NUMERICAL STUDY ON A DIESEL ENGINE FUELED BY EUCALYPTUS BIOFUEL USING CONVERGE CFD SOFTWARE

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Abstract— To meet more stringent norms and standards concerning engine performances and emissions, engine manufacturers need to develop new technologies enhancing the non polluting properties of the fuels. In that sense, the testing and development of alternative fuels such as biodiesel are of great importance. After a brief synthesis on vegetable biofuels as alternatives for compression ignition engines, this paper present a study of a biofuel from the eucalyptus tree leaves. This vegetable source is selected due to its strong availability in South and North of Africa, California, Latin America, Europe and all the Mediterranean countries. Eucalyptus trees are original to the Australian continent where they dominate more than 90% of the forests. In the present study, the Computational Fluid Dynamics (CFD) code CONVERGE was used to model a complex combustion phenomenon in Compression Ignition (CI) engine. The experiments were performed on a single cylinder direct injection diesel engine, with a twice loads condition (50% and 90% of full load) at a constant speed of 1500 rpm. Combustion parameter, such as cylinder pressure is obtained from experimental data. The numerical modelling was solved by CFD code, taking into account the effect of turbulence. For modelling turbulence Renormalization Group Theory (RNG) k-ε model was used. The sub models such as droplet collision model and KH-RT model were used for spray modelling. A good agreement between the modelling and experimental data ensures the accuracy of the numerical predictions collected in this work. Results show that when the engine is supplied with neat eucalyptus fuel, the combustion characteristics are slightly changed compared to neat diesel fuel and the soot species are reduced.

Keywords— Diesel engine; Biodiesel, Eucalyptus; Combustion modelling; Gaseous emissions; numerical simulations.

1. Introduction

During the past few decades worldwide use of petroleum has rapidly increased due to the growth of human population and industrialization, which has resulted in depletion of fossil fuel reserves leading to an increased petroleum prices. The scarcity of petroleum fuel reserves has made renewable energy an attractive alternative source of energy for the future. Furthermore, conventional fossil fuelscontribute to global warming via increased greenhouse gas (GHG) emissions. As a consequence, a move is being made towards alternative, renewable, sustainable, efficient and cost effective energy sources with lesser emissions [1]. Biodiesel is considered as one of the promising alternative resources for diesel engine, especially from non-edible oil feedstock as well as its potential to be a part of a sustainable energy mix in the future. The advantages of non-edible oils as a diesel fuel are liquid nature portability, ready availability, renewability, higher heat content, higher flash point, higher cetane number, lower sulphur and aromatic content as well as biodegradability [2].
Several works are carried out on biodiesel preparation from different vegetable oils and their utilization in compression ignition engines [3, 4]. The early biofuels synthesized from comestible vegetable oils (soybean, sunflower…) are known as the first generation of biodiesels.

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However, when the prices of first biofuel generation products intended for human and animal consumption were affected, scientific researchers tried to find other non-edible vegetable sources known as the second biofuel generation [5]. The second generation biodiesel produced from non-edible feedstock as well as biomass can overcome the socio-economic disadvantages of current biodiesel technology and be able to address many of the challenges of climate change and the energy crisis. Sometimes the biofuels are also produced from waste or cooking oil [6] and animal fats [7]. These alternative energy resources are largely environment-friendly but necessity is to evaluated case-to-case for their advantages, drawbacks and specific applications. Studies have exposed that the injection of vegetable oils in neat form in engines is possible but not preferable [8]. The high viscosity of vegetable oils and the low volatility affect the atomization and spray pattern of in-cylinder injected fuel, leading to imperfect combustion and severe carbon deposits, injector choking and piston ring sticking. These problems can be corrected by different methods reducing the viscosity. These methods are the transesterification [9], the dilution [10], the emulsion [11] and the preheating [12]. After rectification, the corrected oils have properties comparable to conventional diesel and can be injected in CI engines without technical modifications.

Numerical simulation has become one of the most powerful tools to investigate the evolution of combustion and emission of internal combustion engine for better understanding of the complex phenomenon inside the cylinder. CONVERGE, KIVA, STAR-CD, Fire are some similar kind of CFD packages used for IC engine simulations. Therefore, in contrast to experiments, results can often be reached faster and cut-price. This allows detailed information of the pertinent processes and is a prerequisite for their improvement. Furthermore, numerical simulation can be used an effective supplement to experimentation in the study of different phenomena's that take place at different time and length scales or in not accessible locations which cannot be investigated using experimental techniques. Thus, numerical investigations are an interesting way to identify the best engine parameter settings (injection, EGR, swirl, etc…) leading to the best compromise between all the engine characteristics (mechanical performances, BSFC and emissions), for exemple;

Kiat Ng et al. [13] investigated numerically the combustion and emission formation mechanisms with the CFD code FLUENT 6.3.26 integrating chemical kinetic reaction mechanism with 80 species and 303 reactions in diesel engine alimented with different biodiesels (coconut methyl ester CME, palm methyl ester PME and soybean methyl ester SME). This study clarified the necessity of integrating a biodiesel mechanism with CFD code to study the combustion and emission NO and soot evolutions. Divers strategies (spray cone angle, SOI, EGR) are examined. The results demonstrate as after the increasing spray cone angle could significantly reduce NOx emissions of SME to the same level as neat diesel. The EGR and SOI were concluded as the most effective means of reducing NOx emissions for both diesel and biodiesel, but the SME showed greater NOx reduction than neat diesel with the same level of EGR rate. Otherwise, several sub-models were developed for KIVA-3V to include biodiesel in the fuel library. Golovitchev and Yang [14] presented the numerical investigation of the rapeseed methyl ester ($C19H34O2$) combustion in a Volvo D12C diesel engine. A chemical mechanism with 88 species and 363 reactions is used. The Partially Stirred Reactor (PaSR) method which takes into consideration the interaction between turbulence and combustion was used under Kiva-3V environment. The results show that RME combustion gives low soot and NO concentrations with if moderate EGR value were used. The skeletal mechanism developed by Lawrence Livermore National Laboratory (LLNL) in Ref. [15] consists of 115 species and 460 reactions, which consists of methyl decanoate, methyl 9-decenoate and n-heptane, was developed to reduce computational costs for 3-D engine simulations with CONVERGE CFD.The skeletal mechanism was able to predict various combustion characteristics accurately such as ignition delay, flame lift-off length, and equivalence ratio at flame lift-off location under different ambient conditions. The skeletal mechanism features a reduction by a factor of around 30 in size while still retaining good accuracy and comprehensiveness compared with the detailed mechanism that consists of 3299 species and 10806 reactions. Separately, Ren et al. [16] studied the combustion and emission in a diesel engine with the variation of properties of biodiesel fuels. They use the ERC-bio mechanism. This study showed that the chemical properties have a greater influence on the combustion and NOx emissions, and the chemical and physical properties were affected to soot emissions. In order to reduce NOx emission in diesel engine alimented with B20 blend of soybean methyl ester (SME) with different strategies is presented with Al-Dawody and Bhatti [17].The results denote that cooling the intake air temperature from 55°C to 15°C could reduce NOx, soot emissions and BSFC by 10.53%, 24.35% and 6.2% respectively. Furthermore, could also decrease NOx emission with piston bowl small diameter but deeper, but with BSFC increase. An et al. [18] performed a CFD simulation using KIVA 4 coupled with CHEMKIN II code for pure biodiesel combustion. Divers injection strategies (post injection timing, injection angle and urea direct injection) are examined for reduction of NOx emissions. This paper denotes that between the tested four post injection timings (10, 15, 20 and 25 CA ATDC), the 15 CA ATDC give a lower NO emission level. The increases in urea mass fraction reduce the NO emission level. Furthermore, the optimized injection strategy without post urea reduced with 58% the NO emission without severe impacts on the CO emissions and BSFC. The influence of biodiesel on combustion and NOx emissions for the TDI diesel engine type with the CFD code CONVERGE, coupled with a detailed chemical kinetics have identified by Wang et al. [19]. The biodiesel tested use as a mixture with ethyl decanoate, methyl-9-decenoate and n-heptane and their chemical mechanism contains 89 species and 364 reactions. They found that the retarded fuel injection timing reduces NOx emissions as a result of tardy combustion phasing. Som and Longman [20] studied comparison between n-heptane and methyl butanoate biodiesel fuels and concluded that biodiesel has a tendency to reduce soot and NOx emissions. Liu et al [21] presented a 3D CI engine simulations employing a reduced chemistry model for Methyl butanoate/n-heptane blends (145 species and 869 reactions) as a surrogate for biodiesel and showed reasonable agreement with the experimental data suggesting the utility of this model for predictions of combustion and emission characteristics. Jayashankara and Ganesan [22] with the CFD code STAR-CD studied the combustion evolution of a diesel engine by varying the start of injection and intake pressure at an engine speed of 1000 rpm. They reported that the advanced of injection timing can increase in-cylinder pressure, heat release rate, temperature and NOx emissions. They also found that retarded SOI results in a reverse trend. The results show than supercharging increase the NOx emissions due to availability of excess air, on the other hand the soot emissions decrease due to higher rate of soot oxidation. These results are validated with the experimental work of Payri et al. [23]. The impact of fuel ratio and SOI on combustion and emissions for a dual fuel system with port-injection of palm oil methyl ester and DI gasoline fuel is study by Li et al [24] with CFD code coupled KIVA4-CHEMKIN II. The results indicate that the increase in gasoline could also reduce soot emissions indifferent of SOI. Higher soot emissions are detected with advanced SOI compared with conventional SOI. With advanced SOI, spray angle would affect the evaporation process, subsequently, combustion duration and equivalence ratio distributions, and finally, soot emissions. Datta and Mandal [25] investigated the impact of start of injection on performance and emission characteristics of a CI engine fuelled with diesel and methyl soyate and make a comparison between them. The tests were conducted at three different start of injection (17CA, 20CA and 23CA BTDC). The results indicate that advanced SOI decreases the brake thermal efficiency and increases the brake specific fuel consumption for the two fuels. While PM and smoke emissions are less for methyl soyate. On the other hand the emissions of NOx, CO_2 and exhaust temperature are found to be more when compared to diesel.

This paper focuses on well the numerical simulation of a direct injection (DI) diesel engine fuelled with biofuel second generation, eucalyptus biofuel "EB100" to evaluate the combustion and pollutant emissions. The simulation was carriedout using CONVERGE-Computational Fluid Dynamics (CFD) software. The combustion process and emission characteristics of diesel engine are presented and discussed.

2 Experimental setup

A single-cylinder air-cooled Lister Petter TS1 diesel engine developing a power output of 4.5kW at 1500 is utilized for the experimental tests. The schematic of the experimental setup is illustrates in Figure (2), the sensors location, and the engine details are given in Table 1. An electrical dynamometer is used for loading the engine. An orifice meter connected to a large tank is attached to the engine intake manifold to make airflow measurements. In order to get a database experiments are initially carried out using neat diesel and then neat Eucalyptus (EB100) as biofuel. The fuel flow rate is measured on the volumetric basis using a burette and a stopwatch. A chromel-alumel thermocouple in conjunction with a slow-speed digital data acquisition system is used for measuring the exhaust gas temperature. The properties of the Eucalyptus biofuel and neat diesel are presented in table (2). During our tests, the injection timing is set at 20° before TDC. Specifications of the injection data are summarized in table (3).

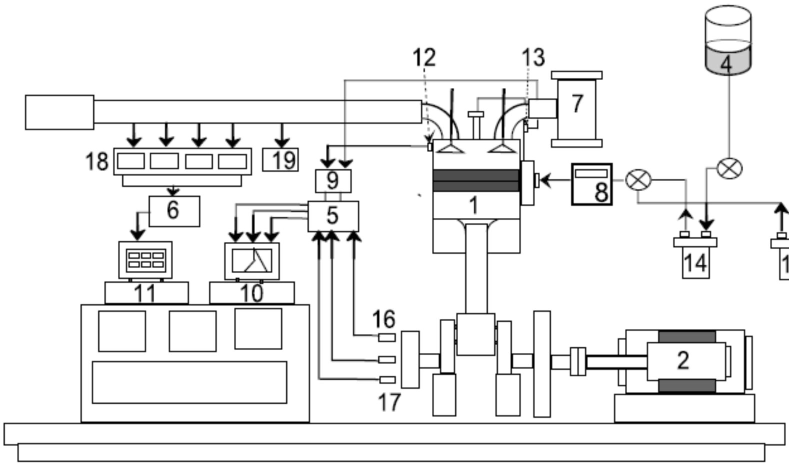


Figure 1schematic of experimental setup

1. Test engine
2. Dynamometer
3. Animal Fat tank
4. Diesel tank
5. A/D card for pressure
6. A/D card for analyser
7. Air tank
8. Burette for fuel
9. Charge amplifier
10. Fast data acquisition system
11. Slow data acquisition system
12. Cylinder pressure sensor
13. Injection pressure sensor
14. Diesel filter
15. Animal fat filter
16. TDC shaft position encoder
17. Speed sensor
18. Exhaust gas analyser
19. Smoke meter

Table 1 *Engine and dynamometer specifications*

Parameters	**Specifications**
Engine type	Lister Petter Type S1
Nozzle opening pressure range	165-245 bars
Governor type	Mechanical centrifugal type
N° of cylinders	Single cylinder
N° of strokes	Four-stroke
Rated Power	5.4 kW (7.3 hp) @ 1800 rev/min
Cylinder Bore	95.25 mm
Upper bowl diameter	45.00 mm

Bowl depth	15.10 mm
Stroke length	88.5 mm
Compression ratio	18:1
Air measurement device	Diaphragm
Eddy current dynamometer	
Model	Eurotherm PARVEX
Type	132 M-G
Maximum power	42 kW

Table 2. Properties of fuels (Neat Diesel and neat Eucalyptus biodiesel).

Average Properties	**EB100**	**Neat diesel**
Density (kg/m3)	896	852
Lower calorific value (MJ/kg)	40	45.76
Viscosity at 40 °C (mPa.s)	2.99	1.57
Cetane number	53	49
Flash point (°C)	105	67

Table3. Fuel System Specifications

Number of Nozzle Holes	3
Nozzle Hole Diameter	0.25 mm
Included Spray Angle	125°
Fuel injection timing	20° BTDC
Injection duration	20°

1. Numerical Simulation

3.1. Computational Modeling

The numerical simulations are carried out under the CFD CONVERGE [26] code environment. CONVERGEV2.1.0 simulate three-dimensional, incompressible/compressible, chemically reacting, transient, and steady-state fluid flows in complex geometries with stationary or moving surfaces. CONVERGE can handle any number of species and chemical reactions, as well as transient liquid sprays, and laminar or turbulent flows. Its sub-models are originally developed for petroleum-based fuels, such as gasoline and diesel. The 3D simulations are performed using the Eulerian–Lagrangian approach. Different sub-models are included in the CONVERGEV2.1.0software as spray injection, atomization and breakup, droplet collision, coalescence and turbulence. The gas-phase flow field is described using the Favre-Averaged Navier–Stokes equations. Details regarding the code and how it works have been mentioned by K. J.Richards [26]. In this work, the physical fuel properties (viscosity, surface tension, density, conductivity, evaporation pressure, latent heat of evaporation and specific heat) of EB100 are determined by Belala [27] and added to the CONVERGE code fuel library (liquid.dat) for modelling the spray, ignition, combustion and NOx emissions in engine combustion chamber. Since intake and exhaust strokes will not be simulated, the valves have been omitted from the computed domain. The CFD code solves the following governing equations.

- Continuity equations for species *m*:

$$\frac{\partial \rho_m}{\partial t} + \nabla(\rho_m . u) = \nabla\left[\rho D \nabla\left(\frac{\rho_m}{\rho}\right)\right] + \dot{\rho}_m^c + \dot{\rho}^s \delta_{m1} \quad (1)$$

Where ρ_m is the mass density of species *m* and ρ is the total mass density, *u* is the velocity $\dot{\rho}_m^c$ is the source term due to the chemistry and $\dot{\rho}^s$ is the source term due to the spray, and δ is the Dirac delta function. By summing the preceding equation over all species we obtain the general fluid density equation:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho . u) = \dot{\rho}^s \quad (2)$$

- The momentum equation is written as a combination of conservative and non-conservative terms as:

$$\frac{\partial \rho u}{\partial t} + \nabla(\rho . u . u) = -\frac{1}{\alpha^2}\nabla p - A_0 \nabla\left(\frac{2}{3}\rho k\right) + \nabla\sigma + F^s + \rho g \quad (3)$$

Where p is the fluid pressure, F^s is the rate of momentum gain per unit volume due to the spray, $A_0 = 1.0$ when the turbulence is activated.

- Finally the internal energy conservation is:

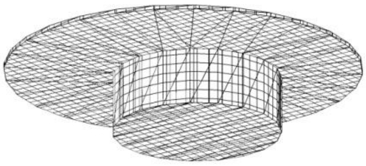
$$\frac{\partial \rho I}{\partial t} + \nabla(\rho u I) = -p\nabla . u + (1 - A_0)\sigma : \nabla . J + A_0 \rho \varepsilon + \dot{Q}^c + \dot{Q}^s \quad (4)$$

Where $I[J/Kg]$ is the internal energy, exclusive of chemical energy. The heat flux vector $J[J/s]$ is the sum of contributions due to heat conduction and enthalpy diffusion terms. $\dot{Q}^s[J/m^3.s]$ and $\dot{Q}^c[J/m^3.s]$ are terms due to spray interactions and chemical heat, respectively.

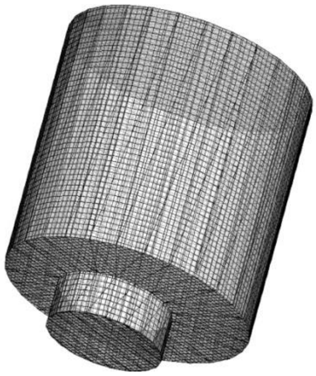
The 3-D simulations were performed using the RANS approach with considering compressible fluid, non-stationary regime and ideal gas assumption in the computational fluid dynamics (CFD) software CONVERGE. It incorporates models for spray injection, atomization and breakup, turbulence, droplet collision, and coalescence. The gas-phase flow field is described using the Favre-Averaged Navier–Stokes equations in conjunction with the RNG *k-ε* turbulence model, which includes source terms for the effects of dispersed phase on gas-phase turbulence. These equations are solved using a finite volume solver. Kelvin Helmholtz (KH) and Rayleigh Taylor (RT) models are used to predict the primary and secondary breakup of the computational parcels [26]. For combustion, the multi-scale CTC (Characteristic Time Combustion) model associated with the Shell ignition model [26] is used. Currently, CONVERGE includes sub-models for both soot and NOx production. For NOx prediction, extended Zeldovich NOx model [26] is adopted. The extended Zeldovich mechanism as off rired by Heywood (1988) is employed to calculate NO formation [28]. Finally, the Hiroyasu soot model is used for soot prediction [26]. Details of these models are described in reference [26], hence only a brief description is provided here.

3.2. Meshing of Computational Domain

CONVERGE is a Computational Fluid Dynamics tool for the simulation of three-dimensional, incompressible/compressible, chemically reacting transient fluid flows in complex geometries with stationary or moving surfaces. The mesh was created such that the boundary was flagged and adjusted to have the real geometry (bore, stroke, etc,…). The adaptive mesh refinement (AMR) technique in CONVERGE enables local mesh refinement according to temperature, velocity and species gradients [26]. The base grid resolution is specified in an input file, thus allowing for grid resolution studies to be performed without making separate meshes. The geometry volume is always correctly calculated, allowing for extremely course meshes to be used while setting up a case. For more information, see Ref [26]. In the actual study, the AMR is activated for velocity and the temperature at 20°C CA BTDC. AMR algorithm adds automatically the setting where the flow field is most under-resolved or where the sub-grid is the largest. Consequently, the cell size and the total number of cells are not constant and change over the simulation process. The total cells number it 7216 at 4 CA BTDC and 85336 at 150 CA ATDC, figure (2). All meshes are created using the pre-processor of CONVERGE2.1.0V. Note that the combustion chamber is not axisymmetric; a 360° full mesh was constructed.



Mesh at 4 CA BTDC



Mesh at 150 CA ATDC

Figure 2 Grid generated in CONVERGE at different times

3.3. Boundary conditions

In order to solve the previous governing transport equations, boundary conditions must be specified for each equation. Three types of wall boundary are chosen. Only the first one (piston head) is set as translate and other boundaries were set as stationary and smooth boundaries. The standard wall law is a logarithmic curve fit of a turbulent boundary layer. In practice, the wall law profile is used to determine the tangential components of the stress tensor at the wall. The gas temperature and velocity at the wall are set as below:

Piston wall: moving wall; $U_{gas}=U_{piston};T_{piston}=435K$;

Cylinder wall: stationary wall (not moving); $U_{gas}=0$; $T_{cylinder\text{-}wall}= 413K$

Cylinder head: stationary wall; $U_{gas}=0$; $T_{cylinder\text{-}head}= 355K$

Initial conditions in the combustion chamber are:

350 K	gas temperature
1.0070e+05 Pa	gas pressure
62.02710 m²/s²	initial turbulent kinetic energy
17.18340 m²/s²	initial turbulent dissipation

The injection profiles are shown in fig 3, 4, details of the injection data are presented in table 3

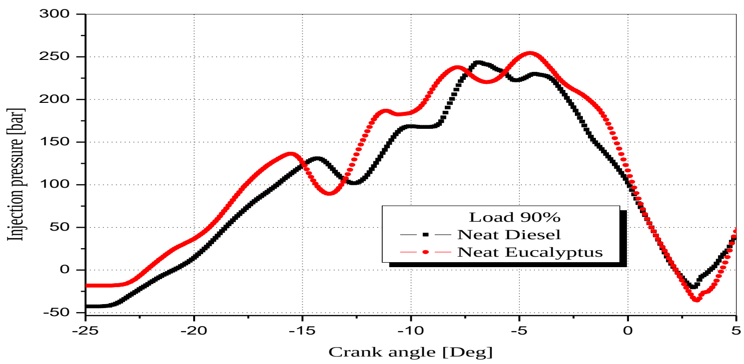


Figure3 Injection pressure for EB100 and neat diesel at load 50%

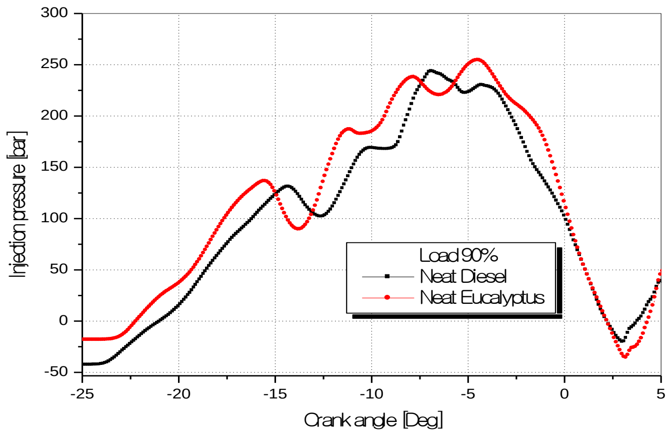


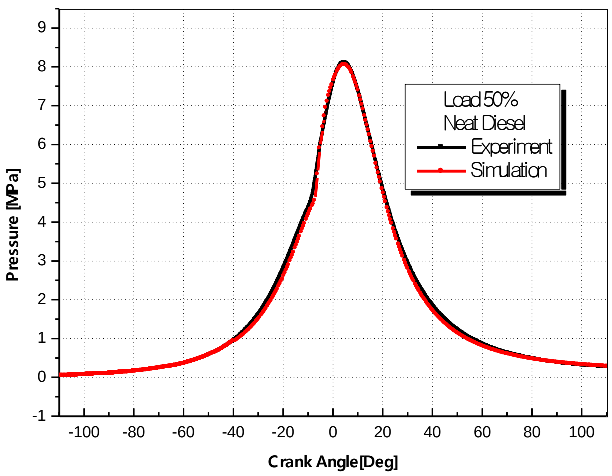
Figure 4 Injection pressure for EB100 and neat diesel at load 90%

2. Results and Discussions

4.1. Combustion characteristics

This section will discuss the combustion characteristics of the studied fuels. The present simulation results are validated by comparing with experimental. The results obtained for average pressure at 50% and 90% of full load for EB100 and neat diesel fuels are compared in Figures (5) to (6). The simulation results are in reasonably good agreement with the experimental results and the average error does not outstrip (3%) for both the cases especially in the region of peak cylinder pressure.In-cylinder pressure evolution is the most extensively used indicator for modeling accuracy relative to experimental data. These results indicate that the sub-models and the boundary conditions used in this investigation reflect correctly the engine operation.

In these figures it is clear that peak pressure of EB100 is higher over 3 bars for 50% load and over 6 bars for 90% of full load and fuel combustion starts at 4–5 crank angle degrees before top dead center for 50% load and 4–5 crank angle degrees after top dead center for 90% of full load.



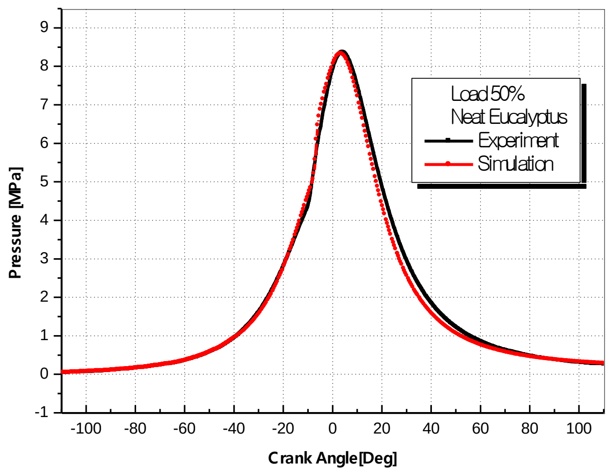


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Figure.5. Comparison of predicted and measured in-cylinder pressure at load, 50%.
a) Neat diesel case. b) EB100 case

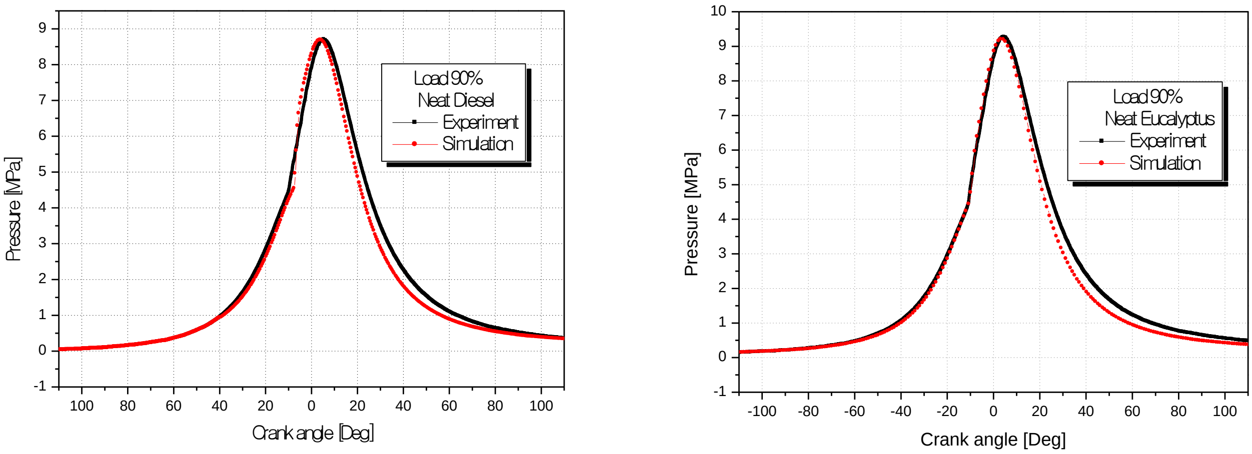

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Figure.6. Comparison of predicted and measured in-cylinder pressure at load, 90%.
a) Neat diesel case. b) EB100 case

Figures (7a) and (7b) illustrate a comparison of heat release rates at 50% and 90% full load. Both for neat diesel and EB100 fuels it is observed a rapid premixed burning followed by diffusion combustion as is typical for a compression combustion engine. After the ignition delay period, the premixed fuel-air mixture burns rapidly releasing heat with a very rapid rate and after this premixed combustion phase, non-premixed combustion regime takes place, where the burning rate is controlled by the availability of fuel-air mixture. By analyzing these diagrams, it can be deduced easily that when engine is fueled with EB100, the combustion starts earlier. These diagrams again reconfirm early onset of heat release for EB100 which shows shorter ignition delay compared to neat diesel. The maximum of the premixed combustion heat release rate is higher for EB100, which is responsible for higher peak pressure. Thus, air entrainment and fuel-air mixing rates along with rapid burning nature with EB100. This leads to more fuel being prepared for rapid combustion with EB100 after the ignition delay despite its higher viscosity level in comparison with the neat diesel fuel. Significantly higher EB100 combustion rates during this phase leads to higher exhaust temperatures. After the top dead center, more burning occurs in the diffusion phase with neat diesel as shown in (7a) and (7b). The figures (8a) and (8b) shows a comparison between EB100 fuel mode and neat diesel mode about the evolution of in-cylinder average temperature at 50% and 90% of full load . It is observed a higher level for EB100 since starting combustion. The maximum difference in temperatures was 73 K at 50% full load. The higher temperatures reached with the EB100 fuel may explain the higher levels of the produced NOx at high and partial loads.

4.2. Emissions

Some results concerning soot and NOx pollutants are respectively presented in the figures (9) to (10) at 50% and 90% of full load. The higher temperature level obtained when engine is fueled by the EB100 lead to higher NOx level at the end of combustion process at 50% load and becomes lower at 90% of full load, figure (16). However, the soot emissions reach to their maximum at the end of the premixed combustion phase but decrease slowly to become lower at the end of the non-premixed phase. It is seen that the soot generated by EB100 fuel are lower than neat diesel case due to oxygen content in the eucalyptus which helps in more complete oxidation of the fuel and reduces soot concentration in the exhaust gas. This result confirms earlier studies on various biofuels studied by several searchers [29-30].

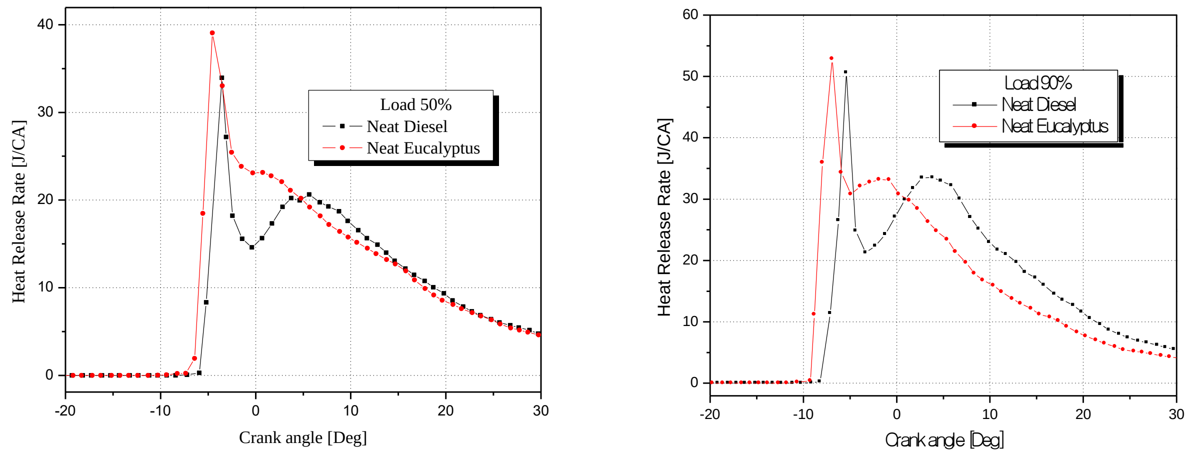

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Figure.7. Comparison of predicted heat release rate profiles for Neat Diesel and EB100 fuels,
a) *At load 50%. b) at load 90%..*

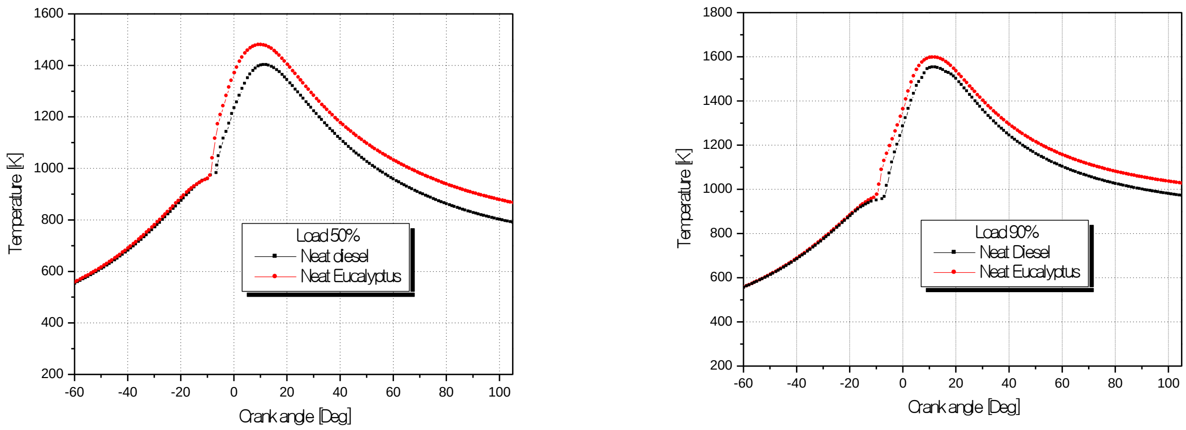

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Figure.8. Comparison of predicted temperature profile for Neat Diesel and EB100 fuels,
a) *at load 50%. b) at load 90%..*

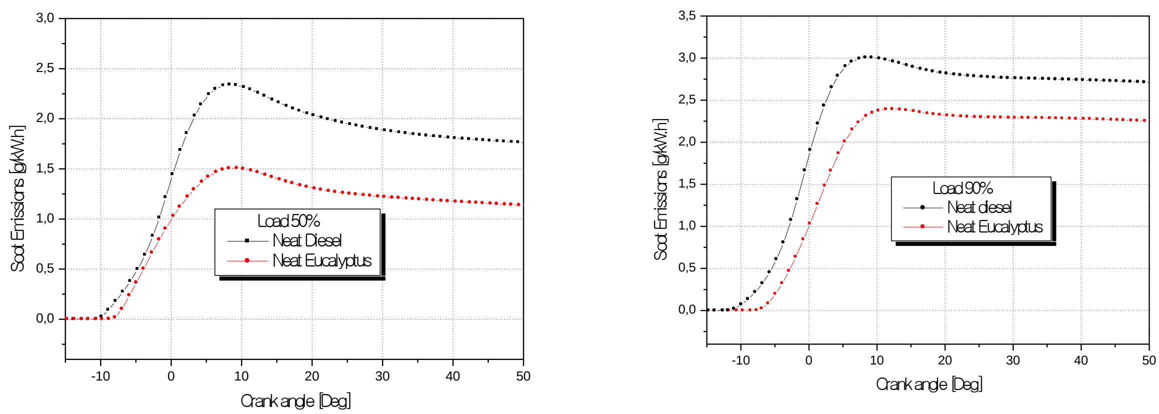

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Figure.9. Prediction of the soot emissions a) at 50% engine load. b) at 90% engine load

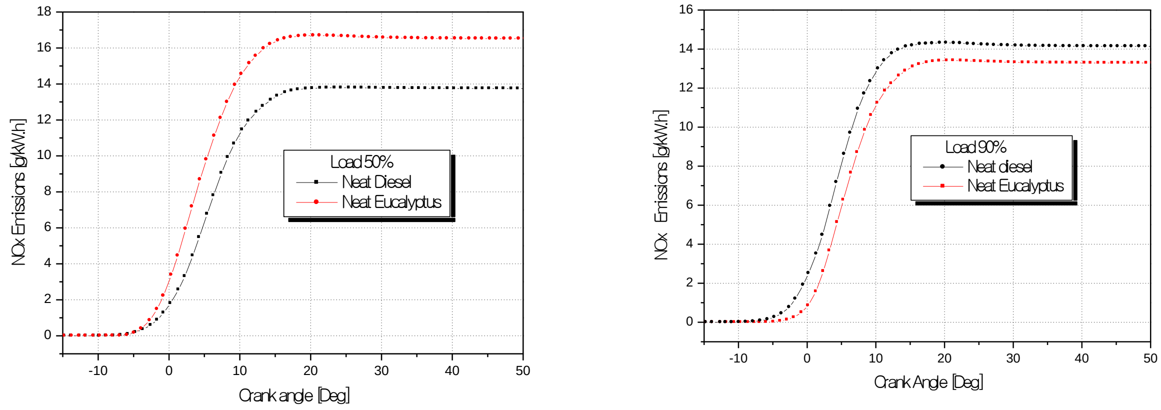

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Figure.10. Prediction of NOx emissions a) at 50% load. b) at 90% load

5. Conclusions

The second generation biodiesels have become the leading raw materials for obtaining biodiesel due to an increase in world's energy demand, competition of edible oil sources for human use and biofuel production as well as environmental pollution. Biofuels synthesized from vegetable oil constitute an interesting alternative to neat diesel fuel. They are biodegradable; nontoxic with low emission, hence they are environmentally beneficial. For the methods of conversion, the transesterification method is the most suitable method among the several possible methods for biodiesel production. In this study, CONVERGE CFD software is utilized for modeling processes in a single cylinder diesel engine fuelled with neat eucalyptus biodiesel. CONVERGE includes advanced numerical techniques and physical models describing processes of spray, turbulence and combustion, and the nonlinear interactions of such processes. The objective of modeling biofuel combustion is to gain better understanding of the combustion and emissions behavior in biofuel engines.

The following conclusions are pulled based on the results of the numerical investigations:

- For the numerical investigations very good agreement was obtained between the predicted and measured in-cylinder pressure at 50% and 90% of maximum load engine.
- Overall, the peak pressure and the ant the heat release rate in EB100 is higher than neat diesel.
- For diesel engine soot and NOx constitute a major problem to investigate. The results show a severe antagonism between soot and NOx. The effect of injection timing on soot and NOx concentrations is investigated in this work and simultaneous reduction of soot and NOx formation are achieved.
- The EB100 fuel decreases the soot emissions but increases the NOx levels at 50% of full load, on the other side, this biofuel decreases the soot and NOx emissions at 90% of maximum load.
- Shorter ignition delays are observed for the EB100 fuel case as compared to the diesel fuel. The heat release rate is higher for EB100 than diesel fuel.

Nomenclature

α	Diffusion coefficient	[$W/m.K$]
F^s	Rate momentum gain per unit volume due to the spray	[$Kg/m^2.s^2$]
p	Pressure	[Pa]
g	Specific body force	[m/s^2]
I	Specific internal energy	[J/Kg]
T	Temperature	[K]
$\dot{Q}^s$	Spray source term	[$J/m^3.s$]
$\dot{Q}^c$	Chemical source term	[$J/m^3.s$]
J	Heat flux vector	[J/s]
k	Turbulent kinetic energy	[m^2/s^2]
ε	Turbulent dissipation rate	[m^2/s^2]
σ	Viscous stress tensor	[N/m^2]
ρ	Density	[Kg/m^3]
ρ_m	Density of chemical species *m*	[Kg/m^3]
ρ_m^s	Mass density source term for species *m* due to Spray	[Kg/m^3.
ρ_m^c	Mass density source term for species *m* due to chemistry	[Kg/m^3.
t	Time	[s]
u	Gas velocity components	[m/s]

Notations

AMR: Adaptive Mesh Refinement.
ATDC: After Top Dead Center
BTDC: Before TDC.
CA: Crank Angle
CFD: Computational Fluid Dynamic.
CI: Compression Ignition.
CTC: Characteristic Time Combustion.
CO: Carbon monoxide
CO2: Carbon dioxide
DI: Direct Injection
EB: Eucalyptus Biodiesel
IC: Internal combustion
RANS: Reynolds-averaged Navier Stokes.
RNG: Re-Normalization Group
TDI : Turbocharged Direct Injection
TKE: Turbulent kinetic energy
NOx: Oxides of Nitrogen
PM: Particulate Matter
RME: Rapeseed methyl ester
SOI: Start of Injection
SME: Soybean Methyl Ester
EGR: Exhaust Gas Recirculation

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