ANALYSIS OF COMBUSTION AND EMISSION CHARACTERISTICS OF A DIESEL ENGINE FUELED WITH ALTERNATIVE FUEL BY THERMODYNAMIC MODELING

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By

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CERTIFICATE

This is to certify that the thesis entitled "**Analysis Of Combustion And Emissions Characteristics Of A Diesel Engine Fueled With Alternative Fuel By Thermodynamic Modeling**" submitted by **Mr. Saroj Ray** in partial fulfillment of the requirements for the award of Bachelor of Technology Degree in Mechanical Engineering at National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by him under my guidance.

To the best of my knowledge the matter embodied in the thesis has not been submitted to any University /Institute for the award of any Degree or Diploma.

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ABSTRACT

With depletion of conventional fuel source at tremendous rate and increasing environment pollution has motivated extensive research in alternative fuel and engine design. Biodiesel being environmental friendly, usually lower exhaust emission and better lubrication properties can be suitable alternative. However the use of biodiesel fuel in diesel engines has associated with some of the drawbacks like higher nitric oxide (NO) emissions, poor cold flow properties and oxidation stability. Several measures have been taken to overcome these demerits for better utilization of biodiesel in diesel engines. In this work an attempt has been made to reduce the NO emissions of biodiesel by emulsifying Wood Pyrolysis Oil (WPO) in lower percentages with Jatropha Methyl Ester (JME). The experiment was performed on a single cylinder, air cooled, direct injection diesel engine for this purpose. The experimental results are validated with a two zone model program developed using a MATLAB software. One zone consists of pure air called non burning zone and the other zone consists of fuel and combustion products called burning zone. The first law of thermodynamics and equations of state are applied in each of the two zones to get the temperature and pressure inside the combustion chamber. Using the two zone combustion model the combustion parameters and the chemical equilibrium composition of all the species were determined theoretically. The extended Zeldovich mechanism is used for predicting the NO emissions and soot density is calculated using a semi-empirical model. The comparisons of experimental and theoretical results are presented in this paper.

CONTENTS

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Symbols/Definitions/ Abbreviations

$\frac{d(mu)}{d\theta}$	rate of change of internal energy of the system of mass m.
$\frac{dQ_r}{d\theta}$	rate of heat release during combustion period
$\frac{dQ_h}{d\theta}$	rate of heat transfer from gases to walls
$\frac{dW}{d\theta}$	rate of mechanical work done by the system on the boundary
P	pressure
V	instantaneous cylinder volume
m	mass
R	gas constant
T	temperature
$\frac{dV}{d\theta}$	change of volume with crank angle
h	heat transfer coefficient
A	heat transfer area
T_w	wall temperature
V_{cl}	cylinder clearance volume
λ	crank radius to piston length ratio
$\frac{dQ}{dt}$	rate of heat release
K	thermal conductivity of the cylinder gas
d	cylinder bore
Re	Reynolds number
T_g	cylinder bulk gas temperature
Cmodel	model constant=1000 kJ/kg degree
f_1, f_2	functions
q	heat losses
m_f	cumulative mass of fuel injected
Q	cumulative heat release rate
LCV	lower calorific value of fuel
C_{rate}	constant for mixing rate=0.002 s
k	density of turbulent kinetic energy
Eu/dθ	total turbulent energy of the fuel jet at a given crank angle instant, θ
$E_{diss}/d\theta$	rate of energy dissipation across the control surface
dEi/dθ	rate of generation of kinetic energy of fuel jet in to the cylinder
C_{turb}	efficiency of conversion of kinetic energy to turbulent energy in the free jet=0.2

$\mathbf{n}$	engine speed
$\boldsymbol{\rho_f}$	density of fuel
$\mathbf{C_d}$	coefficient of discharge of nozzle
$\mathbf{A_n}$	area of nozzle holes
$\frac{d\mathbf{mf}/d\boldsymbol{\theta}}{\boldsymbol{\rho}\mathbf{f}}$	volumetric injection rate of the fuel
$\mathbf{Eu}$	total turbulent energy of the fuel jet at a given crank angle instant
$\mathbf{C_{diss}}$	dissipation constant
$\boldsymbol{\lambda_{diff}}$	excess air ratio for diffusion burning=1.4
$\boldsymbol{\Phi}$	Crank angle
$\boldsymbol{\Phi_{inj}}$	Fuel injection crank angle
$\boldsymbol{\Phi_{exh}}$	Crank angle at exhaust valve opens
CHR	Cumulative Heat Release
WPO	Wood Pyrolysis Oil
JME	Jatropha Methyl Ester
JOE15	Jatropha Oil Emulsion with 15% WPO
TDC	Top Dead Centre
BDC	Bottom Dead Centre
DI	Direct Injection
ID	Ignition Delay

Chapter 1

INTRODUCTION

With depletion of conventional fuel source at tremendous rate and increasing environment pollution has motivated extensive research in alternative fuel and engine design. Experimental works aimed at good fuel economy and lower tailpipe emissions frequently changes the operating parameters which is a time and money consuming method. Alternatively computer simulation of engine with a mathematical model can be done easily to estimate the effects of design and changes in the operating parameters in short period of time and are cheap.

Modeling is the simple representation of complex real world problem. Almost all the real world phenomena are complex and can be simplified by taking some assumptions. It is desired to develop the simplest possible model that incorporates the major features of the phenomenon of interest. Many models have been developed by many researchers to solve the complex heterogeneous combustion process of diesel engines [1-5]. Due to extreme complexity of engine processes and our inadequate understanding at a fundamental level, most engine models are incomplete. Models are used for design purpose, for complete understanding of processes and to predict engine behaviour over a wide range of operating condition.

The theoretical models used in the case of internal combustion engines can be classified into two main groups: thermodynamic models and fluid dynamic models. Thermodynamic models are mainly based on the first law of thermodynamics and are used to analyse the performance characteristics of engines. Pressure, temperature and other required properties are evaluated with respect to crank angle or time. The engine friction and heat transfer are taken into account using

empirical equations obtained from experiments. These models are further classified into two groups namely single-zone models and multi-zone models. On the other hand, multi-zone models are also called computational fluid dynamics models. They are based on the numerical calculation of mass, momentum, energy and species conservation equations in either one, two or three dimensions to follow the propagation of flame or combustion front within the engine combustion chamber.

Two zone model consists of one non burning zone which contain pure air and other zone consist of fuel and combustion products called burning zone. First law of thermodynamics and state equations are applied in each of the two zones to yield cylinder temperatures and cylinder pressure histories. Using the two zone combustion model the combustion parameters and the chemical equilibrium composition were determined. Multi-dimensional models need detail information of many phenomena and large computation time. Single zone model though simple does not account for heterogeneous character of diesel engine. Therefore, it is reasonable to choose a two zone model which is simple and required reasonable computer time.

Biodiesel is defined as mono-alkyl esters of long chain fatty acids derived from vegetable oils or animal fats. It was observed from the literature [6-8] that the use of biodiesel in diesel engine results in a slight reduction in brake power and a slight increase in fuel consumption. However, the lubricant properties of the biodiesel are better than diesel, which can help to increase the engine life. Also the exhaust emission of the biodiesel is lower than the neat diesel operation due to the presence of oxygen in the molecular structure of the biodiesel. Moreover, the biodiesel fuel is environment friendly, because biodiesel does not produce SO_x and also there is no increase in CO_2 emission at global level.

The drawbacks of biodiesel are higher nitric oxide emissions and poor oxidation stability than petroleum-based diesel fuel. This oxidation can cause the fuel to become acidic and to form insoluble gums and sediments that can plug fuel filters [9]. The problem of oxidation stability has to be resolved in order to store the biodiesel fuels for long time storage.

The objective of the present work is to develop a two zone model for a direct injection diesel engine fuelled with diesel and biodiesel blend. It gives a detailed description of model which consists of various sub-models. Moreover, the present work validated the model with experimental investigation.

Chapter 2

LITERATURE REVIEW

The rapid development of computer technology has encouraged the use of complex simulation techniques to quantify the effect of the fundamental processes in the engine systems. The advances achieved by current automotive engines would have been impossible without the simulation models providing these insights [10-11].Lyn et al. [12] analyzed the effects of injection timing, injection velocity and fuelling rate on the delay period. An increase in speed at constant load increases the peak pressure and temperature, due to the decrease in heat transfer, resulting in a slight decrease in delay period as analysed by Wong et al. [13].

For engineering applications a semi-empirical relation based on chemical chain reactions called as Weibe's function is used to find heat release rate [14-15]. However a single Weibe's function is not able to predict the heat release rate during early premixed burning. Thus a double Weibe's function is required for accurate prediction direct injection diesel engine [14,16].

Biodiesel have a relatively low flash point, a high heating value, high density and high viscosity comparable to those of petroleum derived diesel. Many studies show that unburned hydrocarbons (HC), carbon monoxide (CO) and sulfur levels are significantly less in the exhaust gas while using biodiesel as fuel. However, a noticeable increase in the oxides of nitrogen (NO_x) levels is reported with biodiesel [17-20]. Biodiesel blends reduce levels of global warming gases such as CO_2. Its additional advantages include outstanding lubricity, excellent biodegradability, higher combustion efficiency and low toxicity as compared to other fuels [21].

The combustion process in diesel engine is extremely complex due to transient and heterogeneous nature of combustion which is mainly controlled by turbulent mixing of fuel and air. High speed photography studies and in- cylinder sample collection have revealed some interesting characteristic of combustion phenomena [10].

Ignition delay in diesel engines has a direct effect on engine efficiency, noise and exhaust emissions. A number of parameters directly affect the ID period, among them cylinder pressure and temperature, swirl ratio and misfire. A number correlations use an Arrhenius expression similar to that proposed by Wolfer [22] in 1938 where he measured the ignition delay as a function of pressure and temperature. Watson [16] developed an ID correlation using a diesel engine under steady state conditions which is still widely used. Later Assanis et. al. [23] developed an ignition delay correlation for predicting the delay period in a heavy-duty turbocharged direct injection diesel engine running under both steady state and transient operation.

Chapter 3

MODEL DESCRIPTION

The main calculation procedure is based on the integration of the first law of thermodynamics and the perfect gas state equation combined with the various sub-models. The following assumptions were made for the analysis:

(i) The cylinder contain are present in two zones: one zone consists of pure air called non burning zone and other zone consist of fuel and combustion products called burning zone.

(ii) Pressure and temperature in each zone inside the cylinder are uniform and vary with crank angle.

(iii) The contains of each zone obeys the perfect gas state law.

Energy Conservation Law and State Equation

During compression stroke, only one zone (of pure air) exists. Then, the first law of thermodynamics for a closed system is applied together with the perfect gas state equation (Heywood). The change in internal energy can be expressed as follows:

$$\frac{d(mu)}{d\theta} = \frac{\mathrm{dQr}}{d\theta} - \frac{\mathrm{dQh}}{d\theta} - \frac{\mathrm{dW}}{d\theta} \qquad (1)$$

By replacing the work transfer term $\frac{\mathrm{dW}}{d\theta}$ with P $\frac{\mathrm{dV}}{d\theta}$ or by the ideal gas law PV = m RT, the above equation (1) can be rearranged as

$$\frac{d(mu)}{d\theta} = \frac{\mathrm{dQr}}{d\theta} - \frac{\mathrm{dQh}}{d\theta} - P\frac{\mathrm{dV}}{d\theta} \qquad (2)$$

where, V is the instantaneous cylinder volume with respect to crank angle which is given by

$$V=V_{cl}+\left(\frac{\pi D^2}{4}\right) r[1+\lambda^{-1}-cos\phi-\sqrt{\lambda^{-2}-sin^2\phi}] \quad (3)$$

In the above equations, the term du is given as fourth order polynomial expressions of the absolute temperature T, including the enthalpy of formation at absolute zero [10].

Internal energy calculation as a function of temperature:

$$h_i/(R_{mol}T)=a_{i1}+ a_{i2} T/2+a_{i3} T^2/3+ a_{i4} T^3/4+a_{i5} T^4/5+a_{i6}/T \quad (4)$$

$$u_i = h_i – RT$$

For the surrounding air zone, which only loses mass (air) to the burning zone, the first law of thermodynamics for the unburned zone is written as

$$dE = dQ – pdV – h_a dm_a \quad (5)$$

The burning zone not only receives mass from the air zone, but also there is an enthalpy flow from the fuel which is ready to be burned in the time step. So, the first law of thermodynamics for the burning zone becomes

$$dE = dQ - pdV + h_a dm_a + h_f dm_f \quad (6)$$

The first law of thermodynamics for the combustion in time step dt is

$$f(E)=E(T2) – E(T1) – dQ + dW + dmf\ Qvs =0 \quad (7)$$

If f(E) is greater than the accuracy required new value of T_2 is calculated using the Newton-Raphson numerical method. The unburned zone temperature is calculated using the equation.

$$T_u=T_{soc}\left(\frac{P}{P_{soc}}\right)^{(\gamma-1)/\gamma} \quad (8)$$

Heat Transfer Model

The amount of heat exchange between gases to wall as well as wall to gas is significant as it directly affects the engine performance and its life. Heat transfer between the cylinder trapped mass and surrounding walls was calculated using the formula of Annand [24]. Annand formula to calculate heat loss from the cylinder is given by

$$\frac{dQ}{dt} = aK\frac{\mathrm{Re}^{b}}{d}\left(\mathrm{T_g} - \mathrm{T_w}\right) + c\left(\mathrm{T_g}^{4} - \mathrm{T_w}^{4}\right) \quad (9)$$

In this equation 'T_w' is the cylinder wall temperature which is assumed as 450 K and a, b, c are constants. The constant values are taken as, a=0.2626, b=0.6, c=5.67×10^{-8} W/m^2 K^4. The first term in the above equation is convective heat transfer term and the major parameter affecting convection is Reynolds number. The second term in the above equation is a radiation term assuming grey body radiation.

Ignition Delay

The time delay between the start of injection and the start of combustion is defined as the ignition delay period. Determination of the start of combustion (SOC) by selecting the proper method is a key issue in ignition delay studies. In the combustion model the ignition delay is also taken into account. The ignition delay period is calculated by integrating Wolfer's relation using trapezoidal rule.

$$\int_{\mathrm{tinj}}^{\mathrm{tign}} \frac{dt}{t(p,T)} = \frac{1}{K\ \mathrm{tinj}} \int_{\mathrm{tinj}}^{\mathrm{tign}} \frac{dt}{(p(t))^{-q}\exp(E/RT(t))} = 1 \quad (10)$$

The values of various constants corresponding to D.I. diesel engines are K =2272; q =-1:19; E/R = 4650.t_{inj}.

Chapter 4

WIEBE'S COMBUTION MODEL

Wiebe function is used to predict the mass fraction burn and burn rate in internal combustion engines operating with different combustion systems and fuels [14]. Wiebe linked chain chemical reactions with the fuel reaction rate in internal combustion engines and his approach was based on the premise that a simple one-step rate equation will not be adequate to describe complex reacting systems such as those occurring in an internal combustion engine. Moreover, developing and solving rate equations which account for the simultaneous and sequential interdependent chain and chain branching reactions would be time consuming and tedious task. He argued that for engineering application the details of chemical kinetics of all the reactions could be bypassed and a general macroscopic reaction rate expression could be developed based on the concept of chain reactions.

The Wiebe functions for the non-dimensional burn fraction x and its derivative w (burn rate) as functions of degrees crank angle can be written as

$$x=1-e^{-6.908(\phi/\phi d)^{m+1}} \tag{11}$$

$$w=\frac{dx}{d\phi}=\frac{6.908(m+1)}{\phi d}\left(\frac{\phi}{\phi d}\right)^{m}e^{-6.908(\phi/\phi d)^{m+1}} \tag{12}$$

or the non-dimensional burn fraction x and its derivative w (burn rate) as functions of time t can be written as

$$x=1-e^{-6.908(t/td)^{m+1}} \tag{13}$$

$$w=\frac{dx}{dt}=\frac{6.908(m+1)}{td}\left(\frac{t}{td}\right)^{m}e^{-6.908(t/td)^{m+1}} \tag{14}$$

The time it takes to reach maximum burn rate t_m can be found by differentiating equation (14) and equating the result to zero

$$t_m=t_d\left(\frac{m}{6.908(m+1)}\right)^{1/(m+1)} \quad (15)$$

The corresponding burn fraction is

$$X_m=1-\exp(-6.908(t_m/t_d)^{m+1}) \quad (16)$$

From above equations (13) and (14)

$$X_m=1-\exp(-m/(m+1)) \quad (17)$$

Wiebe suggested the physical meaning of the exponent m which was based on equation (15), which shows that for a given combustion duration the time it takes for maximum burn rate to be reached is determined solely by the magnitude of m, which, in turn, determines the magnitude of the maximum burn rate (equation (15)).When calculating the heat release, prior knowledge of actual overall equivalence ratio is necessary. The term equivalence ratio is defined as the ratio of actual air-fuel ratio to the stoichiometric air-fuel ratio. This helps in fixing the mass of fuel to be admitted.

Chapter 5

CHEMISTRY OF COMBUSTION

In a combustion process, fuel and oxidizer react to produce products of different composition. The theory of combustion is a complex process and has been a topic of intensive research for many years. Let us represent the chemical formula of a fuel as $C_\alpha H_\beta O_\gamma N_\delta$. In the present case it was consider that 10 species were present in combustion product and the combustion equation is given by

$$\epsilon\varphi C_\alpha H_\beta O_\gamma N_\delta + 0.21O_2 + 0.79N_2 \longleftrightarrow \nu_1 CO_2 + \nu_2 H_2O + \nu_3 N_2 + \nu_4 O_2 + \nu_5 CO + \nu_6 H_2 + \nu_7 H + \nu_8 O + \nu_9 OH + \nu_{10} NO$$

From atomic balance of each species C- H- O- N the following 4 equations.

C $\quad \epsilon\varphi\alpha = (y_1+y_5)N1 \qquad (18)$

H $\quad \epsilon\varphi\beta = (2y_1+2y_6+y_7+y_9)N1 \qquad (19)$

O $\quad \epsilon\varphi\gamma+0.42 = (2y_1+y_2+2y_4+y_5+y_8+y_9+y_{10})N1 \qquad (20)$

N $\quad \epsilon\varphi\delta+1.58 = (2y_3+y_{10})N1 \qquad (21)$

By definition total number of mole fraction is unity.

$$\sum_{i=1}^{10} yi - 1 = 0 \qquad (22)$$

The chemical reaction consider in equilibrium are as follows:

$0.5H_2 \longleftrightarrow H$

$0.5O_2 \longleftrightarrow O$

$0.5H_2+0.5O_2 \longleftrightarrow OH$

$0.5O_2+0.5N_2 \longleftrightarrow NO$

$H_2+0.5O_2 \longleftrightarrow H_2O$

$CO+0.5O_2 \longleftrightarrow CO_2$

Use of equilibrium constant is identical to maximizing the entropy of the gas. This method is similar when considering restricted species list such as the present case. Compositions of all the species were found using method develop by Olikara and Borman (1975). Once the composition is known the thermodynamic properties of interest can be computed like enthalpy, entropy, specific volume and internal energy.

Nitric oxide formation model

The current approach to modeling NOx emissions from diesel engines is to use the extended Zeldovich thermal NO mechanism and neglects other sources of NOx formation [25]. The extended Zeldovich mechanism consists of the following mechanisms.

$O + N_2 \longleftrightarrow NO + N$

$N + O_2 \bullet \longleftrightarrow NO + O$

$N + OH \longleftrightarrow NO + H$

The change of NO concentration is expressed as follows:

$$\frac{d(NO)}{dt} = 2\,(1-\alpha^2)\,\frac{R1}{1+\alpha R1/(R2+R3)} \qquad (23)$$

Where R_i is the one-way equilibrium rate for reaction i, defined as

$R1 = k_{1f}\,(N)_e(NO)_e$, $R2 = k_{2f}\,(N)_e(O2)_e$,

$R3 = k_{3f}\,(N)_e(OH)_e$, $\alpha = (NO)/(NO)_e$

Net Soot formation model

The exhaust of CI engines contains solid carbon soot particles that are generated in the fuel rich regions inside the cylinder during combustion. Soot particles are clusters of solid carbon spheres with HC and traces of other components absorbed on the surface. They are generated in the combustion chamber in the fuel rich zones where there is not enough oxygen to convert all carbon to CO_2. Subsequently as turbulence motion continue to mix the components most of these carbon particles find sufficient oxygen to react and form CO_2.Thus soot particles are formed and consumed simultaneously in the combustion chamber.

The net soot formation rate was calculated by using semi-empirical model proposed by Hiroyasu et. al. (1983). According to this model the soot formation rate (index sf) and soot oxidation rate (index so) was given by

$$\frac{dm_{sf}}{\mathrm{dt}} = A_{sf} dm_f^{0.8} p^{0.5} \exp\left(-\frac{E_{sf}}{R_{mol}T}\right) \tag{24}$$

$$\frac{dm_{so}}{dt} = A_{so} m_{sn} \left(\frac{p_{o_2}}{p}\right) p^{n} \exp\left(-\frac{E_{so}}{R_{mol}T}\right) \tag{25}$$

Where pressure are expressed in bar, dm_f is the unburned fuel mass in kg to be burned in time step dt. Therefore the net soot formation rate is expressed as

$$\frac{dm_{sn}}{dt} = \frac{dm_{sf}}{dt} - \frac{dm_{so}}{dt} \tag{26}$$

A computer program using MATLAB software was generated with all above mentioned equations and considering all the values of constants, in order to predict the combustion attributes like in cylinder pressure, heat release rate, heat losses, combustion temperature, NO_x emission and net soot formation rate.

Chapter 6

EXPERIMENTAL ANALYSIS

The engine used in the present study was a kirloskar single cylinder, direct injection, diesel engine. The schematic diagram of the experimental setup is shown in Figure 1 and the detailed specification of the test engine is given in Table 1. The experimental setup consists of a single cylinder, four stroke, air cooled and direct injection diesel engine coupled to an electrical dynamometer. The data-acquisition system consists of a Kistler piezoelectric pressure transducer model: 5395A, a charge amplifier, a computer and a crank angle encoder.

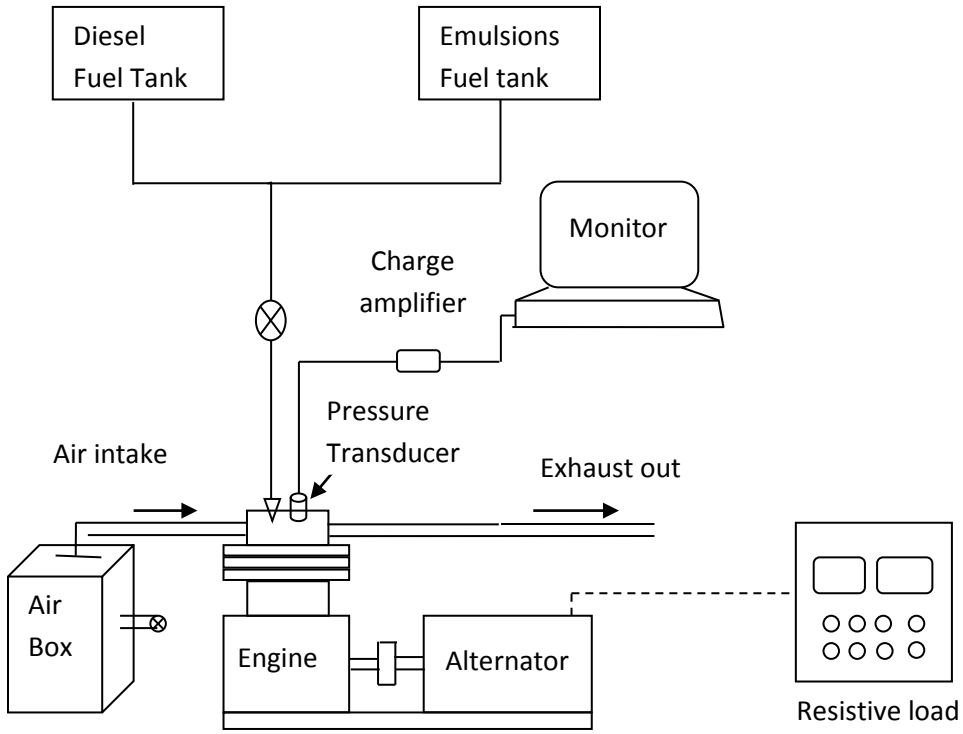

Figure 1. Schematic diagram of experimental setup [24]

Fuel consumption is measured with the help of a solenoid controlled automatic burette. Air consumption is measured by a differential pressure sensor fitted in the air box. A surge tank is used to damp out the pulsations produced by the engine, for ensuring a steady flow of air through the intake manifold. A non-contact type sensor is connected near the flywheel of engine to measure the speed. Twenty three channel signal analyzer is used for data acquisition and the acquired data is transferred through ethernet cable and stored in a personal computer for offline analysis.

Table 1. Technical specifications of engine [24]

Make	Kirloskar
Model	TAF 1
Bore x Stroke	87.5 x 110 mm
Compression ratio	17.5:1
Piston type	Bowl-in-piston
Number of valves	2
Rated power	4.4 kW
Rated speed	1500 rpm
Type of fuel injection	Pump-line-nozzle injection system
Nozzle type	Multi hole
No. of holes	3
Needle lift	0.25 mm
Spray-hole diameter	0.25 mm
Cone angle	110^{o}
Start of injection	23^{o} CA bTDC
Nozzle opening pressure	200 bar
Inlet valve opening	4.5^{o} CA bTDC
Inlet valve closing	35.5^{o} CA aBDC
Exhaust valve opening	35.5^{o} CA bBDC
Exhaust valve closing	4.5^{o} CA aTDC

The samples were collected at a rate of 20 kHz (20,000 samples/sec) for 20 continuous cycles. The samples are separated with reference to the TDC signal, resulting in an array of data for each cycle. There are 20 arrays of data pertaining to each cycle. These 20 arrays are averaged to make 1 array of data, i.e., averaged cycle data. The combustion attributes of the diesel engine are obtained by means of the pressure transducer which is flush mounted on the cylinder head, and the crank angle encoder is fixed on the output shaft of the engine.

Table 2. Comparison of fuel properties of WPO, JME and JOE15 with diesel [24]

Properties	ASTM method	Diesel	WPO	JME	JOE15
Specific gravity at 15 °C	ASTM D 4052	0.83	1.15	0.88	0.9267
Net calorific value (MJ/kg)	ASTM D 4809	43.8	20.58	39.1	36.32
Flash point (°C)	ASTM D 93	50	98	118	156
Fire point (^{o} C)	ASTM D 93	56	108	126	-
Pour point (°C)	ASTM D 97	-6	2	-1	-
Cloud point (^{o}C)	ASTM D 2500	-	10	4-10	-
Carbon residue (%)	ASTM D4530	0.1	12.85	-	-
Kinematic viscosity at 40 °C (cSt)	ASTM D 445	2.58	25.3	4.6	7.28
Cetane number	ASTM D 613	50	25	51	-
Moisture content (wt %)	ASTM D4442	0.025	15-30	0.03	-
Final boiling point (^{o}C)	ASTM D 86	344	250-280	342	-
Empirical formula	ASTM D3239	$C_{12}H_{26}$	$C_{1.13}H_{2.92}N_{0.01}S_{0.01}O_{1.62}$	$C_{7.56}H_{13.89}N_{0.01}O_{0.81}$	$C_{2.08}H_{4.13}N_{0.01}S_{0.01}O_{1.17}$
Molecular weight	ASTM D5296	170	42.86	117.6	48.27
Stoichiometric A/F ratio	-	15	3.39	12.4	7.26

The fuels considered are diesel ($C_{12}H_{26}$), jatropha methyl ester ($C_{7.56}.H_{13.89}N_{0.01}O_{0.81}$) and JOE15 ($C_{2.08}H_{4.13}N_{0.01}S_{0.01}O_{1.17}$). The volumetric composition of JOE15 fuel used in this investigation is 15% WPO, 83% JME and 2% surfactant Span 80.

Experiments were conducted in the diesel engine fuelled with diesel, JME and JOE15 emulsion. Combustion parameters such as pressure, mass fraction burn, heat release rate and cumulative heat release were obtained with the help of data acquisition system. The experimental results were validated with simulated results and compared.

Chapter 7

RESULTS AND DISCUSSION

Pressure-crank angle diagram

Figure 2 shows the measured and computed pressure trace and crank angle histories for the fuels tested at full load condition. In a compression ignition engine, cylinder pressure depends on the burned fuel fraction during the premixed burning phase which is the initial stage of combustion and the ability of the fuel to mix well with air and burn. High peak pressure and maximum rate of pressure rise are corresponding to large amount of fuel burned in premixed stage.

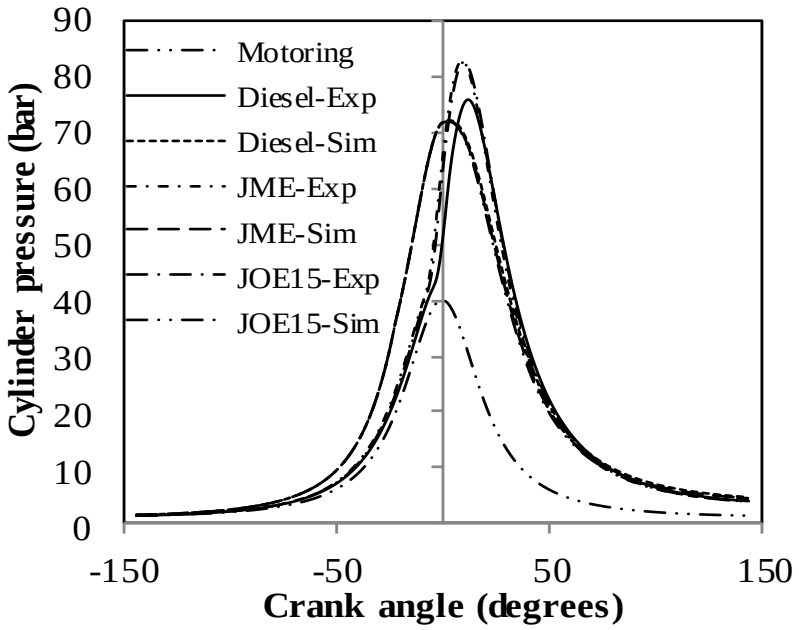

Figure 2: Variation of cylinder pressure with crank angle at full load.

It is observed from the experimental results that the peak pressure for diesel, JME and JOE15 are 75.72, 82.61 and 82.40 bar respectively. For the simulated conditions the peak pressure values are 71.76, 71.99 and 71.85 bar respectively. In both the cases the combustion of JME

and JOE15 starts earlier than that of diesel fuel. Also the peak cylinder pressure of JME and JOE15 is marginally higher than that of diesel, as a result of high viscosity and low volatility.

Apparent or Net heat release rate

Figure 3 depicts the variation of apparent heat release with respect to crank angle for different fuels tested. The term apparent or net heat release rate is determined by deducting the heat transfer to cylinder walls, crevice volume, blow-by and the fuel injection effects from heat energy liberated by burning the fuel.

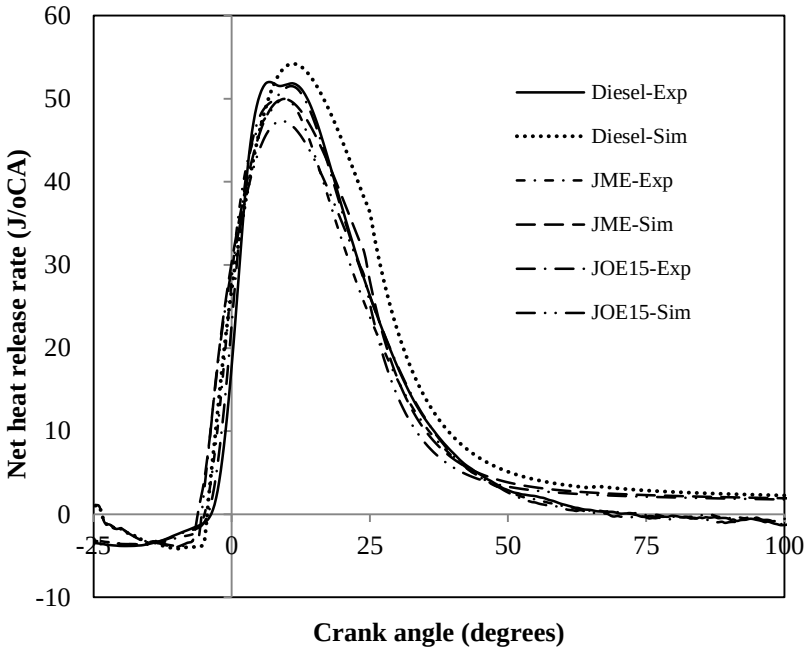

Figure 3: Variation of maximum heat release rate with crank angle.

The experimental results of maximum net heat release rate for diesel, JME and JOE15 are by about 52.01, 49.97 and 48.51 J/^{o}CA respectively at full load condition. At simulated conditions the values are by about 54.20, 49.93 and 47.31 J/^{o}CA for diesel, JME and JOE15 respectively. The intensity of premixed combustion phase for diesel is found to be more and whereas, this is lower in the case JME and JOE15. It is also seen that the quantity of diffusive combustion are found to be shorter for JME and JOE15 emulsion due to faster burning

characteristics. Oxygen present in JME and quick evaporation nature of emulsified fuel JOE15 are the causes for faster burning process [27].

NO emissions

In diesel engine exhaust, NOx is predominantly composed of NO, with lesser amounts of NO_2. Other oxides of nitrogen, such as N_2O, N_2O_5, NO_3 are negligible. In general, NOx formation mechanisms are described as thermal NOx, prompt NOx and fuel NOx. Under most diesel engine combustion conditions, thermal NOx is believed to be the predominant contributor to total NOx. At high temperatures, occurring within the combustion chamber of a diesel engine, N2 and O2 can react through a series of chemical steps known as the Zeldovich mechanism. NOx formation occurs at temperatures above 1500 °C, and the rate of formation increases rapidly with increasing temperature [28].

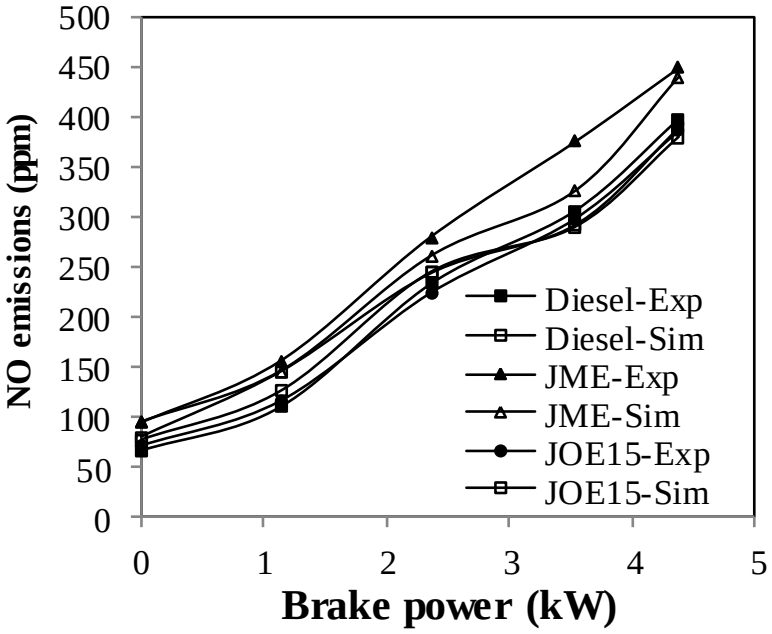

Figure 4: Variation of NO emissions with brake power.

Figure 4 depicts the comparison of NO emissions for the tested fuels in both experimental and simulated conditions. The experimental results of NO emissions at full load condition for diesel, JME and JOE15 are 398 ppm, 449 ppm and 386 ppm. In case of simulated conditions the NO values are 388 ppm, 440 ppm and 380 ppm respectively for the above said fuels. It can be observed from the figure that the NO emissions of the JME operation are higher compared to JME-WPO emulsions as well as diesel operation.

The presence of oxygen molecule in biodiesel causes an increase in combustion gas temperature resulting in a marginal increase in NO emissions [29]. Also it can be observed that the NO emission decreases with the addition of WPO content in the emulsions. This may

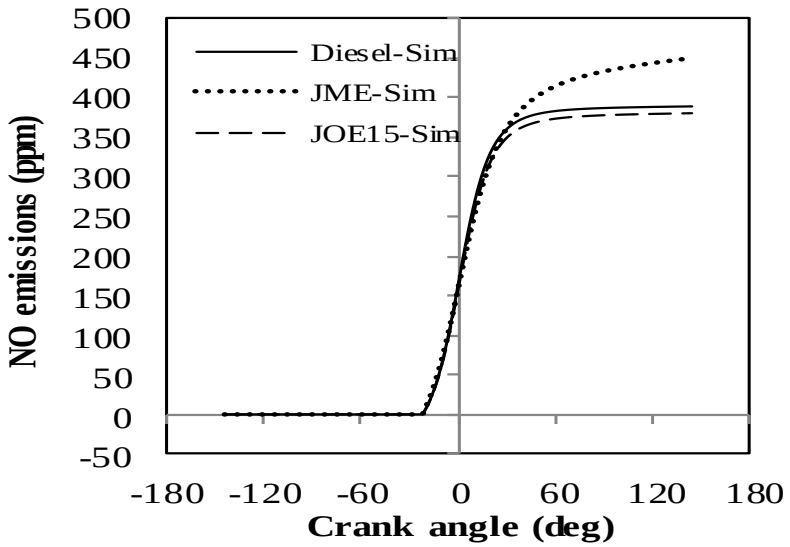

Figure 5: Variation of NO concentration with crank angle at full load.

be due to the water content present in the WPO which will reduce the combustion temperature [30].

Figure 5 depicts the NO concentration with respect to crank angle at full load condition for simulated conditions.

It can be seen that the NO emission increases after the ignition and reaches maximum around 27^{o} after TDC then lasting until EVO (exhaust valve opening). Compared to diesel readings, the NO concentration is higher in the case of JME operation and it is lower in the case of JOE15 operation. The reduction in the NO concentration of JOE15 may be due to following

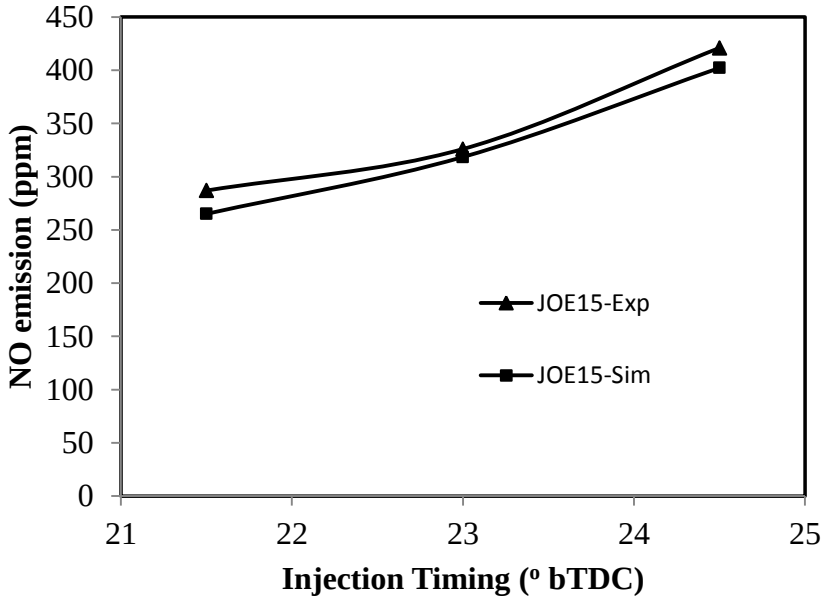

Fig. 6 Variation of NO emission with injection timing

two reasons. First, the in-cylinder temperature decreases because of higher specific heat capacity of water available in the WPO. Secondly, the water content in the WPO results in reduction of N_2 and O_2 concentrations and the improvement of local rich oxygen [31].

Figure 6 depicts the comparison of NO emissions for the JOE15 fuel in both experimental and simulated conditions. The experimental results of NO emissions at full load condition for JOE15 at injection timings 21.5^{o},23^{o} and 24.5^{o} bTDC are 287 ppm, 326 ppm and 421 ppm. In case of simulated conditions the NO values are 265.1 ppm, 318.5 ppm and 402.2 ppm respectively for the above said injection timings. It can be observed from the figure that the

NO emissions of the JOE15 increase with advancing the injection timing and decreases with retarding of injection timing.

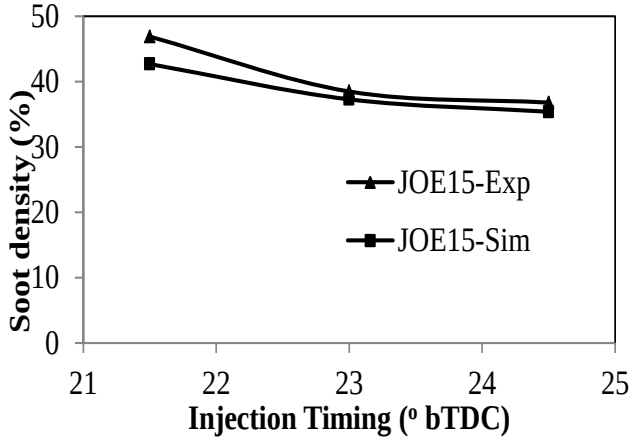

Fig. 7 Variation of soot density with injection timing

Figure 7 depicts the comparison of soot density for the JOE15 fuel in both experimental and simulated conditions. The experimental results of soot density at full load condition for JOE15 at injection timings 21.5°,23° and 24.5° bTDC are 46.9 %, 38.5 % and 36.8 %. In case of simulated conditions the soot density values are 42.7 %, 37.3 % and 35.4 % respectively for the above said injection timings. It can be observed from the figure that the soot density of the JOE15 decreases with advancing the injection timing and increases with retarding of injection timing.

Chapter 8

CONCLUSIONS

In the present work a comprehensive two zone model has been developed for diesel engine to validate the experimental results obtained with three different fuels, viz. diesel, JME and JOE15. The experimental investigation was conducted on a DI diesel engine, it is revealed that the developed model predicts adequately well with the combustion parameters for the three fuels examined. The main results obtained from this study:

- The experimental results of the peak cylinder pressure of JME and JOE15 is marginally higher than that of diesel, and similar results are obtained with simulated conditions.
- The maximum heat release rate of JME and JOE15 are lower than that of diesel fuel in both experimental and simulated conditions.
- The NO emissions of diesel fuel are increased with load both in experimental and simulated conditions. Similar trends have been obtained with JME and JOE15.
- The NO emissions of JOE15 are increased and soot density is decreased with advancing of injection timing and the values are in vice versa in retarded conditions.
- The presented model can predict the combustion and emission characteristics such as cylinder pressure, heat release, NO emissions and soot densities which are in good agreement with the experimental results.

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