SECOND LAW ANALYSIS OF A DIESEL ENGINE FUELLED WITH DIESEL-ETHANOL BLEND USING DIFFERENT IGNITION DELAY CORRELATIONS

GAURAV P CHAUDHARI[1], M M DESHMUKH[2]

[1] *M.Tech Student, Thermal Engineering, Govt. College of Engineering, Amravati (M.S), India*
[2] *Assossiate. Professor, Department of Mechanical Engineering, Govt. College of Engineering, Amravati (M.S), India*

Abstract - *A mathematical model is developed by using different ignition delay models for compression ignition engine fuelled with the diesel-ethanol blend under different blend percentages. 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 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 modelled the pressure and temperature are predicted for every crank position. The diesel engine is considered as closed system for thermodynamics analysis. The cylinder gases are assumed as ideal gas.The different developed ignition delay model is validated against the data obtained by experimentation at laboratory. This study again elaborated how properties of cylinder charge gases varying with the crank angle position. By performing the experimentation on diesel engine, experimental results have been used with models given by Arrhenious, Wolfer, Watson and Hardenberg. At Last the different availabilities of engine cylinder at varying blend percentages and at constant speed are calculated.It is found that out of total fuel availability supplied, the 40-45% availability is linked with work availability and remaining is the availability loss in the heat transfer and irreversibilities generated during the combustion. As the blend percentages goes on increasing the combustion irreversibilities reduces and heat transfer loss is increased.*

***Key Words*: Second Law Analysis, Ignition Delay, Diesel Engine Modelling , Diesel-Ethanol Blend, Availability, Irreversibility, Heat Release Rate, Pressure Prediction.**

1.INTRODUCTION

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

The second-law analysis provides a more critical and thorough insight into the engine processes by defining the term of availability destruction or irreversibilities 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 irreversibilities 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 irreversibilities term. It was shown that combustion duration, heat release shape, i.e. premixed burning fraction, and injection timing only marginally affect combustion irreversibilities (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 irreversibilities [9]. Second-law application to internal combustion engines are mainly applied for:

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

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. Internal Combustion Engine simulation and mathematical modeling has long been established as an effective tool for studying engine performance and contributing to evaluation and new development. In order to analyse engine performance – that is, evaluate the inefficiencies associated with the various processes – second-law analysis must be applied [3].

1.1.1 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

The need for improving these models was also established by incorporating developments happening in engine designs. The combustion in modern DI diesel engines is mainly divided in two phases (a) a small ignition delay event in which pre-flame activities take place followed by (b) main heat release event in which actual combustion happens. These events are modelled differently considering prominent role of chemical kinetics in ignition delay event and physical mixing rate in heat release event.

(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 that predicts satisfactorily the important instantaneous parameters used by a designer e.g. heat transfer, fuel consumption, and the performance turbocharger and piston. On the same tenor, with the drop in norms for HC and NOx, ignition delay, however small it may be, cannot be neglected while estimating emissions.

(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. Early DI diesel engines operated at relatively low compression ratios and low injection pressures. Hence, they demanded very advanced injection timings in commensurate with the large ignition delay. With the advent of new norms, reduction in ignition delay held the key to solving emission and noise problems. Higher temperature at the beginning of injection by increased compression-ratio reduced the delay period substantially.

(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. Detailed 2-dimensional axisymmetric spray calculations are attempted using the mixing of the injected fuel with the surrounding air entrained due to high shear velocity of the jet as shown in figure 1.1.1. A criterion of stoichiometric burning of the fuel in ignitable elements has been used in these models by spray-mixing approach.

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. The convolution integral of the heat release from the individual packets summed neatly to the net heat release rate as shown in fig. 1.1.2. Due to the absence of universal decay constants for elemental heat-release rates in different types of engines and their operating conditions, the application of this elegant idea posed difficulty.

(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

1.1.2 Availability of a 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 thermomechanical 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 [6]

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

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 brought to p_0 and T_0.

Availability is an extensive property with a value 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 exists 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.

1.1.2 General Availability Balance Equation

For an open system experiencing mass exchange with the surrounding environment, the following equation holds for the total availability on a time basis:

$$\frac{dA_{CV}}{dt} = \int \left(1 - \frac{T_0}{T_j}\right) Q_j - \left(W_{cv} - p_0 \frac{dV_{cv}}{dt}\right) + \sum_{in} m_{in} b_{in} - \sum_{out} m_{out}\, b_{out} - I$$

where:

(a) $\frac{dA_{CV}}{dt}$ is the time rate of change in the exergy of the control volume content (i.e. engine cylinder, or exhaust manifold, etc.).

(b) $\int \left(1 - \frac{T_0}{T_j}\right) Q_j$ is the availability term for heat transfer, with T_j the temperature at the boundary of the system, which in general, is different from the temperature level of a process (although these two temperatures are the same when applying the most usual simulation approach of internal combustion engines operation, i.e. single-zone modeling), and Q_j represents the time rate of heat transfer at the boundary of the control volume. This equation shows that increasing the temperature of a specified energy stream also increases its availability or, the ability of the stream to produce work. This statement is very useful when studying internal combustion engines (particularly compression ignition engines), since here an increase in the fuel–air equivalence ratio ϕresults in an increase in exhaust gases temperatures due to the lean mixtures involved, and thus their potential for work production. Moreover, this equation denotes
that there is a limitation imposed by the second-law of thermodynamics as regards operation and efficiency of thermal engines.

(c) $W_{cv} - p_0\left(\frac{dV_{cv}}{dt}\right)$is the availability term associated with (mechanical or electrical) work transfer.

(d) $\sum_{in} m_{in} b_{in}$ and $\sum_{out} m_{out}\, b_{out}$ are the availability terms associated with inflow and outflow ofmasses, respectively. In particular, the terms b_{in} and b_{out} in Eq. above refer to the flow or stream availability (or exergy) of the incoming and the outgoing cylinder mass flow rates, respectively, given by (neglecting kinetic and potential energy contribution):

$$b = b^{tm} + b^{ch} = h - T_0 s - \sum_i x_i\, \mu_i^0$$

with s_0 the entropy of (cylinder) flow rate were it brought to p_0 and T_0. Flow availability is defined as the maximum work output that can be obtained as the fluid passes reversibly from the given state to a dead state, while exchanging heat solely with the environment.

(e) I is the rate of irreversibility production inside the control volume due to combustion, throttling, mixing, heat transfer under finite temperature difference to cooler medium, etc. Another relation often applied is $I = T_0 S_{irr}$ based on an entropy balance, with S_{irr} denoting the rate of entropy creation due to irreversibilities.[5]

2. Literature Review

H. An, W.M. Yang, J. Li have done numerical modeling on a diesel engine fueled by biodiesel–methanol blends. A modeling study was conducted to investigate the impact of methanol addition on the performance, combustion and emission characteristics of a diesel engine fueled by biodiesel. Good agreements in terms of ignition delay, cylinder pressure and heat release rate predictions were obtained.[5] The simulation results revealed that with partial replacement of biodiesel by methanol, tangible improvement on the cylinder pressure was observed under 10% load condition especially for the case with 5% methanol blend ratio. Whereas, under 50% and 100% engine load conditions, only comparable cylinder pressure curves were seen. In terms of performance characteristics, almost linearly increased indicated thermal efficiency with respect to methanol blend ratio were observed under all the engine load conditions.

C.D. Rakopoulos, E.G. Giakoumis [1] have done second-law analyses applied to internal combustion engines operation. This paper surveys the publications available in the literature the application of the second-law of thermodynamics to internal combustion engines. The availability (exergy) balance equations of the engine cylinder and subsystems are reviewed in detail providing also relations concerning the definition of state properties, chemical availability, flow and fuel availability, and dead state. A detailed reference is made to the findings of various researchers in the field over the last 40 years concerning all types of internal combustion engines, i.e. spark ignition, compression ignition (direct or indirect injection), turbocharged or naturally aspirated, during steady-state and transient operation. Main differences between the results of second- and first-law analyses are highlighted and discussed.

D.B. Lata, Ashok Misra[9] have done analysis of ignition delay period of a dual fuel diesel engine with hydrogen and LPG as secondary fuels. In this study, experiments were performed on 4 cylinder turbocharged, intercooled with 62.5 kW gen-set diesel engine by using hydrogen, liquefied petroleum gas (LPG) and mixture of LPG and hydrogen as secondary fuels. The experiments were performed to measure ignition delay period at different load conditions and various diesel substitutions. The experimental results have been compared with ignition delay correlation laid down by other researchers for diesel and dual fuel diesel engine. It is found that ignition delay equation based on pressure, temperature and oxygen concentration for a dual fuel diesel engine run on diesel-biogas gives variation up to 6.56% and 14.6% from the present experimental results, while ignition delay equation for a pure diesel engine gives 7.55% and 33.3% variation at lower and higher gaseous fuel concentrations, respectively. It is observed that the ignition delay of dual fuel engine depends not only on the type of gaseous fuels and their concentrations but also on charge temperature, pressure and oxygen concentration.

Timothy Bodisco, Philipp Treondle, Richard J. Brown [11] have done inter-cycle variability of ignition delay in an ethanol fumigated common rail diesel engine. An experimental study has been performed to investigate the ignition delay of a modern heavy-duty common-rail diesel engine run with fumigated ethanol substitutions up to 40% on an energy basis. The ignition delay was determined through the use of statistical modelling in a Bayesian framework this framework allows for the accurate determination of the start of combustion from single consecutive cycles and does not require any differentiation of the in-cylinder pressure signal. At full load the ignition delay has been shown to decrease with increasing ethanol substitutions and evidence of combustion with high ethanol substitutions prior to diesel injection have also been shown experimentally and by modelling. Whereas, at partial load increasing the ethanol substitutions have increased the ignition delay. A threshold absolute air to fuel ratio (mole basis) of above ~110 for consistent operation has been determined from the inter-cycle variability of the ignition delay, a result that agrees well with previous research of other in-cylinder parameters and further highlights the correlation between the air to fuel ratio and inter-cycle variability.

2.1.1 Parameter Calculation

1) Specific heat ratio

Specific heat ratio varies with charge temperature and composition and is known to have a very significant effect on the calculated heat release energy. Ideally, gamma would be varied with fuel specification, air to fuel ratio (AFR), exhaust gas recirculation (EGR), charge pressure and charge temperature but for present calculation gamma is usually made a function of temperature only.
Gamma (γ) is the ratio of specific heats. A low value of gamma produces heat release value that is too high and a heat release rate that is negative after the completion of combustion.

$\gamma = 1.35 - 6\times10^{-5}*T + 10^{-8}*T^2$

γ is also dependent on equivalence ratio, Φ. The effect of ignoring this term is an error of up to +0.015 in gamma ($0.8 < \Phi < 1.2$).

2) Calculating cylinder volume

Hairuddin et al. [12] used following formulas for obtaining cylinder volume from crank angle for a slider -crank mechanism.

$$V = \frac{V_d}{r-1} + \frac{V_d}{2} \times (R^2 + 1 - cos\Theta - \sqrt{R^2 - sin\Theta^2})$$

Where,
Vd is the displacement volume,
r the compression ratio,
Θ is the instantaneous crank angle position (Θ=0 is at TDC) and
R is the ratio of the connecting rod length to the crank radius.

3) Instantaneous surface area is given by

$$A = \Pi/2 \times B^2 + \Pi \times B \times L/2 \times (R + 1 - cos\Theta + \sqrt{R^2 - sin\Theta^2})$$

4) Mean gas temperature
The mean gas temperature is required for the calculation of heat release.
For a polytropic process
PV^n = constant

$$\frac{T_2}{T_1} = (\frac{V_1}{V_2})^{n-1} = (\frac{P_2}{P_1})^{\frac{n-1}{n}}$$

Hence

$$T_2 = T_1 \times (\frac{V_1}{V_2})^{n-1} = T_1 \times (\frac{P_2}{P_1})^{\frac{n-1}{n}}$$

For a known reference location, such as inlet valve closure:

$P_{ref} \times V_{ref} = m \times R \times T_{ref}$

To calculate the temperature at an arbitrary position between inlet valve closure and exhaust valve opening

$P_{cal} \times V_{cal} = m \times R \times T_{cal}$

5) Rate of pressure change

The rate of pressure rise is calculated using a simple numerical differentiation:

$$\frac{dP}{d\theta} = \frac{(P(i+1)-P(i-1))}{(\theta(i+1)-\theta(i-1))}$$

6) Convective heat transfer coefficient

If apparent net heat release rate so obtained already provides only an approximate answer to the thermal energy conversion, the combustion efficiency evaluation through the calculation of fuel burning rate is still less reliable because the uncertainty related to heat transfer modelling which in this case is involved. Heat transfer through cylinder walls is generally modelled by means of semi -empirical correlations carried out from experiments and based on the laws of similarity for turbulent tube flow. In this way heat flux is expressed as:

$dQ_w/d\theta = h_i \times A_i \times (T_i - T_w)$

The heat transfer coefficient h is calculated by Hohenberg's correlation as follows:

$h = 3.26 \times P^{0.8} \times T^{-0.4} \times V^{-0.06} \times (V_p + 1.4)^{0.8}$

Where,
P = Pressure
T = Temperature
V = Volume of the cylinder
V_p = Mean piston speed

$V_p = 2LN/60$

7) Wall temperature

Wall temperature for heat transfer calculations is determined by equivalence ratio

$T_{wall} = 400$ K for $\Phi < 0.8$

8) In order to calculate the apparent net heat release rate, equation can be solved by assigning as an instantaneous known term pressure measured datum and its calculated rate of change together with volume and its derivative versus crank angle.

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} \times P \times \frac{dV}{d\theta} + \frac{V}{\gamma-1} \times \frac{dP}{d\theta} + \frac{dQ_w}{d\theta}$$

9) Pressure can be predicted using heat release rate by integrating above equation for the pressure.

$$\frac{dP}{d\theta} = \frac{\gamma-1}{V}\left(\frac{dQ}{d\theta} - \frac{dQ_w}{d\theta}\right) - \gamma \times \frac{P}{V} \times \frac{dV}{d\theta}$$

10) Mass Fraction Burn

Mass fraction burn is calculated on the concept that the difference in pressure in the firing and motoring pressure is in the proportion of mass fraction burn. The difference is calculated as follows.

$$dP_i = P_i - P_{i-1}(\frac{V_{i-1}}{V_i})^n$$

$$dP_{total} = \sum(dP)$$

$$MFB = x_b = \frac{dP_i}{\Sigma(dP)}$$

2.1.2 Ignition Delay Correlations

For diesel fuels a reasonable estimate of the delay, ID is achieved by Wolfer. The time taken for visible fire to appear in the pre-mixed zone of spray is a strong function of pressure and temperature of the ambient.

For diesel fuels a reasonable estimate of the delay, ID is achieved by Wolfer, [6]:-
ID=3.45 $\exp(2100/T_m)*P_m^{-1.02}$

Where, T_m and P_m are the mean temperature and pressure of the ambient during ignition delay.

Classical Arrhenius type model for Ignition Delay and its extension to other fuels values of pressure and temperature are necessary to predict ignition delay. If this equation has to be developed as design tool, then it is necessary to predict pressure and temperature precisely for required engine operating condition. Considering pressure and temperature at TDC position will ignore effect of injection timing.
The Arrhenius type of equation is used to describe ignition delay [1]:-

$$\zeta_{id} = A.\phi^{-k}.P^{-n}.exp\left(E_a/R_u T_{cyl}\right)$$

Where, ζ_{id} = Ignition delay
Φ = Equivalence ratio
E_a = Activation energy
T_{cyl} =Cylinder charge temperature
R_u = Gas constant
A , k and n = Empirical constants.

The Cetane number of biodiesel is better than that of diesel. Therefore, the Cetane number truly represents the compression ignition quality of fuel. Therefore, this parameter needs to be incorporated while developing a new model for predicting the ignition delay especially with fuels containing oxygen. Here, the correlation developed by Hardenberg and Hase can be employed. It is given by, [3]

$$\zeta_{id}\ (A) = (0.36 + 0.22\overline{S_p})exp[E_A(\frac{1}{\bar{R}.T}) - \frac{1}{17190}).(21.2/(P-12.4))^{0.63}]$$

Where, $\overline{S_p}$ = piston speed (m/s)

$\bar{R}$ = Universal gas constant (8.3143 J/mol)

E_A = Apperent activation energy

$$E_A = \frac{618840}{CN+25}$$

Where, CN = Cetane number

Watson developed equations for fuel energy release appropriate for diesel engine simulations. In their development, the combustion process starts from a rapid premixed burning phase (represented by function f_1), followed by a slower mixing-controlled burning phase (represented by function f_2), with both functions empirically linked to the duration of ignition delay (~τid) and the duration of combustion ($\Delta\theta_{comb}$).

The Watson model is given by [3]

$$\zeta_{id} = A(BP)^{-N}.P_{SOC}^{-B}.exp(\frac{E}{RuTsoc})$$

The constants A, N and B are adjustable.

3. System Development

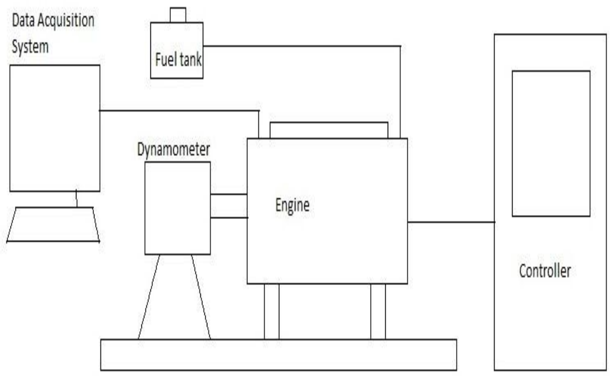

Fig. 3.1 Block diagram of Experimental setup

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

Table 3.2 Engine Specification

3.1 Experimental procedure

The aim of the experiment is to obtain the pressure and crank angle data for varying blend percentages at constant speed. Constant fuel supply of Diesel-Ethanol Blend was made through the fuel tank which later passed through the fuel conditioning unit.

The blend percentages is varied through by 10,20 & 30% and 100% pure diesel is used.The pressure-theta data is obtained from the computer which is connected to the controllers and pressure theta sensors.

For each blend percentages 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.

4 Results

4.1 Engine cylinder pressure at various loads

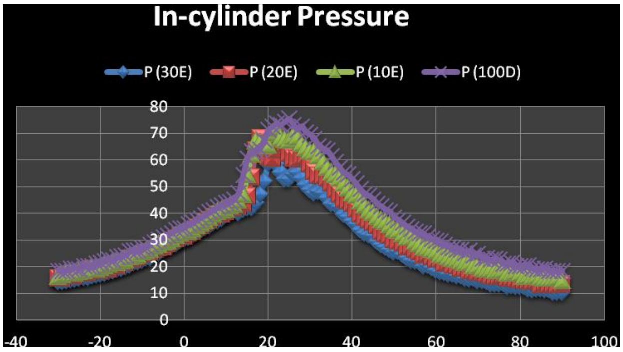

Fig 4.1 Engine cylinder pressure Vs Crank Angle

It is observed that along with increase in load the pressure inside the cylinder is increased. This is because of increase in the fuel supply for the increased load. 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 load. The deviation from the motoring curve marks the start of combustion. It is also seen start of combustion point collies closer to the TDC for lowering the load and it is because of the ignition delay that is decreases with increased in the load.

Since with increase in blend percentage the pressure inside the cylinder decreases. This is because of increase in the fuel supply for the increased blend percentage. As, in Power/Expansion stroke, the piston moves from Top Dead Center to Bottom Dead Center inside the engine cylinder, the high temperature, high pressure gases pushes the piston down which forces the crank-shaft to rotate. Thus due to ethanol-diesel blend, pressure inside the cylinder decreases which leds to decrease in incylinder temperature causes reduction in work availability.

4.2 Specific heat ratio

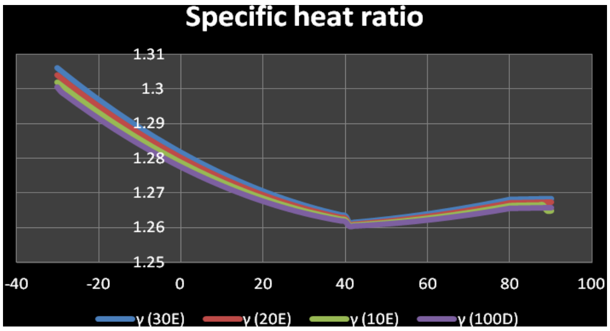

Fig 4.2 Sp. Heat ratio Vs Crank Angle

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 4.2 with increasing crank angle, the value of ratio of specific heat is goes on decreasing. Upto 40° ATDC.

4.3 Rate Of Heat Release

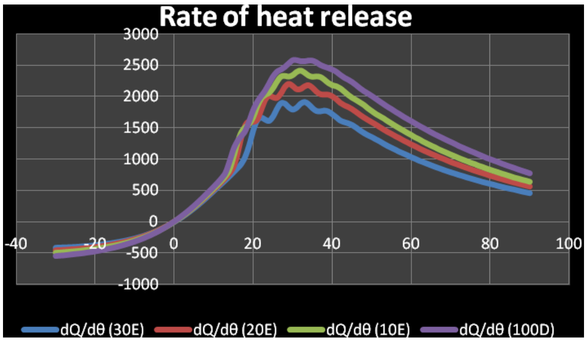

Fig 4.3 Heat Release Rate Vs Crank Angle

Rate of heat release is modelled by Weibe Function.From the figure 4.3, it is observed that,the rate of heat release is large for full load condition.

This is due to increase in fuel consumption rate with increase in the loadind over the engine.Thus, for more fuel consumption and increased load, there will be proportionate raise in heat release rate.

This is due to decrease in fuel consumption rate with increase in the blend percentage in the engine cylinder.Thus, for less fuel consumption and increased blend percentage, there will be proportionate fall in heat release rate. Since, Heating value of our commercial diesel fuel used is approximately 44800KJ/Kg and that of the Ethanol is 29700 KJ/Kg. The Blend of these fuels in any quantity would leads to the cumulative decrease in the Heating value as compared to the pure diesel fuel.

Therefore, the heat release rate is more for pure diesel fuel combustion as compared to the blended fuel.

4.4 Heat Transfer Coefficient

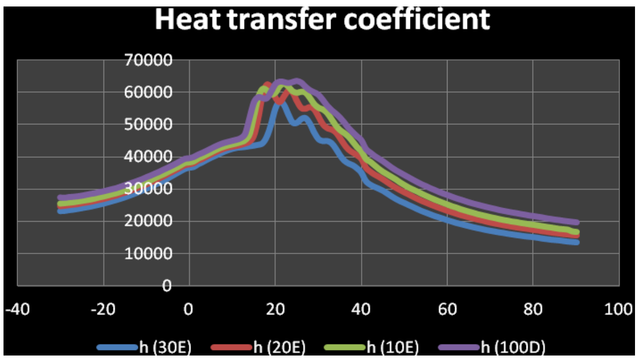

Fig 4.4 Heat Transfer coefficient Vs Crank Angle

Heat transfer coefficient is calculated by Hohenberg's heat transfer coefficient. As seen from fig., as load increases, the heat transfer coefficient increases upto 20° ATDC and beyond that it tends to decrease.Since There will be increased in the heat flux with increasing load.

4.5 Work Availability Rate

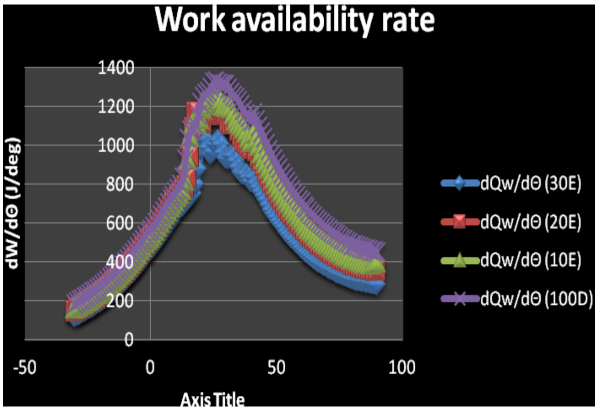

Fig 4.5 Work Availability Rate Vs Crank Angle

4.6 Cumulative Work Availability

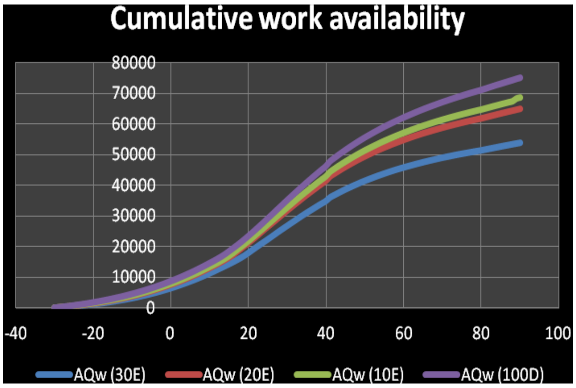

Fig 4.6 Cumulative Work Availability Vs Crank Angle

Figure 4.5 and figure 4.6 shows 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 otherwise up to top dead centre just after the TDC at the start of expansion stroke and due to combustion also the work availability rises. In this region the fuel availability mainly converts in to work availability

With increasing load 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 load the % of work availability to the fuel availability remains constant or decreases.

The blended ethanol in Ethanol-Diesel blend decreases the maximum temperature 'T_3' at the start of Expansion/Power Stroke as compared to the maximum temperature 'T_3' attainted by pure diesel fuel. Due to decrease in 'T_3', the difference (T_3-T_4) also decreases .Where, 'T_4' is the Temperature at the end of Expansion or at the initialization of Exhaust Stroke. Thus the Work availability decreases proportionality with increase in the percentage of ethanol-diesel blend. Thus, there would be more availability destruction with increase in the percentage of ethanol blend.

4.7 Rate Of Availability Loss

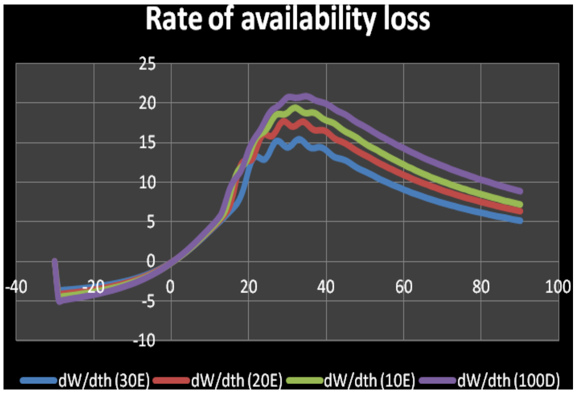

Fig 4.7 Rate of Availability Loss Vs Crank Angle

4.8 Cumulative Availability Loss

Fig 4.8 Cumulative Work Availability Vs Crank Angle

Figure 4.7 and figure 4.8 depicts the cumulative availability loss in heat transfer process. It is clear that the as the load is increased the pressure and temperature in the cylinder increase consequently the heat transfer through the cylinder walls increase.

In fact heat loss from engine cylinder walls contains a significant amount of availability, which is almost completely destroyed only after this is transferred to the cooling medium.

As, the maximum temperature 'T_3' in power stroke decreases with increase in the percentage of the blend, the difference (T_3-T_4) decreases. These decrease in (T_3-T_4) implies the heat loss in cylinder wall during power stroke.Thus, the availability loss and cumulative availability loss increases with in the percentage of ethanol-diesel blend.

4.9 Pressure Prediction by Various Ignition Delay Correlations

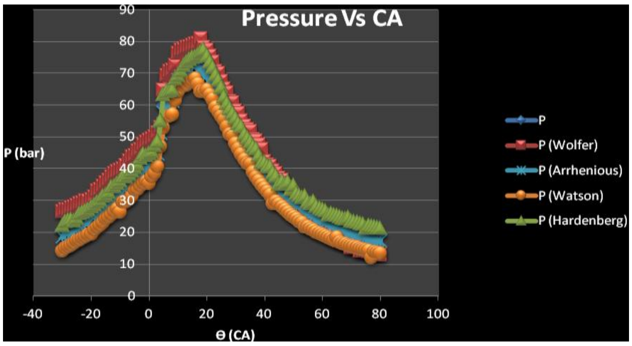

Fig 4.9 Cylinder Pressure (Predicted And Experimental) Vs Crank Angle

The figure above shows the average experimental pressure and variations of predicted pressures from Ignition delay correlations.

It can also be seen that either K or x affect the main combustion period, i.e. the diffusion controlled regime, having a marked effect on the peak pressure or so the work output achieved. Observing the fuel preparation rate equation, one can see that high values of K and low values of the exponent x correspond to higher preparation rates, a fact leading to improved combustion and so higher peak pressures in a rather smooth way.

The Watson model consistently over-predicted the length of the ignition delay under transient conditions, even though the initial steady state values were coinciding with the predictions of the new model.

5. CONCLUSIONS

Present study deals with experimental calculation and simulation of rate of heat release and pressure for the diesel engine fuelled with 10%,20%,30% diesel-ethanol blend and pure diesel fuel. 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 Wolfer model was found to be in very close relation with experimental values, as it is more calibrated than the other models. It gives the result values near to the accurate.

2) The Arrhenious model is also giving good relationship with experimental values. It uses equivalence ratio term which has good impact on the accuracy.

3) Various Availability Terms are studied with their dependence over In-cylinder Pressure and percentages ethanol-diesel blends.

REFERENCES

[1} C.D. Rakopoulos, E.G. Giakoumis, "Second-law analyses applied to internal combustion engines operation", Progress in Energy and Combustion Science 32 (2006) 2–47

[2] Maro JELIĆ and Neven Ninić, "Analysis of internal combustion engine thermodynamic using the second law of thermodynamics", ISSN 0562-1887, Strojarstvo 50 (2) 85-94 (2008)

[3] Aysegul Abusoglu, Mehmet Kanoglu, "First and second law analysis of diesel engine powered cogeneration systems", Energy Conversion and Management 49 (2008) 2026–2031

[4] Maher M Abou Al-Sood, Mahmoud Ahmed and Yousef M Abdel-Rahim, "Rapid thermodynamic simulation model for optimum performance of a four-stroke, direct-injection, and variable-compression-ratio diesel engine", International journal of energy and environment Engineering 2012, 3:13

[5] Frank A. Albinil and Elizabeth D. Reinhardt, "Modeling Ignition and Burning Rate of Large Woody Natural Fuels", Int. J. Wikiiand Fire 5(2): 81 -9I.1995

[6] C. D. Rakopoulos and E. G. Giakoumis, "Speed and load effects on the availability balances and irreversibilities production in a multi-cylinder turbocharged diesel engine", Applied Thermal Engineering Vol. 17, No. 3. pp. 299-313, 1997

[7] Zhang, Zuohua Huang, Liangjie Wei, Jiaxiang Zhang, Chung K. Law, "Experimental and modeling study on ignition delays of lean mixtures of methane, hydrogen, oxygen, and argon at elevated pressures", Combustion and Flame 159 (2012) 918–931

[8] D.B. Lata, Ashok Misra, "Analysis of ignition delay period of a dual fuel diesel engine with hydrogen and LPG as secondary fuels", International journal of hydrogen energy 36 (2011) 3746-3756

[9] Fadila Maroteaux, Charbel Saad, "Diesel engine combustion modeling for hardware in the loop applications: Effects of ignition delay time model", Energy 57 (2013) 641-652

[10] Fadila Maroteaux, Charbel Saad , Fabrice Aubertin, "Development and validation of double and single Wiebe function for multi-injection mode Diesel engine combustion modeling for hardware-in-the-loop applications", Energy Conversion and Management 105 (2015) 630–641

[11] Claude Valery Ngayihi Abbe, Robert Nzengwa, Raidandi Danwe, Zacharie Merlin Ayissi, "A study on the zero Dimensional phenomenological model for diesel engine simulation: Application to combustion of Neem methyl esther biodiesel", Energy Conversion and Management 89 (2015) 568–576

[12] John B. Heywood, "Internal Combustion Engine Operation", McGraw-Hill, 2007