Second Law Analysis of Compression Ignition Engine fuelled with Ethanol Blending by Using Heat Transfer Models

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Abstract - *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. The heat transfer model is one of them, which can be very useful to understand engine processes. A single zone, zero dimensional models is developed for simulation. Four different types of heat transfer models are taken to compare to find out the best of them. The diesel engine fuelled with ethanol blends like 10%E, 20%E, 30%, and pure diesel 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. It is found that Hohenberge correlation is best to calculate the engine cycle results. Eichelberg's model can also be a good choice. Tuning is required for the Woschni model to have good accuracy.*

Key Words: **Second law analysis, Heat Transfer Models, Diesel Engine, Engine Modelling**

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 [3]. 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 [1]. 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 useful 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 [10].

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 [9]. 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 [10].

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 [23]:

(1) Developing a more complete understanding of the process under study from the discipline of formulating the model;
(2) Identifying key controlling variables to provide guidelines for more rational and therefore less costly experimental development efforts;
(3) 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;
(4) Providing a rational basis for design innovation.

Model differently considering prominent role of chemical kinetics in ignition delay event and physical mixing rate in heat release event. [24]

(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.

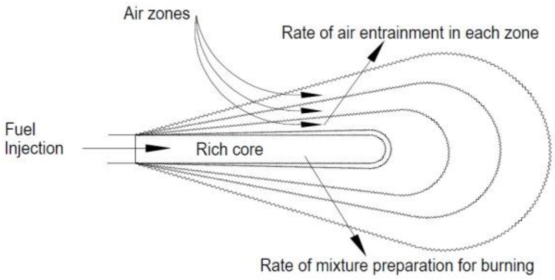

Figure 1.1.1: Multi Zone Spray Model[24]

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 [16],

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

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.

1.3Availability 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 (5)

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 consists 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. 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 wall temperatures, we are deal 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 and has continually been further, developed. Heat transfer to the walls of the combustion chamber, that is the cylinder liner, the piston, and the cylinder head, is calculated from:

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

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

2. LITERATURE REVIEW

Modelling of combustion process in CI engine is carried out by many authors and with their continuous research they have given some correlations which are highly useful for the new research scholars. The modelling of combustion process is mainly divided in following parts:

2.1 Modelling of combustion processes

2.1.1 Ignition Delay

Ignition delay in direct injection diesel engines is of great interest to researchers and engineers because of its direct impact on the intensity of heat release immediately following auto ignition, as well as its indirect effect on engine noise and pollutant formation. The delay period is composed of a physical delay, encompassing atomization, vaporization, and mixing, coupled with a chemical delay, a result of pre-combustion reactions in the fuel/air mixture. The two time scales are not simply additives, but are occurring simultaneously. Detailed ignition models exist (eg. Agarwal and Assanis, 1998), but due to the complexity of the in cylinder physical and chemical processes, can only provide ignition delay trends for practical fuels. Numerous steady state ignition delay correlations have been proposed based on experimental data in constant volume bombs, steady flow reactors, rapid compression machines and engines. Many of those correlations use an Arrhenius expression similar to that proposed by Wolfer:

$$\tau = A_p^{-a} Exp[\frac{E_a}{R_u T}]$$

Where p and T are pressure and temperature averaged between start of injection and start of combustion, E_a is activation energy, R_u is gas constant, and A, n are adjustable constants.

2.1.2 Heat transfer coefficient

Over the years many papers have been published aiming to quantify the heat transfer coefficient to easily measured or derived engine parameters. Some of the most common functions used are implemented in the combustion analysis software and are presented below.

1) Annand

$$h = \frac{a\lambda}{B} R_e^{0.7} + c\frac{(T^4 - T_{wall}^4)}{T - T_{wall}}$$

where, 0.35<a<0.8, c=0 during intake and compression, c=0.576σ for CI engine combustion and expansion, c=0.075σ for SI engine combustion and expansion, σ=5.67 x 10^{-8} $W.m^{-2}.K^{-4}$

2) Woshni

$$h = 129.8B^{-0.2}P^{0.8}T^{-0.53}(C_1 V_p + C_2 \frac{V_s T_{ref}}{P_{ref} V_{ref}}(P - P_{motored}))^{0.8}$$

where, C_1=6.18 in scavenging period, C_1=2.28 in compression, combustion and expansion, C_2=0 in scavenging period and compression, C_2=3.24 x 10^{-3} in combustion and expansion, C_2=6.22 X 10^{-3} in combustion and expansion(IDI engines), V_s displacement per cylinder, $P_{motored}$ cylinder pressure of the motored engine[bar], T_{ref} temperature in the cylinder at intake valve closing(IVC), P_{ref} pressure in the cylinder at IVC[bar]

3) Hohenberge

$$h = 130V^{-0.06}P^{0.8}T^{-0.04}(v_p + 1.4)^{0.8}$$

4) Eichelberge

$$h = 2.185 C_m^{0.33}(PT)^{0.5}$$

Where, C_m is the mean piston speed of the engine in m/s

2.2 Review from research papers; first and second law analysis

C. D. Rakopoulos and E. G. Giakoumis [1] have studied speed and load effects on the availability balances and irreversibility's 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 irreversibility's. 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 [2] 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.

E.G. Giakoumis [3] completed the work on cylinder wall insulation effects on the first and second-law balances of a turbocharged diesel engine operating under transient load conditions. To include the second-law balance, an experimentally checked transient diesel engine simulation code has been expanded. The improved model of Annand and Ma is employed to simulate heat loss to the cylinder walls. It's disclosed that second-law analysis should be applied to evaluate the inefficiencies related to the assorted processes.

Orhan Durgun and Zehra Sahin [4] completed the work on multi-zone combustion modeling for the prediction of diesel engine cycles and engine performance parameters. In this, quasi-dimensional phenomenological combustion model developed by Shahed and Ottikkutti are used and developed with new assumptions. Annand's correlation is employed to calculate the instantaneous total heat transfer from the cylinder contents to the combustion chamber walls.

F. Payri, P. Olmeda, J. Martín and A. García [5] have done a complete 0D thermodynamic predictive model for direct injection diesel engines. In this, a variation of the expression given by Woschni is employed to calculate the heat transfer constant. To improve the original model of Woschni, many efforts are taken. Finally the justification have been given to include the proposed sub-models for correct prediction of the engine performance and it's been quantified the error within the simulation results if these sub-model weren't enclosed.

Mahmoud Ahmed and Maher M Abou, Al-Sood Yousef M Abdel-Rahim [6] has done the work on developed rapid thermodynamic simulation model for optimum performance of a four-stroke, direct-injection, and variable-compression-ratio diesel engine. For the performance of a four-stroke, DI diesel engine a thermodynamic simulation model is developed. The Eichelberg model as changed by Rakopoulos and Hountalas is employed. The comparisons between foreseen and experimental results for various engines, in operation beneath different conditions show that there's a decent concurrence between foreseen and measured values.

Junnian Zheng and Jerald A. Caton [7] have done second law analysis on a low temperature combustion diesel engine to study the effect of injection timing and exhaust gas recirculation. They work on engine cycle simulation incorporating the second law of thermodynamics to gauge the energy and exergy distribution of different processes in the low temperature combustion diesel engine. The Hohenberg correlation is employed to calculate the cylinder heat transfer. By comparison to the standard injection timing cases, they observed that the late injection timing cases show lower proportion of heat transfer exergy and better proportion of net flow exergy, which means for the exhaust recovery process for the late injection cases a lot of exergy can be used.

Claude Valery Ngayihi Abbe, Robert Nzengwa, Raidandi Danwe, Zacharie Merlin Ayissi, Marcel Obonou [8] have done a study on the 0D phenomenological model for diesel engine simulation with application to combustion of Neem methyl esther biodiesel. The Woschni approach is planned to formulate heat transfer model. They state that the model is ready to predict engine operation characteristics in several in operation points with fair accuracy and few adjusting numerical coefficients. The results obtained were found satisfactory in terms of accuracy, simplicity and computer price.

C.D. Rakopoulos and E.G. Giakoumis [9] have done second-law analyses applied to internal combustion engines operation. This paper surveys the publications offered within the literature regarding the applying of the second-law of thermodynamics to IC engines. A close reference is created to the findings of different researchers within the field over the last forty years regarding all sorts of IC engines, i.e. SI, CI, turbocharged or naturally aspirated, throughout steady-state and transient operation. Main variations between the results of second and first law analyses are highlighted and mentioned. It's believed that engine operation optimization supported the second-law of thermodynamics will function as a strong tool to the engine designer.

Maro Jelić and Neven Ninić [10] have done analysis of internal combustion engine thermodynamic using the second law of thermodynamics. They studied work of various authors. They expressed that one in every of the foremost appropriate ways in which in analysis of energy degradation is application of the second law of thermodynamics in analysis of the method in combustion engines and by applying the numerical simulations in modeling the IC engine processes in conjunction with the analysis by the second law of thermodynamics, we tend to get an awfully potent tool for good insight and improvement of SI and CI engines achieving lower fuel consumption and lower emissions.

Aysegul Abusoglu and Mehmet Kanoglu [11] have done first and second law analysis of diesel engine powered cogeneration systems. In this article, the thermodynamics analysis of the prevailing diesel engine cogeneration system is performed. The exergy analysis is aimed to gauge the exergy destruction in every element and in exergetic efficiencies also. They explicated that such data are often employed in development of the new energy economical systems and for increasing existing system's efficiency. This elaborated analysis give a strong and systematic tool for characteristic all price sources and for optimization of design of cogeneration systems powered by diesel engine.

A. Aziz Hairuddin, Andrew P. Wandel and Talal Yusaf [12] have done effect of different heat transfer models on a diesel homogeneous charge compression ignition engine. The mechanism of chemical kinetics influences the combustion with some cylinder wall heat losses. The result of various heat loss models in diesel HCCI engine should be investigated more. A single-zone model was utilized in this study together with three different heat loss models: Woschni, modified Woschni, and Hohenberg correlations. They found that the distinction in heat loss models ends up in a giant distinction within the heat flux, and therefore the modified Woschni model has the best heat flux among these models. The study revealed that the modified Woschni model created additional correct results, whereas the Woschni and Hohenberg models need additional standardization of constants before they will utilized in a diesel HCCI engine.

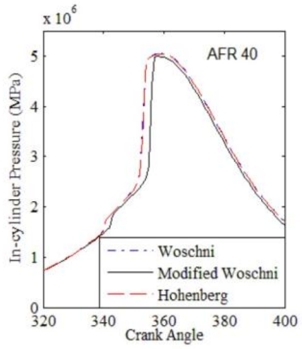

Fig. 2.2.1: In-cylinder pressure comparison with different AFRs and different heat transfer coefficient models [12]

Adrian Irimescu [13] worked on convective heat transfer equation for turbulent flow in tubes applied to internal combustion engines operated under motored conditions. He developed an equation for the case of turbulent flow in tubes was applied for the study of convective heat transfer in IC engines. Calculated average heat flux values below motorised conditions were compared to measurements, results offered within the literature obtained by employing a CFD code and also the models of Woschni and Annand. They found out that Annand's model was more accurate than the Woschni model.

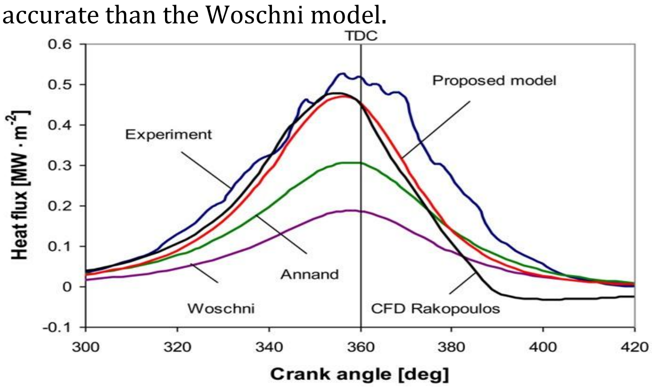

Fig.2.2.2: Comparison of measurements with calculated heat flux using different models [13]

Junfeng Zhao and Junmin Wang [14] have studied Control-oriented multi-phase combustion model for biodiesel fueled engines. In this paper, the combustion characteristics of Diesel and biodiesel fuels are investigated and compared. Based on the experimental information obtained from a medium-duty diesel engine, a multi-phase combustion model, that is applicable to each Diesel and biodiesel fuels, is developed and it shows satisfactory accuracy at intervals of test range during this study. The Hohenberg correlation is used. Through a grey-box parameter identification approach, a group of Wiebe coefficients, that are partially thermodynamics-based, are found; then the model will have an affordable accuracy for the vary of experiments conducted.

R. Sindhu, G. Amba Prasad Rao and K. Madhu Murthy [15] have done the work on thermodynamic modeling of diesel engine processes for predicting engine performance. They focus on computational studies. The paper deals with the modeling of diesel engine processes considering heat losses, and variable specific heats using double-Wiebe function for the heat release. The heat transfer coefficient is given by Woschni model. It was found that early injection timing leads to higher levels of pressure and temperature in the cylinder.

Junseok Chang, Orgun Güralp, Zoran Filipi, and Dennis Assanis [16] presented the paper on new heat transfer correlation for an HCCI engine derived from measurements of instantaneous surface heat flux. An experimental study had been carried out to provide qualitative and quantitative insight into gas to wall heat transfer in a gasoline fueled Homogeneous Charge Compression Ignition (HCCI) engine. Heat flux measurements were used for assessing several existing heat transfer correlations. Woschni, Hohenberg, and Annand & Ma models are examined. Among the three global models tested, Hohenberg appears to be the closest to the measured profile, while Woschni seems to be the least accurate, since it is under-predicting heat flux before combustion and over-predicting during main combustion.

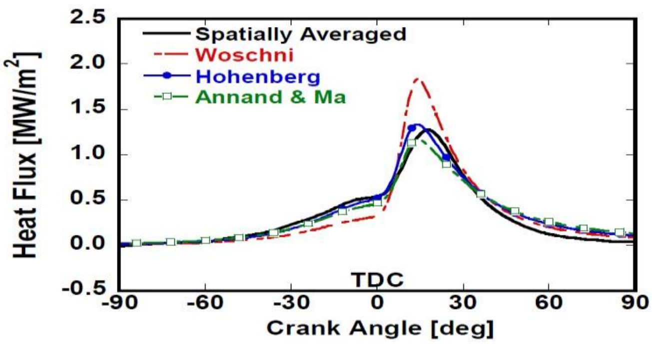

Fig. 2.2.3: Comparison of measured spatially-averaged heat flux and predictions of from several global models [16]

John Agudelo, Elkin Gutiérrez and Pedro Benjumea [17] has study the experimental Combustion Analysis of a HSDI Diesel Engine Fuelled with Palm Oil Biodiesel-Diesel Fuel Blends. In this work a detailed combustion diagnosis was applied to a turbocharged automotive diesel engine operating with neat palm oil biodiesel (POB), No. 2 diesel fuel and their blends at 20 and 50% POB by volume (B20 and B50 respectively). Heat transfer was calculated using the correlation proposed by Woschni adjusting the constants to the engine by means of energy balances. Additionally, brake thermal efficiency, combustion duration, maximum mean temperature, temperature at exhaust valve opening and exhaust gas efficiency decreased; while the peak pressure, exergy destruction rate and specific fuel consumption increased.

Mohand Said Lounici, Khaled Loubar, Mourad Balistrou and Mohand Tazerout [18] have done the work on investigation on heat transfer evaluation for a more efficient two-zone combustion model in the case of natural gas SI engines. Two-zone model was one of the most important engine simulation tools, especially for SI engines. However, the pertinence of the simulation depends on the accuracy of the heat transfer model. The fuel energy was transformed to heat loss from the chamber walls. Also, knock appearance is closely related to heat exchange. From the experimental measurements were carried out for comparison and validation. The effect of correlation choice has been first studied. The most known correlations have been tested and compared, like Eichelberg, Woschni, Hohenberg, Sitkei, Annand. It is found that Hohenberg's correlation is the best choice.

3. SYSTEM DEVELOPMENT

3.1 Experimental Model

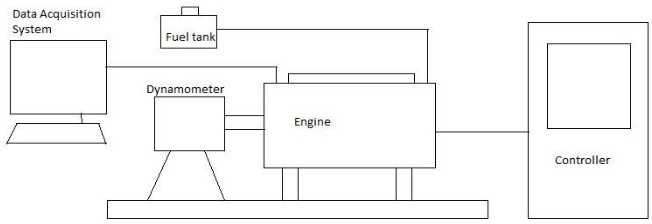

Figure 3.1.1: Block diagram of Experimental setup

Specification 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 3.1.2. 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 load 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 load, 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 loads 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 load is calculated.

4. PERFORMANCE ANALYSIS

4.1 Experimental Analysis

For the engine combustion study, the engine cylinder pressure plays an important role. So, at beginning the engine cylinder pressure at constant loads (50%) and at constant speed(1500 rpm) is studied for the ethanol blending likes 10%E, 20%E, 30%E and 100%D while the fuel consumption per hour is varying. The following graphs show the experimental analysis on ethanol blends using single cylinder four stroke diesel engines.

4.1.1 Engine cylinder pressure vs. Crank angle

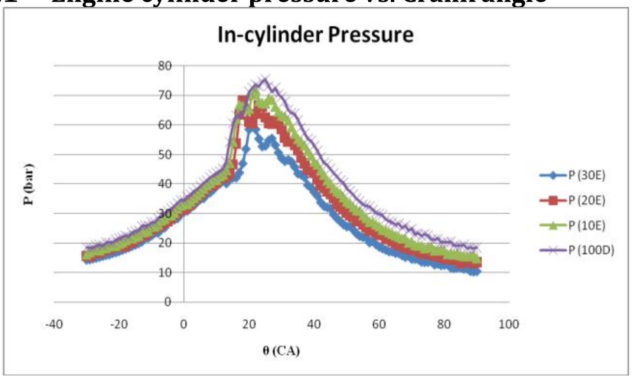

Figure 4.1.1: Experimental pressure vs. crank angle

It is observed that along with increase in ethanol blends, the pressure inside the cylinder is decreased. This is because of increase in the fuel supply for the increased ethanol blends are shown in fig. It is noted that the peak of firing pressure curve is shifted slightly towards the positive side of TDC at constant load and at 1500 rpm. 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 lowering blends and it is because of the ignition delay decreases with at constant speed.

4.1.2 Determining availability terms

While studying engine cylinder availabilities, various availability terms such as fuel availability, work availability, availability loss in heat transfer through engine cylinder etc., have to be considered.

Work availability

Figure 4.1.2 and figure 4.1.3 shows the term rate of work availability and cumulative work availability with the crank angle. From below 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.

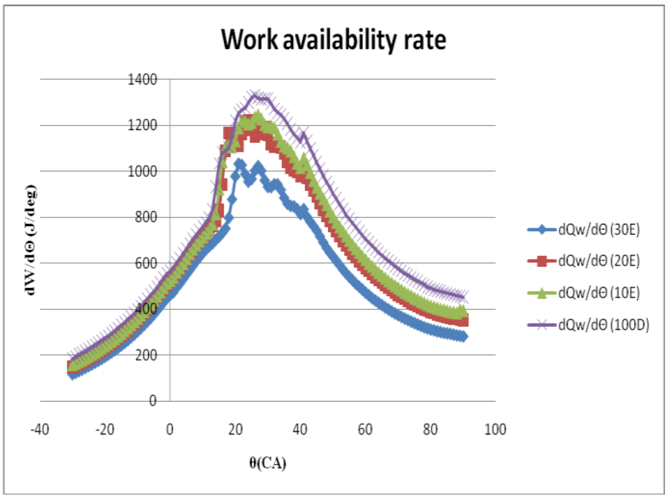

Figure 4.1.2: Work availability rate vs. crank angle

From the above fig4.1.2, increasing the blend the pressure inside the cylinder increases and results in the work availability for every cycle. Though the work availability increases with increase in the blends the % of work availability to the fuel availability remains constant or decreases.

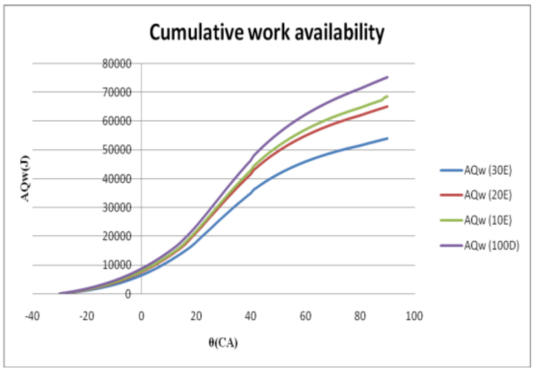

Figure 4.1.3: Cumulative work availability vs. crank angle

From the above fig. Shows cumulative work availability vs. Crank angle, after the top dead centre the cumulative work availability increases with increasing the crank angle. The maximum cumulative work availability shows for the pure diesel and less for the 30%E blends.

4.1.4 Availability loss in heat transfer through engine cylinder

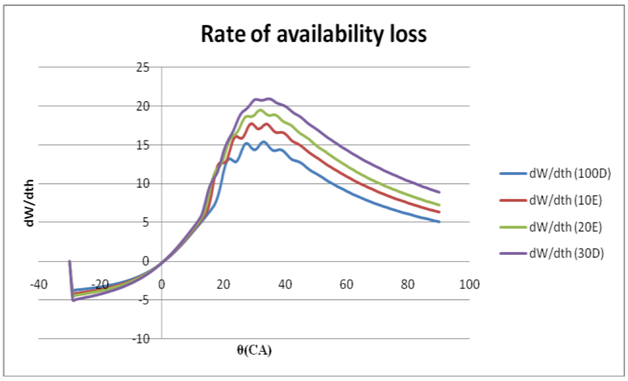

Figure 4.1.4: Availability loss through heat transfer vs. crank angle

Figure 4.1.4 and figure 4.1.5 depicts the cumulative availability loss in heat transfer process. It is clear that the as the fuelled ethanol blends are increased the pressure and temperature in the cylinder increase consequently the heat transfer through the cylinder walls increase. As in this study the speed of the engine was kept constant so the time duration for heat transfer was same at constant load condition and at constant speed the heat transfer is only the function of charge temperature and convective heat transfer coefficient which in turn again depends on the pressure and temperature of in-cylinder gases. Therefore parameters which cause the increase in the temperature and pressure of the in-cylinder gases show the increase in the heat transfer loss from engine cylinder.

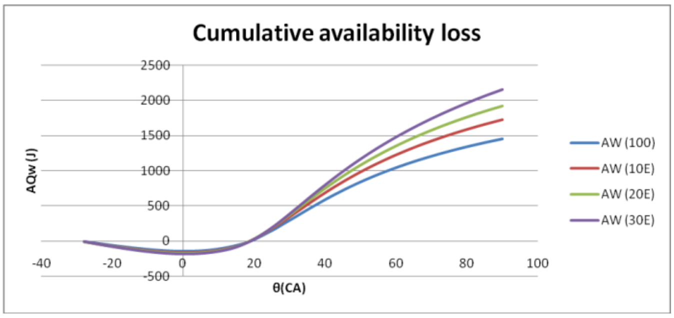

Figure 4.1.5: Cumulative availability loss through heat transfer vs. crank angle

Fig.4.1.5: shows cumulative availability loss vs. Crank angle, the different fuelled ethanol blend the maximum cumulative availability loss shows in 30%E blends. For the pure diesel shows the less availability than the others 10%E, 20%E,etc.

4.1.6 Heat transfer rate vs. Crank angle

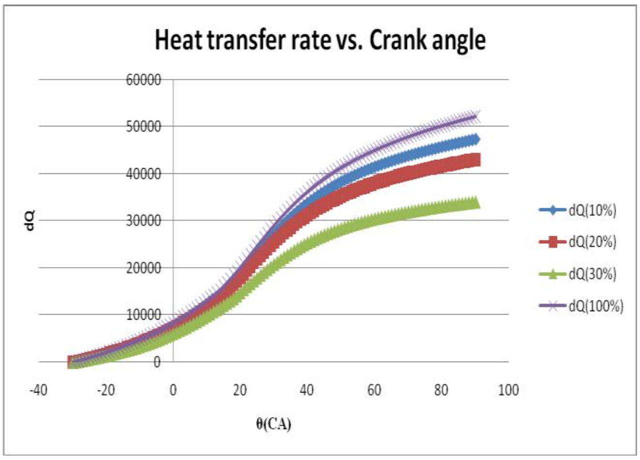

Fig.4.1.6 Heat transfer rate vs. Crank angle

The heat from the combustion chamber is transferred mainly by convective mode. The fig shows the heat transfer rate vs. Crank angle.

4.1.7 Heat transfer coefficient vs. crank angle

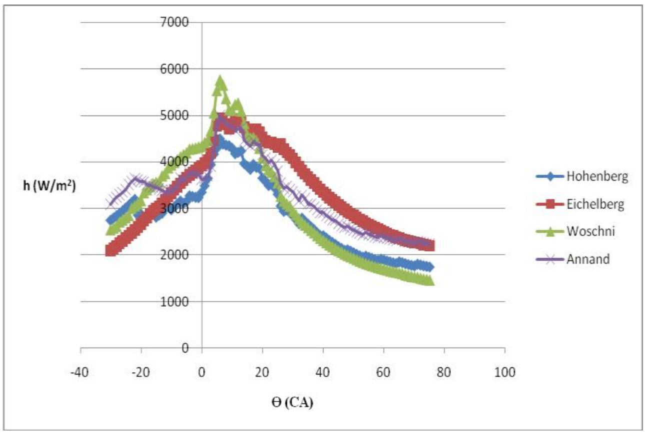

Figure 4.1.7: Heat transfer coefficient vs. crank angle

The Woschni correlation has the highest heat transfer coefficient, compared with the other two models, as a result of the difference in the temperature exponent used in the model. The heat flux decreases when the piston is in a downward motion, and during the intake process, the heat flux is minimum. This shows that heat is being added to the chamber during the intake process, when the wall temperature is slightly higher than in-cylinder temperature. However, the heat flux increases when the piston is in compression process. A high heat transfer coefficient causes too much heat loss to the cylinder wall, and this shows that this model causes too much energy to be wasted when piston is at TDC. Therefore, a high scaling factor is not suitable for Woschni model. Improper characteristic velocity causes incorrect heat loss to the cylinder wall. The piston is in a downward and upward motion, so the piston's instantaneous velocity is not the same across the crank angle ranges. The instantaneous piston speed is at minimum when the piston is at TDC and BDC, and it is in maximum when the piston is in the middle of stroke. Therefore, in this case, the characteristic velocity could be difference across the stroke range. However, the Woschni and Hohenberg models assumed that characteristic velocity is constant for all the crank angle ranges. The modified Woschni equation, on the other hand, uses a different approach, where the characteristic velocity varies across the engine cycle.

5. CONCLUSION

5.1 Conclusions

Present study deals with experimental calculation and simulation of rate of heat release and pressure for the diesel engine fuelled with ethanol blends 10%E, 20%E, 30%E and100%D. 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 availability which mainly depends on the internal energy increases with the increase in the ethanol blends because of increase temperature.
2) Hohenberge correlation is found to be best among all four correlations studied. It is predicting the results better than other correlations.
3) Annand's model can also be a good choice if adjustment of constants is properly done.
4) The Woschni correlation over-estimates the heat transfer coefficient during compression and underestimates it during combustion. Therefore the proper adjustment of constants is necessary for this model.

5.2 Future Scope

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

1) The homogeneous charge compression ignition engine also can use for heat transfer model to better result.
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.
5) The computational fluid dynamics can be used to predict the heat transfer from the engine cylinder.
6) The different types of engines can be used to study the models.

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

5.3 Applications

1) The simulation model developed in this study can be used to analyse the diesel engine with the slight changes in the parameters of predicted heat release curve.
2) Simulation model can be used to analyse the heat release rate at effect of various parameters such as inlet pressure and temperature, injection timings, speed, load, air fuel ratio, compression ratio, etc.

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