Reduced order modeling of combustion in SI engine

A thesis submitted in partial fulfilments of the requirements for award of the degree

Of

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Declaration of Originality and Compliance
Of
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I hereby declare that the thesis entitled '**Reduced order modeling of combustion in SI engine'** contains literature survey and original research work by the undersigned candidate, as part of Master of Engineering (Automobile Engineering under Mechanical Engineering Department) studies during academic session 2020-2022. All information in this document has been obtained, supervised and presented in accordance with academic rules and ethical conduct. I also declare that, as required by the rules and conduct, I have fully cited and referred all Material and results that are not original to this work.

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CERTIFICATE OF RECOMMENDATION

This is to certify that the thesis entitled "**Reduced order modeling of combustion in SI engine**" is a bona-fide work carried out by Ajit kumar under our supervision and guidance in partial fulfilment of the requirements for awarding the degree of Master of Engineering in Automobile Engineering under Department of Mechanical Engineering, Jadavpur University during the academic session 2020-2022.

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CERTIFICATE OF APPROVAL

The foregoing thesis entitled "**Reduced order modeling of combustion in SI engine**" is hereby approved as a creditable study of an engineering subject carried out and presented in a satisfactory manner to warrant its acceptance as a prerequisite for the degree of "Master of Automobile Engineering**"** under Department of Mechanical Engineering, Jadavpur University, Kolkata 700032, for which it has been submitted. It is understood that by this approval the undersigned do not necessarily endorse or approve any statement made, opinion expressed or conclusion drawn there in but approve the thesis only for the purpose for which it is submitted.

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ABBREVATIONS

CA – Crank angle

CN -Cetane number

CI -Compression Ignition

CO2 - Carbon dioxide

CO - Carbon monoxide

NOx - oxides of nitrogen

BSFC -Break specific fuel consumption

HRR - Heat release rate

BTE - Break thermal efficiency

Deg -Degree

RPM - Revolution per minutes

R - Gas constant

Cp - Specific heat at constant pressure

Cv - Specific heat at constant volume

IC – Internal combustion

SI – Spark ignition

TDC – Top dead center

BDC – Bottom dead center

AFR – Air fuel ratio

MFB – Mass burn fraction

BMEP – Brake mean effective pressure

T – Temperature

P – Pressure

B – Bore diameter

L – Stroke length

l - Connecting rod length

r – Compression ratio

V – Volume

R – Gas constant

m – Total mass

m_f – mass of fuel

LHV – Lower heating value

P_o- Intake pressure

Greek:-

$\boldsymbol{\theta}$= Crank angle

$\boldsymbol{\gamma}$= Ratio of specific heats or adiabatic index (Cp/Cv)

CHAPTER 1

Introduction

1.1 BACKGROUND

For many years, the majority of the world's transportation systems and small power stations have relied mostly on internal combustion (IC) engines. Since the previous several decades, the usage of IC engines has grown significantly as a result of the world's fast expansion. The development of the IC engine has created several challenges. The naturally occurring fossil fuels used by the IC engines are non-renewable. In the 1960s, the automobile started to be associaciated with several problem such as smog, air pollution, and ozone layer depletion. Currently, emissions of gases like methane and carbon dioxide are identified as the cause of global warming since they increase the greenhouse effect. Fuel shortages and price increases are results of the oil crisis. It is predicted that the stocks of fossil fuels will be totaly exhausted in the next 50 to 70 years considering the current consumption of these fuels. The government's tightening of the emission standards has had an influence on all of these factors, which have an impact on engine development. The two main objectives of modern engine development are efficiency improvements (efficient use of fuel chemical energy) and emissions reduction, which are being driven by today's regulations.

1.2 IC ENGINES

A heat engine is a device which transforms the chemical energy of a fuel into thermal energy and utilises this thermal energy to perform useful work.
Heat engines are broadly classified into :

- Internal combustion engines
- External combustion engines

The fuel and air are burned outside the engine cylinder in external combustion engines, and the products of fuel combustion do not act as the engine cylinder's working fluid. In internal combustion engines, the fuel is either burned inside the engine cylinder or the combustion products are supplied into the engine cylinder as a working fluid.

It can be said that the IC engine plays a crucial role in our transportation sector. The reason is that, even now, we rarely have any alternatives to IC engines to replace them (because to availability, compatibility, heating value, CN number, cost of fuel, etc.) [3]. The ongoing improvement of IC engines, on the other hand, makes them more adaptable and useful in our daily lives. ICE becomes the lead in the cases of both light duty vehicles and heavy duty vehicles .When we discuss about Internal combustion engines, there are mainly two kinds of internal combustion engines available i.e. Spark ignition engine and compression ignition engine. Most of the light duty IC engines are spark ignition (SI) engines fuelled with gasoline fuel which mostly runs on stoichiometric air-fuel mixture . The air fuel mixture is prepared prior to the suction stroke. The ignition is initiated by the help of spark plug. Here the ignition is initiated at the tip of the spark plug having spherical nature of shape and the generated flame propagates to the other end of the cylinder. Here the combustion can be said as propagation mode of combustion. On the other hand, the diesel engine intakes fresh air in the suction stroke and compressed in the compression stroke. At the end of the compression stroke, when the temperature of the intake air is higher than the auto ignition of diesel, the fuel is injected few degrees before the TDC and combustion is initiated. Here the combustion can be said as explosive mode of combustion. The two major cycles used in IC Engines are Otto and Diesel.

As discussed earlier, there are mainly two kinds of IC engines available i.e. Spark ignition engine and compression ignition engine. The basic CI cycle consists of four major parts namely 1) suction, 2) compression, 3) combustion and 4) exhaust. The basic difference between them is the combustion characteristics. In SI engine the combustion is initiated with a spark at the end of compression stroke and the in case of CI engines, the combustion is initiated by compressing the charge. In SI engine, the air fuel mixture is previously done as the fuel is sucked into the cylinder in the suction stroke with air but in case of CI engine, the fuel is injected at the end of compression stroke, few degrees before the piston reaches TDC. So, the mixing phenomena are very important in case of CI engine [2].

For these different types of combustion phenomena, the knocking phenomenon is different for those two types of engines. Knocking takes place in SI engine due to the ignition at multiple places inside the cylinder, when two flame fronts travel towards each other and collide with each other. On the other hand, in case of CI engine, the fuel accumulates in the ignition delay period. When actual combustion starts, the accumulated fuel burns at a time which leads to knocking. However, the modern engines are modified to and those drawbacks are reduced to the minimum [4].

1.3 SPARK IGNITION ENGINE

As already discussed, Spark ignition engine have four stages, namely suction , compression, combustion , expansion and exhaust. The fuel and air are mixed in the intake system of a conventional spark-ignition engine before being inducted via the intake valve into the cylinder, where they mix with residual gas, and then compressed. Under normal operating conditions, an electric discharge at the spark plug initiates combustion at the end of the compression stroke. A turbulent flame develops after inflammation, propagates through this essentially pre-mixed fuel, air, and burnt gas mixture until it reaches the combustion chamber walls, and then goes extinguished. [2]

A nearly homogenous mixture of fuel and air is prepared in the carburettor of spark-ignition engines. This results in the formation of a homogeneous mixture outside the engine cylinder, and the combustion process begins within the cylinder before TDC during compression stroke. [4]

One of the innovative technologies that can enhance the efficiency and emissions of an IC engine is lean burn combustion. In an IC engine, a lean burn is when there is more air being burned with the fuel. The AFR required for gasoline to burn stoichiometric is 14.6:1. Lean burning is defined as the having combustion with an AFR higher than 14.6:1. Lean burning may increase thermal efficiency and lower exhaust emissions when it is within limited range. The cyclic fluctuation in the total combustion rate, however limits it.

Combustion in SI engine can classified into two major types, normal combustion and abnormal combustion.

In normal combustion, there are three stages of combustion, 1. Ignition lag 2.Propagation of flame 3. Flame termination.

In ignition lag, flame is being prepared where there is nucleus of flame which are self-propagating grows and have chemical process which depends upon both pressure and temperature, fuel and residual gases. In propagation of flame stage, there is propagation of flame and it is of physical nature. It can be observed where first measurable rise is seen on indicator diagram. During this heat is released and rate of it depends upon turbulence intensity and lastly in third stage, this started after maximum pressure is achieved .Rate of combustion reaches on low and there is no pressure rise.

There is abnormal combustion due to several factors like combustion chamber deposits, operating conditions, composition of fuel, engine design etc. There are two types, Knock and surface ignition. The most significant abnormal combustion phenomena is knock. The unburned mixture in front of the flame, known as end gas, is compressed as the flame moves throughout the combustion chamber, rising its temperature, pressure and density. Prior to proper combustion, some of end gas combination undergo chemical reactions and exceeds self-ignition temperature and ignitions happen. This results in harsh metallic noise-producing sound in the cylinder. Surface ignition is other abnormal combustion phenomena. It is ignition of charge by hot spots inside cylinder like overheated spark plugs or valves, deposits on cylinder.

1.4 Reduced order modelling

Modeling is the process of developing mathematical model in which physical phenomena characteristics is computed with the mathematical expressions considering feasible assumptions required and used to examine the system.

When it is done on CFD (engine system is divided into many number of 3D cells and each of them is computed individually of mass, momentum, energy etc.) with the help of Ansys analysis , it requires high computational timing , more powerful computer specifications and high programming skills but if we do reduced order modeling with the help of MATLAB scripting , it requires very less computation timing and about nearby results.CFD is most preferably used to compute detailed engine analysis during combustion in SI engine. The parameters of IC engine and their performance can also evaluated by modeling engine but it is costly process to conduct experiments on different operating conditions, so numerical simulation is easy and less costly method, which designers preferred to design optimised and better efficient engine. It also helps in understanding the phenomenon of system.

There are three types of combustion models, (i). Zero dimensional models (ii). Quasi-dimensional models (iii). Multi-dimensional models

The most suitable and simplest to observe empirical variations effect in operating condition of engine for heat release rate and cylinder pressure is zero dimensional models. It does not involve flow field dimensions.

Zero D models are further classified into (i).Single zone models,(ii).Two zone models and (iii).Multi zone models.

In single zone models, engine is modelled as a thermodynamic system where fluid property considers uniform in whole system as single entity and exchanges mass and/or energy with its surroundings. The first law of thermodynamics is then applied to the system to calculate the energy released during combustion.

In two zone models, system consists of two zone, unburned and burned zone. They have distinct thermodynamic properties and cylinder wall as common surrounding. By applying 1st law of thermodynamics to the two zones, properties are calculated.

1.5 Motivation

Even though alternate fuel engine technology has shown remarkable possibilities, diesel and SI engine still the most often used IC engines in the modern world. Over their entire operating range, diesel engines typically run with an overall lean equivalency ratio. As a result, it has a higher thermal efficiency but emits more smoke and particulate matter as exhaust. It is possible to reduce exhaust emissions, but this needs specialised equipment that is expensive. In contrast, a gasoline engine functions with an equivalency ratio of 0.8 to 1.2. Due to the limited working range, it performs poorly under part load conditions. Utilizing a common technology, 3-way catalytic converters, exhaust emissions may be reduced to an extremely low level.

The goal of recent technical advancements is to increase efficiency of engine and lower exhaust emissions to acceptable levels. Lean AFR gasoline engines may have advantages similar to those of diesel engines (better efficiency) and stoichiometric gasoline engines (lower exhaust emissions).

As was briefly said above, students need to have a thorough understanding of the fundamentals of an IC engine in order to stay up with recent technical advancements in IC engines. In this situation, numerical simulation is simple and affordable method to groom the research work. Despite the fact that many software is accessible, it cannot reveal what is really occurring in the background.

1.6 Literature review

Salah et al. [1] have studied thermodynamic model for simulating a single curved-cylinder crank-rocker engine, which operates on gasoline fuel. The single and two-zone heat release models serve as the foundation for the thermodynamic model. As a function of engine speed and combustion efficiency at various operating points, friction losses were included to the overall model. To estimate engine performance and efficiency on a motorbike engine, a model was created in MATLAB. Plotting several thermodynamic parameters and engine performance in relation to crank angle are done using the estimated data. At various crank angle settings, the engines volume, indicated cylinder pressure, heat release rate, pressure-volume diagram, and engine braking torque have been evaluated. The results from the simulation model were plotted and compared with existing data. It was found that when both utilized the same injection timing for the same engine capacity, the performance of the crank rocker engine was better than the slider-crank engine.

Melih et al. [5] states that combustion modelling is crucial for properly predicting in-cylinder pressure development and engine performance in an engine simulation. The Wiebe function, which represents mass fraction burned (MFB) as a function of crank angle position, is frequently used to predict the combustion process. This work presents a predictive zero dimensional (Zero-D) single zone engine simulation of a SI engine supplied by methane and a methane-hydrogen blend. In this work, the Wiebe function's single and double form were used to simulate and predict the combustion process. For this aim, the parameters of the single and double-Wiebe functions were determined by least squares fitting to the MFB curves generated from experimental pressure data. The Zero-D single zone engine model created for the methane and methane-hydrogen blend fuelled SI engine was then modified to use these Wiebe functions in order to provide in-cylinder pressure development and gross indicated mean effective pressure (GIMEP) for the prediction of engine performance. The study showed that the double-Wiebe Function model fits the data more accurately than the single-Wiebe Function model. Additionally, the GIMEP prediction for the methane-hydrogen blend-fuelled SI engine modelling rather than the methane-fuelled modelling has been significantly improved by the fitted double-Wiebe function.

Ashish et al.[6] evaluated the performance of spark ignition engines by considering a single-zone zero-dimensional model for any hydrocarbon fuel based on Wiebe heat release function ,has been developed in Simulink. For simulating the engine cycle, Annand's model for convective heat losses is used. Experimental results from published research are used to validate the Simulink results. Similar to the experimental result, the peak pressure during combustion decreases as intake pressure decreases. The brake thermal efficiency decreases from 25% to 17% when speed increases from 1000 rpm to 4000 rpm, yet the indicated efficiency remains to be approximately constant about 30%.

M.A. ceviz et al. [7] evaluated the expression for ratio of specific heat dependent on the varying temperature and air fuel mixture for unburnt and burnt mixture. Specific heat ratio gets decreased when temperature is raised whereas increases on the rising of air fuel mixture. The function of Gawoski , Brunt and Egnell for specific heat ratios are plotted and compared with new function. The specific heat ratio values of burnt mixes are higher than those of unburned mixtures. For burned and unburned mixes, the Gatowski linear function provides the highest specific heat ratio, but the Egnell work's function provides a good match for a burned mixture. It seems that the function of Brunt is close to the outcomes of the equilibrium combustion model for a burnt mixture for a limited range of high temperatures. The variation of the mass fraction burnt was used to calculate the global specific heat ratio. The results suggest that implementing of a function dependent on temperature and equivalence ratio significantly lowers the inaccuracy resulting from a temperature-only dependent specific heat ratio under lean engine operation.

E.Abu nada et al.[8] modelled the zero dimension single zone SI engine heat model considering the effect of temperature change on specific heat ratio and its effect on pressure and temperature inside the engine. It depicts the change in cylinder pressure vs volume for a SI engine running at piston speeds of 2000 and 5000 rpm at a 15:1 air/fuel ratio. It is evident that greater engine speeds result in increased cylinder pressure, at engine speeds of 2000 and 5000 rpm, respectively, a maximum pressure of 48 and 53 bars is observed. It is significant to observe the influence of temperature-dependent specific heat has on the temperature of the gas exit. Higher engine speeds and lower air-fuel ratios produce higher maximum temperature values. The influence of temperature-dependent specific heat is highly significant in determining the maximum gas temperatures. Higher engine speeds and lower air-fuel ratios provide higher BMEP values. BMEP was not significantly impacted by temperature-dependent specific heat, in comparison to temperatures. It was observed that there is no significant variance in efficiency between constant and variable specific heat models at low engine speeds. However, the selection of constant specific heat versus variable specific heat models matters when the engine is operating at high speeds; specifically, the variable specific heat model is more important at higher engine speeds.

Stepenenko et al.[9] states about the single zone, double zone and multi zone model of thermodynamics analysis of combustion inside the SI engine cylinder. The accuracy of estimate for harmful emissions is decreased by single-zone calculations since they require averaging the temperature inside the cylinder throughout the combustion time. By employing a two-zone as well as multi-zone model that enables for more precise engine toxicity prediction, this drawback can be overcome. In single zone, It has been concluded that the engine performance estimation is not much affected by the cylinder's average temperature and allows to predict the release rate of heat with higher accuracy. The two-zone model is most frequently used because it enables the good accuracy of engine operational parameters for a numerical description of the combustion process and analysis of the effects of different characteristics during the combustion process and emission.

M.reyes et al.[7] studied on the effects of utilising natural gas and hydrogen in various fractions on a spark ignition engine's combustion. In order to determine the flame radius value respect to the burned mass , surfaces for heat dissipation through wall and head, the two-zone model takes into account of flame front having spherical nature starting from centre of the spark plug and provides the flame front intersection to the piston, cylinder head, and cylinder wall.

Musab et al.[8] study analyses the suitability for hydrous ethanol as a clean, affordable, and environmentally friendly alternative fuel for spark ignition engines and examines the chemical and physical characteristics of hydrous ethanol and gasoline fuels in comparison. Spark-ignition (SI) engines' performance is impacted by the large variations in the characteristics of gasoline and hydrous ethanol fuels, which are enough to cause a major shift during operations of engine during combustion phase. The impacts of hydrous ethanol and its gasoline fuel mixes on the cycle-to-cycle fluctuations, emission characteristics and engine performance parameters of SI engines have also been highlighted. The hydrous ethanol uses as fuel significantly improved combustion properties and improved engine performance, according to nearly all available engine testing data. Additionally, there was a considerable reduction in emissions of carbon monoxide and nitrogen oxides. It is also important to note that using hydrous ethanol resulted in lower emissions of carbon dioxide and unburned hydrocarbons but uncontrolled emissions of formaldehyde and acetaldehyde significantly soared.

Siti et al.[9] analyses the engine performance of LPG in SI engine with the help of mathematical model using MATLAB. It is based on zero D model with the single zone model approach. The simulation findings demonstrated the differences in engine performance between LPG and gasoline in terms of torque, brake power, and specific fuel consumption. Gasoline recorded a higher peak cylinder pressure than LPG. Compared to gasoline, LPG has a greater heat transfer rate. When utilising LPG fuels in gasoline engines, LPG will burn 10 degrees earlier than gasoline at the same crank angle. This is a result of LPG's higher flame speed. Compared to LPG, gasoline releases more energy as a result of full combustion. Since gasoline releases more energy during burning, its torque will be greater than that of LPG.

Sumit et al. [12]. investigates the effects of gasoline with ethanol added on the emissions characteristics of single-cylinder SI engine at various compression ratios. In the lab, the three mixes E5, E10, and E15 were created. Lever is used to change the compression ratio of SI engine. The lower heating value of ethanol is approximately 25% less than the values of gasoline. There were no significant issues, such knocking, during engine running. A smoke metre and exhaust gas analyser were used to measure the emission testing. The findings of the experiments showed that mixing improved brake thermal efficiency which is more than by using fuel, like gasoline. According to the results of the emission testing, CO emissions somewhat dropped, HC emissions raised moderately, and CO_2 and NO_x emissions significantly increased. The results of the experiment show that using ethanol may effectively assist in compensating for the usage of fossil fuels and can also enhance the emission characteristics.

Hakan et al.[13] develops an empirical formulae for combustion duration. For this objective, the Spark ignition engine cycle model has been used to estimate the impacts of changes in compression ratio, engine speed, fuel/air equivalency ratio, and spark advance on combustion time. The fuel used was C3H8. Turbulent combustion model was used to determine burn durations under various engine operating circumstances. A 2nd degree polynomial equation were used to represent how the theoretical findings' combustion duration variations with respect to each operational parameter. These functions have been used to create empirical formula for the burn duration. Combustion durations observed from experimental research and a combustion model were in good agreement.

Elliot et al[15] states about the advanced multi-mode combustion that uses spark ignition, homogenous charge compression ignition, spark-assisted CI and homo-charge compression ignition, operating methods is captured by a reduced-order model in this study. Such a model, which was previously unobtainable, was needed to improve engine system models' already-existing capabilities and to make large-scale parametric investigations of advanced combustion process possible. The model is based on two distinct thermodynamic zones that are separated by an infinitely thin flame interface. Using a new 0 D formulation of coherent combustion model, where flame propagation is observed and end gas auto ignition is simulated on MATLAB using a method combining a semi-empirical burn rate model and chemical kinetics .The ability of the results to reproduce heat release characteristics for flame propagation and auto-ignition during spark-assisted CI combustion was one of the results that showed overall strong trend-wise agreement with the experimental data. A sizable parametric analysis was used to evaluate the calibrated model, and the operating zones for homogenous charge CI and spark-assisted CI under naturally aspirated conditions that were anticipated were reflective of those observed during engine testing. Comparing actual advanced combustion techniques to idealised engine models revealed that efficiency gains of up to 30% over traditional spark-ignition operation are attainable. According to the study, inefficient combustion and pumping effort are the main causes of losses of efficiency for advanced combustion systems considered.

Mohand et al [16] studies to analyse the impact of the selection of the heat transfer correlation and burnt zone heat transfer area calculation methods and propose an optimum option for a more effective two-zone thermodynamic model. A simulation is created for this purpose. For comparison and validation, experimental measurements are taken. First, the impact of correlation preference was investigated. They examined and compared relationships which are the taken from different literatures. For correlation selection, experimental pressure findings are validated by a literature review of several previous research based on observed heat transfer rates for various SI engines. Hohenberg's correlation is determined to be the best option. It produces the more precise outcomes. It is simple to use and computation time is reduced to a minimum.

T. morel et al. [17] states that the design of engines must take engine heat transfer into consideration. Due to energy losses, it impairs engine performance. It also influences in-cylinder temperature of gas during combustion, which in turn impacts emissions (particularly NOx), knock, and other characteristics. Additionally, it has a direct impact on how thermally loaded engine structural components are. This is a significant challenge for component-oriented design analyses of IC engines in addition to adding uncertainty and inaccuracy to estimations for performance and emissions. The model represents propagation of flame as spherically increasing zone coming from the spark point during combustion.

The computation of combustion characteristics heavily relies on the specific heat ratio employed in heat release calculations. In this investigation, a spark ignition engine employing natural gas and gasoline fuels is used to examine the impacts of an assumed γ on heat release analysis of engine pressure data.

The experiments were conducted with the engine speed set at N = 3300 rpm and the spark timing set to the timing for the highest braking torque. The 1st law of thermodynamics is used to calculate heat release rate during a cycle and to determine the combustion parameters.

The results demonstrated the great sensitivity of the combustion parameters to the fluctuation and first derivative of the specific heat ratio. The results also demonstrated that for natural gas operation, the specific heat ratio's first derivative has a stronger effect on the combustion characteristics than it does for gasoline operation. Additionally, while determining the combustion parameters, the specific heat ratio's first derivative should not be disregarded.

Yelina et al.[18] has taken pressure and volume data, 1D single zone and two-zone studies have been used to determine the mass fraction burnt in an engine using ethanol/gasoline mixed fuels. The blend of air and ethanol fuel were stoichiometric, hence experiments was done with a CFR engine while maintaining IMEP at 330 kPa. Four ethanol-gasoline fuel mixes were used in this investigation as a baseline: E20, E40, E60, and E84. The study showed that for the five different fuels tested, the three models consistently provide profiles of mass fraction burnt that are identical. Additionally, the two-zone model suggested a 3 percent increase in combustion efficiency over single zone model and a 17 percent increase over the apparent release of heat approach when using gasoline with gamma dependent on temperature.

Shehal et al. [19] states that ammonia can be utilised as composite fuel for powering existing IC engines when combined with hydrocarbon fuels. The feasibility of creating gasoline-ammonia fuel mixes and the ethanol as an emulsifier to increase the solubility for ammonia in gasoline are examined. . For produced fuel, engine dynamometer's test were performed. A volume-basis of 17.35 percent of ammonia in liquid phase is possible with gasoline containing 30% ethanol. According to engine dynamometer studies, ammonia-rich fuels provide more torque and power, mostly on higher engine speeds. The optimum blend of gasoline for an existing SI engine is observed to have 12.90% ammonia and 20% ethanol by volume.

German et al. [20] presented an experimental investigation of the impacts with methane and hydrogen proportions, compression ratio and equivalence ratio on knock occurrence crank angle , combustion duration (CD), and compression polytropic coefficient (n). The experiments were carried out in a CFR engine, where the influence of input pressure on KOCA was also investigated. The statistical research found that the intake pressure has no effect on KOCA but that the examined engine operating conditions have a significant impact on the combustion characteristics. Additionally, it was suggested to use validated empirical correlation to calculate the Critical Compression Ratio, KOCA, and CD as functions of engine operating parameters. It was shown that efficiency and form factor of the Wiebe's function, a = 2.8 and m = 2.4, can be used for actual burnt mass fraction curves. These values for blends of hydrogen and methane differ from the commonly used values for conventional fuel, a = 5.0 and m = 2.0.

1.7 OBJECTIVE:

This research thesis has an objective to develop a mathematical model of SI engine using MATLAB scripting to study the effects of different parameters like burning duration, change in specific heat ratio, compression ratio etc. on the combustion of four stroke SI engine and to study to study the behaviour of in-cylinder parameters like pressure and temperature changes.

As discussed earlier, it will be best for designers to develop and compute the characteristics by investing lesser duration of time as compare to other simulations like CFD by simple reduced order modeling without detailed analysis (chemical analysis, turbulence analysis, squish effect etc.), and achieve a nearly accurate result within a very short time.

CHAPTER 2

PROBLEM FORMULATION

2.1 THEORY

SI engines comprises of four processes. Ideally those are isentropic compression followed by constant volume heat addition after that isentropic expansion process and lastly constant volume heat rejection. In it, air fuel mixture is mixed outside the cylinder and charge is prepared which is inducted through the suction valve. In the compression process, the temperature of air inside the cylinder is increased and at the end of compression stroke, spark is ignited which results in combustion inside the cylinder. Next expansion stroke takes place where piston goes back to the bottom dead center.Ideally, It is considered that heat is added at constant pressure and there is rapid vertical increase of pressure and temperature of air but in actual case, spark is ignited before the top dead canter and temperature rise takes time due to the chemical delay. Ignition delay has two parts, naming physical delay and chemical delay. In physical delay, it consists of atomisation and vaporisation of fuel but in SI engine, air-fuel mixture is already prepared outside the cylinder so it is considerable. Chemical delays occur because it takes time for the chemical process between fuel and air to initiate combustion. When sparking starts, the spark plug makes contact with the closest fuel particle, and chemical reactions start. Its temperature rises as a result of chemical reaction, and subsequent fuel particles will also rise in temperature as a result of heat transfer. The first flame front will occur when the temperature of the next fuel particle reaches the firing temperature of the fuel at the conclusion of the chemical reaction. The transposition rate and reaction rate is responsible for flame propagation. The process is a pure chemical reaction where the flame reaches the unburned charge. The mass transfer rate, or transposition rate, is a result of the physical movement of flame front caused by the pressure difference between the burning charge and the unburned charge. In the compression stroke, the temperature takes place due to the combined effect of compression and the fuel combustion but in the expansion stroke, the phenomenon is somewhat different. Here the temperature decreases due to expansion and temperature rises due to fuel combustion. So in the expansion part, the change of temperature is due to the combine effect of those two. Now the pressure and temperature is directly related to the instantaneous volume of the cylinder and the instantaneous volume of the cylinder depends on the piston position at a particular instant, i.e. we need instantaneous position of the piston at every crank angle to find out the instantaneous volume.

The pressure change follows the some trend as temperature. Here the temperature rise due to the fuel combustion can be determined by energy balance inside the cylinder.

2.2 Formulation

All the engine parameters of an IC engine can be obtained from two main data, one is pressure and another is temperature. All the output parameters like BP, IP, BMEP, IMEP etc. can be obtained from those to data. Here we need to mainly formulate four stages, 1) compression, 2) combustion and 3) expansion. In a complete cycle of -360 deg to 360 deg, compression takes place between -180 to 0 deg crank angle and expansion stroke takes place from 0 deg crank angle to 180 deg crank angle.

Here we have assumed system as zero dimension single zone. In it, pressure and temperature are same throughout the system. In single zone models, engine is modelled as a thermodynamic system where fluid property considers uniform in whole system as single entity and exchanges mass and/or energy with its surroundings. The first law of thermodynamics is then applied to the system to calculate the energy released during combustion.

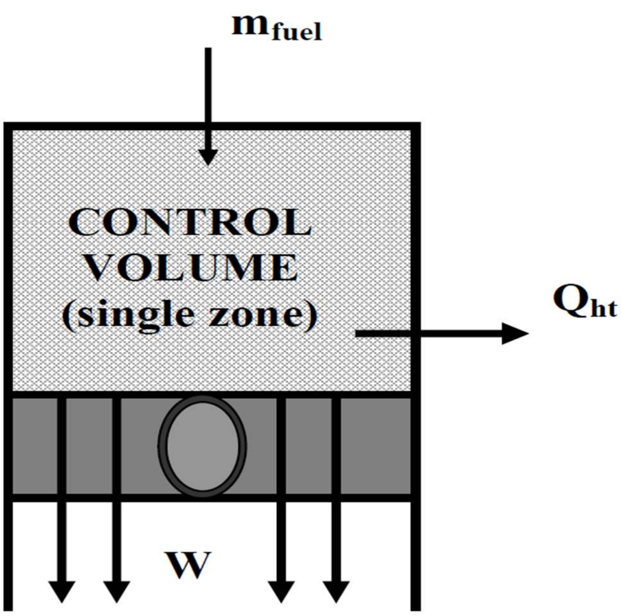

Fig. (1) Representation of single zone model

A single zone model has been developed to calculate various thermodynamic parameters using the following assumptions:

- The cylinder charge (air-fuel mixture) is assumed homogeneous.
- The engine uses gas carburettor to supply fuel into the engine cylinder.
- The reactant (air-fuel mixture) and combustion product mixture are assumed to be ideal gas.
- The specific heat of the gaseous species varies with the local temperature.
- The model has been developed for spark ignition engine without turbo or super-charger.
- There is no effect of combustion chamber design.

- Initial pressure at intake stroke and final pressure at exhaust stroke is assumed at atmospheric.

- Negligible effect of Exhaust system

Chemical reaction

Isooctane is the fuel and ambient air is the oxidizer used in this modelling. To model the burning process, chemical reaction is provided as:

$C_8H_{18} + 12.5(O_2+3.71\ N_2) \rightarrow 8\ CO_2 + 9\ H_2O + 46.428\ N_2$

$(A/F)_{Stoi}=15.05$

Combustion duration

The amount of mass burned in total and the amount of heat released are not dependent on how long the combustion process lasts. In contrast, if the same amount of heat is released over a shorter period of time, heat release rate will be higher and the pressure peak will be higher than it would be over a longer period of time.[11]

$$\Delta\theta = f_1(r) * f_2(N) * f_3(\varphi) * f_4(\theta s) * \Delta b1$$

$$f1(r) = 3.2989 - 3.3612 * (r/r1) + 1.08(r/r1)^2$$

$$f2(N) = 0.1222 + 0.9717 * (N/N1) + 1.08(N/N1)^2$$

$$f3(\varphi) = 4.3111 - 5.6383 * (\varphi/\varphi 1) + 2.3040(\varphi/\varphi 1)^2$$

$$f4(\theta s) = 1.0685 - 0.2545 * (\theta s / \theta s1) - 0.2902(\theta s / \theta s1)^2$$

where r1, N1, $\theta s1$ and φ_1 are reference value as per the engine specification.

Heat release during combustion

Weibes function gives empirical expression from which mass of fuel burnt with per degree of CA is calculated.

$Xb(\theta) = 1\text{-exp} \{-a(\frac{\theta-\theta_b}{\Delta\theta})^{m+1}\}$

Where 'a' is combustion completeness factor and tells about the efficiency of combustion. It depends upon the intensity of charge motion, engine design and falls in the range of $2 \le a \le 6$.

$\Delta\theta$ is the combustion duration and m is the form factor that affects the shape of mass burned profile. These parameters determine the shape of wiebe function and hence its accuracy to predict the actual heat release during combustion.

Differential of fuel mass burnt fraction wrt CA is written as,

$$\frac{dx_b}{d\theta} = \frac{a(m+1)}{\Delta\theta} * \frac{(\theta-\theta s)m}{(\Delta\theta)m} * \exp(-a * \frac{(\theta-\theta s)}{(\Delta\theta)} m + 1)$$

Qch is Heat released during combustion, then

$$\frac{dQc}{d\theta} = 0 \qquad ,\text{for } \theta ivc < \theta \le \theta o \text{ and } \theta > \theta eoc$$

$$\frac{dQch}{d\theta} = Q_{max} \frac{dx_b}{d\theta} \text{ , for } \theta s < \theta \le \theta eoc$$

Convective heat transfer from wall

Heat transfer influences performance, efficiency and emissions of engine. More heat transfer from the wall will result in lower gas temperature and pressures for a given quantity of fuel within the cylinder. This will also result in less work is being supplied to the piston every cycle. The amount of engine heat transfer affect the specific power and efficiency.[14]

This sub model predicts the heat transfer from the wall through convective mode of transfer.

$\dot{Q}ht = h_C A(T-T_W)$

Here hc is a convective heat transfer coefficient, which is calculated by using Annand's correlation for convective heat transfer which is as follows[11].

$\frac{\text{hc B}}{\text{k}} = a(\frac{\rho \text{SpB}}{\mu})^\wedge b$

$\text{Sp} = \frac{2aN}{60}$, mean piston speed, $\mu = \frac{\mu \text{air}}{1+0.027^{\varphi}} = \frac{3.3*10^{-7}*T^\wedge 0.7}{1+0.027^{\varphi}}$

$k = \frac{9\gamma-5}{4} \mu Cv$,   b=0.7

Here a is intensity of charge motion coefficient and design of engine and in between the range $0.35 \leq a \leq 0.8$.

Volume change due to piston displacement

Piston displacement,

$x(\theta)= (l+a)-(a\cos\theta +((l^2-a\sin\theta)a\sin(\theta)^\wedge 0.5))$ (1)

Volume change due to movement,

$V(\theta) = \frac{Vs}{cr-1} + \frac{Vs}{2}((R+1) - cos\theta - (R^2 - \sin^2\theta)^{0.5})$

Differential of volume change wrt CA,

$\frac{dV}{d\theta} = Vs/2*\sin\theta+((\sin\theta\cos\theta)/((R^\wedge 2-\sin\theta * \sin\theta)^\wedge(0.5)))$

Compression, combustion and expansion model

by ideal gas equation,

PV=mRT

By taking log on both sides,

lnP + lnV = ln(mR)+lnT

$$\frac{1}{P}\frac{dP}{d\theta} + \frac{1}{V}\frac{dV}{d\theta} = \frac{1}{T}\frac{dT}{d\theta}$$

From 1ST law of thermodynamics,

dq=du+dw

du=dq-dw

$$mCv\frac{dT}{d\theta} = \frac{dQ}{d\theta} - P\frac{dV}{d\theta}$$

By dividing LHS by mRT and RHS by PV,

$$\frac{Cv}{RT}\frac{dT}{d\theta} = \frac{1}{PV}\frac{dQ}{d\theta} - \frac{1}{V}\frac{dV}{d\theta}$$

R=Cp-Cv and Cv/R = $1/(\gamma-1)$

$$\frac{1}{T}\frac{dT}{d\theta} = (\gamma\text{-}1)[\frac{1}{PV}\frac{dQ}{d\theta} - \frac{1}{V}\frac{dV}{d\theta}]$$

$$\because [\frac{dQ}{d\theta} = \frac{dQch}{d\theta} - \frac{dQht}{d\theta}]$$

$$\frac{1}{P}\frac{dP}{d\theta} + \frac{1}{V}\frac{dV}{d\theta} = (\gamma\text{-}1)[\frac{1}{PV}\frac{dQ}{d\theta} - \frac{1}{V}\frac{dV}{d\theta}]$$

$$\frac{dP}{d\theta} = -\gamma\frac{P}{V}\frac{dT}{d\theta} + (\gamma - 1)\left[\frac{mf*LHV}{V}\frac{dxb}{d\theta} - \frac{hcA(T-Tw)}{V}\frac{60}{2\pi}\right] \quad (2)$$

From solving equation (2), we can get Pressure variation wrt CA.

To get Temperature variation inside the combustion chamber, it can be calculated by solving this equation (3),

T=PV/mR (3)

In this mathematical modeling, firstly engine specification is taken from the research paper as no experiment work is involved. This is zero dimensional single zone modeling so properties like temperature, pressure, density etc. are uniform throughout inside the cylinder. Here, we need to formulate an equation by taking account of pressure and temperature changes. During compression, combustion and expansion, system is assumed as the close system and fluid inside the system assumed as the nature of ideal gas. 1^{st} law of thermodynamics is applied considering the effect from chemical energy released from fuel and heat transfer through the wall. The weibes function gives the relation between mass of fuel burn related to the crank angle change and combustion duration is calculated through the given empirical relation. MATLAB is used to solve the equation 1 to evaluted the pressure inside the cylinder.Firstly,it is attempted to solve the equation through the runge kutta order 4 method but result is not good .After that, change in crank angle is considered as 0.1 and tried to solve it.

There is second approach which we can also follow is to considering the compression process as isentropic and take the value of pressure and temperature variation with the expression $TV^{\gamma-1}$=constant and PV^{γ}=constant where γ represents adiabatic index. Now here we can see that temperature and pressure depends on instantaneous volume and the volume changes with every single crank angle inside the cylinder. So, to calculate the pressure and temperature, we need to calculate the instantaneous volume at every single degree of crank angle by solving equation(1).

Now we can calculate the volume at every single crank angle and the pressure and temperature can be calculated. The calculation has been carried out using a MATLAB code. In the combustion zone, weibes equation is applied to calculate the heat release rate by the fuel so temperature rise due to fuel can be calculated by applying the Q=mCvdT . Spark is ignited nearby the end of compression stroke so we need to account the temperature change due to compression plus temperature change due to heat released by the fuel. So, after adding both temperature rise, we get resultant temperature at respective crank angle. Pressure is calculated by considering the isentropic pressure-temperature relation till the end of compression. During the expansion stroke, it is checked at which duration burning of fuel is expected to end and till that crank angle, change in temperature is applied by considering the effect of expansion where temperature is decreased and by burning fuel where temperature is increased, and pressure is calculated by considering isentropic process and after the end of fuel burning , pressure and temperature is accounted by only considering the effect of expansion.

CHAPTER 3

PROBLEM VALIDATION

3.1 Validation

(Fuelled with gasoline)

Engine specification:

Bore, B (in m) -	0.0794
Connecting rod length, l(in m) -	0.2334
Stroke length, L (in m) -	0.1112
Compression ratio (r) -	7.4
Fuel used –	gasoline, C_8H_{18}
Start of fuel injection -	-25 CA BTDC
Rated speed -	4000 RPM
Equivalence ratio -	1
LHV of fuel -	44000kJ/Kg

Here, experimental data is obtained from Ashish et al.[6] and plotted it with the MATLAB result predicted during this thesis work. The result obtained for Pressure vs Crank angle is varied for various inlet pressure of 0.91 bar, 0.81 bar, 0.71 bar and 0.61 bar individually as shown in figure, all others conditions and specifications of engine is same. After predicting Pressure vs crank angle curve, mass burn fraction vs crank angle, heat release rate vs crank angle and volume vs crank angle curve is plotted with the literature result. The graph is also plotted for combustion duration vs factors on it, it is dependent like equivalence ratio, compression ratio, engine speed and validated with the research paper.

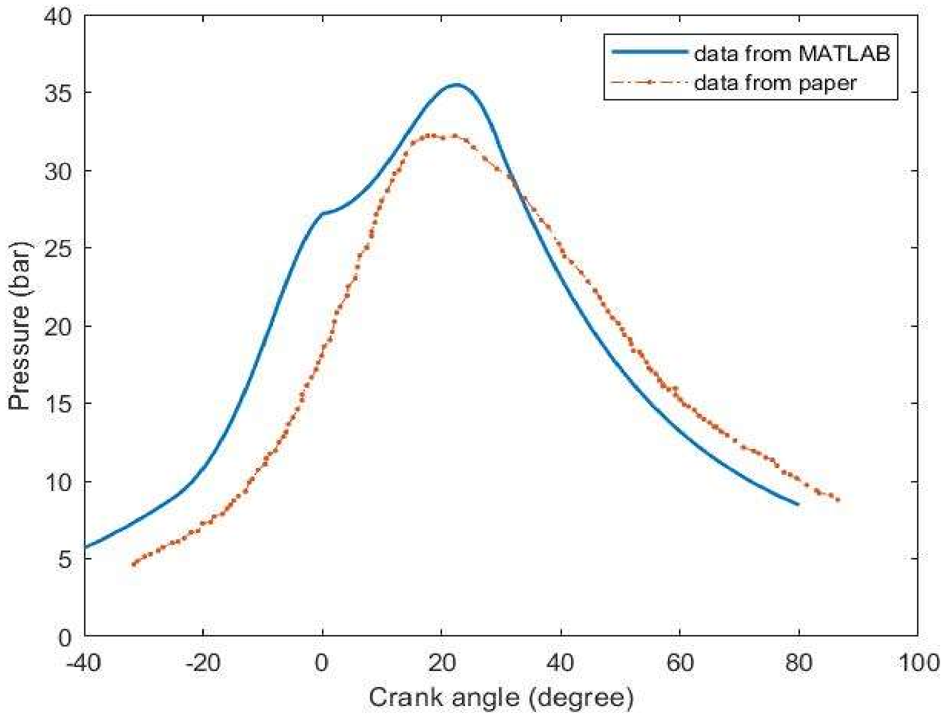

Fig. (2) Pressure vs Crank angle (degree) for Pin=0.91 bar

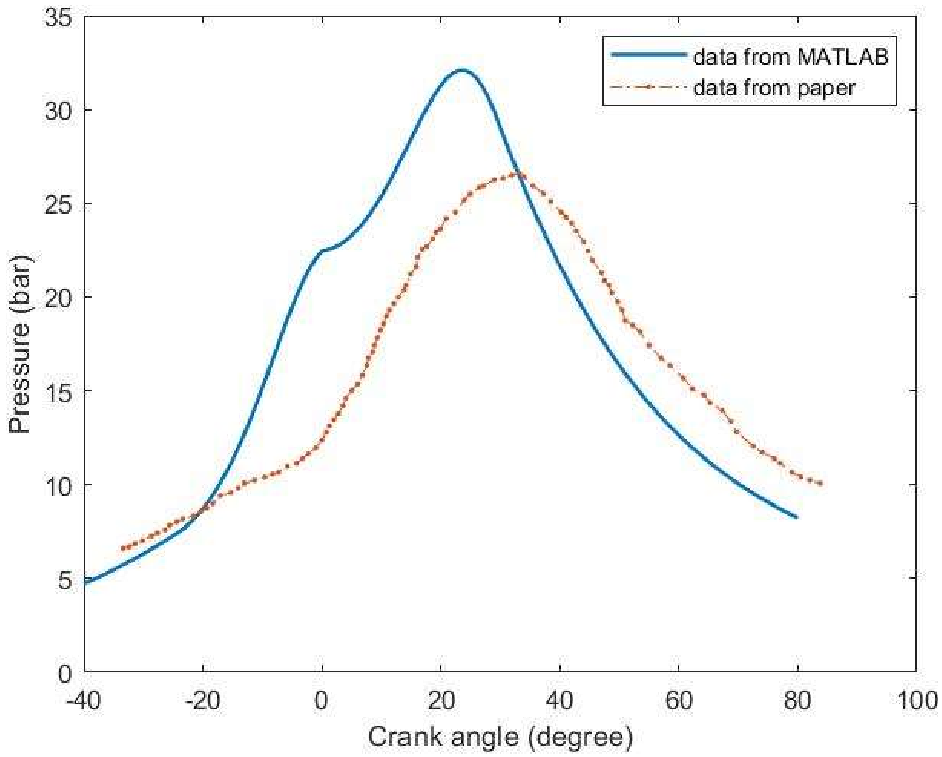

Fig. (3) Pressure vs Crank angle (degree) for Pin=0.81 bar

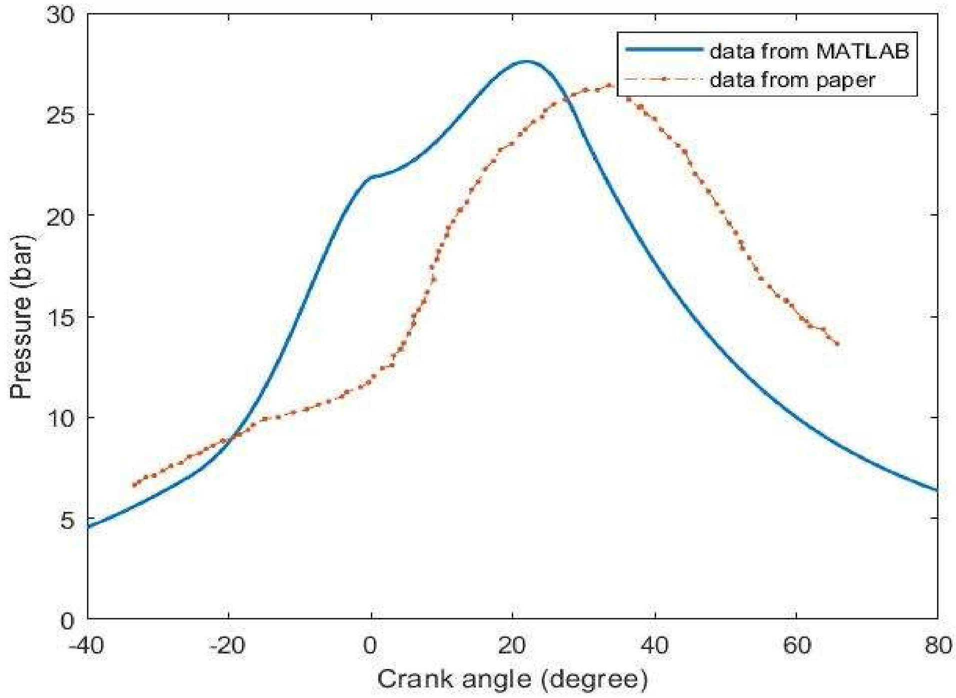

Fig. (4) Pressure vs Crank angle (degree) for Pin=0.71 bar

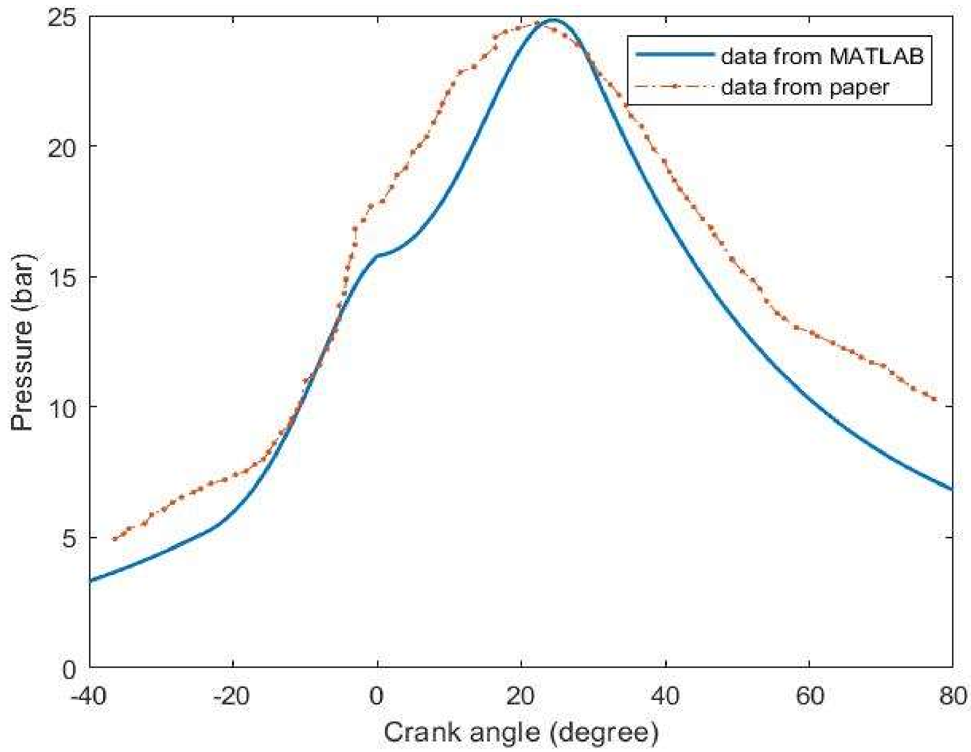

Fig. (5) Pressure vs Crank angle (degree) for Pin=0.61 bar

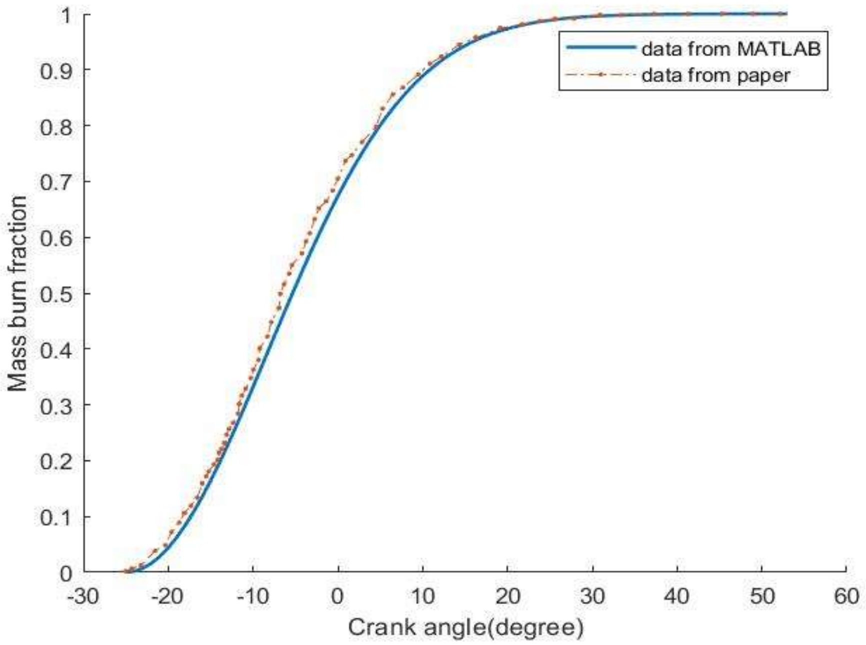

Fig. (6) Mass burn fraction vs Crank angle (degree) for Pin=0.91 bar

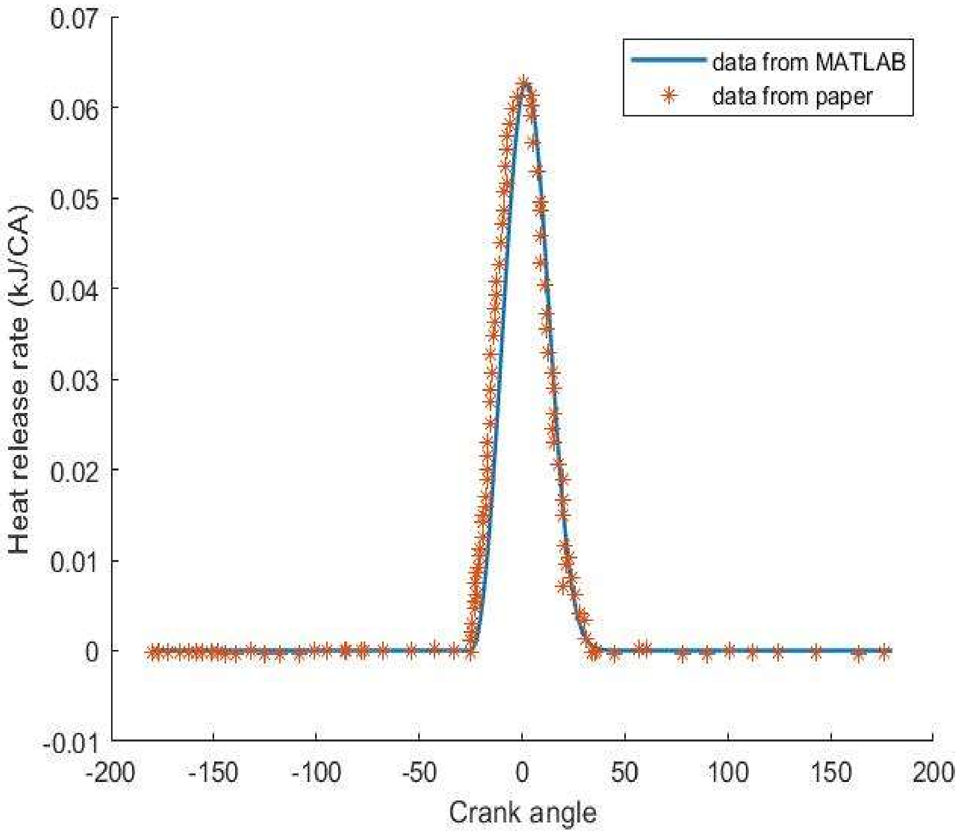

Fig. (7) Heat release rate vs Crank angle (degree)

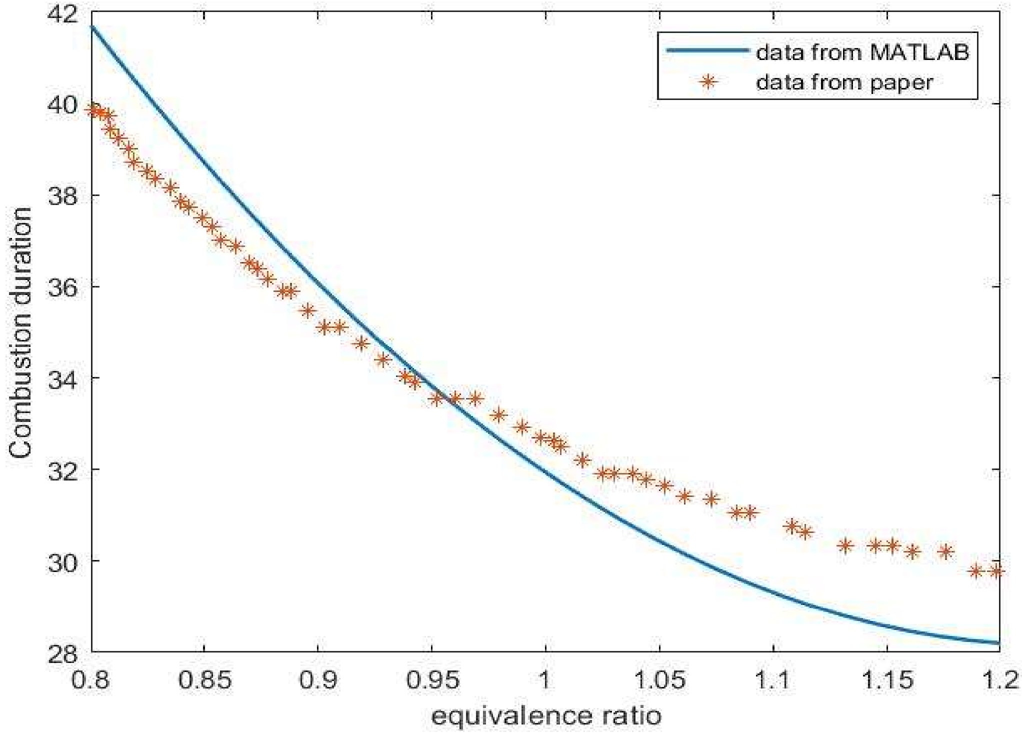

Fig. (8) Combustion duration vs Equivalence ratio

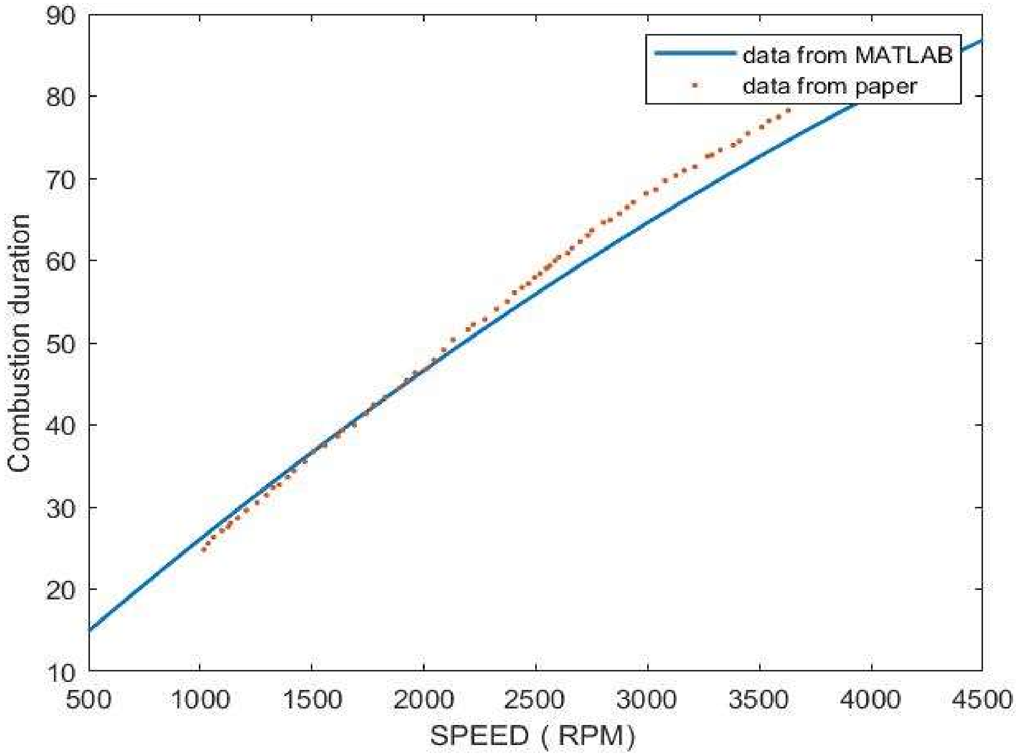

Fig. (9) Combustion duration vs Speed (RPM)

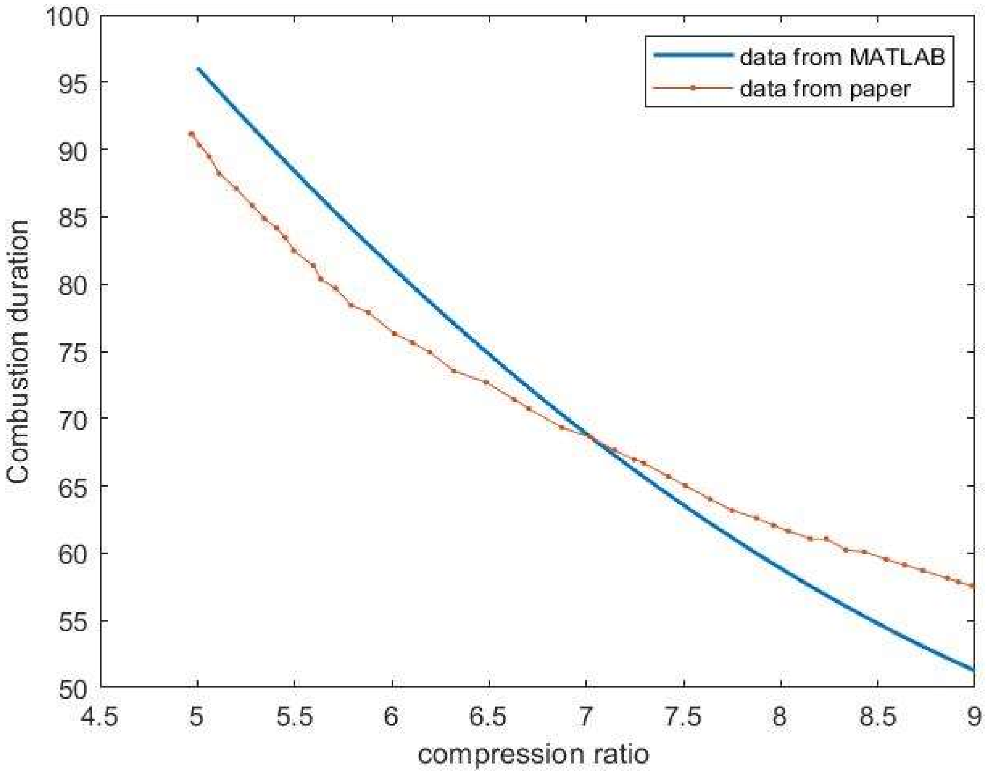

Fig. (10) Combustion duration vs Equivalence ratio

Here, we have plotted the pressure data with respect to angle for different suction pressure, the solid line represents the data plotted by MATLAB estimations and pattern is used to show the results from the research paper. The peak pressure here is higher in case of MATLAB results than the paper results. This may be due to the fact that, we have not accounted any kind of losses. The result for burn fraction and heat release rate with respect to crank angle is about the same compared to literature results but there is some fluctuation in combustion duration results.

Chapter 4

RESULTS AND DISCUSSION

Engine specifications

Bore (B), in m	0.0864
Stroke (L), in m	0.0674
Connecting rod (l), in m	0.13
Compression ratio (r)	8.3
Speed (N), rpm	3000 @ 1.8 HP
Spark timing	25° bTDC
Intake pressure (P_0),	1
Relative Fuel-air ratio	1
Fuel	C_8H_{18} (Gasoline)

Here, we have plotted the graph for pressure with respect to crank angle with 1 degree of interval. Gasoline is taken as fuel with stoichiometric air-fuel ratio 15.05. In this section, we have gone through different parameters of combustion like heat release rate, combustion duration, burn fraction etc. to understand their effects and their significance on the combustion.

In our program, we have consider the rate of fuel burn through the weibes expression, so value of rate of heat release is different at different crank angle .Peak pressure of data from MATLAB is higher than the data from literature . It is due to the fact that heat losses like surrounding losses, blow losses are not considered during the estimation.

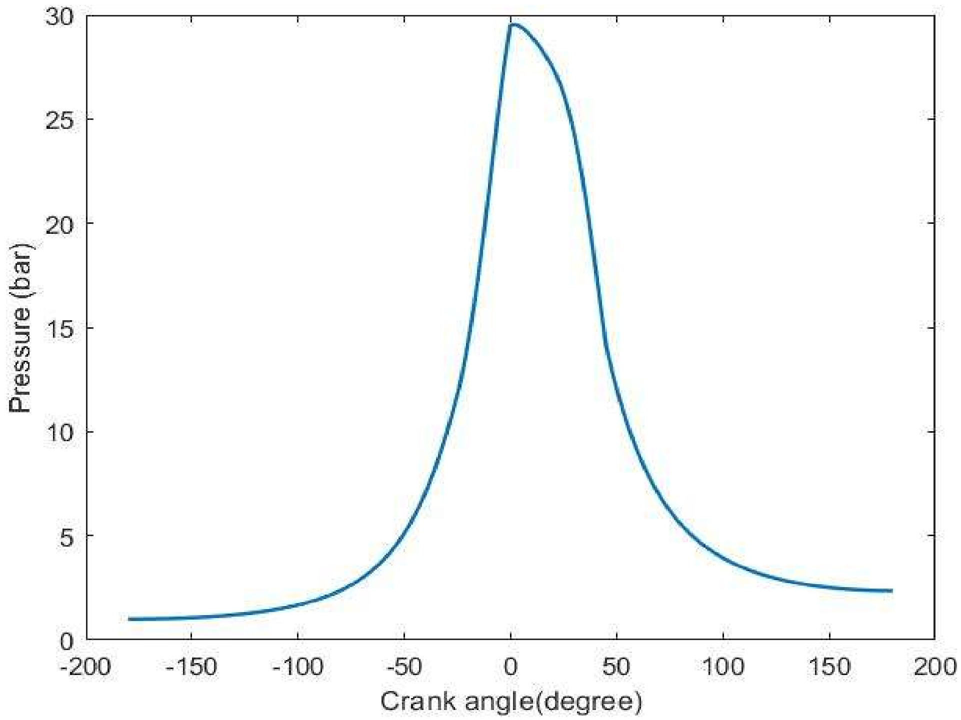

Fig. (11) Pressure (bar) vs crank angle for P_0=1 bar

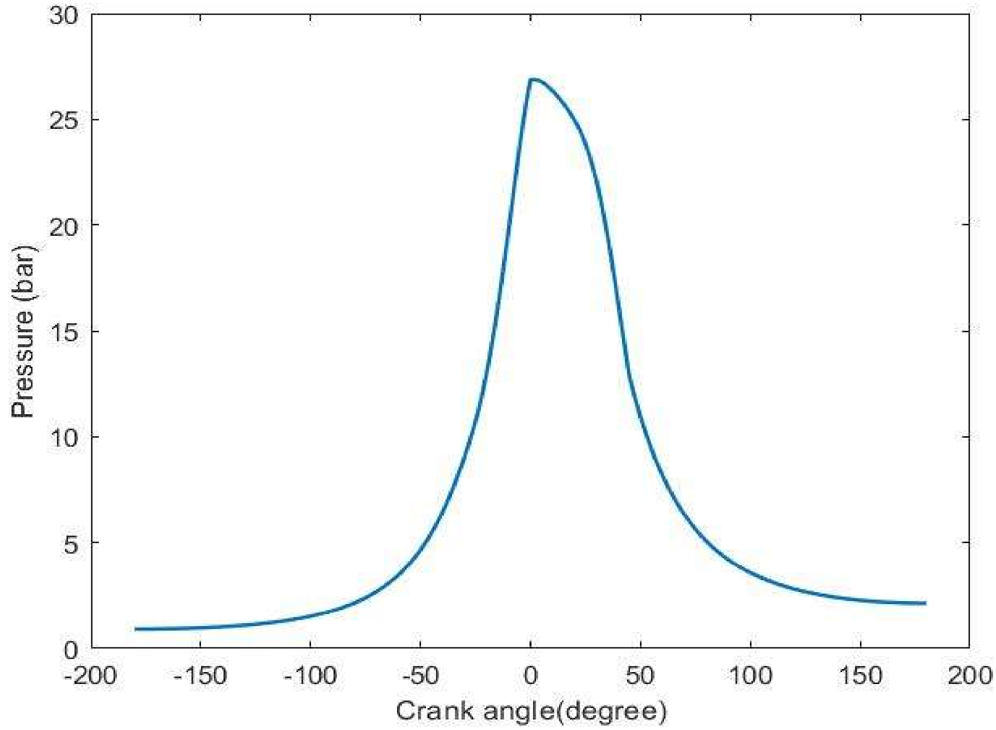

Fig. (12) Pressure (bar) vs crank angle for P_0= 0.91 bar

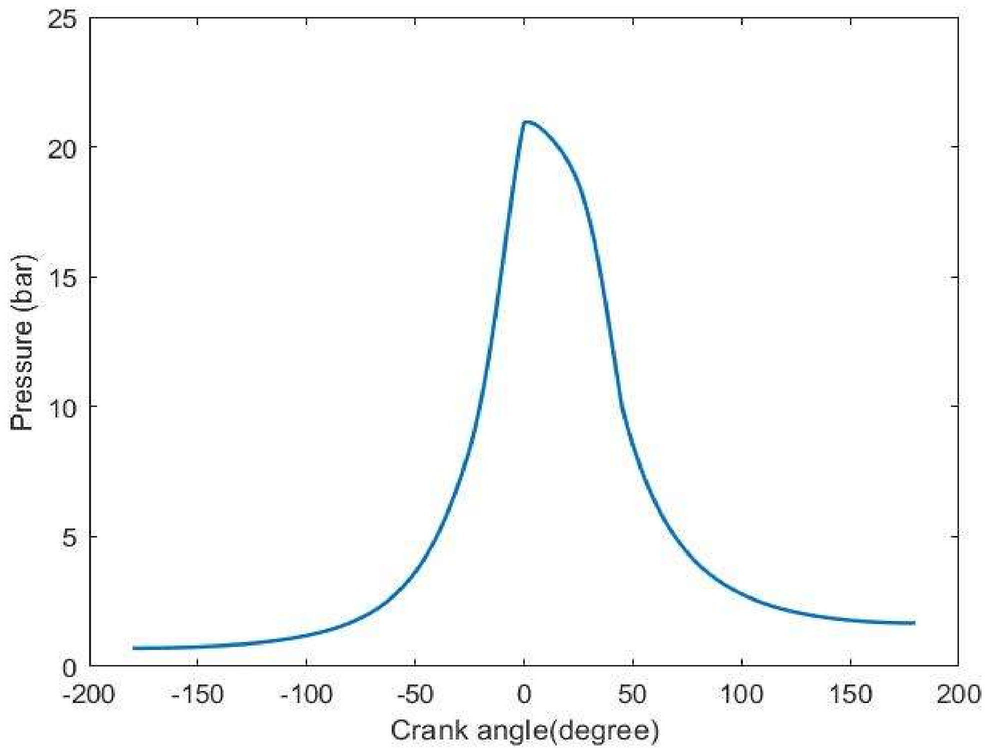

Fig. (13) Pressure (bar) vs crank angle for P_0= 0.71 bar

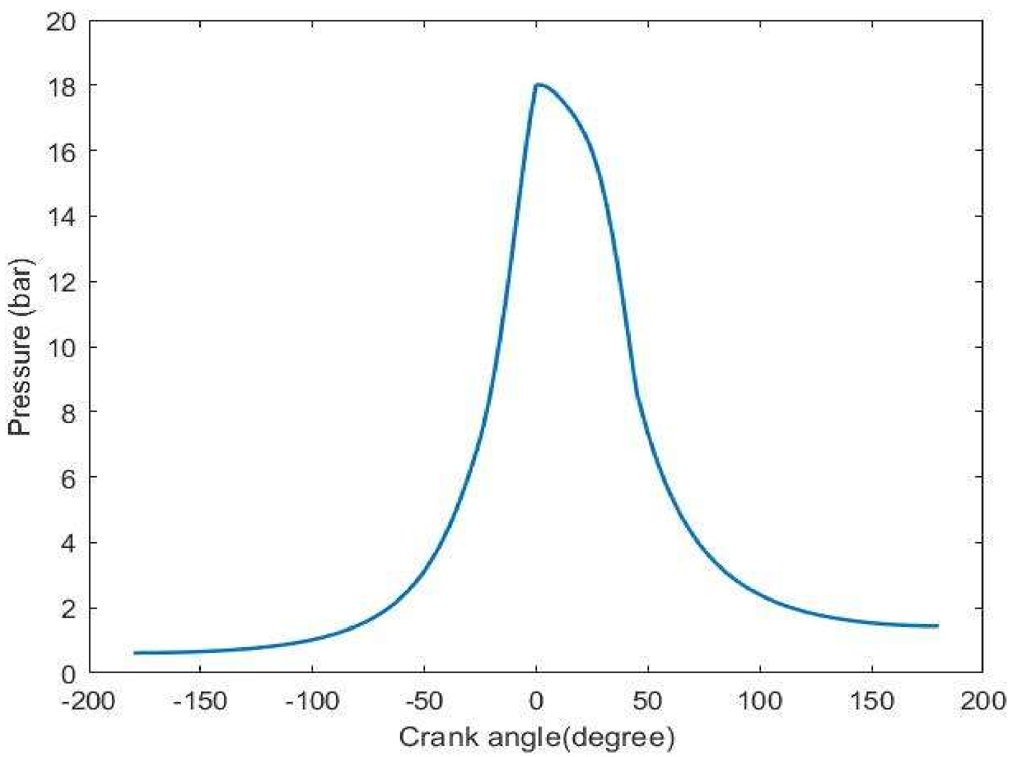

Fig. (14) Pressure (bar) vs crank angle for P_0=0.61 bar

As we can observe that from figures (11, 12, 13 and 14), peak pressure is declining with decrease in inlet pressure. Before the spark initiation at -25 degree before TDC, there is slow increase in pressure with respect to crank angle but after spark initiation, there is steep increase in pressure due to fuel burn. In earlier stage after the spark, it is seen that there is slow increase, as time is required by fuel to burn and flame to propagate because of chemical delay.

Effect of combustion duration

It has been discuss about the empirical expression of combustion duration in the formulation section. The combustion duration depends upon the compression ratio, engine speed, and equivalence ratio and spark advancement of spark ignition. Here we are plotting the graph with the help of MATLAB and checking the dependence of these factors individually.

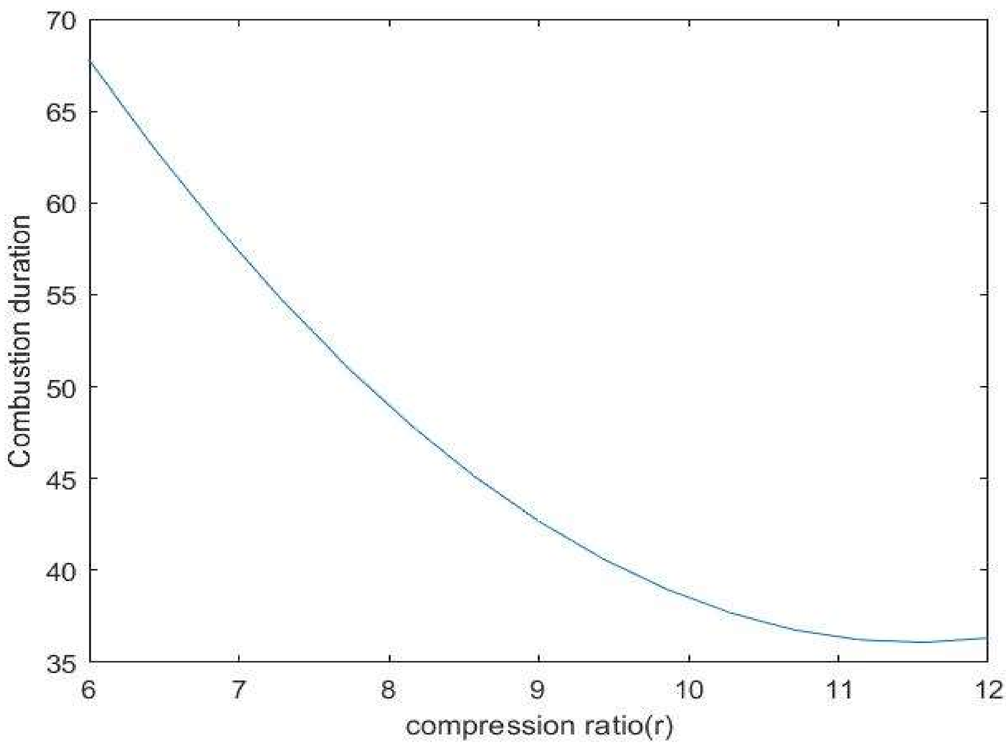

Fig. (15) Effect of compression ratio on combustion duration

As we can conclude from the graph, combustion duration is dependent on the compression ratio. As compression ratio is increasing, combustion duration is following the trend to decrease.

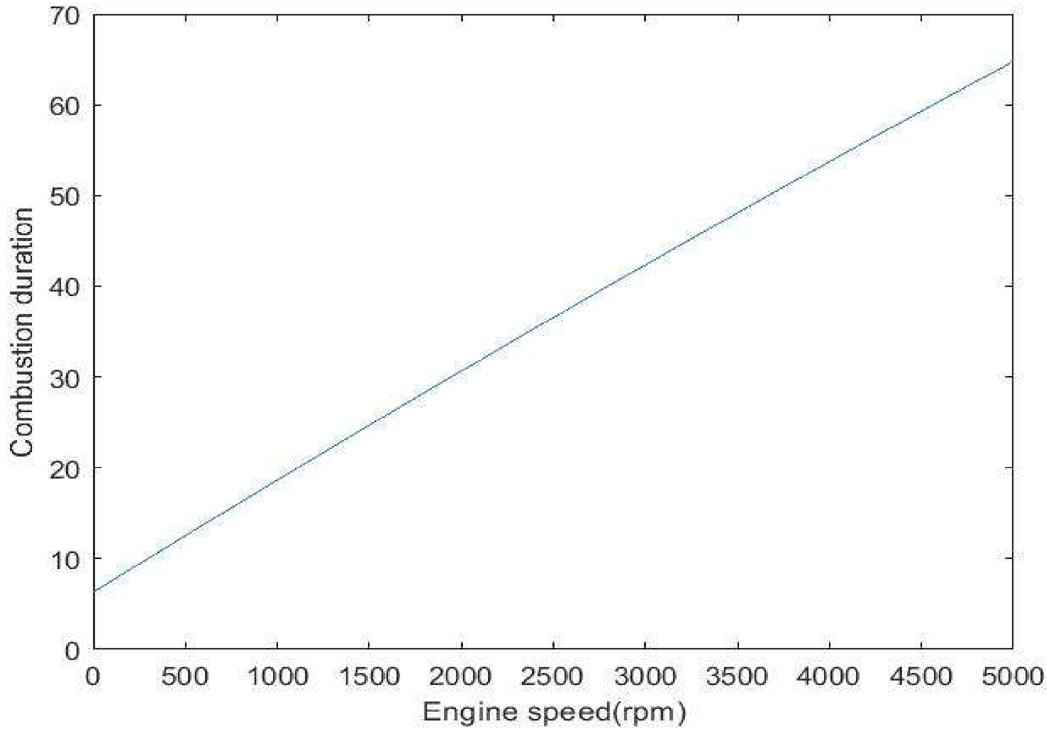

Fig. (16) Variation in combustion duration wrt N

As we can conclude from the graph, combustion duration is dependent on the engine speed. As engine speed is increasing, combustion duration is following the trend to increase. It seems like combustion duration is following the linear trend to increase but its empirical expression have the square terms of engine speed.

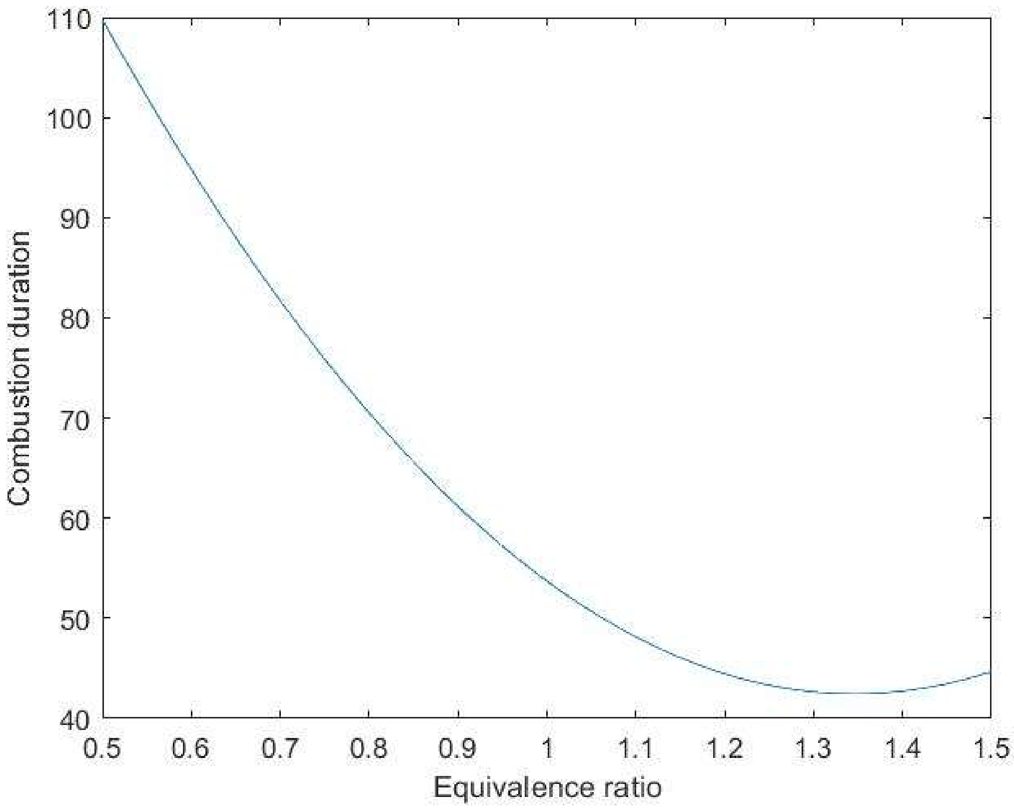

Fig. (17) Effect of equivalence ratio on combustion duration

As we can conclude from the graph, combustion duration is dependent on the equivalence ratio. As equivalence ratio is increasing, combustion duration is started declining and after φ=1.3, very less increment is observed.

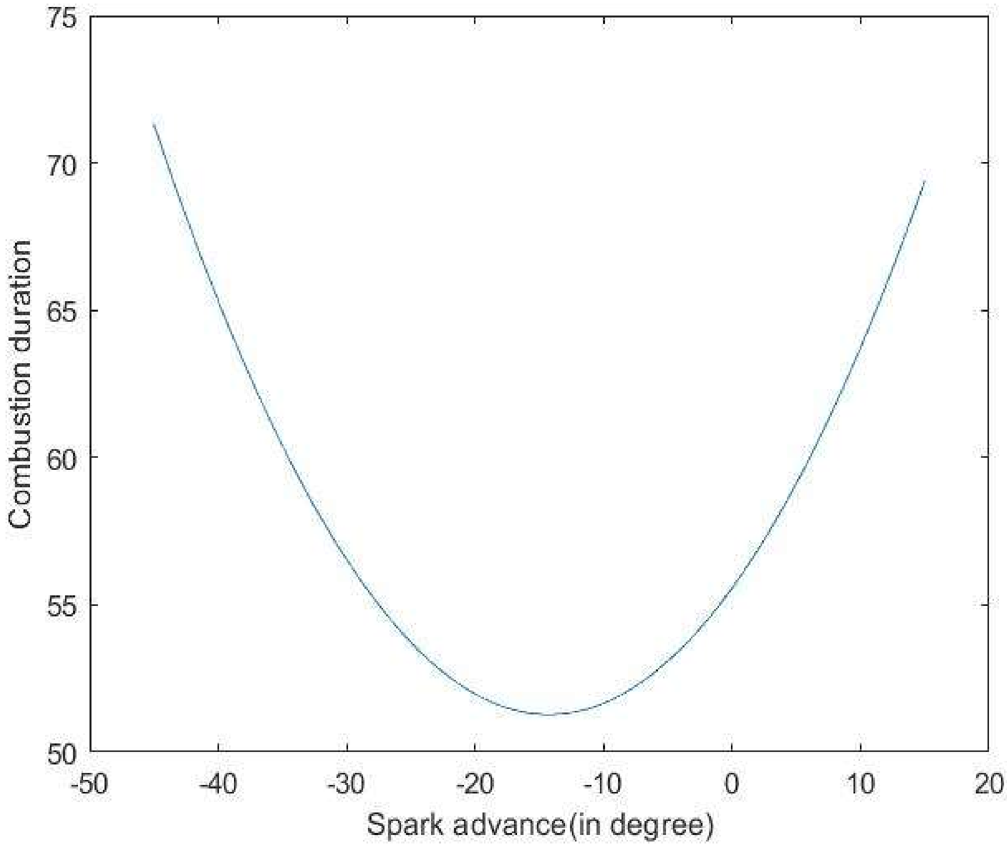

Fig. (18) Effect of spark advancement on combustion duration

As we can conclude from the graph, combustion duration is dependent on the engine speed. As spark advancement is given up to -15 degree before TDC, combustion duration is continuously decreasing. After -15 degree before TDC, It started following the trend to increase.

Burn fraction

As previously discussed, burn fraction is a part of fuel is burning during combustion wrt CA. It is given by Wiebes expression.

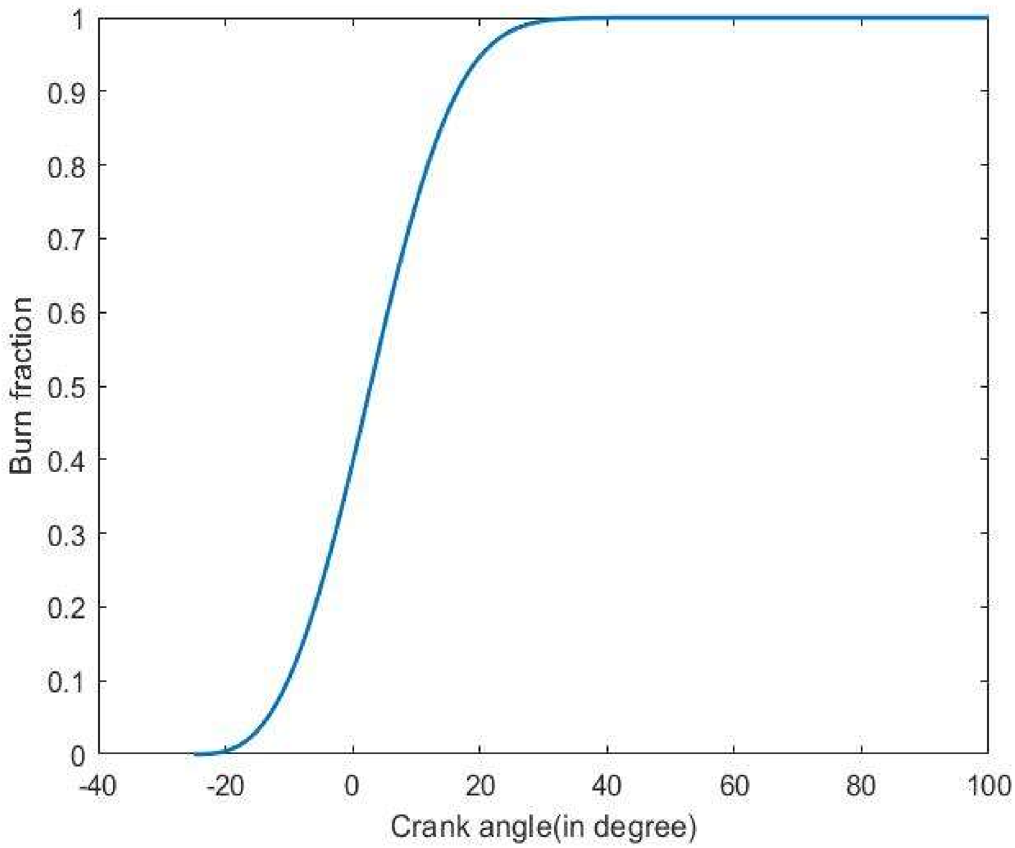

Fig. (19) Burning fraction vs Crank angle (in degree)

As spark ignition timing is -25 before TDC, fuel is started burning and rate of burning is exponentially increases wrt to CA and after complete burning of fuel till the combustion duration, it reaches the saturation i.e. No fuel is left to burn after it achieves burning fraction values to 1.

Heat release rate

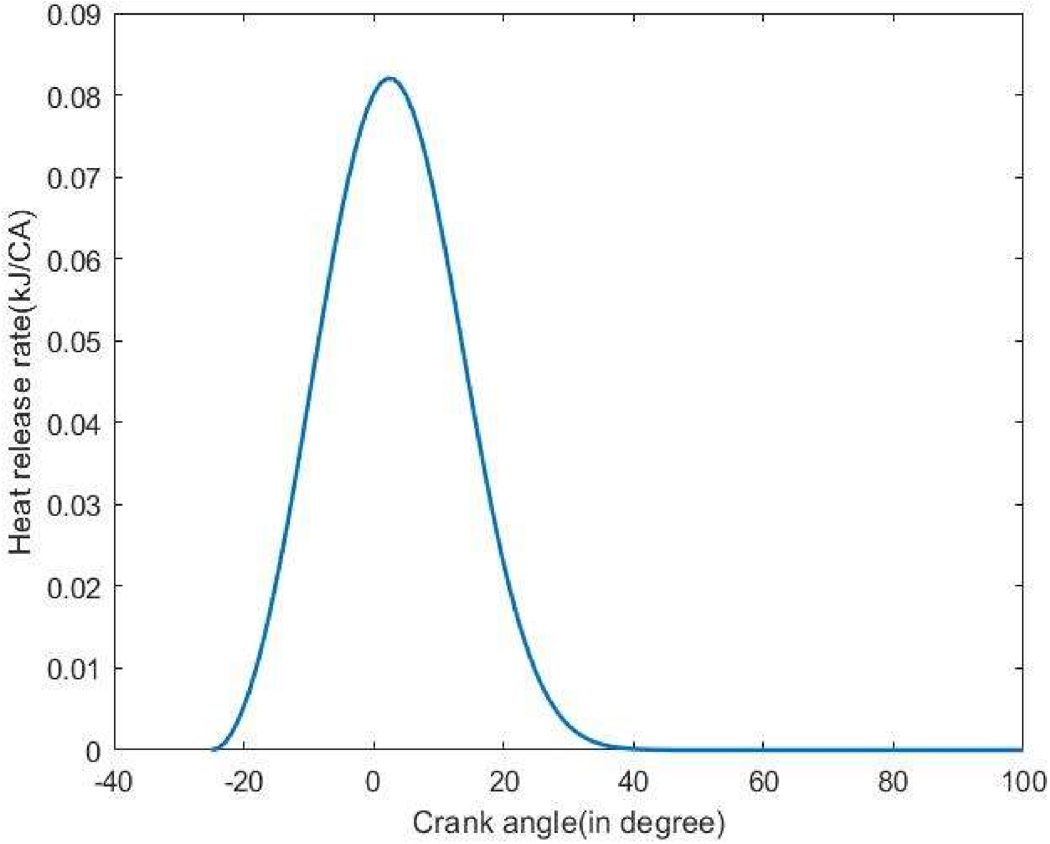

Fig. (20) Heat release rate vs Crank angle

The graph for Heat release rate with respect to crank angle is plotted in figure 17. Here we can observe that heat release due to chemical process is started nearby the spark ignition timing i.e. -25 before TDC reaches maximum at the point where peak pressure is observed. As it is heat release per degree of crank angle, if we need to calculate the total heat release, we have to integrate heat released at every crank angle and sum it up to the end of combustion. It is dependent on calorific value of fuel and combustion duration.

Volume change with respect to crank angle

Here we have plotted volume change in cylinder wrt CA by using MATLAB.

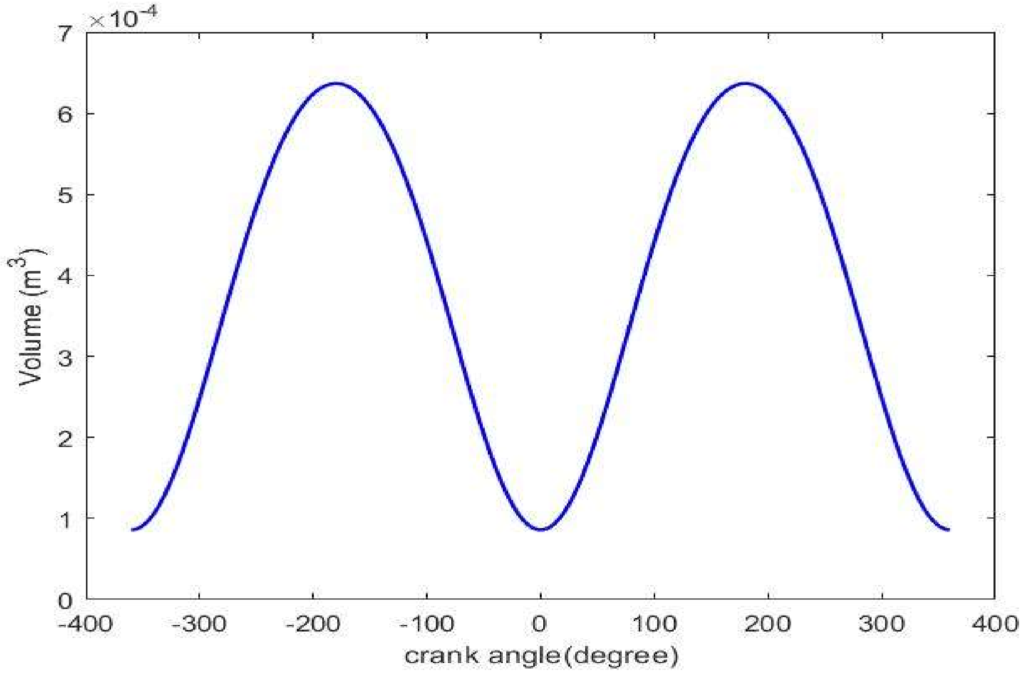

Fig. (21) Volume change vs crank angle

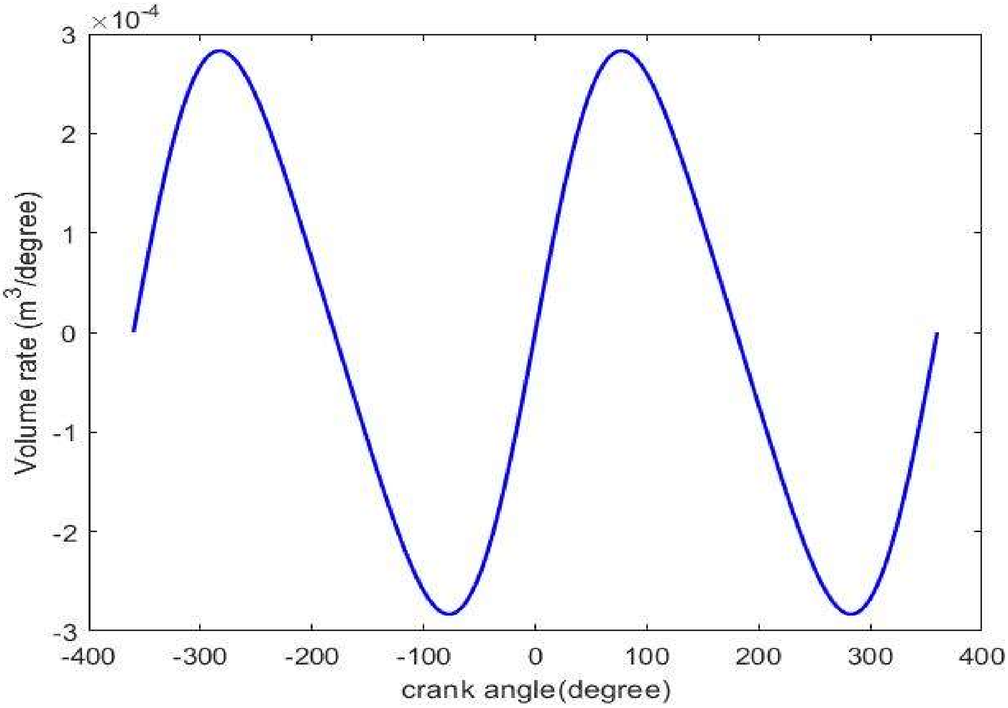

Fig. (22) Volume rate change vs crank angle

CHAPTER 5

CONCLUSIONS

5.1 Conclusions

Here we have studied the reduced order modeling of combustion process in a spark ignition engine fuelled with gasoline. The validation is done with the experimental predicted result obtained by Amit et al. [6].Firstly, we have tried to formulate the combustion phenomena by assuming the system as zero dimension single zone model. In it, formulation is divided into three part as compression, combustion and expansion. We have successfully formulate this mathematically and consider the effect of heat transfer and rate of fuel burn also but when we code this in MATLAB then it is not getting good result for the compression and expansion part so we have tried another approach by consider the fluid following the ideal law and compression and expansion process as isentropic. The burning of fuel is also taken in consideration by following the weibes expression and heat release is calculated for each degree of crank angle. The results follows the same trend as the validated paper but the MATLAB prediction have error relative to the validated paper result due to reason that heat transfer through wall and other losses are not considered in the prediction. The result for burnt mass fuel, heat release rate and combustion duration is good and have very less error as compared to paper result.

5.2 Scope of future work

Here firstly, we tried to formulate the combustion in SI engine by taking the zero dimension single zone model and successfully formulate the expression for it, but when it comes to formulate it on the MATLAB, result was not in the good term. Therefore, we tried to formulate the pressure change inside the cylinder as single zone but following the ideal nature and used isentropic equation to evaluate it. The result was nearby good but we have not considered any losses (turbulence, swirl, NOx formulation etc.) which take place inside the cylinder. We also considered that the values of Cp and Cv is constant irrespective of temperature but in real scenario, Cp and Cv values changes with temperature. All this considerations contributes in the error. With accounting all the above considerations, the result will be more accurate to the experimental results. We can also study the details and run our program for supercharger and turbocharger.

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