Second Law analysis of single cylinder four stroke diesel engine on reduced valve lift utilizing residual trapped gases

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Abstract – *According to the optimization of fuel consumption and necessity of abiding vehicle emission regulations, demands for the Variable Valve Lift (VVL) mechanism as one of efficient approach in automotive industry in order to enhance engine performance is observed increasingly. The paper thus presents second law analysis of single cylinder four stroke diesel engine on reduced valve lift.*

Modeling activities can make major contributions to engine engineering at different levels of generality or detail, corresponding to different stages of model development. The modeling of engine processes continues to develop as our basic understanding and knowledge of the physics and chemistry of the phenomena of interest steadily expands. The model developed is single zone zero dimensional model. Once the heat release rate is modeled the pressure and temperature are predicted for every crank position. A single zone, zero dimensional model is developed for simulation. The diesel engine is considered as closed system for thermodynamics analysis. The cylinder gases are assumed as ideal gas. The developed model is validated against the data obtained by experimentation at laboratory. Both experimental and mathematical data are compared.

Key Words: **Diesel Engine, Valve Lift, Second Law analysis, Engine modelling, Availability**

1. INTRODUCTION

Diesel Engines often during combustion result into accumulation of residual trapped gases in the engine cylinder. In compression ignition engines, high nitrogen oxides (NOx) and their particulate matter (PM) emissions are the biggest handicap. It is not possible to reduce both of NOx and PM emissions simultaneously in conventional compression ignition combustion. Complex and expensive after treatment exhaust emission systems are required to eliminate these emissions. However, emission norms get harder and harder with each passing day. The current after treatment systems may be insufficient in the near future to meet stringent emission regulations. Another problem of the after treatment systems such as diesel particulate filters, lean NOx trap and selective catalytic reduction is cost and durability. Therefore, it is required to improve combustion process. For this reason a great deal of importance is given to the in cylinder combustion processes.

Presently, compression ignition engines have attracted a great interest due to their high thermal efficiency, reliability, durability and high specific power compared to spark ignition engines. In spite of these advantages of compression ignition engines, high nitrogen oxides (NOx) and their particulate matter (PM) emissions are the biggest handicap. It is not possible to reduce both of NOx and PM emissions simultaneously in conventional compression ignition combustion. Complex and expensive after treatment exhaust emission systems are required to eliminate these emissions. However, emission restrictions get harder and harder with each passing day. The current after treatment systems may be insufficient in the near future to meet stringent emission regulations. Another problem of the after treatment systems such as diesel particulate filters, lean NOx trap and selective catalytic reduction is cost and durability. Therefore, it is required to improve combustion process. For this reason a great deal of importance is given to the in cylinder combustion processes.

1.1 Introduction

Thermodynamic models of the real engine cycle have served as effective tools for complete analysis of engine performance and sensitivity to various operating parameters on the other hand, It has long been understood that traditional first-law analysis, which is needed for modeling the engine processes, often fails to give the engineer the best insight into the engine's operation. In order to analyze engine performance – that is, evaluate the inefficiencies associated with the various processes – second-law analysis must be applied [1]. Second-law analysis with its more 'interior' study of what is happening during a process contributes a new way of thinking and studying thermodynamic processes, a fact providing more flexibility and field for improvement to the engineer [2]. The use of second law analysis is not necessarily intended for general performance computations but for understanding the details of the overall thermodynamics of engine operations. The second law of thermodynamics is a powerful statement of related physical observations that has a wide range of implications with respect to engineering design and operation of thermal systems. The second law can be used to determine the direction of process, establish the condition of equilibrium, to specify the maximum possible performance of thermal systems and identify those aspects of processes that are significant to overall performance [3].

The second-law analysis provides a more critical and thorough insight into the engine processes by defining the term of availability destruction or irreversibility and assigning different magnitude to the exhaust gases and heat losses terms. By so doing, it spots specific engine processes and parameters, which can improve the engine performance by affecting engine or subsystems irreversibility's and the availability terms associated with the exhaust gases (to ambient) and heat losses to the cylinder walls. Most of the analyses so far have focused on the dominant combustion irreversibility's term. It was shown that combustion duration, heat release shape, i.e. premixed burning fraction, and injection timing only marginally affect combustion irreversibility's (although the latter's impact on work, heat transfer and exhaust gases availability is significant), the combustion irreversibility production rate is a function of fuel reaction rate only, and also an increasing pre-chamber volume increases the amount of total combustion irreversibility's [4]. The majority of different reports and studies have investigated the influence of heat transfer, combustion, friction and mixing processes on availability destruction suggesting different options to reduce energy degradation and increase portion of energy available for useful work [3].

Objectives of second-law application to internal combustion engines are:

1. To weigh the various processes and devices, calculating the ability of each one of these to produce work.
2. To identify those processes in which destruction or loss of availability occurs and to detect the sources for these destructions.
3. To quantify the various losses and destructions.
4. To analyze the effect of various design and thermodynamic parameters on the exergy destruction and losses.
5. To propose measures/techniques for the minimization of destruction and losses, to increase overall efficiency.
6. To propose methods for exploitation of losses most notably exhaust gas to ambient and heat transfer to cylinder walls now lost or ignored.
7. To defined efficiencies so that different applications can be studied and compared, and possible improvements measured.

1.2 Modeling In Diesel Engine

In engineering, modeling a process has come to mean developing and using the appropriate combination of assumptions and equations that permit critical features of the process to be analyzed. Modeling activities can make major contributions to engine engineering at different levels of generality or detail, corresponding to different stages of model development, by [5]:

Developing a more complete understanding of the process under study from the discipline of formulating the model;

(1) Identifying key controlling variables to provide guidelines for more rational and therefore less costly experimental development efforts;
(2) Predicting engine behavior over a wide range of design and operating variables to screen concepts prior to major hardware programs, to determine trends and tradeoffs, and, if the model is sufficiently accurate, to optimize design and control;
(3) Providing a rational basis for design innovation.

Models differently considering prominent role of chemical kinetics in ignition delay event and physical mixing rate in heat release event are considered. [6]

(1) Combustion Model

The combustion starts almost at the onset of fuel injection because the ignition delay in modern DI diesel engines is very small with high compression ratio and highly retarded injection timing, which enable substantial reduction in noise, NOx and HC. . The heat release estimated with this assumption predicts satisfactorily the important instantaneous parameters used by a designer e.g. heat transfer, fuel consumption, and the performance turbocharger and piston.

(a) Ignition delay

The ignition delay in a diesel engine is defined as the time interval between the start of injection and the start of combustion. This delay period consists of (a) physical delay, wherein atomization, vaporization and mixing of air fuel occur and (b) of chemical delay attributed to pre-combustion reactions. Both physical and chemical delays occur simultaneously.

(b) Heat Release

i. Models based on Fluid dynamics

These types of models are often called as multidimensional models due to their inherent ability to provide detailed geometric information on the flow field based on solution of the governing equations.

ii. Phenomenological models

In these types of models, details of different phenomenon happening during combustion are added to basic equation of energy conservation. It found that the understanding of the spray structure offered the clue to better heat release predictions.

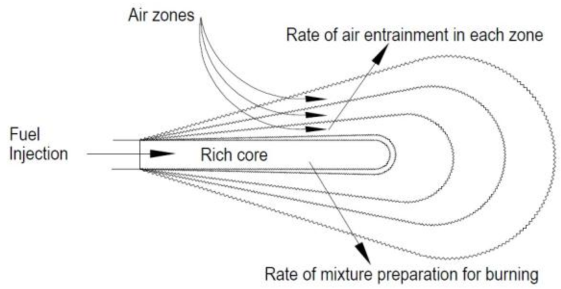

Figure 1.1: Multi Zone Spray Model [6]

iii. Zero-dimensional models

This type of models is more attractive due to their simplicity as they use simple algebraic equations to describe heat release rate. The rate of injection diagram was subdivided into elemental fuel packets emanating as rectangular pulses, which results in exponentially decaying heat energy function.

(2) Emission models

DI diesel engines emit different emissions from exhaust. Smoke, Hydrocarbons, Nitric Oxides, Carbon monoxide and particulate matter are mainly regulated. They are formed in different phases of combustion as described below.

(a) Hydrocarbons
(b) Oxides of Nitrogen
(c) Particulate Matter

A variety of studies on engine cylinder heat transfer measurement and modeling have been made. The studies of heat transfer can be separated into compression and expansion processes, combustion process, gas exchange process and whole process. Repeated attempts have been made to provide an empirical formula for the estimation of instantaneous transfer rate. These formulas can be divided into four groups.

(a) The formula by mainly experimental analysis,
(b) The formula based on steady turbulent flow,
(c) The formula from energy analysis of unsteady thermal boundary layer and Radiative heat transfer.

The global heat transfer coefficient can be written as [7],

$$h_{global}(t) = \alpha_{scaling} . L(t)^{m-1}.k/\mu^{m}.p(t)^{m}.T(t)^{-m}.v(t)^{m}$$

The global heat transfer coefficient depends on characteristic length, transport properties, pressure, temperature, and characteristic velocity. A scaling factor α scaling is used for tuning of the coefficient to match specific engine geometry. A value for the exponent *m* has been proposed by several different authors.

The heat transfer coefficient by Nusselt is given by [26],

$$h = 0.99(1+1.24C_m).\sqrt[3]{P^2.T} + \frac{0.362}{(1+\xi_g+\frac{1}{\xi_w}-1)} * \frac{(\frac{T}{100})^4-(\frac{T_W}{100})^4}{T-T_W}$$

Where, ξ_g – Emissivity of gas, ξ_w – Emissivity of a wall, C_m – Mean piston speed, P – Pressure.

1.3 Availability of the system

The availability of a system in a given state can be defined as the maximum useful work that can be produced through interaction of the system with its surroundings, as it reaches thermal, mechanical and chemical equilibrium. Usually, the terms associated with thermo mechanical and chemical equilibration are differentiated and calculated separately. For a closed system experiencing heat and work interactions with the environment, the following equation holds, for the thermo mechanical availability.

$$A^{tm} = (E - U_0) + P_0(V - V_0) - T_0(S - S_0)$$

Where, $E = E_{kin} + E_{pot} + U$, with E_{kin} the kinetic and E_{pot} the potential energy, P_0 and T_0 are the fixed pressure and temperature of the environment; and U_0,V_0 and S_0 are the internal energy, volume and entropy of the contents were they bought to P_0 and T_0.

Availability is an extensive property with a value of greater than or equal to zero. It is obvious that availability is a property, the value of which depends not only on the state of the system, but also on the ambient properties. As stated above, there is no availability in a system when thermal, mechanical, and chemical equilibrium exist with the environment. Thermal equilibrium is achieved when the temperature of the system is equal to the temperature of the surrounding environment. In the same way, mechanical equilibrium is achieved when there is no pressure difference between the working medium and the environment.

1.4 Heat Transfer From the engine cylinder

The description of heat transfer in the internal combustion engine places the highest demands on modeling and is usually based on a global inspection of very complex relations. Heat transfer consist of a radiation component.

$$\frac{dQ_w}{dt} = \frac{dQ_h}{dt} + \frac{dQ_r}{dt}$$

Usually the radiation component dQ_r/dt is added to the convective heat transfer coefficient, although the maxima of dQ_h/dt and dQ_r/dt actually appear phase shifted with reference to the crank angle. Proceeding from the Newtonian method, for the description of the wall heat flow:

$$\frac{dQ_w}{dt} = \sum h_i A_i (T_w - T_{gas})$$

We usually subdivide the combustion chamber thereby into three areas :

1. The piston
2. The cylinder head
3. The portion of the liner released by the piston including deck height and piston junk.

The valves are usually calculated with the cylinder head or in very detailed modeling as their own area. The surfaces for the pistons and the cylinder head are usually larger than the cylinder cross-section surface. The calculation of heat transfer with the help of the Newtonian approach and the heat transfer coefficients requires an exact description of gas and all temperatures. The mean gas temperature results from the local averaging of gas temperature in the combustion chamber. Since the combustion chamber system is usually seen as an ideally mixed volume, the mean gas temperature is easy to determine from the condition equation for an ideal gas. At respective wall temperatures, we are dealing with the internal wall temperature averaged over one working cycle. For the piston and the cylinder, usually local constant temperatures are used. In the case of the liner, the wall temperature depends a lot on the engine type and on whether the liner is completely or only partially surrounded by the water jacket. In giving the temperature for the liner, we usually subdivide it into several areas, or we provide a temperature profile over the length of the liner. The temperatures can either be determined by measuring, or we can use a simple, iterative method for the calculation of internal wall temperature for stationary operating points. For the calculation of heat transfer coefficient, semi-empirical methods are usually used, since many influence factors can only be determined by experimentally. The method of Woshni, which was constructed for diesel engine in 1969, has continually been further developed. The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from:

$$Q_w = h_i A_i (T_g - T_w)$$

Where, Q_w wall heat flow (cylinder head, piston, liner), A_i surface area, h heat transfer coefficient, T_g temperature in the cylinder, T_w wall temperature.

2. LITERATURE REVIEW

S. V. Lokhande and M. M. Deshmukh [9] have given the review of this experiment and analysis. In this paper different mathematical models and their equation are studied. After the complete analysis of the papers, experimentation was done to get the experimental data. With the help of experimental data, Second analysis of the diesel engine on reduced valve lift is done.

Tianyou Wang, Daming Liu, Gangde Wang, Bingqian Tanand ZhijunPeng [10] have given the Effects of Variable Valve Lift on In-Cylinder Air Motion. The experimental work conducted in the modified four-valve optical spark ignition (SI) test engine with three different MVL. Particle image velocimetry (PIV) was employed for measuring in-cylinder air motion and measurement results were analyzed for examining flow field, swirl and tumble ratio variation and fluctuating kinetic energy distribution. Results of ensemble-averaged flow fields show that reduced MVL could produce strong swirl flow velocity, and then resulted in very regular swirl motion in the late stage of the intake process. The strong swirl flow can maintain very well until the late compression stage. The reduction of MVL can also increase both high-frequency and low-frequency swirl flow fluctuating kinetic energy remarkably.

Zhilong Hu, Yong Gui, Min Xu, Kangyao Deng, Yi Cui and Jiayong Dou [11] have done a design of a variable valve hydraulic lift system for diesel engine. The hydraulic mechanism, which consists of a driving plunger, a driven plunger, a hydraulic cylinder, and a hydraulic oil tank, is the key part of the VVL mechanism. Simulation is conducted to study the relationship between maximum valve lift and rotation angle of driving plunger. Calculation results indicate that the maximum valve lift decreases with increasing rotation angle. A prototype was manufactured and successfully tested in a single-cylinder diesel engine.

Emad Monemian, M.H. Shojaeefard, Amir Hossein Parivar [12] have demonstrated the Potentials of Variable Valve. Lift for improving full load operation in gasoline engine. Based on the final results, VVL with association of variable valve timing (VVT) could result in 3% increase in brake torque and 15.5% in nitrogen oxides emission averagely and also upto 4.5% reduction of BSFC at full load operation.

C. D. Rakopoulos and E. G. Giakoumis [4] have studied speed and load effects on the availability balances and irreversibilities production in a multi-cylinder turbocharged diesel engine. They used a multi-cylinder, turbocharged, in direct injection diesel engine from a view based on second-law analysis. A single-zone thermodynamics model is developed following the filling and emptying modeling technique. In all parts of the diesel engine plant a second-law analysis is performed, that provides all the present availability terms and accounts for the analysis of each component's irreversibilities. The results by first law and second law are compared. To simulate the heat loss to the cylinder walls for each the main chamber and therefore the pre-chamber, the model of Annand is employed. They highlighted however the two basic engine operation parameters, i.e. speed and load, have an effect on the operation of this engine below a second-law perspective.

C. D. Rakopoulos and E. G. Giakoumis [8] developed a computer model for studying the first- and second-law (availability) balances of a turbocharged diesel engine, operating under transient load conditions. They use the model of Annand to simulate the heat loss to the cylinder walls. They declared that second-law analysis results don't invariably go together with the first-law ones, strengthening the idea that a joint improvement of first and second-law could also be an awfully sensible choice for establishing best engine performance.

J. C. Dent and Pramod S. Mehta [14]A phenomenological model is presented for prediction of the combustion characteristics of a Quiescent Chamber Diesel engine. Predictions with the model have shown acceptable agreement with a range of experimental data. The major physical processes controlling combustion have been characterized, and the dominant role of air entrainment and turbulent mixing confirmed quantitatively.

N. Watson and M. Kamel [15]The lower efficiency of the indirect-injection diesel engine, with respect to the direct injection type, is due to additional heat transfer from the combustion chamber, gas pumping between chambers, later injection timing and a different burning rate schedule. The paper describes a project to isolate and quantify these reasons for low efficiencies, using a mathematical model, supported by experimental verification. The model is based on a two-zone system (main and pre-chamber), with component models for heat transfer, gas flow through the passage etc. Experimentally derived heat release schedules in main and pre-chamber are used. It is shown that for a 0-1 m bore engine, with a Ricardo Comet Vb pre-chamber, the different burning rate schedule is the major contribution to the difference in efficiency, with injection timing and gas pumping being the least significant.

W. T. Lyn and E. Valdmanis [16] The effects of physical factors on ignition delay have been studied on a motored research engine using a single injection technique. The fuels used included a high cetane number reference fuel, gas oil and M.T. 80 petrol. The primary factors investigated are those pertaining to the fuel spray, such as injection timing, quantity, and pressure (affecting drop size, velocity and injection rate)j hole diameter (affecting drop size and injection rate) and spray form (nozzle type); and those pertaining to the engine, such as temperature, pressure and air velocity. Engine operating variables such as speed and load affect the ignition delay because they change the primary factors such as injection pressure, compression temperature, pressure and air velocity. It has been found that inner normal running conditions, compression temperature and pressure are the major factors. All other factors have only secondary effects. Under starting conditions, when ignition is marginal, mixture formation becomes as important as compression temperature and pressure. Such factors as air velocity and spray form which affect the mixing pattern can have a very pronounced effect on ignition delay. Published data on ignition delay are compared with those obtained in the present investigation and a generalization of the data is recommended for engine design and computational work.

Donald J. Patterson[17] Cylinder pressure variation is a fundamental and widespread combustion problem in spark-ignited engines. The basic factors causing this problem are variations both in the start and in the rate of combustion. These variations occur not only from cycle to cycle within each cylinder but may also show up as consistent differences between cylinders. Our test results indicate that the major cause of cyclic combustion rate variation is the mixture velocity differences that exist within the cylinder near the spark plug at the time of ignition. As yet we do not know how to reduce the cyclic mixture velocity variations and thus reduce the problem at its origin. However, it is possible to circumvent some effects of cyclic pressure variation by increasing the average combustion rate.

H. Hiroyasu and T. Kadota [18] In order to determine spray droplet size in a diesel engine, fuel was injected into high-pressure, room-temperature gaseous environments with a diesel engine injection system. Droplet size was measured using the liquid immersion sampling technique with a mixture of water-methylcellulose solution and ethanol used as an immersion liquid for diesel fuel oil. The volume distribution of diesel spray droplets is well correlated with chi square distribution with freedom, ø = 8, in the range of this investigation. The Sauter mean diameter increased with increasing back pressure, with the amount of fuel in a spray, and with decrease in pump speed. An empirical correlation was developed between effective injection pressures, air density, the quantity of the fuel delivery, and the Sauter mean diameter of spray droplets, the swirl atomizer, air blast atomizer, and rotating-cup atomizer has been measured by numerous investigators, but only a few studies of droplet size in the diesel combustion chamber have been made. The lack of sufficient data of droplet size distribution in diesel spray is due to the difficulty in measurement which arises from the large number of droplets to be sampled, intermittent characteristics of the spray, and its liquidity.

3. SYSTEM DEVELOPMENT

3.1 Experimental Model

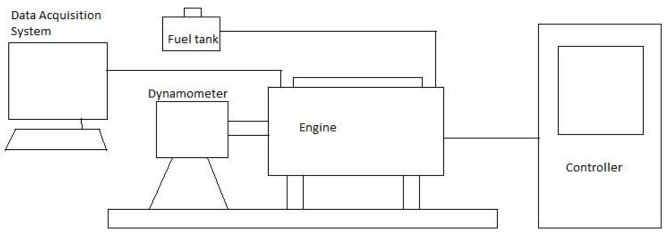

Figure 3.1: Block diagram of Experimental setup

Table 3.1.1: Engine Specification

Engine Model	Single Cylinder Four Stroke Air Cooled Diesel Engine
Engine Make	Comet
Maximum Output	5Bhp / 3.7kW@ 1500 rpm
Bore	80mm
Stroke	110mm
Compression ratio	16:1

Table 3.1.2: Test Bed Specification

Eddy Current Dynamometer	Water Cooled Eddy Current Dynamometer
	Maximum BHP: 10@1500 rpm
Air flow rate Transmitter	Anemometer : Hot wire type
	Output: 4-20mA
Load-cell Transmitter	S type, Range: 0-25 Kg
	Output: 4-20mA
Fuel Sensor Transmitter	Output: 4-20mA
	Range: 105 gm.
Pressure (Sensor Measuring Range)	0-250bar

3.1.1 Actual experimental setup description

The complete experimental setup is shown in figure 3.1. The engine is connected with water cooled eddy current dynamometer, which operates at maximum 10 bhp at 1500 rpm. The control panel is attached, which shows readings of different sensors. The data acquisition is done using ICET software by Niyo Engineers. Different data from sensors is being loaded in the software which can be also saved. Engine is supplied with diesel from fuel tank. The arrangement is made on the engine for the measurement of pressure and temperature. Inside the engine cylinder, the engine is attached to the eddy current dynamometer for the performance study at the varying load. Engine is to be equipped with several measuring instruments. The main part of engine and measuring equipments are explained as follows.

3.1.2 Test engine

The test engine foe experimental purpose is single cylinder, four stroke, air cooled, direct injection diesel engine. The engine is connected to the dynamometer which can give the maximum of 10 bhp of power at 1500 rpm. The engine has dimension of 110 × 80 mm of stroke to bore value.

3.1.3 Fuel measurement

This is done by using a specially designed arrangement using Ultrasonic sensor. The amount of fuel consumed is determined by software/hardware combination by deducting the initial reading from final reading at a regular pre-determined interval.

3.1.4 Air flow measurement

Air flow is measured using an anemometer placed inline of suction air.

3.1.5 Exhaust gas heat loss measurement

Exhaust gases from the engine passes through the flexible hose to the calorimeter. The calorimeter is mounted on a stand and supports. Exhaust gas enters into the calorimeter through the calorimeter exhaust gas inlet .Heat is exchanged by circulating water through a pipe in the calorimeter. Sensors mounted at various position measures the temperatures at that point.

3.1.6 Temperature measurement

The temperature at different points is measured and displayed on PC. The points are,

1) Calorimeter exhaust gas inlet temperature
2) Calorimeter exhaust gas outlet temperature
3) Calorimeter water inlet temperature
4) Calorimeter water outlet temperature
5) Ambient temperature

3.1.7 Speed measurement

The speed of an Engine is measured by a sensor and it is displayed on PC.A PNP type inductive proximity sensor is used to detect the speed. The sensor gives one pulse per revolution. And the frequency of these pulses is directly proportional to the speed.

3.1.8 Load measurement

Load cell is mounted on the dynamometer to measure the load on the engine.

3.1.9 P-θ Measurement

Pressure Sensor along with Signal Conditioner is used for cylinder pressure measurement. Angle and TDC are marked by encoder. Pressure Sensor: It generates an electrical voltage in response to pressure. Encoder: It generates pulse as per crank angle. It also generates pulse for TDC.

3.1.10 Experimental procedure

The aim of the experiment is to obtain the pressure and crank angle data for varying valve lift at constant speed. Constant fuel supply of diesel was made through the fuel tank which later passed through the fuel conditioning unit. The load is varied by the tuner which is placed at the controller display. The pressure theta data is obtained from the computer which is connected to the controllers and pressure theta sensors. For each valve lift, the number of pressure theta data was obtained and the average of all cycle was calculated. This is done to avoid local variations in measurement. The same procedure is repeated for all valve lifts and the data for pressure and crank angle was obtained. The fuel consumption is measured by specific fuel consumption meter which calculate fuel consumption in every 30 seconds and from that data, the fuel consumption per hour for every valve lift is calculated.

4. EXPERIMENTAL ANALYSIS

For the engine combustion study, the engine cylinder pressure plays an important role. So, at beginning the engine cylinder pressure at 75% load load at constant speed of 1500 RPM is studied with 20 % blend of ethanol in diesel

4.1 In-cylinder pressure

It is observed that along with increase in valve lift the pressure inside the cylinder is increased. This is because of eddies created inside the engine cylinder due to the increased valve lift. It is noted that the maximum value of firing pressure curve that is peak of firing pressure curve is shifted slightly towards the positive side of TDC for lowering in the in valve lift. The deviation from the motoring curve marks the start of combustion. It is also seen that start of combustion point gets closer to the TDC for reduced valve lift and it is because of the ignition delay that decreases with increase in the valve lift.

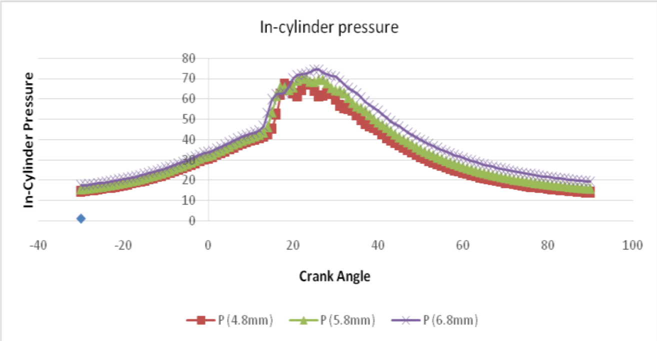

Fig -4.1: In-Cylinder pressure

4.2 Mass burn fraction calculation

Mass fraction burned is calculated on the concept that the mass burn fraction of fuel will be in the proportion of difference in the pressure due to change in volume (motoring pressure) and the actual pressure. And it is found that for almost all valve lifts the pattern of mass fraction burned was same. The mass fraction burned curve obtained from actual pressure data is very smooth and does not show any abrupt changes. The graph shows the very fast burning up to 25º after TDC and after that the heat release rate slows substantially. The first half burning at very fast rate shows that it is burning with abundance of oxygen. At the beginning before TDC the curve goes down to some negative values and just, before TDC it started rising. That rising point before the TDC marks the start of heat release rate

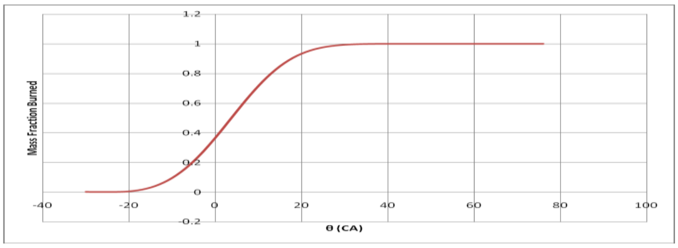

Figure 4.2: Mass fraction burn

4.3 Specific heat ratio

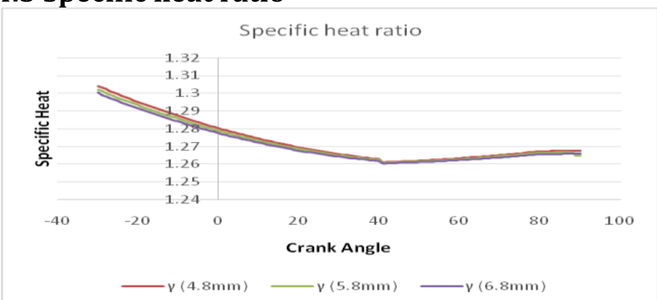

Fig -4.3 Specific heat ratio

Since the specific heat, ratio γ has a great influence on the heat release peak and on the shape of the heat release curve many researchers have elaborated different mathematical equations to describe the dependence of from temperature. It is evident from the figure with increasing crank angle as temperature increases; the value of ratio of specific heat goes on decreasing.

4.4 Heat Transfer Co – efficient

The heat transfer coefficient used is the Hohenberg heat transfer coefficient because of the difference in the temperature exponent used in the model. The heat transfer coefficient decreases when the piston is in a downward motion and before the start of combustion it is increasing.

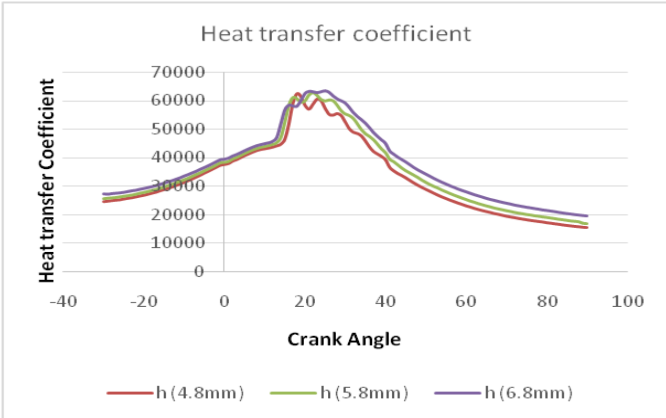

Fig -4.4: Heat Transfer Coefficient

4.5 Heat release rate

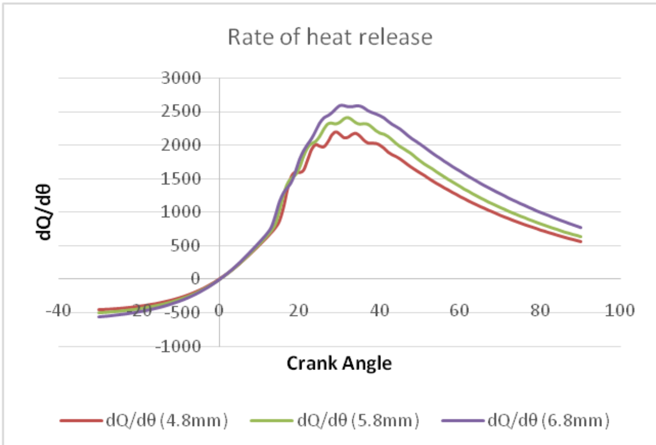

Fig -4.5: Heat release rate

Heat release rate is modelled using Weibe model for heat release production. Once heat release rate is predicted, engine cylinder pressure curves for various heat release pattern can be calculated.The heat release rate decreases when the piston is in a downward motion, and during the intake process, the heat flux is minimal.

4.6 Work Availability Terms

While studying engine cylinder availabilities we have to consider various availability terms such as fuel availability, work availability, etc.

4.6.1 Work availability

Figure4.6.1 and figure 4.6.2 show the term rate of work availability with the crank angle. From above figures, it is observed that during the compression process the work availability is transferred to the engine charge so it does not show any rise but fall in it up to top dead centre just after the TDC as piston starts expanding and the due to combustion also the work availability rises. In this region the fuel availability mainly converts in to work availability.

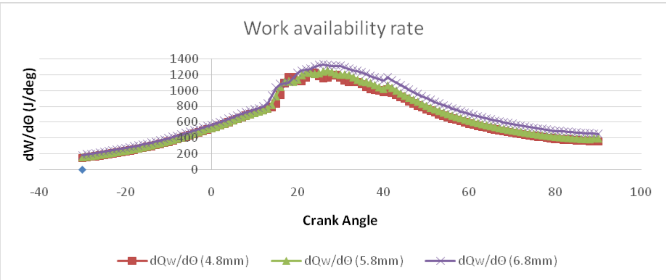

Fig -4.6.1: Work Availability Rate

4.6.2 Cumulative work availability

With increasing valve lift the pressure inside the cylinder increases and results in increase in the work availability for every cycle. Though the work availability increases with increase in the valve lift the percentage of work availability to the fuel availability remains constant or decreases.

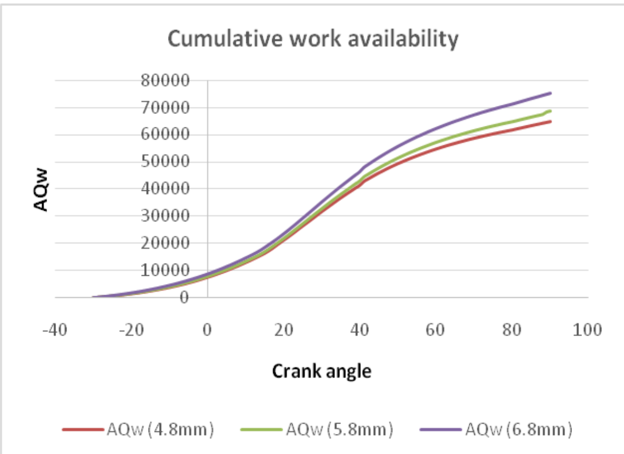

Fig -4.6.2: Cumulative work availability

4.7 Availability Loss terms

While studying engine cylinder availabilities we have to consider various availability terms such as fuel availability, work availability, etc.

4.7.1 Rate of availability loss

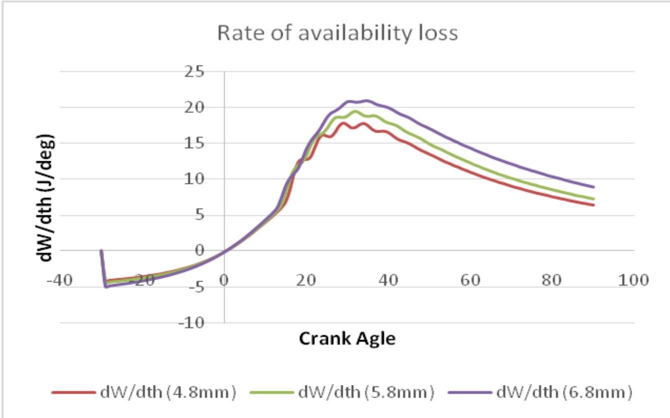

Fig -4.7.1: Rate of Availability loss

4.7.2Cumulative availability loss

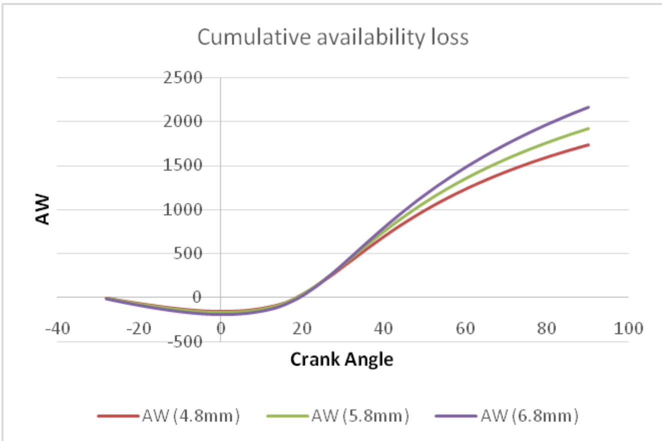

Fig -4.7.2: Cumulative availability loss

As the crank angle is advanced the in-cylinder temperature increases which reduce the combustion irreversibility, but significantly increase the availability loss associated with the heat transfer to the cylinder walls.

5. CONCLUSION

Present study deals with experimental calculation and simulation of rate of heat release and pressure for the diesel engine. The single zone zero dimensional models for direct injection diesel engine for closed cycle for combustion process has been successfully developed. The model is effectively used to estimate the performance of the engine for given operating conditions. Detailed equations are given for the calculation of state properties. Following conclusion can be drawn from present study:

1) The valve lift variation did have effect on the pressure variation inside the engine cylinder.
2) The Wiebe model has given good relationship with the experimental values.
3) The reduction in the valve does enhance the performance of the single cylinder diesel engine

6. FUTURE SCOPE

The existing model that has been presented in this dissertation can be modified and improve in the following items.

1) In this dissertation, constant valve lifts were used as a parameter to do the analysis. A further implementation of this result can be used to make mechanisms to vary valve lift for enhanced engine performance.
2) Single zone model used in this study follows experimental trends of the performance parameter. However program should be developed by using multi zone model to take into account special variation and to get more accurate results in terms of exact values of the output.
3) Using multi zone model, this study can be further extended to predict the effect of considered operating parameters on emission formation.
4) The accuracy in the model can be enhanced by tuning and verifying the parameter must be done.

If some of these improvements are done the complexity increases and therefore the computational time and simulation time will increase.

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