated Simulations" -- which are shaping up to be the new frontier in CAE simulations. Advanced graphical interface GT-ISE, based on object-oriented data model. This software is unique post-processing tool GT-POST, which excels in intuitive integration with GT-SUITE and automates most standard data analysis tasks. The output data is stored in a commercial MySQL database, and in XML format. This software is quite easy to use, with minimal learning curve when moving from one product to another.

To develop this model, there are many parameters that should be determined. In developed the model, first for the project it must used the GT-VTrain. This is a multi-purpose simulation tool for the mechanical design analysis of valve train systems. It has a well proven and validated track record and performs, in a single environment, the following valve train analysis functions:

a) Cam design
b) Valve train mechanism kinematics
c) Quasi-static rigid analysis, with all inertias plus valve springs; computes all forces, separation speeds
d) Valve train multi-body dynamics
e) Contact (for example cam-follower) tribology: Hertzian stress, deformation, oil film thickness, wear load
f) Valve train frictional forces and power loss
g) Camshaft torsional vibrations (time/freq. domain)
h) Camshaft bending dynamics
i) Camshaft bearing orbits, friction torques
j) Gear, chain and belt drive dynamics

Before starts run the model, the cam data is the important part. Cam Kinematics is the ObjectName of the 'KinemCam*' object containing cam and cam-follower kinematics information for each cam. Cam Position is cam axial position, measured from drive side bearing for each cam (dimensions CAMPOS (1), CAMPOS (2)... in figure below). Cam Angle is angle shift (delay) for each cam (see figure below). Valve train Branch Angleis for this attribute defines the angle between the Global Coordinate System and the zero Reference Angle (see figure below).

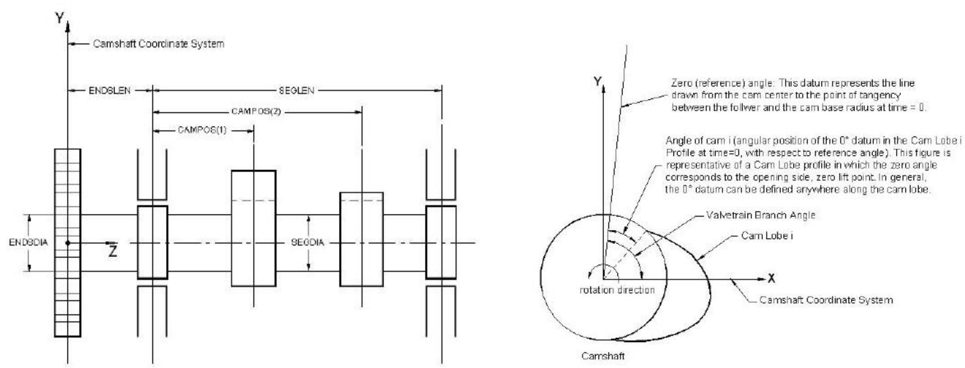
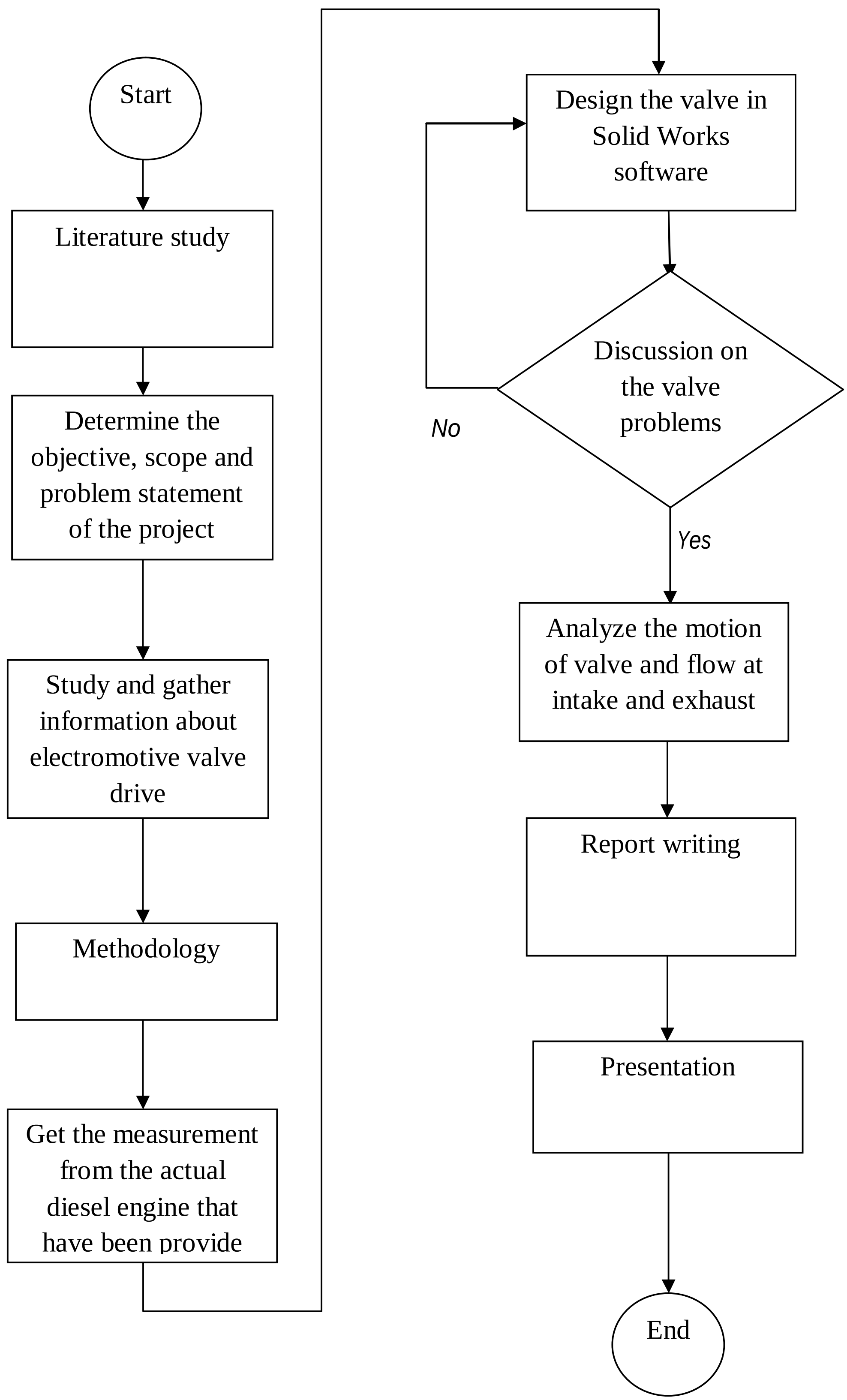



**Figure 3.23** Parameter that needed in GT-Power software

This attribute allows for the correct force direction to be applied for each valve train attached to the camshaft for bending calculations and bearing analysis. Among other applications, this attribute allows the user to model camshafts for V-Engines where there is a common camshaft that has pushrod valve train branches that go to both cylinder banks. In these situations, the lifter followers, for valve trains in different banks, will not be in the same plane and thus the forces that they apply to the camshaft will not be in the same direction. This attribute can be set to "def" and 90 degrees will be used.

## 3.6 FLOWCHART



## 3.7 GANTT CHART

### 3.7.1 Final year project 1

| Project Activities | W1 | W2 | W3 | W4 | W5 | W6 | W7 | W8 | W9 | W10 | W11 | W12 | W13 | W14 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Literature study | | | | | | | | | | | | | | |
| Identify problem statement | | | | | | | | | | | | | | |
| Define objective and scope of study | | | | | | | | | | | | | | |
| Detailed methodology | | | | | | | | | | | | | | |
| Proposal preparation | | | | | | | | | | | | | | |
| Presentation preparation | | | | | | | | | | | | | | |
| FYP1 presentation | | | | | | | | | | | | | | |

### 3.7.2 Final year project 2

| Project Activities | W1 | W2 | W3 | W4 | W5 | W6 | W7 | W8 | W9 | W10 | W11 | W12 | W13 | W14 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Literature study | | | | | | | | | | | | | | |
| Identify problem statement | | | | | | | | | | | | | | |
| Define objective and scope of study | | | | | | | | | | | | | | |
| Detailed methodology | | | | | | | | | | | | | | |
| Proposal preparation | | | | | | | | | | | | | | |
| Presentation preparation | | | | | | | | | | | | | | |
| FYP2 presentation | | | | | | | | | | | | | | |
| Report Submission | | | | | | | | | | | | | | |

# Chapter 4

# Result and Discussion.

## 4.1 Engine Specifications.


**Figure 4.1** Actual four stroke diesel engine.

**Table 1** Engine specification

| | |
|---|---|
| **Type** | 4-stroke, vertical cylinder, air-cooled diesel engine |
| **No. of Cylinders** | 1 |
| **Bore x Stroke (mm)** | 86 x 75 |
| **Displacement (litres)** | 0.435 |
| **Continuous Rated Output** | |
| **r/min$^{-1}$** | 3600 3000 |
| **kW** | 6.6 5.7 |
| **PS** | 9.0 7.7 |
| **Max Rated Output** | |
| **r/min$^{-1}$** | 3600 3000 |
| **kW** | 7.4 6.5 |
| **PS** | 10.0 8.8 |
| **High Idling r/min$^{-1}$** | 3800 3175 |
| **Dry Engine Weight with Electric Start (lbs/kg)** | 118 / 53.5 |
| **Dry Engine Weight without Electric Start (lbs/kg)** | 107 / 48.5 |
| **Cooling System** | Forced Air by Flywheel Fan |
| **Lubrication System** | Forced lubrication with Trochoid Pump |
| **Starting System** | Electric Start / Recoil Start |
| **Dimensions (inches/mm)** | |
| **Length x Width x Height** | 16.2 / 412 x 18.5 / 471 x 19.4 / 494 |
| **Engine Oil Pan Capacity** | |
| **Dipstick Upper Limit** | 1.7 qt / 1.6 litres |
| **Dipstick Lower Limit** | 1.06 qt / 1.0 litres |
| **Fuel Tank Capacity** (recommended) | 5.0 qt / 4.7 litres |

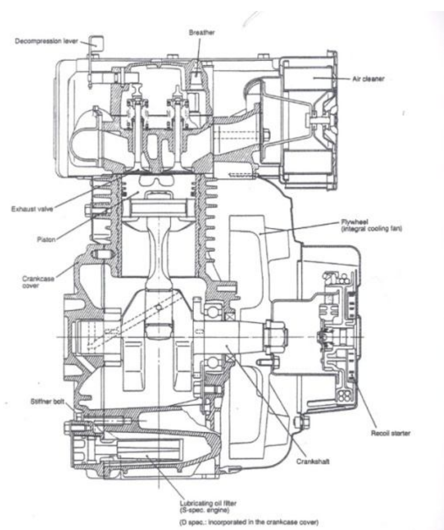



**Figure 4.2** The diesel engine configuration (front view)

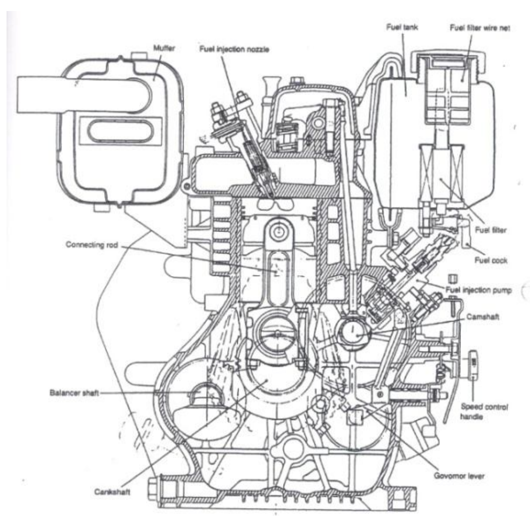



**Figure 4.3** The diesel engine configuration (side view)

This engine using is direct injection (DI) diesel engine. The injection nozzle is placed inside the combustion chamber and the piston incorporates a depression (often toroidal) where initial combustion takes place. Direct injection diesel engines are generally more efficient and cleaner than indirect injection engines. Some recent gasoline engines utilize direct injection as well. This is the next step in evolution from multi-port fuel injection and offers another magnitude of emission control by eliminating the wet portion of the induction system. By virtue of better dispersion and homogeneity of the directly injected fuel, the cylinder and piston are cooled, thereby permitting higher compression ratios and more aggressive ignition timing, with resultant enhanced output. More precise management of the fuel injection event also enables better control of emissions. Finally, the homogeneity of the fuel mixture allows for leaner air/fuel ratios, which together with more precise ignition timing can improve fuel economy. Some direct-injection systems incorporate piezo-electronic injectors. With their extremely fast response time, multiple injection events can occur during each power stroke of the engine

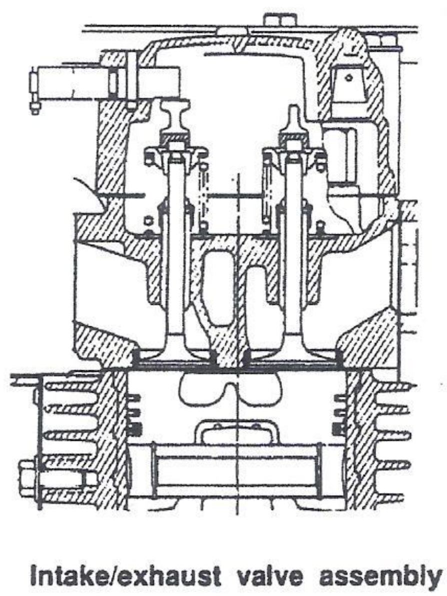



**Figure 4.4** The cut view of cylinder head

In order to design the EMVD according to the proposed, there are some difficulties in modeling it. There are many parameters to consider before build the model in Solidworks. First the proposed EMVD from the journal is for gasoline engine. Secondly, if the electric motor is at the top of cylinder head it is difficult to design it. The reason is in this four stroke diesel engine, the location for the camshaft is at bottom of the cylinder head. So the motor should situate at the bottom of the cylinder head.

## 4.2 Model configuration

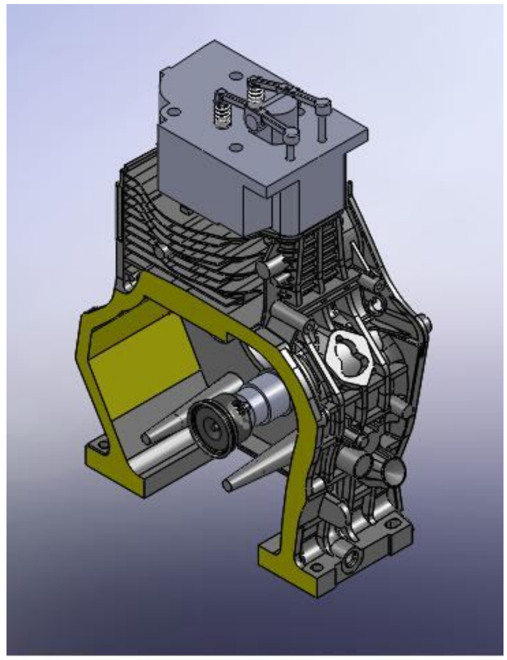


**Figure 4.5** The Full assembly of  EMVD system.

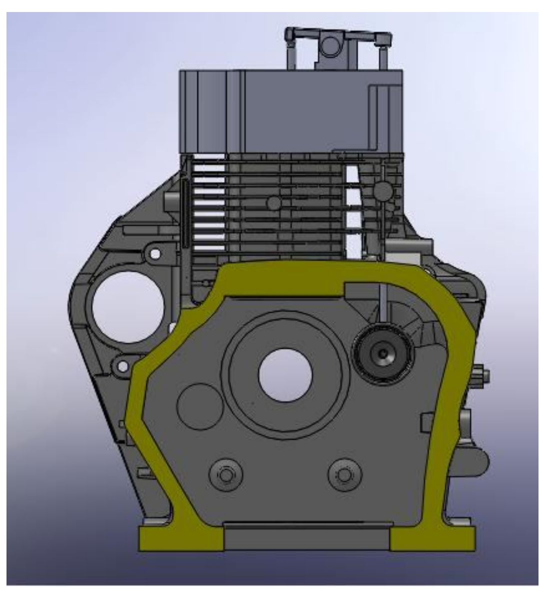


**Figure 4.6** The front view of the EMVD system.

In calculating the opening valve, a ruler has been put at the cylinder head. The valve will be measured by using the ruler. The maximum of intake and exhaust valve will be determined by the ruler. The figure below will show the location of the ruler at the cylinder head.

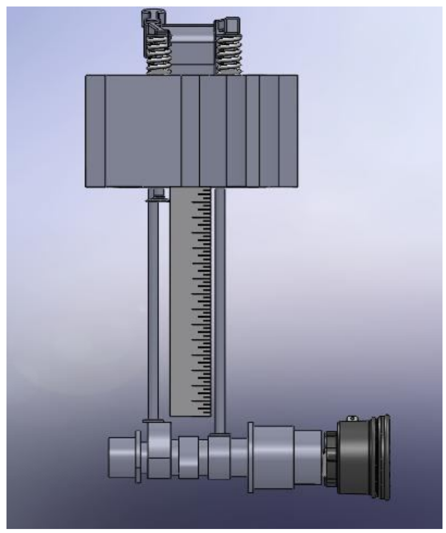


**Figure 4.7** Location of the ruler

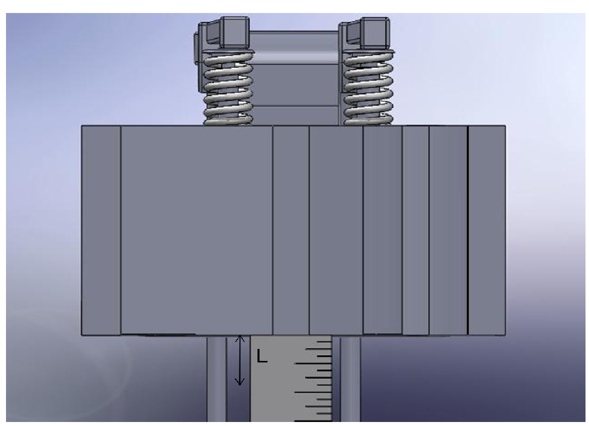
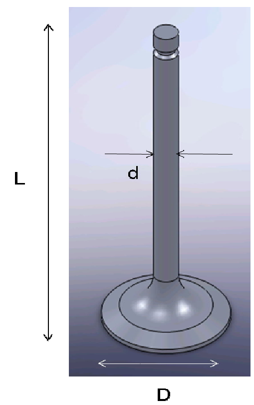



**Figure 4.8** Focusing at the cylinder

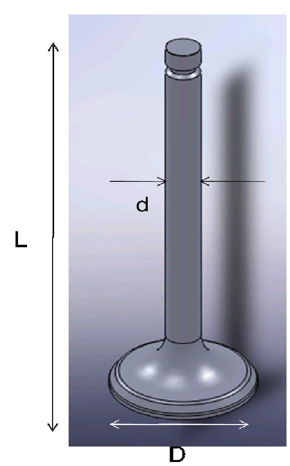
## 4.3 Dimension of the parts

### 4.3.1 Intake valve



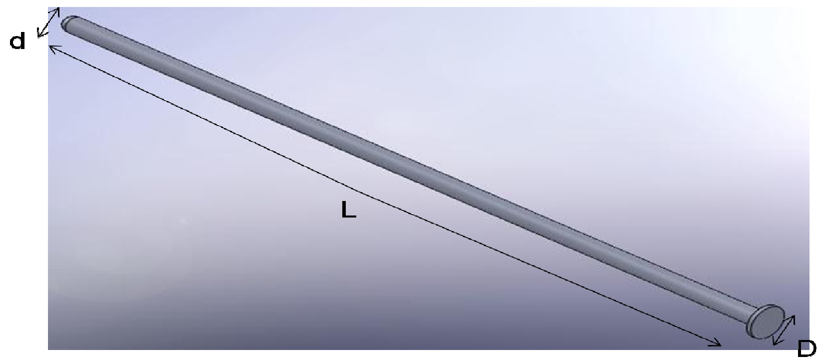

- Diameter (D) : 32.38 mm
- Length (L) : 100.55mm
- Diameter (d) : 6 mm

### 4.3.2 Exhaust valve



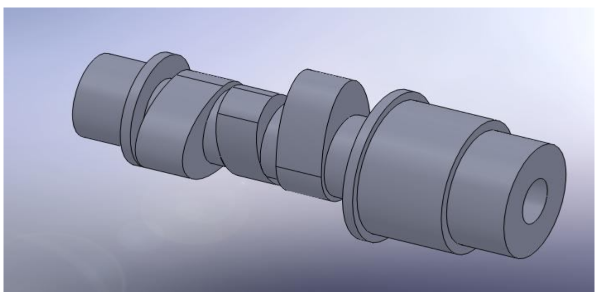

- Diameter (D) : 16.4mm
- Length (L) : 100.55mm
- Diameter (d) : 6 mm

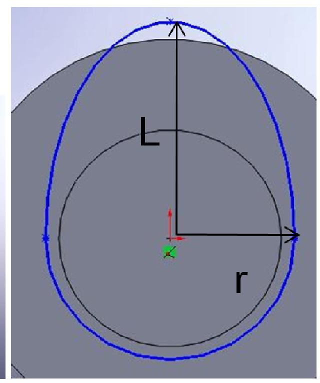
### 4.3.3 Push rods




- Diameter (D) : 12.4 mm
- Length (L) : 257.5mm
- Diameter (d) : 6 mm

### 4.3.4 Camshaft






a. Intake cam

- Base radius (r) :14 mm
- Ellipse length (L) : 25 mm

b. Exhaust cam

- Base radius (r) :14 mm
- Ellipse length (L) : 23 mm

To get computational valve opening, the EMVD has been modeled in GT Power Suite. From this software, the outcomes are the valve maximum opening and also the velocity of the valves. This software offers the true "virtual engine or power train" tool, capable of integrated simulations of the total engine and power train system.

But in this project, the consideration is only at the first point that is the design of the cam. And for the others parameters the value will be ignored and using the software default.

## 4.4 Cam's profile.

### 4.4.1 Intake cam

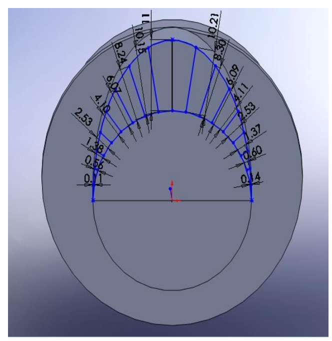


**Figure 4.9** Intake cam profile

To get the value to be entered in the GT Power software, foe every 10 degree, the length from the radius point will be calculated. The length will be minus with the base radius that is 14 mm. Table below is for intake and exhaust cam profile.

**Table 2** Intake cam profile

| Degree | Distance (mm) |
|---|---|
| 10 | 0.11 |
| 20 | 0.56 |

| | |
|---|---|
| 30 | 1.38 |
| 40 | 2.53 |
| 50 | 4.10 |
| 60 | 6.07 |
| 70 | 8.24 |
| 80 | 10.15 |
| 90 | 11 |
| 100(80) | 10.21 |
| 110(70) | 8.30 |
| 120(60) | 6.09 |
| 130(50) | 4.11 |
| 140(40) | 2.53 |
| 150(30) | 1.37 |
| 160(20) | 0.6 |
| 170(10) | 0.14 |

### 4.4.2 Exhaust cam

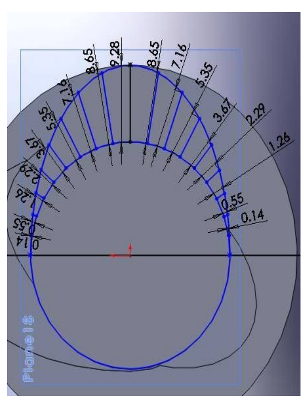


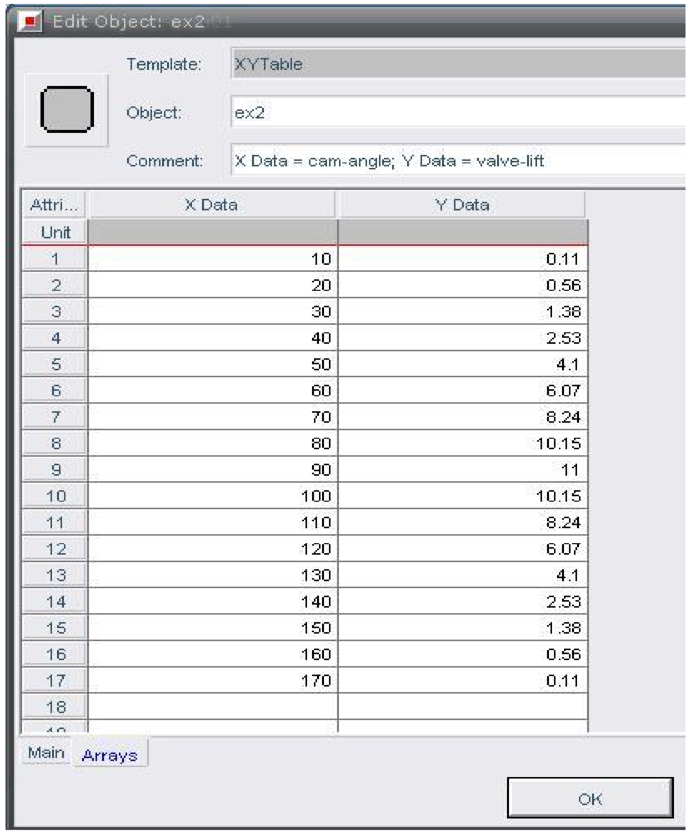

**Figure 4.10** The exhaust cam profile

**Table 3** Exhaust cam profile

| Degree | Distance (mm) |
|---|---|
| 10 | 0.14 |
| 20 | 0.55 |
| 30 | 1.26 |
| 40 | 2.29 |
| 50 | 3.67 |
| 60 | 5.35 |
| 70 | 7.16 |
| 80 | 8.65 |
| 90 | 9.28 |
| 100(80) | 8.65 |
| 110(70) | 7.16 |
| 120(60) | 5.35 |
| 130(50) | 3.67 |
| 140(40) | 2.29 |
| 150(30) | 1.26 |
| 160(20) | 0.55 |
| 170(10) | 0.14 |

### 4.5 GT-Power modeling

All of this value is needed to be entered in array for the cam parameter. The value will be entered according to the value of the length minus the base radius and also the cam angle.




| Attri... | X Data | Y Data |
|---|---|---|
| Unit | | |
| 1 | 10 | 0.11 |
| 2 | 20 | 0.56 |
| 3 | 30 | 1.38 |
| 4 | 40 | 2.53 |
| 5 | 50 | 4.1 |
| 6 | 60 | 6.07 |
| 7 | 70 | 8.24 |
| 8 | 80 | 10.15 |
| 9 | 90 | 11 |
| 10 | 100 | 10.15 |
| 11 | 110 | 8.24 |
| 12 | 120 | 6.07 |
| 13 | 130 | 4.1 |
| 14 | 140 | 2.53 |
| 15 | 150 | 1.38 |
| 16 | 160 | 0.56 |
| 17 | 170 | 0.11 |
| 18 | | |

**Figure 4.11** Parameters will be enter in this section

The parameters for others also must be entered before the model run such as the motor parameters, the push rods and etc. Because of the free piston engine does not have the crankshaft, so it has been replaced by the electric motor. In this software there is no model for electric motor. So, the model for the crankshaft will be used to act like the motor. For the push rods, the actual value will be entered.

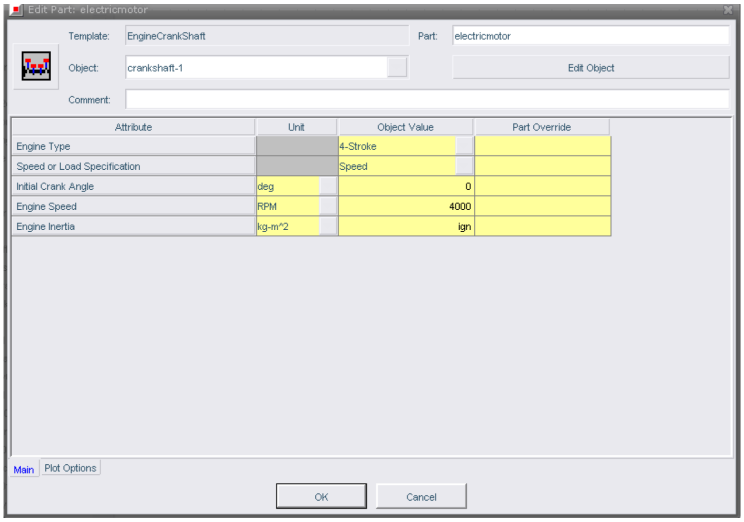



**Figure 4.12** Electric motor parameter

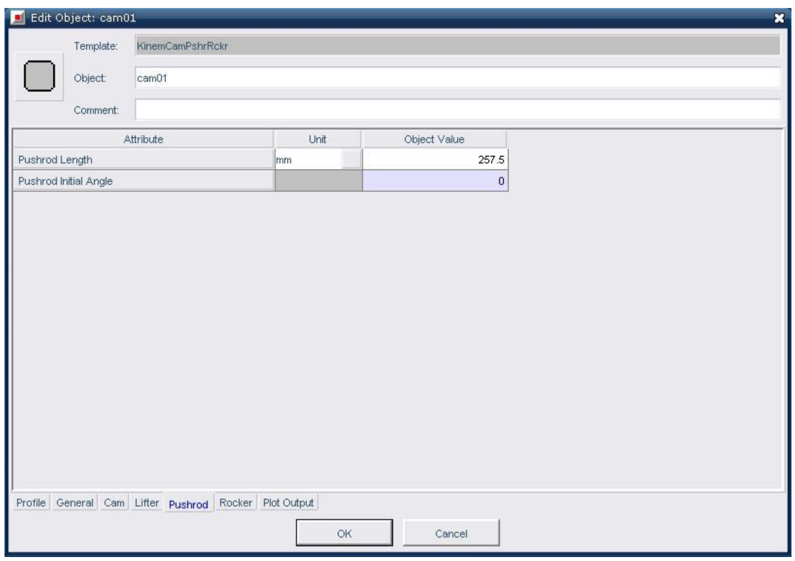



**Figure 4.13** Push rod parameter


**Figure 4.14** Parameter for rocker arm

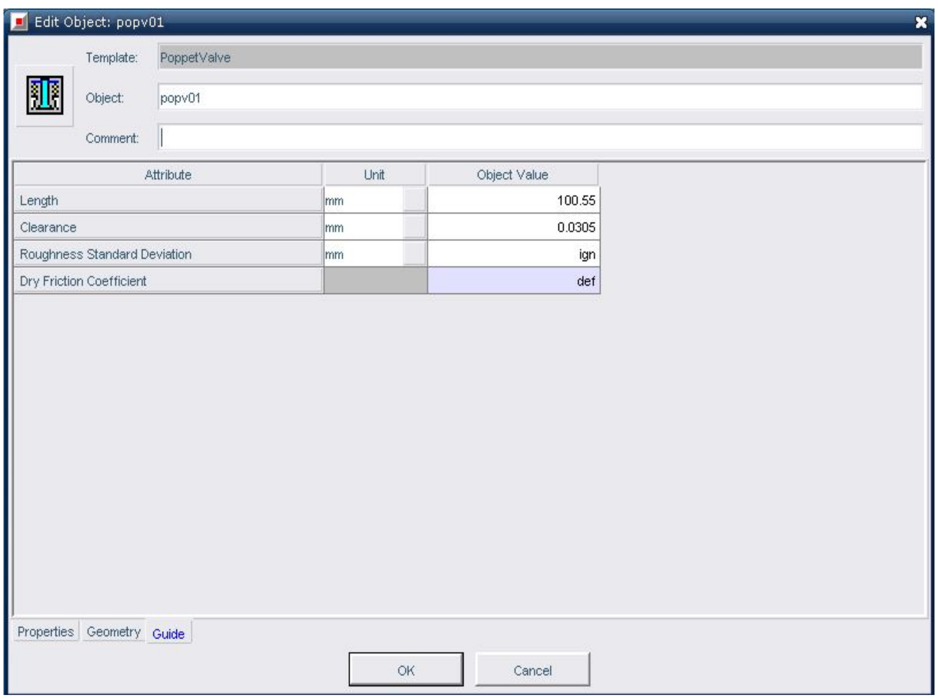



**Figure 4.15** The valve parameters.

Roughness and dry coefficient that required in this software will be ignored

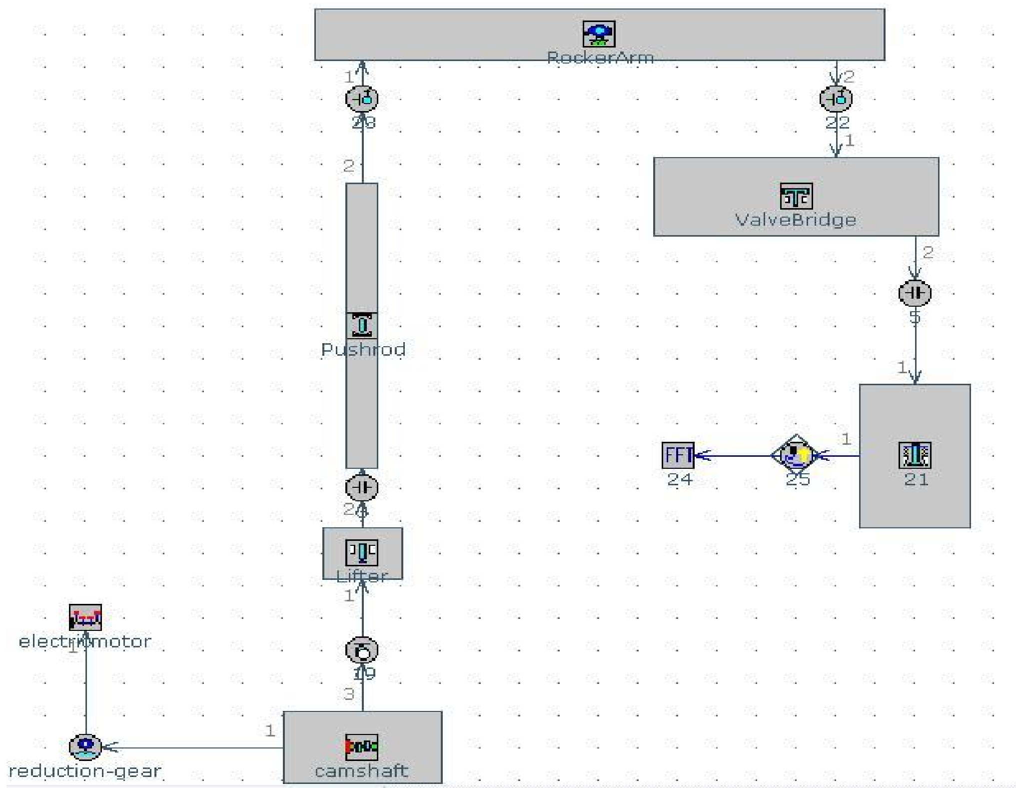



**Figure 4.16** the valve train model

Figure above is the model for the valve train. This model will be run at 4000 rpm. From this model running, the results that will be determined are the valve velocity and the maximum valve opening. All the results that needed must be tick before running the model. And after running, the result will be appeared at other file that is GT-Post document. In this document there are many graphs that needed after run it.

## 4.6 Valve timing

The timing for the valve open and close is very important part. Valve timing is made up of timing events measured lift per degree of crank rotation. Most conversations and articles always seem to focus on the valve duration and maximum valve lift. The thing that must be remember, is that those lifts, durations, and lobe separation angles come from the timing events, and not the other way around. Four functions of engine valve timing are outlined as follows.

- Small overlap of valve timing in idling and light load decreases reversing blow of exhaust gas into air intake manifold, thereby improves stability of idling revolution and starting performance of engine.
- Large overlap of valve timing in middle load flows unburned gas in exhausting stroke into air intake manifold thereby increases internal EGR effect purifying exhaust gas.
- Early closing of intake valve in high load in the ranges of low- and middle-engine revolutions increases volumetric efficiency of intake air, thereby increases torque.
- Delayed closing of intake valve in high-engine revolution increases volumetric efficiency of intake air, thereby increases torque.

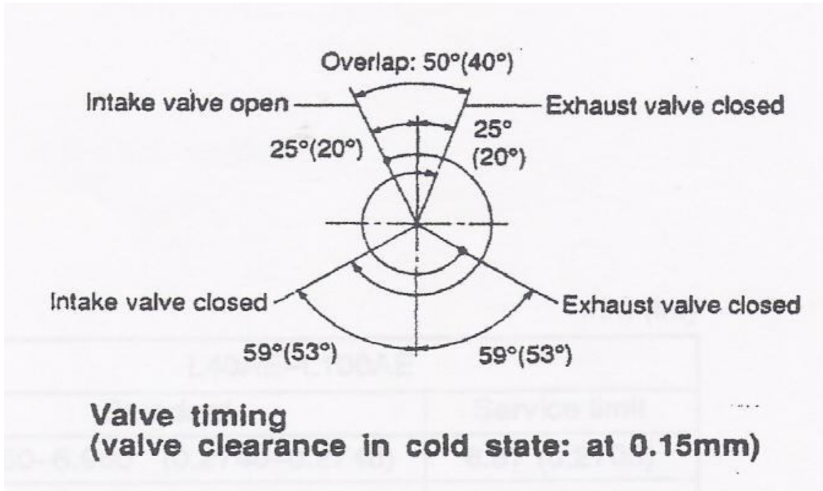
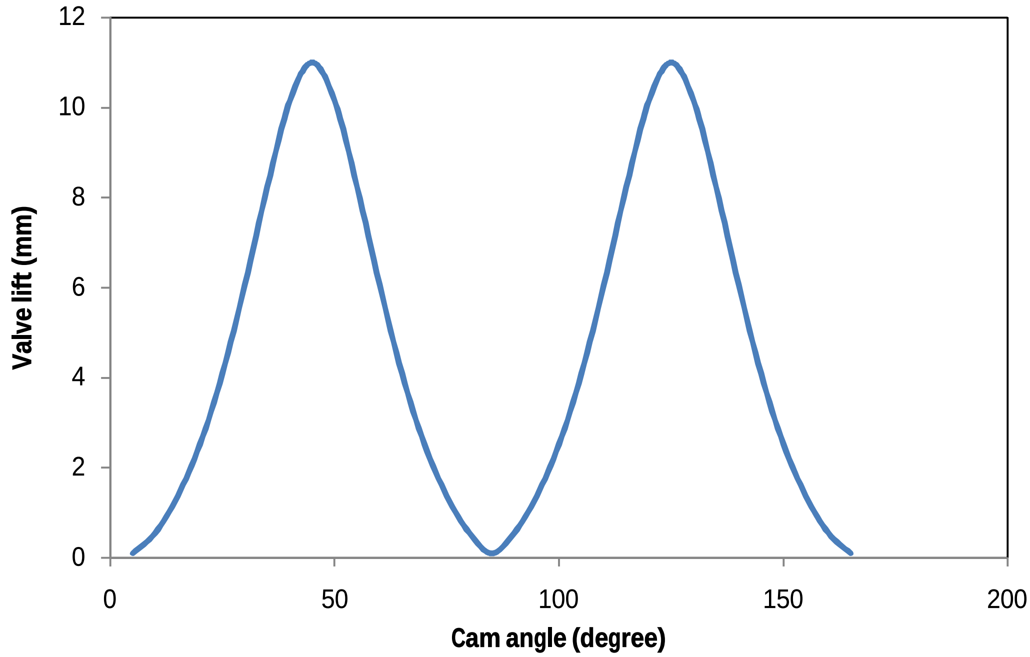



**Figure 4.17** Valve timing actual four stroke diesel engine

For the valve timing above, should know that in modeling it must follows as the actual valve timing so that the model can have the same result as the actual diesel engine. For the intake valve timing, the valve will open when at 20 degree before top dead center of the piston movement. And it will close at 53 degree after bottom dead center. For the exhaust valve, the valve open at 53 degree before bottom dead center and close at 20 degree after top dead center.

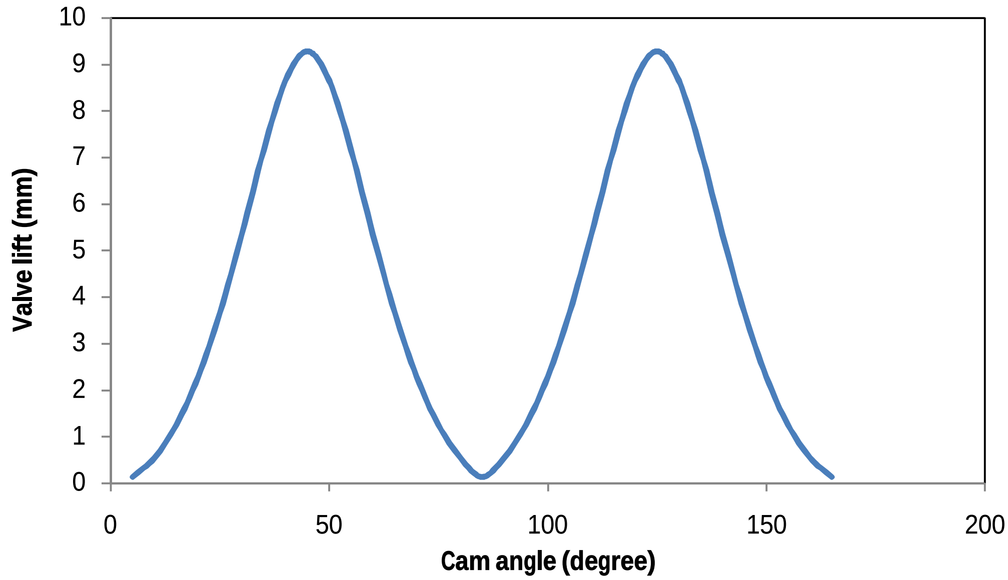
### 4.6.1 Intake valve




**Graph 1** Valve lift versus cam angle at 4000 rpm

Figure above shows that the result after running the GT-Power model. It shows that the maximum valve opening is 10.15 mm. From the graph also shows the valve start open the valve at 20 degree and start to close at 53 degree of cam angle. Intake valve opening typically occur at 10 to 25 degree before top center. Engine performance is relatively insensitive to this timing point. It should occur sufficiently before top center so that cylinder pressure does not dip early in the intake stroke. Inlet valve closing usually falls in the range of 40 to 60 degree after bottom center, to provide more time cylinder filling under conditions where cylinder pressure is below the intake manifold pressure at bottom center.

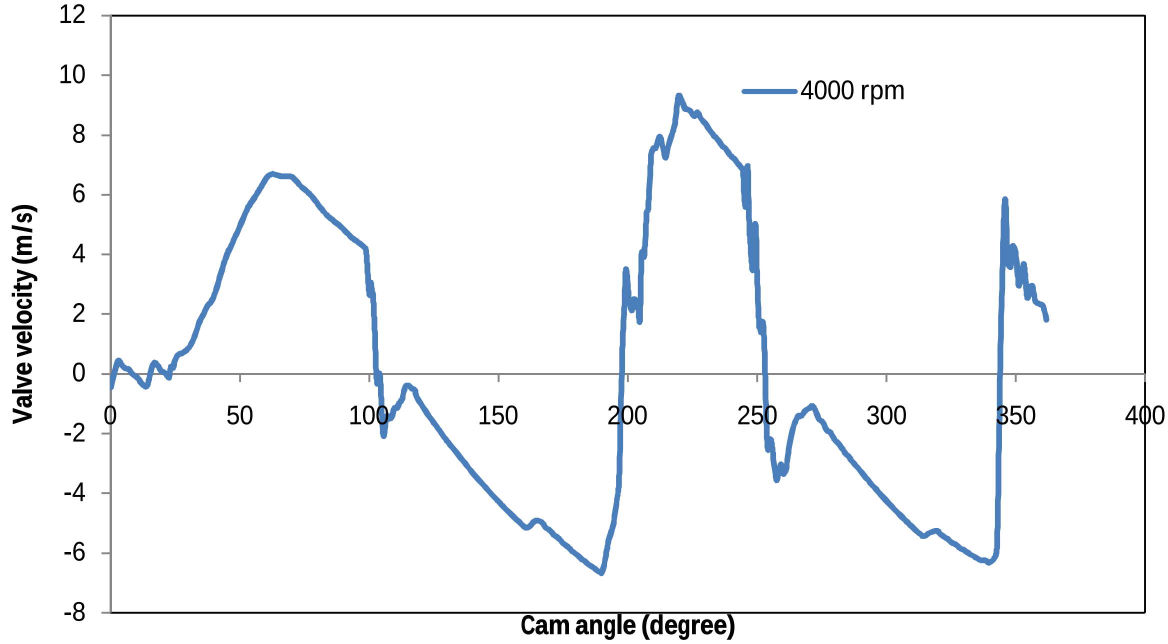
### 4.6.2 Exhaust valve




**Graph 2** Exhaust valve lift versus cam angle

The graph above is shows the maximum opening exhaust gas and the timing. The maximum opening exhaust valve is 9.28 mm. The maximum opening is at 53 degree. This graph is same like the desired characteristic from the established journal. Exhaust valve opening occur s 50 to 60 degree before bottom center, well before the end of the expansion stroke, so that blowdown can assists in expelling the exhaust gases. The goal is to reduce cylinder pressure to close to the exhaust manifold pressure as soon as possible after bottom center over the full engine speed range. The timing of exhaust valve opening affects the cycle efficiency since it determines the effective expansion ratio. Exhaust valve closing typically falls in the range 8 to 20 degree after top center. Exhaust valve closing timing should occur sufficiently far after top center so that the cylinder pressure does not rise near the end of the exhaust stroke. Late exhaust valve closing favors high power at the expense of low-speed torque and idle combustion quality.

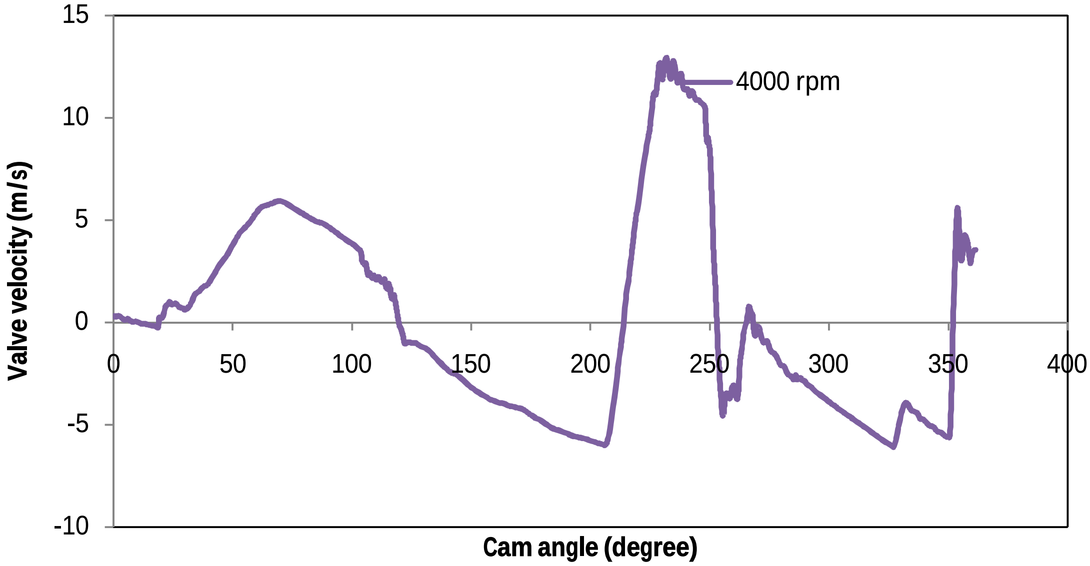
### 4.6.3 Valve velocity for intake valve




**Graph 3** Valve's velocity versus intake cam angle

From the graph above, it shows the velocity of the valve. The maximum velocity is 8.864 m/s. The velocity of a valve increases is because the pushrods are going down the slope at the camshaft. Actually, the velocity does not have positive and negative. The positive and negative sign is to show whether the valve is moving upward or downward. The positive sign is for the valve going downward and negative sign is for the valve moving upward.

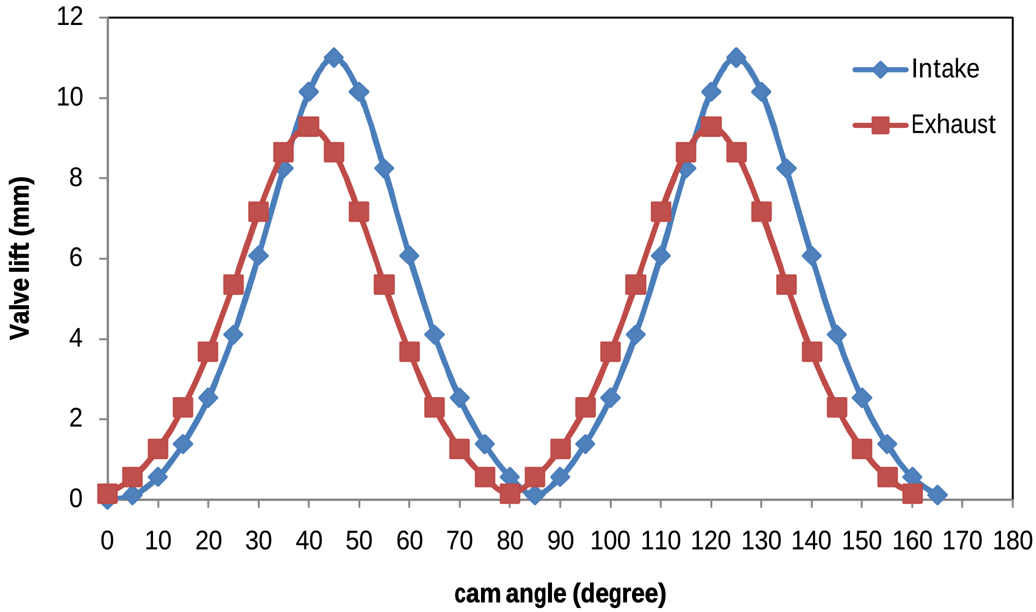
### 4.6.4 Valve velocity for exhaust valve




**Graph 4** Valve's velocity versus cam angle for exhaust

From the graph above, it shows the velocity of the valve for exhaust. The maximum velocity is 12.62 m/s. The velocity of a valve increases is because the pushrods are going down the slope at the camshaft. Actually, the velocity does not have positive and negative. The positive and negative sign is to show whether the valve is moving upward or downward. The positive sign is for the valve going downward and negative sign is for the valve moving upward. The graph is not too smooth is because error in parameter at GT-Power. Because, for some parameters the value is set to be default or be ignored.

### 4.6.5 Valve lift for intake and exhaust valve




**Graph 5** Valve lift for intake and exhaust versus cam angle

The graph above shows the valve lift for intake and exhaust versus with cam angle. The broad peaks occurring at maximum piston velocity reflect the fact that valve flow area is constant at this point. The peaks close to TC result from the exhaust valve closing and intake valve opening profiles. The peak at the end of the exhaust strokes is important since it indicates a high pressure drop across the valve at this point, which will result in higher trapped residual mass. The peak pressure decreased when the exhaust valve closing timing was retarded (or extended). The result is due to the smaller amount of residual gases remaining inside the cylinder as a result of the increased amount of exiting gases with more retarded exhaust valve closing timing.

# CHAPTER 5

# CONCLUSION

## 5.1 CONCLUSION

The objectives of this research were to construct an EMVD on single cylinder four stroke free piston diesel engine, design the model in 3D computer aided design and to analyze the motion of both valves that is intake and exhaust. One of the most critical components in the project was the motor. Unfortunately, the motor can’t be easily implemented in an actuation system on an IC engine head. Smaller motors will be required to implement the proposed EMVD on an engine cylinder head. For an example, a very small off-the-shelf dc motor capable of actuating engine intake valves and small enough to be implemented on an IC engine head was more recently obtained.

According to the journal, the obtained results confirm that this EMVD can achieve the valve timing while maintaining the characteristics of the internal combustion engine valve drives. The operation of the EMVD can be broken into three modes: initial, transition and holding. In the initial mode, the valve is moved from its resting position (the middle of the stroke) to one end of the stroke (closed or open) upon system startup.

The valve is moved from one end of the stroke to the other end during the transition mode, and is held at the arrival end during the holding mode. With optimized control and design, a much lower power consumption and peak torque have been achieved besides a faster transition time, which make it possible to fit in a smaller motor.

## 5.2 RECOMMENDATION FOR FUTURE WORK

For recommendation for the future works, to construct an EMVD apparatus in the laboratory, integrate it into a computer-controlled experimental test stand, and carry out experiments to verify the operation of this apparatus. In addition, the experimental results offer key insights on how to improve the apparatus in the future. In particular, in the near future, a smaller dc motor, a gas force simulator, and a lash-adjustment device will be incorporated into the EMVD experimental test stand. Adding the controllable motor should be considered for the future works. If the controllable motor added, the engine’s speed can be adjusts to speed that desired.

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