PAPER • OPEN ACCESS

Performance and emission of generator Diesel engine using methyl esters of palm oil and diesel blends at different compression ratio

To cite this article: M Aldhaidhawi *et al* 2016 *IOP Conf. Ser.: Mater. Sci. Eng.* **147** 012135

View the article online for updates and enhancements.

You may also like

- The effect of compression ratio on the performance and emission characteristics of C.I. Engine fuelled with corn oil biodiesel blended with diesel fuel.
  Abdulrahman Shakir Mahmood, Haqi I. Qatta, Saadi M. D. Al-Nuzal et al.

- Effect of Variable Compression Ratio on Performance of a Diesel Engine Fueled with Karanja Biodiesel and its Blends
  Rahul Kumar Mishra, Tarun soota and Ranjeet singh

- Synthesis of Biodiesel from Low-Quality Crude Palm Oil with Heterogeneous Catalyst Cao-ZnO
  Aman Santoso, C.F Aghnia Hanindita, Sumari et al.

Performance and emission of generator Diesel engine using methyl esters of palm oil and diesel blends at different compression ratio

M Aldhaidhawi[1, 2], R Chiriac[1], V Bădescu[1], H Pop[1], VApostol[1],A Dobrovicescu[1], M Prisecaru[1], A A Alfaryjat[1], M Ghilvacs[1] and A Alexandru[1]

[1]Applied Thermodynamics, Engines, Thermal and Refrigerant Equipment Department University POLITEHNICA of Bucharest Faculty of Mechanical Engineering and Mechatronics, 313 Splaiul Independentei, sector 6, Bucharest, 060042 Romania
[2]Ministry of Higher Education and scientific research, Al-Furat Al-Awsat Technical University, Najaf, Iraq

E-mail: mohanadhamazah@gmail.com

Abstract. This study proposes engine model to predicate the performance and exhaust gas emissions of a single cylinder four stroke direct injection engine which was fuelled with diesel and palm oil methyl ester of B7 (blends 7% palm oil methyl ester with 93% diesel by volume) and B10. The experiment was conducted at constant engine speed of 3000 rpm and different engine loads operations with compression ratios of 18:1, 20:1 and 22:1. The influence of the compression ratio and fuel typeson specific fuel consumption and brake thermal efficiency has been investigated and presented. The optimum compression ratio which yields better performance has been identified. The result from the present work confirms that biodiesel resulting from palm oil methyl ester could represent a superior alternative to diesel fuel when the engine operates with variable compression ratios. The blends, when used as fuel, result in a reduction of the brake specific fuel consumption and brake thermal efficiency, while NOx emissions was increased when the engine is operated with biodiesel blends.

1. Introduction

Today, energy use in the most fundamental requirement for human existence, for tasks such as food production, agricultural production, and generation of electricity, thermal power plants, transportation sector, and it is actually the most important requirement for the industrial sector. The majority of the global overall energy output is generated from conventional fuels such as gasoline and diesel. It is acknowledged by everyone the fossil fuels are limited in quantity, and although the world is not endowed with new sources of fossil fuels, experts have carried out research on the depletion of existing sources. Moreover, it has raised the risk of instability of supply and increased public awareness about the effects of fossil fuel emissions on the environment and the potential health risks of governments around the world, and this has led to restrictions being imposed on emissions resulting from fossil fuel combustion.

During the past 20 years, much more information has become available on biodiesel as a fuel for diesel engines. New and innovative fuel sources are currently being developed in attempt to find alternative energy sources to replace conventional fossil fuels. One of the most promising sources of biodiesel has attracted much attention as a potential resource for the production of an alternative to

diesel fuel, which depends on vegetable oil and animals fats .The advantage of using biodiesel as fuels in diesel engine is that biodiesel has higher lubricity than standard diesel fuel reducing the wear and tear on engines, higher cetane number help to improve the combustion, in addition, biodiesel consider renewable energy source and non-toxic. Whereas there are some obstacles associated with the use of these types of fuels such as higher viscosity, higher density and lower calorific value compared to the diesel fuel.

There have been several studies in literature [1, 2, 3] reporting that the higher viscosity of biodiesel fuel is considered the major disadvantage of biodiesel, as this effects fuel flow, especially in cold weather conditions. This property influences atomization quality, poor evaporation and slower mixing with air, exhibits a lower spray cone angle, increased average droplet diameter, and longer tip penetration of the sprayed injected fuel. Many studies [4, 5] have primarily focused on engine output, exhaust gas emissions and engine coke-up when using biodiesel, especially in higher percentage mixes with diesel, and they reported observing further degradation of the engine output, increased emissions due to combustion quality deterioration and engine coke-up after long engine running times, this behaviour being explained by biodiesel oil's higher viscosity and lower heating value.

Recently, researchers have shown an increased interest in mathematical modelling, owing to the continuously increasing improvements in computational power, which enabled new research in the field of internal combustion engines by using computer simulations. Numerical methods are techniques by which mathematical problems are formulated so that they can be solved with arithmetic operations [6]. Diesel engines occupy a prominent role in the present transportation and power generation sectors. Many methods have been tried and are in use to reduce pollutant emissions from diesel engines. The main options to reduce pollutants are the usage of biofuels and adopting some modifications to the combustion process [7]. Diesel engine simulation models can be used to understand combustion performance; these models can reduce the number of experiments.

Jagadish D et al. [8] developed a theoretical model for the combustion performance of a single cylinder direct injection diesel engine fuelled by biofuels, with options such as supercharging and exhaust gas recirculation. They observed that the model is successful in predicting the engine performance with biofuels when compared with the experimental approach. Patil S [9] developed thermodynamic modelling for performance analysis of a compression ignition engine fuelled with Biodiesel and its blends with Diesel, and validation was done by comparing the predicted parameters, such as brake thermal efficiency and in-cylinder pressure, with the experimental results, which were found to be a close approximation.

The main objective of this work is to model engine performance based on a single zone closed system for a single cylinder compression ignition engine, and to compare it with the experimental results when using palm oil biodiesel B7 (7% palm oil +93 Diesel) and B10 by volume as a fuel, at a constant speed of 3000 rpm at different loads and varied compressions of 18:1, 20:1 and 22:1.

2. Mathematical modeling
The study presents a developed modeling to predict the performance and exhaust gas emissions of a compression ignition engine fuelled by palm oil methyl ester at different blends. This modeling is carried out on the basis of the first law of thermodynamics. A thermodynamic model is developed with the aid of Matlab code, Zero-dimensional flow conditions inside the cylinder, single zone, a closed cycle model. The assumptions used in this model are: The gas in the cylinder moves through the equilibrium states, no gas leakage through the valves and piston rings so that the mass remains constant.

2.1. Energy conservation
The energy balance equation for a closed system, according to the first law of thermodynamics can be written as:

$$\frac{dQ}{d\theta} = m\frac{dU}{d\theta} + \frac{dw}{d\theta}.$$ (1)

The rate of heat release is split into the heat released due to combustion (dQ_c) of the fuel and the heat transfer that occurs to the cylinder walls (dQ_h). The equation (1) can be written as:

$$\frac{dQ_c}{d\theta} - \frac{dQ_h}{d\theta} = mCv\frac{dT}{d\theta} + p\frac{dV}{d\theta}.$$ (2)

After simplifying equation (2) and taking into consideration the ideal gas law and rate of heat transfer, we obtain an equation to determine the temperature as a function of the crank angle:

$$\frac{dT}{d\theta} = \frac{1}{mCv}\frac{dQc}{d\theta} - \frac{Ah(T_g - T_w)}{mCv} - \frac{RT}{CvV}\frac{dV}{d\theta}.$$ (3)

The pressure was calculated by the following equation [8]:

$$\frac{dp}{d\theta} = \left(\frac{dQ_c}{d\theta} - \frac{dQ_h}{d\theta} - \frac{\gamma}{\gamma-1}p\frac{dV}{d\theta}\right)\frac{\gamma-1}{V}$$ (4)

Where: Cv is the specific heat at constant volume, $\frac{dQ_c}{d\theta}$ the rate of heat release due to combustion of fuel, $\frac{dQ_h}{d\theta}$ rate of heat transfer to the wall, m mass (kg), A cross section area, h heat transfer coefficient, T_g gas temperature, T_w wall temperature.

2.2 Instantaneous cylinder volume V (θ)
The engine cylinder volume (instantaneous volume) at any crank angle is calculated from the given equation [10].

$$V(\theta) = v_s\left[\frac{Cr}{Cr-1} - \frac{1-\cos\theta}{2} + \frac{L}{s} - \frac{1}{2}\sqrt{\left[\left(2\frac{L}{s}\right)^2 - \sin^2\theta\right]}\right]$$ (5)

Where, Cr – Compression ratio L – Connecting rod length s - Stroke θ – Crank angle position

2.3 Combustion Process
The combustion process in a diesel engine is considered very complex and heterogeneous. Combustion is modeled using a single zone approach, which is based on a uniformly distributed heat releasing phenomenon. The Wiebe function is used to calculate the heat release rate due to fuel combustion, the mass burned fraction (x_b) which is expressed in terms of [11]:

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^{m+1}\right].$$ (6)

Then the heat release rate is determined by differentiating the mass fraction x_b with respect to $\frac{\theta - \theta_s}{\Delta\theta}$ and multiplying $\frac{Q_{av}}{\Delta\theta}$

$$\frac{dQ_c}{d\theta} = a(m+1)\left(\frac{Q_{av}}{\Delta\theta}\right)\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^m \exp\left[-a\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^{m+1}\right]$$ (7)

Where: θ_s start of combustion, $\Delta\theta$ combustion duration. The shape parameter in Vibe function was set at 2 for this purpose and the Vibe parameter at 6.9[15]. Q_{av} is the heat released per cycle.

2.4 Ignition delay
Ignition delay is defined as the period between the start of fuel injection and start of combustion. The empirical formula is used in the present model as [12]:

$$\tau_{id} = \left(0.36 + 0.22U_p\right)\exp\left[EA\left(\frac{1}{RT} - \frac{1}{17190}\right)\left(\frac{21.2}{p-12.4}\right)^{0.63}\right]$$

(8)

Where: τ_{id} ignition delay time, T is the mean temperature in Kelvin, p is the mean pressure in bar and U_p piston speed (m/s), E_A is apparent activation energy

2.5 Heat transfer
Anand's equation was used to determine the heat transfer between gases produced due to the chemical process inside the combustion chamber and the cylinder wall, which is shown in the following equation [13]:

$$\frac{dQ_h}{d\theta} = A\left[a\frac{k}{B}\left(\frac{\rho UpB}{\mu}\right)^b \left(T_g - T_w\right) + c\left(T_g^4 - T_w^4\right)\right]$$

(9)

Where:
k Thermal conductivity, ρ gas density, μ Viscosity, Up mean piston speed, B cylinder bore,
a Parameter value for combustion equal to $0.35 \leq a \leq 0.8$,b Parameter value equal to 0.7 , Parameters value where c=0 for the compression period and otherwise c=3.3 x10^{-8}W/$m^2 K^4$.

2.6 Frictional power PF.
The study of engine frictionhas been a topic of research for many years were has an effect on engine efficiency. The total losses of power due to friction in different moving parts are calculated by using the following empirical relation [14]:

$$FP = C + 1.44\frac{C_m *1000}{B} + 0.4*\left(C_m\right)^2$$

(10)

Where: *FP* is the total friction power loss and C is a constant, which depends on the engine type C= 75 kPa for direct injection engine [14].

2.7 NOx emissions
The mechanism of the formation of NO has been modeled based on the extended Zeldovich mechanism [12].
- O + N2 = NO + N
- N + O2 = NO + O
- N + OH = NO + H

The NO formation rate is given as:

$$\frac{d[NO]}{dt} = \frac{6x10^{16}}{T^{\frac{1}{2}}}\exp\left(\frac{-69090}{T}\right)[O_2]_e^{1/2}[N_2]_e$$

(11)

The $\dfrac{d[NO]}{dt}$ dependence on temperature in the exponential term is evident. High temperatures and high oxygen concentrations result in high NO formation rates.

The characteristic time for the NO formation process is:

$$\tau_{NO} = \frac{8.10^{-16}\, T \exp\left(\dfrac{58300}{T}\right)}{p^{1/2}} \tag{12}$$

By multiplying this time with NOx formation rate we can get the NOx in to PPM as:

$$NOx(PPM) = \left(\frac{d[NO]}{dt}\right)\!.\tau_{NO}.10^6.\left(\frac{1000}{3600}\right) \tag{13}$$

Where: *PPM* mean parts per million.

3. Methodology

For the purposes of observing and conducting the present work, it is important to explain how this study was conducted. Firstly, internal-combustion engine is simulated by using the Matlab program. Secondly, experiment was be carried out on one cylinder four stroke diesel engine at constant load, 3000 rpm engine speed and different compression ratios, the numerical results were validate against the experimental result.

3.1. Simulation

A computer program has been developed by using Matlab software to produce the numerical solutions of the equations used in the thermodynamic model which were described in thesection 2. Brake specific fuel consumption and brake thermal efficiency based on a thermodynamic model which was developed by using the first law of thermodynamics. During the investigation, theoretical results are predicted for various blends of palm oil methyl ester with diesel namely D, B7 and B10 (pure Diesel, 7% palm oil +93% Diesel, 10% palm oil +90 Diesel respectively) by volume at compression ratios (22:1 ,20:1 ,18:1).

3.2. Experimental

An experiment to examine performance was conducted on a four stroke, single-cylinder diesel engine, the main engine specifications are given in Table 1. The experimental test was done in the Automotive Development Center in University of Technology of Malaysia (UTM)   for a variety of fuels such as diesel fuel (D) and, B7 and B10 palm oil methyl ester. The major properties that were tested in the Chemical Engineering, University of Technology of Malaysia (UTM) for the diesel and blends are listed in Table 2. Fuel consumption was measured by determining the time taken by the diesel engine to consume a certain amount of fuel. The engine's RPM was also monitored by using a tachometer. Two fuel tanks were used to supply fuel to the test engine, one for diesel fuel and the other for biodiesel blends. The engine is coupled to an eddy current dynamometer, the inlet and outlet temperatures measured by using a thermocouple. A pressure transducer was placed inside the cylinder head to measure the pressure inside the cylinder. The computer based data acquisition system is the SPECTRUM (MI.3112CA) card installed on a DEWE-5000 portable data acquisition system to collect and analyze the results, The TELEGAN emission analyzer was used to measure the exhaust gas emission.The engine was initially fuelled with diesel fuel until it achieved engine operation stability. All tests were done with diesel fuel in order to provide the baseline data and then the fuel was switched to B7 and B10 biodiesel blend fuels. Before stopping the test engine after each test with biodiesel blend fuels, the engine was switched back to diesel fuel operation until all the biodiesel based blend was purged from the fuel lines, injection pump and injector, to avoid clogging. The engine

compression ratio (CR) was changed from 22:1, 20:1 (original designed by manufacture) and 18:1. The change in compression ratio was done by changing the cylinder head gasket thickness (made from copper) to achieve the appropriate compression ratio.

Table 1. Yanmar l70N6-MTRIYJ engine specifications made in Italia.

	4 stroke, vertical cylinder diesel
No. of cylinder	1
Bore x stroke	78 x 67mm
Displacement	0.320 L
Combustion system	Direct injection
Cooling system	Forced air by flywheel fan
Maximum engine speed	3600(rpm)
Starting system	Electric start/Recoil start
Max Rated output(KW]	4.9KW @3600 rpm

Table 2. Palm oil Biodiesel and Diesel fuel properties.

	Diesel	B7	B10
Viscosity @40 ℃ [cst]	5.17	5.048	5.019
Density@15 ℃ [kg/m3]	844.3	845.3	846.9
Cetane Number	47	50.8	51.7
Calorific value [MJ/Kg]	45.652	45.235	45.012

4. Result and discussion

A numerical investigation on the effect of changing the compression ratio on engine performance and exhaust gas emission has been conducted on a single-cylinder four stroke diesel engine. The numerical result was compared with the experimental result listed in Table 3 to test the usefulness of this model. The brake specific fuel consumption, brake thermal efficiency and nitrogen oxide (NOx) are studied. All data was taken at constant engine speed (3000rpm), varied compression ratios (CR) of 18:1, 20:1 and 22:1. 70% engine loads were applied to each CR and fuel supplies of pure diesel, B7 and B10, where they were varied in order to represent the actual application of the engine throughout its CR range.

Table 3. Experimental result obtained from test engine.

| Compression ratio	ALPHA %	N RPM	BSFC g/kW.h	BTE %	NOx ppm
Pure Diesel					
22:1	70	3000	370.396	22.7	220
20:1	70	3000	403.17	22.3	191
18:1	70	3000	456.69	21.9	145
Biodiesel B7 (7% palm +93%Diesel)					
22:1	70	3000	355.502	22.4	253
20:1	70	3000	413.583	22.1	242
18:1	70	3000	464.86	21.3	233
Biodiesel B10 (10 % palm +90%Diesel)					
22:1	70	3000	373.4	21.7	298
20:1	70	3000	411.54	21.5	220
18:1	70	3000	467.393	20.6	200

4.1 Cylinder Pressure

Experimental data for B10 biodiesel fuel relative to the crank angle at 70% load and 3000 rpm and compression ratio 22:1 has been used to fit the simulated cylinder pressure curve. The resulting model is then validated with the experimental pressure data. Figure 1 shows the cylinder pressure and crank angle. It can be seen that the model shows a good agreement between experimental versus simulated data for 70% load at constant speed, particularly for maximum pressure and respective crank angle. The same trend is observed for all the test fuels and compression ratios.
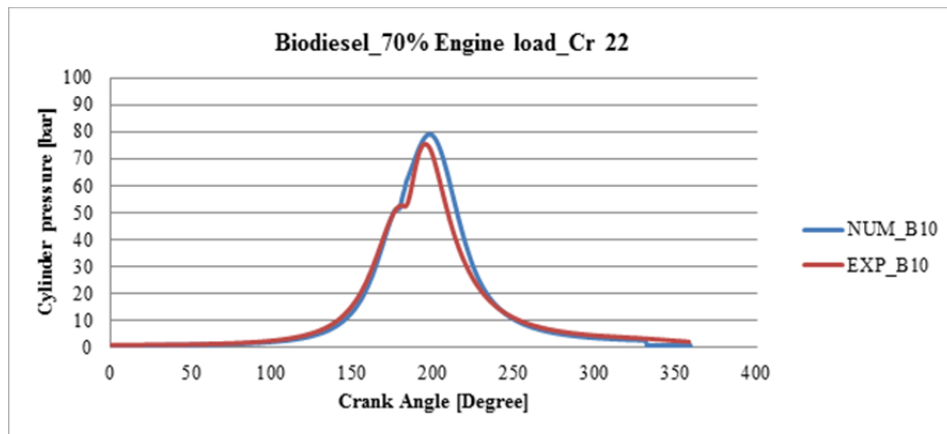
Figure 1. Comparison between experimental and simulation pressure traces
for 70% load, 3000 rpm speed.

4.2 Brake Specific Fuel Consumption (BSFC)

Figures (1-4) compare the variation of Brake Specific Fuel Consumption (BSFC) with compression ratio for Diesel fuel (D), Biodiesel B7 and B10 numerical and experimental results. It can be observed that the BSFC is a clear indication of the efficiency with which the engine develops power. The smaller BSFC indicates the more effective use of fuel to generate power. For all fuels tested, the BSFC achieves an optimum value at CR 22. This is due to an increase in temperature in the combustion chamber, leading to complete combustion. The same trend is observed for all the test fuels. The max relative error between experimental and numerical result was 8.2 % at CR 20, biodiesel B7 fuel, while the minimum error was 1.3 at CR 18, Diesel.
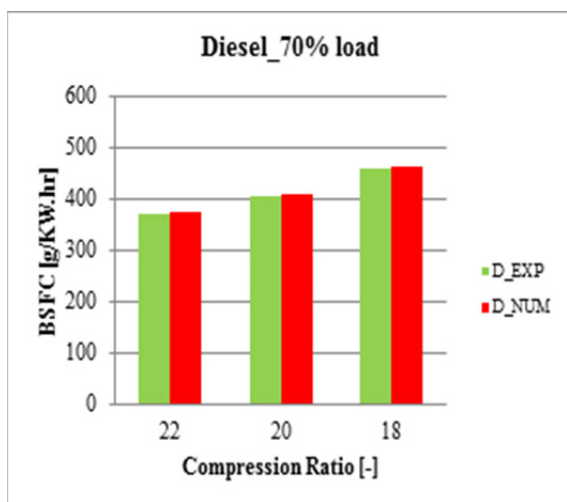
Figure 2: BSFC comparison values for diesel (D).
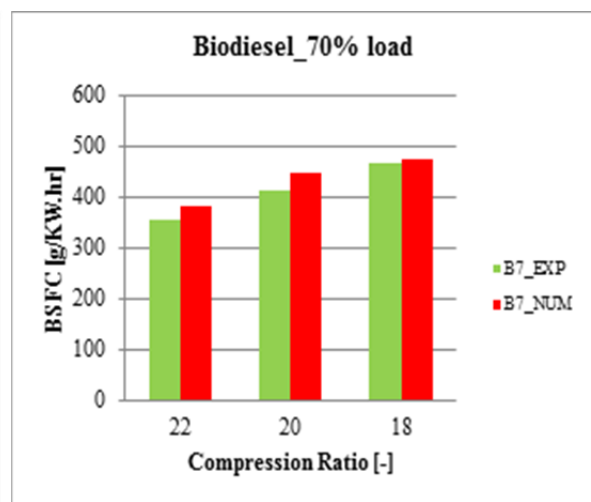
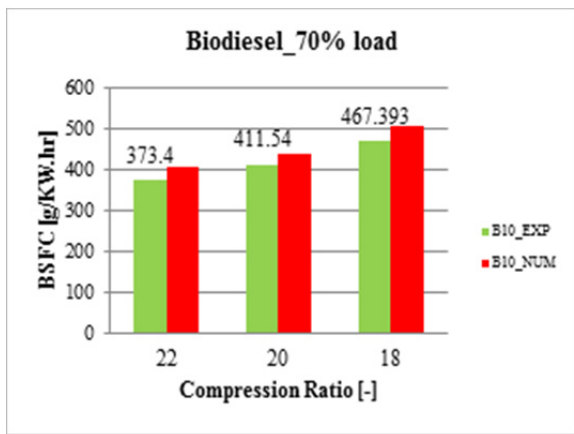**Figure 3**: BSFC comparison values for Biodiesel (B7).
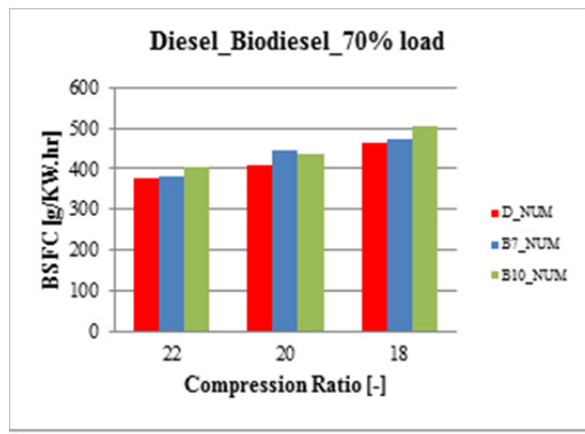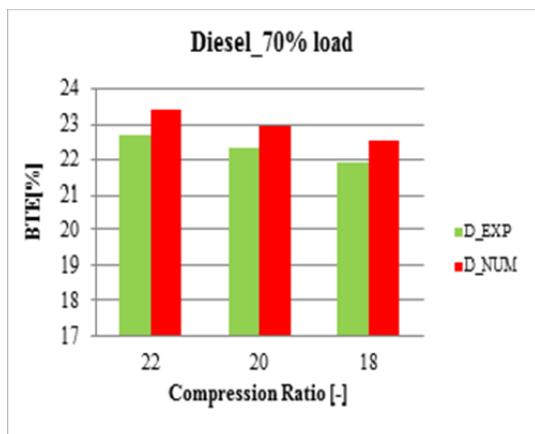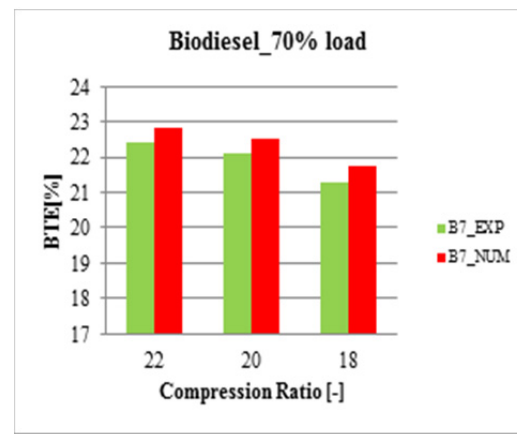
Figure 4: BSFC comparison values for Biodiesel (B10).	Figure 5: BSFC comparison values for (D, B7, B10).

4.3. Brake Thermal efficiency (BTE)

The variation in brake thermal efficiency (BTE) with CR for diesel and blends fuels tested is shown in figures (6-9). From these figures we can clearly observe that the BTE of biodiesel at all blends was found to be lower than diesel for all CR monitored. This might be due to lower fuel heat value and so higher fuel consumption of the biodiesel blends to produce the same power.

Figure 6: BTE comparison values for Biodiesel (D).	Figure 7: BTE comparison values for Biodiesel (B7).
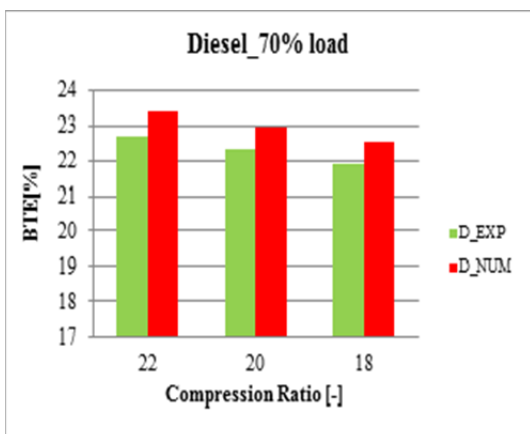
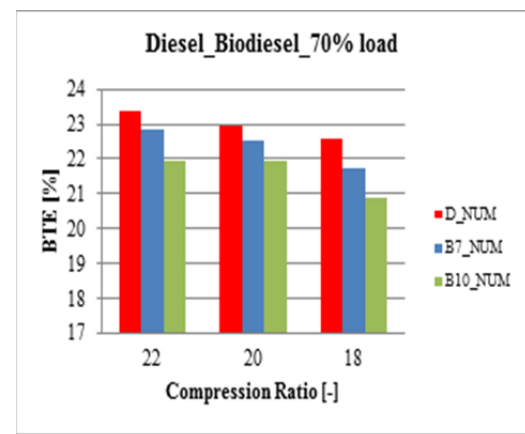| Figure 8: BTE comparison values forBiodiesel (B10).	Figure 9: BTE comparison values for (D, B7, B10).

The max relative error between experimental and numerical results was 7% at CR 20, Diesel fuel, while the minimum error was at CR 20, Biodiesel B10. The same trend is observed for all the test fuels   and   compression   ratios.

4.4. Oxide of Nitrogen (NOx)
The variation of Nitrogen Oxide (NOx) with respect to CR for different blends at engine speed 3000 rpm and constant load for the experimental and numerical studies is shown in figures (10-13). The NOx emission for diesel and other blends increased when the CR is increased and it was observed that NOx increased when the biodiesel percentage increases. This is due to the fact that biodiesel has 12% more oxygen content than diesel, with higher peak temperatures inside cylinder. The max relative error between experimental and numerical result was 19% at CR 22, B10 fuel, while the minimum error was 6% at CR 18, diesel. The same trend is observed for all the test fuels and compression ratios.
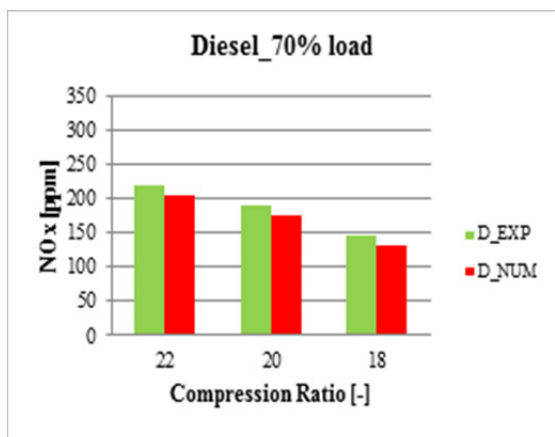

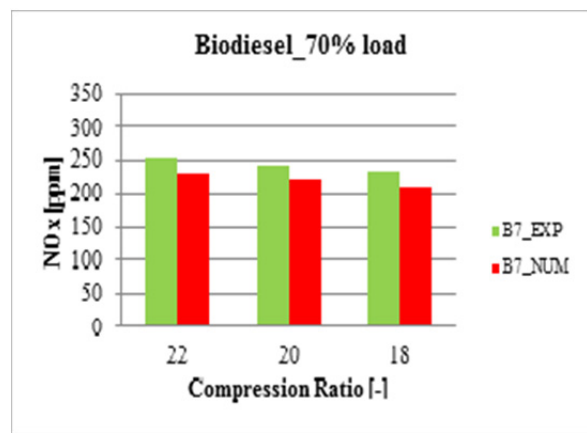**Figure 10:** NOx comparison values for diesel (D).

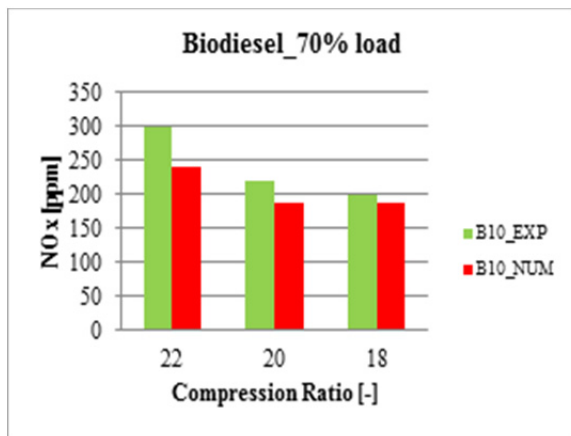**Figure 11**: NOx comparison values for Biodiesel (B7).

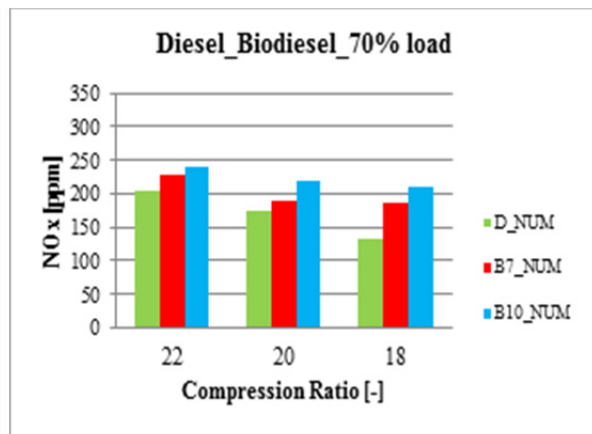**Figure 12:** NOx comparison values for Biodiesel (B10).

Figure 13: NOx comparison values for (D, B7, B10).

5. Conclusion
A thermodynamic model was developed for analyzing brake specific fuel consumption, brake thermal efficiency and NOx emission. The simulation model was validated by comparing it with the experimental results conducted on a direct injection single cylinder diesel engine to evaluate the performance and exhaust emission for different fuel blends and compression ratios. The main conclusions of this study are: The model showing good agreement between experimental versus simulated data, for all compression ratios at constant load particularly for maximum pressure and

respective crank angle; The same trend is observed for the parameters (cylinder pressure, BSFC, BTE and NOx) for all the test fuels and compression ratios; The BSFC and thermal efficiency were lower with biodiesel blends at all operation conditions due to the fact that biodiesel has lower heating value and higher density; NOx emission was higher with biodiesel blends; this may be to the advance in combustion phasing, higher combustion temperature and oxygen content of biodiesel; Hence, it is concluded that this model can be used for the prediction of the performance characteristics and exhaust gas emission of the compression ignition engine fueled with biodiesel fuel.

6. References

[1]   Menkiel B, Donkerbroek A, Uitz R, Cracknell R and Ganippa L 2014 Combustion and soot processes of diesel and rapeseed methyl ester in an optical diesel eng. *Fuel* **118** pp 406-415

[2]   Esteban B, Riba J , Baquero G, Rius A and Puig R 2012 Temperature dependence of density and viscosity of vegetable *oils biomass and bioenergy* **42** pp 164-171

[3]   Naik S, Vaibhav V, Prasant K and Ajay K 2010 Production of first and second generation biofuels: A comprehensive review *Renew. and Sustainable Energy Reviews* **14** pp 578-597

[4]   Mofijur M, Atabani A E, Masjuki H H, Kalam M A and Masum B M 2013 A study on the effects of promising edible and non-edible biodiesel feedstock on engine performance and emissions production: A comparative evaluation *Renewable and Sustainable Energy Reviews* **23** pp 391-404

[5]   Labeckas G and Slavinskas S 2006 Performance of direct-injection off-road diesel engine on rapeseed oil *Renewable Energy* **31** pp849-863

[6]   Geetha N and Bridjesh P 2015 Numerical Computation of Performance of Diesel Engine Using Matlab *International Journal of Research (IJR)***ISSN 2348-6848** pp 238-247

[7]   Buyukkaya E 2011 Effects of biodiesel on a DI diesel engine performance, emission and combustion characteristics *Fuel* **89** pp 3099-3105

[8]   Jagadish D, Puli R and Murthy K 2011 Zero Dimensional Simulation of Combustion Process of a DI Diesel Engine Fuelled With Biofuels *International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering* **5** pp 1621-1626

[9]   Potdukhe1 S , Deshmukh M M 2013 Modeling and Energy Analysis of a Diesel and Biodiesel Fuelled Engine *International Journal of Science and Research* **(IJSR) ISSN** pp 2319-7064

[10] Zucrow M, Hoffman J 1976 *Gas dynamics* (New York (USA): John Wiley & Sons).

[11] Heywood JB 1988 *Internal combustion engine fundamentals* (New York: McGraw- Hill)

[12] Savva N, Hountalas D 2014 Evolution and application of a pseudo-multi-zone model for the prediction of NOx emissions from large-scale diesel engines at various operating conditions *Energy Conversion and Management* **85** pp 373-388

[13] Gogoi T, Baruah D 2010 A cycle simulation model for predicting the performance of a diesel enginefuelled by diesel and biodiesel blends *Energy* **35** pp 1317-1323

[14] Singh D, Gu F, Fieldhouse J,Singh N, Singal S 2014 Prediction and Analysis of Engine Friction Power of a Diesel Engine Influenced by Engine Speed, Load, and Lubricant Viscosity *Advances in Tribology* **2014** pp 1-9

[15] Lesnik L, Iljaz J, Hribernik A and Kegl B 2014 Numerical and experimental study of combustion, performance and emission characteristics of a heavy-duty DI diesel engine running on diesel, biodiesel and their blends *Energy Conversion and Management* **81** pp 534-546