Two zone thermodynamic model for diesel engine cycle simulation running with biodiesel

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Abstract*— *The aim of the present work is to investigate the possibility of using Pistacia Lentiscus biodiesel and its blends with diesel fuel as an alternative fuel for diesel engines. Pistacia Lentiscus oil is converted to biodiesel with ethanol using sodium hydroxide as a catalyst. The characterization of the obtained biodiesel shows that the thermo-physical properties are in the range recommended by American Standard (ASTM D6751). Innovative biodiesel development tests on the diesel engine require a lot of time and efforts. Here, a mathematical model, based on the two zones thermodynamic model, is developed to analyze the combustion characteristics, performance characteristics (brake power, brake thermal efficiency and specific fuel consumption) and pollutant emissions (CO and NOx) of a DI diesel engine.

Keywords*— *Pistacia lentiscus, biodiesel, diesel engine, thermodynamic model

I. INTRODUCTION

Nowadays investigations about alternative sources of energy from non-edible species having the ability of growing in almost all land types is becoming an important research task due to the gradual depletion of petroleum and the environmental degradation. The use of biofuels such as bioethanol, biogas, hydrogen, animal fat, vegetable oil and biodiesel in internal combustion engines is significant in this context [1–3]. The use of vegetable oil as fuel requires the decrease of its viscosity to a value close to the viscosity of diesel fuel. Previous researches have shown that the most suitable technique to improve vegetable oil properties is the transesterification [4]. This technique consists of a catalyzed chemical reaction involving vegetable oil and alcohol and producing esters (biodiesel) and glycerol. It was found [5] that the yield is optimum for a base catalyzed transesterification at atmospheric pressure, temperature between 55 and 60°C, reaction time of about 45 min to 1 h and molar ratio 6:1 (alcohol to oil). Several studies have focused on diesel engine fuelled by biodiesel or its blends with diesel. It has been reported that biodiesel or its blends with diesel are of comparable performance to diesel fuel. Moreover, a significant reduction in the carbon monoxide emissions, hydrocarbon and smoke were observed.

For biodiesel however, a slight increase in both NOx emissions and specific fuel consumption were depicted [6–9]. The numerical simulations of the diesel engine cycle based on thermodynamic models (single zone, two zones or multi zones) is of great interest for numerous reasons: predicting trends and providing for engineers more data than experiments to develop new concepts. However the numerical simulation of biodiesel fuelling IC engine is still a restricted area of research and the relevant literature remains slight. Rakopoulos et al. [10] developed a thermodynamic multi-zone model evaluating both performance and pollutant emissions of a DI diesel engine running with either vegetable oil or its biodiesel. Using spray formation modeling they showed that spray formation, combustion mechanisms and related emission formation are significantly affected by the physical properties of the fuels being considered. Gogoi and Baruah [11] developed a single zone thermodynamic model for predicting brake power and brake thermal efficiency of a diesel engine running with diesel and its blends with biodiesel derived from Karanja oil under various speeds and compression ratios. They concluded that the model predicts similar performance with diesel, 20% and 40% blending. However, with 60% blending, it gives better performance in terms of brake power and brake thermal efficiency.

In the present work, a two zone thermodynamic model under various equivalence ratio values is developed in order to evaluate the performance (brake power, specific fuel consumption and brake thermal efficiency) and pollutant emissions of a DI diesel engine fuelled by Pistacia Lentiscus biodiesel, diesel fuel and their blends, containing 75% (EB75), 50% (EB50) and 25%(EB25) biodiesel by volume. Using this model, pressure, temperature and other required properties are computed numerically for every crank angle step chosen. The engine friction and heat transfer computations are also incorporated in the model using empirical models. The ignition delay is also taken into account in the combustion model. Initially the engine is operated with diesel fuel, subsequently Pistacia Lentiscus biodiesel and its blends are used and then the results are compared and analyzed. The developed model is highly compatible for simulation work with biodiesel as a suitable alternative fuel instead of diesel.

TABLE I
Properties of tested fuels

Property	EB100	EB75	EB50	EB25	Diesel
MW (g/mol)	310	275	245	213	148
SAF	12.67	12.5	11.88	11.83	15
LHV (MJ/kg)	37.32	38.69	40.06	41.43	42.80
SG at 15°C	896	891	884	876	852
ν at 40°C(cSt)	2.99	2.62	2.36	1.91	1.57
FP (°C)	105	84	76	71	67
BP (°C)	323	-	-	-	210
CI	53	52.25	51.5	50.75	50

2. DESCRIPTION OF THE MODEL

A two zone thermodynamic model has been developed to study the performance of single cylinder, four stroke, air cooled direct injection, compression ignition engine fuelled by diesel, biodiesel and their blends in various proportions. The properties of the Pistacia Lentiscus biodiesel and its blends, measured according to ASTM Standards, are given in Table 1. The model is based on the first law of thermodynamics equation. It is assumed that there is spatial uniformity of pressure, temperature and composition of the cylinder content at each crank angle. Specific heats, internal energy and enthalpy of the gaseous mixture are calculated as a function of temperature. It is also assumed that the air fuel mixture is lean and this leads to temperatures at which dissociation of products does not have much effect on engine performance [11]. The input parameters used in the theoretical model are the engine and operational specifications given in Table 2, the stochiometric air-fuel ratio for each fuel, and the equivalence ratio ER. The outputs of the modelling program instantaneous pressure, temperature, volume and the performance parameters that includes brake power, brake specific fuel consumption and brake thermal efficiency.

A. *Energy equation*

Applying the thermodynamic first law on the in-cylinder control volume, the following general equations can be written as:

Energy equation:

$$\frac{d(mu)}{dt} = \sum \frac{dm_i}{dt} h_i - P\frac{dV}{dt} + \sum \frac{dQ}{dt} \qquad (1)$$

State equation:

$$\frac{1}{P}\frac{dP}{dt} + \frac{1}{V}\frac{dV}{dt} = \frac{1}{m}\frac{dm}{dt} + \frac{1}{r}\frac{dr}{dt} + \frac{1}{T}\frac{dT}{dt} \qquad (2)$$

Mass conservation:

$$\frac{dm}{dt} = \sum\left(\frac{dm}{dt}\right)_{in} - \sum\left(\frac{dm}{dt}\right)_{out} + \frac{dm_{f_{inj}}}{dt} \qquad (3)$$

where $[dmu/dt]$: Rate of change of internal energy,

$[dQ/dt] = [(dQ_c/dt) - (Q_h/dt)]$: Net heat release rate,

$[dQ_c/dt]$: Heat release rate due to combustion of fuel,

$[dQ_h/dt]$: Heat transfer rate,

$[dm_i/dt]$: Intake, exhaust and fuel mass flow rate,

h_i: Intake, exhaust and fuel enthalpy.

According the engine cycle evolution, Eq. (1), (2) and (3) are rearranged to give:

Air intake stroke:

$$\begin{cases} \dfrac{dT_{cyl}}{dt} = \left[\left(-\dfrac{dQ_p}{dt} + \dfrac{dm_{ad}}{dt}h_{air} - \dfrac{dm_{ech}}{dt}h_{cyl} - u\dfrac{dm_{cyl}}{dt} - p\dfrac{dV}{dt}\right)\dfrac{1}{m_{cyl}}\right] \Big/ C_{v_{cyl}} \\ \dfrac{dp_{cyl}}{dt} = p_{cyl}\left(\dfrac{1}{m_{cyl}}\dfrac{dm_{cyl}}{dt} + \dfrac{1}{r}\dfrac{dr}{dt} + \dfrac{1}{T_{cyl}}\dfrac{dT}{dt} - \dfrac{1}{V}\dfrac{dV}{dt}\right) \\ \dfrac{dm_{cyl}}{dt} = \dfrac{dm_{ad}}{dt} + \dfrac{dm_{ech}}{dt} \end{cases} \qquad (4)$$

Compression stroke:

$$\begin{cases} \dfrac{dT_{cyl}}{dt} = \left[\left(-\dfrac{dQ_p}{dt} - p\dfrac{dV}{dt}\right)\dfrac{1}{m_{cyl}}\right] \Big/ C_{v_{cyl}} \\ \dfrac{dp_{cyl}}{dt} = p_{cyl}\left(\dfrac{1}{r}\dfrac{dr}{dt} + \dfrac{1}{T_{cyl}}\dfrac{dT}{dt} - \dfrac{1}{V}\dfrac{dV}{dt}\right) \\ \dfrac{dm_{cyl}}{dt} = 0 \end{cases} \qquad (5)$$

Combustion and detente stroke : Equations expressed for 2 zones: burned and unburned zones

$$\begin{cases} m_{gf} C_{v_{gf}} \dfrac{dT_{gf}}{dt} + p\dfrac{dV_{gf}}{dt} = -\dfrac{dQ_{p_{gf}}}{dt} - u_{gf}\dfrac{dm_{gf}}{dt} + h_{fuel}\dfrac{dm_{fuel_{inj}}}{dt} - h_{gf}\dfrac{dm_{gb}}{dt} \\ m_{gb} C_{v_{gb}} \dfrac{dT_{gb}}{dt} + p\dfrac{dV_{gb}}{dt} = -\dfrac{dQ_{p_{gb}}}{dt} - u_{gb}\dfrac{dm_{gb}}{dt} + h_{gf}\dfrac{dm_{gb}}{dt} \\ \dfrac{1}{p}\dfrac{dp}{dt} + \dfrac{1}{V_{gf}}\dfrac{dV_{gf}}{dt} - \dfrac{1}{T_{gf}}\dfrac{dT_{gf}}{dt} = \dfrac{1}{m_{gf}}\dfrac{dm_{gf}}{dt} + \dfrac{1}{r_{gf}}\dfrac{dr_{gf}}{dt} \\ \dfrac{1}{p}\dfrac{dp}{dt} + \dfrac{1}{V_{gb}}\dfrac{dV_{gb}}{dt} - \dfrac{1}{T_{gb}}\dfrac{dT_{gb}}{dt} = \dfrac{1}{m_{gb}}\dfrac{dm_{gb}}{dt} + \dfrac{1}{r_{gb}}\dfrac{dr_{gb}}{dt} \\ \dfrac{dV_{gf}}{dt} + \dfrac{dV_{gb}}{dt} = \dfrac{dV_{cyl}}{dt} \end{cases} \quad (6)$$

Subscripts: gf for unburned zone and gb for burned zone

Exhaust stroke:

$$\begin{cases} \dfrac{dT_{cyl}}{dt} = \left[\left(-\dfrac{dQ_p}{dt} + \dfrac{dm_{ad}}{dt}h_{air} - \dfrac{dm_{ech}}{dt}h_{cyl} - u\dfrac{dm_{cyl}}{dt} - p\dfrac{dV}{dt}\right)\dfrac{1}{m_{cyl}}\right] \Big/ C_{v_{cyl}} \\ \dfrac{dp_{cyl}}{dt} = p_{cyl}\left(\dfrac{1}{m_{cyl}}\dfrac{dm_{cyl}}{dt} + \dfrac{1}{r}\dfrac{dr}{dt} + \dfrac{1}{T_{cyl}}\dfrac{dT}{dt} - \dfrac{1}{V}\dfrac{dV}{dt}\right) \\ \dfrac{dm_{cyl}}{dt} = \dfrac{dm_{ad}}{dt} + \dfrac{dm_{ech}}{dt} \end{cases} \quad (7)$$

Cv stands for specific heat at constant volume, T for instantaneous temperature, p for instantaneous pressure, and V for instantaneous volume of the cylinder content.

The instantaneous cylinder volume V is given by,

$$V(\theta) = V_{clear} + \frac{\pi D^2}{4} L\left[1 + R_c(1-\cos\theta) - \sqrt{1-(R_c\sin\theta)^2}\right] \quad (8)$$

where V_{clear} is the clearance volume, and D, L, R_c are respectively the cylinder bore, the rod length and the ratio of connected rod length to crank radius.

B. *Heat transfer*

The heat transfer rate is expressed as:

$$\frac{dQ_h}{d\theta} = hA(T - T_w)\left(\frac{1}{\omega}\right) \quad (9)$$

where T_w is the wall temperature, A the heat transfer area, and h the convective heat transfer coefficient. The latter is given by the Woschni model [12]:

$$h = 3.26 D^{-0.2} p^{0.8} T^{-0.55} w^{0.8} \quad (10)$$

Where D is the cylinder bore, and w is velocity of the burnt gases. w may be evaluated with the following formula [12]:

$$w(\theta) = 2.28\overline{U_p} + C_1 \frac{V_d T_r}{p_r V_r}(p - p_m) \quad (11)$$

Where T_r, V_r and p_r are reference state properties. p_m is the pressure at the same position with the engine in motored conditions. The constant C_1 takes the value 0 for the compression process and 0.00324 for both combustion and expansion processes. $U_p = 2NS/60$ is the average piston velocity, S is the engine stroke and N the engine speed.

C. *Heat release* analysis

The combustion process is taken to occur in two zones, namely premixed zone and diffusive zone. The two stage behaviour of combustion heat release rate curves is modeled using a single Wiebe function for each zone [12].

$$\frac{dQ_c}{d\theta} = \sum_{i=1}^{2} 6.9\left(\frac{Q_i}{\theta_i}\right)(m_i+1)\left(\frac{\theta}{\theta_i}\right)^{m_i} exp\left[-6.9\left(\frac{\theta}{\theta_i}\right)^{m_i+1}\right] \quad (12)$$

where i refer to premixed and diffusion phases of combustion.

θ_i represents the combustion duration of each zone, Q_i the integrated energy release for each zone and m_i adjustable parameters. The six parameters are to be identified by the least squares method to match experimental data. For the current study, these values are reported in table 3.

TABLE II
SPECIFICATIONS OF THE TEST ENGINE

Make	LISTER-PETTER
General details	4 stroke, DI, air cooled, single cylinder engine
Bore and stroke	95.3 mm × 88.9 mm
Compression ratio	18
Displacement volume	630 cc
Connecting rod length	165.3 mm
Fuel injection timing	20° BTDC
Rated power output	4.5 kW at 1500 rpm
Inlet and exhaust valve diameter	42 mm × 35 mm
Maximum valve lift	10.61 mm

TABLE III
WIEBE'S CORELATION PARAMETERS

	ER	θ_1	m_1	Q_1	θ_2	m_2	Q_2
EB100	0.2	7.44	1.98	90.18	31.37	0.91	276.33
	0.34	7.65	1.73	52.20	34.72	1.01	481.28
	0.41	7.31	1.52	51.14	37.13	1.03	560.30
	0.52	6.26	1.74	47.50	42.16	0.96	643.32
EB75	0.2	7.82	1.88	88.23	31.36	0.98	269.90
	0.34	7.74	1.98	63.43	32.46	1.10	452.64
	0.41	7.15	1.61	40.63	38.82	1.00	590.25
	0.52	6.29	1.78	34.02	43.08	0.94	661.57
EB50	0.2	7.58	2.00	83.10	32.52	0.94	295.95
	0.34	8.28	1.86	53.64	33.54	1.10	474.53
	0.41	6.76	1.64	27.49	38.07	1.02	585.94
	0.52	5.89	1.63	25.61	42.50	0.94	661.45
EB25	0.2	7.65	1.88	75.62	31.43	0.95	284.01
	0.34	7.53	1.99	47.49	33.11	1.06	476.23
	0.41	7.02	1.46	33.24	38.95	1.00	603.38
	0.52	5.58	1.97	21.32	44.04	0.91	677.34
Diesel	0.2	7.52	2.13	71.05	35.08	0.76	282.58
	0.34	7.93	2.17	44.86	34.00	1.06	483.50
	0.41	6.63	1.96	33.06	37.35	1.04	574.92
	0.52	5.80	1.76	21.33	42.39	0.98	669.48

D. Ignition delay

The ignition delay (*ID*) is the time between fuel injection start and combustion start. Ignition is initiated when the following integral becomes equal to one [13]:

$$\int_{t_{inj}}^{t_{inj}+ID} \frac{1}{ID} dt = \int_{t_{inj}}^{t_{inj}+ID} \frac{1}{2.64 P^{0.8} ER^{-0.2} exp(\frac{16550-20CI}{RT})} dt \quad (13)$$

where R is universal gas constant, CI the fuel cetane number and ER the fuel air equivalence ratio.

E. Flow through valves

During the intake and exhaust processes, intake and exhaust mass flow rate terms ($[dm_{in}/d\theta]$ and $[dm_{ex}/d\theta]$), which are added to the energy equation, are calculated using Saint-Venant equation:

$$\frac{dm}{d\theta} = S_m C_d P_{up} \sqrt{\frac{2\gamma}{(\gamma-1)RT_{up}} \left(R_p^{\frac{2}{\gamma}} - R_p^{\frac{\gamma-1}{\gamma}}\right)} \quad (14)$$

where p_{up} and T_{up} are upstream stagnation properties, and C_D a discharge coefficient. The value of C_D for the intake process is 0.6 and for the exhaust process 0.5 [12]. R_p is the ratio of upstream to downstream stagnation pressures. S_m is the instantaneous valve area and depends upon the valve lift and the geometric features of the valve head, seat and stem. As valve lift increases, three separate stages of flow area development will appear [12].

For $0 < L_v < 0.004$:

$$S_m = \pi L_v \cos\beta \left(D_v - 2w + \frac{L_v}{2}\sin 2\beta\right) \quad (15)$$

For $0.004 < L_v < 0.11$:

$$S_m = \pi (D_v - w)\sqrt{(L_v - w\tan\beta)^2 + w^2} \quad (16)$$

For $L_v > 0.11$:

$$S_m = \frac{\pi}{4}\left[\left(\frac{D_v}{2}\right)^2 - \left(\frac{D_v}{4}\right)^2\right] \quad (17)$$

F. Combustion reaction and gas property calculation

The number of moles of combustion products (exhaust gas constituents) is calculated from the equivalence ratio and molecular formula of the used fuel. The air fuel mixture is lean and this leads to temperatures at which dissociation of products does not have much effect on engine performance [11]. Therefore, dissociation of products of combustion is neglected in order to keep the analysis simple. The gas properties depend on temperature and on composition. Internal energy u, enthalpy h, specific heat at constant pressure C_p and specific heat at constant volume C_v of the gaseousmixture are calculated on the basis of charge composition and temperature. The above-mentioned gaseous mixture properties are calculated as follows [14]:

$$C_{p_i} = \frac{R}{M_i}(a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4) \quad (18)$$

$$C_{v_i} = C_{p_i} - \frac{RT}{M_i} \quad (19)$$

$$h_i = \frac{R}{M_i}T\left(a_1 + a_2\frac{T}{2} + a_3\frac{T^2}{3} + a_4\frac{T^3}{4} + a_5\frac{T^4}{5} + \frac{a_6}{T}\right) \quad (20)$$

G. Frictional power

A certain quantity of generated power within the engine cylinders is lost in friction form, with a reduction in the resulting brake power obtained at the crankshaft. Consequently, knowledge of friction power is required to relate the engine combustion characteristics, which influence the indicated power and the brake power. The power mean effective losses due to moving parts friction are calculated using the following empirical relation [12]:

$$FMEP(kPa) = C_1 + 48\left(\frac{N}{1000}\right) + 0.4\overline{U_p}^2 \quad (17)$$

H. Pollutant emissions model

Nitric oxide formation sub-model

As the consideration of chemical equilibrium cannot predict correctly the NO concentration, the generally accepted kinetics formation scheme proposed by Zeldovitch model is used. The equations, which describe the above model, together with their forward reaction rate constants k_i (in cm^3/mol.s), are [12]:

$$N + NO \overset{1}{\leftrightarrow} N_2 + O$$ $k_{+1}=1.6\ 10^{13}$, $k_{-1}=7.6\ 10^{13}exp(-38000/T)$ (21)

$$N + O_2 \overset{2}{\leftrightarrow} NO + O$$ $k_{+2}=6.4\ 10^{9}\ T\ exp(-3150/T)$, $k_{-2}=1.5\ 10^{9}\ T\ exp(-19500/T)$ (22)

$$N + OH \overset{3}{\leftrightarrow} NO + H$$ $k_{+3}=4.2\ 10^{13}$, $k_{-3}=2.0\ 10^{14}exp(-23650/T)$ (23)

The change of (NO) concentration is expressed as follows:

$$\frac{d[NO]}{dt} = -k_1[NO][N] + k_{-1}[N_2][O] + k_2[N][O_2] - k_{-2}[NO][O] + k_3[N][OH] - k_{-3}[NO][H] \quad (24)$$

CO formation sub-model

The CO emissions are the intermediary combustion products formed when temperatures are enough increased to break CO_2 molecule bonds. The dissociation reactions occurring in CO formation are:

$$CO + OH \overset{1}{\Leftrightarrow} CO_2 + H$$ $k_1=1.5\ 10^{7}\ T^{1.3}\ exp(387/T)$ (25)

$$CO + O_2 \overset{2}{\Leftrightarrow} CO_2 + O$$ $k_2=2.8\ 10^{12}\ exp(-22057/T)$ (26)

The change of (CO) concentration is expressed as follows:

$$\frac{d[CO]}{dt} = (R_1 + R_2)\left(1 - \frac{[CO]}{[CO]_e}\right) \quad (27)$$

With $R_1=k_1\,[CO]_e\,[OH]_e$ and $R_2=k_2\,[CO_2]_e\,[O]$.

III. Results and Discussion

The developed model was used to investigate the effect of variation of equivalence ratio (ER) on engine performance (Brake Power and brake thermal efficiency) fuelled by diesel, biodiesel and blends of diesel and biodiesel. To demonstrate the reliability of the mathematical model, the ER values are selected from experiments carried out on a Lister- Petter DI diesel engine fuelled with diesel, biodiesel and their blends at various engine power output values (0.9, 2.25, 3.15 and 4.05 kW). The input parameters used in the theoretical model are the compression ratio, the stoichiometric air-fuel ratio for each fuel, and the equivalence ratio ER.

Fig.1, displays the typical pressure profile versus crank angle for diesel fuel. With the present model the predicted cylinder pressure values are comparatively closer to the experimental, because Wiebe's function parameters are adequately handled and in addition, the Woschni heat transfer coefficient is adequate to this model.

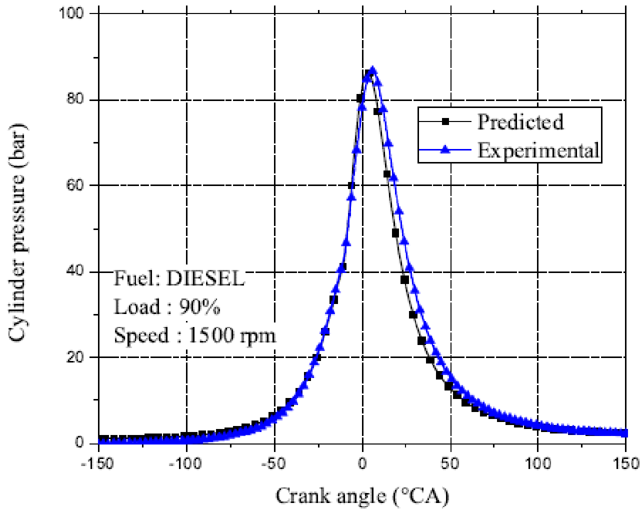

Fig. 1. Cylinder pressure comparison

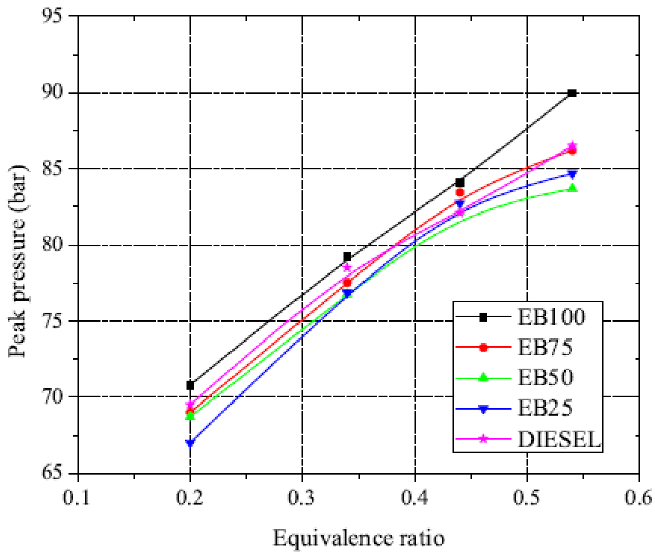

Fig. 2. Predicted peak pressure for different ER

The variation of predicted maximum cylinder pressure with equivalence ratio for diesel, EB100 and their corresponding blends is shown in Fig. 2. It can be seen that EB100 results in higher peak pressure as compared to all the other fuels for all tested equivalence ratios. Also, Fig.2 shows that peak pressure increases with the increase of the amount of biodiesel in the blend. This is due to the enhanced combustion rate as a result of rapid combustion of biodiesel at the premixed combustion period [15].

The Brake specific fuel consumption (BSFC) of a Diesel engine depends on the relationship among volumetric fuel injection system, fuel specific gravity, viscosity and heating value. Fig. 3 represents the BSFC variation versus ER for diesel fuel, biodiesel and their blends. For any ER, the BSFC value increases with the increase of the amount of biodiesel in the blend. More biodiesel and its blends are needed to produce the same amount of energy due to its higher specific gravity and lower heating value in comparison with Diesel fuel. For instance, for ER=0.2, the BSFC value of diesel is 6% lower than for biodiesel. When ER increases, the BSFC decreases sharply for all fuels. As an example, for ER=0.34 representing 50% of load, the BSFC of the pure diesel is found 23%, 18%, 3.5% and 0.6% respectively lower than EB100, EB75, EB50 and EB25. This is due to the fact that the power output of engine at a given engine speed increases together with an increase in ER. At high ER (0.44 or 0.54), EB25 and EB50 are found to be the blends that give lower BSFC.

The relationship between brake thermal efficiency (BTE) and equivalence ratio is presented in figure 4 as regards the engine speed of 1500rpm. The BTE of a Diesel engine is inversely proportional to its BSFC and the heating value of the fuel. Since the BSFC values of the biodiesel and its blends are higher than those with Diesel fuel, the higher BTE with the Diesel fuel is an expected result, which is seen for medium ER (0.34 and 0.44). For *ER* = 0.54, the BTE value of the Diesel fuel is decreased and those of the other fuels are increased. The increase in such a load, especially under the higher ER operating conditions, requires a larger amount of fuel. The amount of air entering the chamber is not sufficient for the the larger amount of Diesel fuel injected. As a result of this, the combustion process deteriorates beyond these operating conditions. However, when the biodiesel or its blend is injected, there is no pronounced effect due to the presence of oxygen in the fuel molecule. The maximum BTE of EB75, EB50 and EB25 blends is around 32% obtained at ER=0.54 against the 30% for diesel.

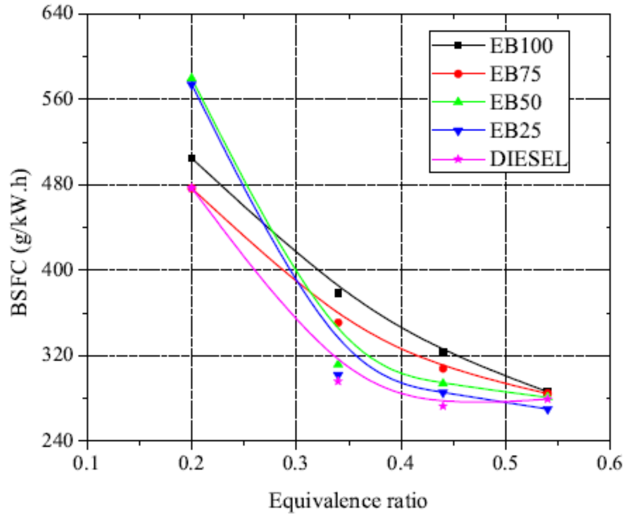

Fig. 3. Brake SFC versus ER

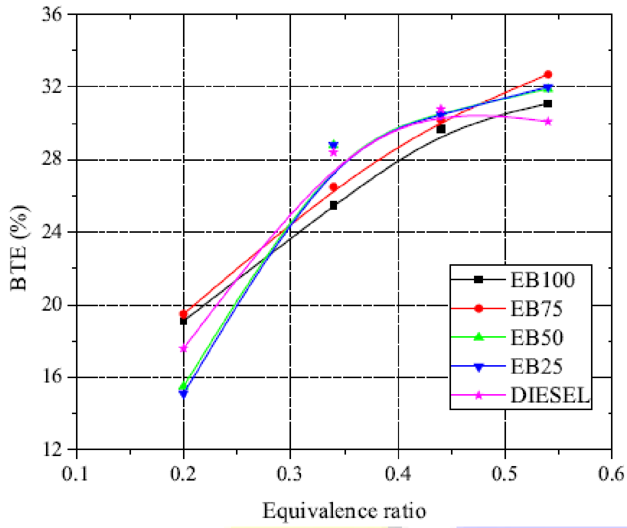

Fig. 4. Thermal efficiency versus ER

The plot of Fig. 5 shows the variation of carbon monoxide (CO) emissions for neat diesel fuel, neat biodiesel and their blends at various equivalence ratios, corresponding to 20%, 50%, 70% and 90% of full engine load conditions. The CO emissions are found to be increasing with the increase in equivalence ratio. The results show that the engine emits less CO using neat biodiesel and its blends as compared to that of diesel fuel under all loading conditions. In addition, increasing biodiesel content in the blends leads to a decrease in CO emissions for the reason that amount of oxygen content in biodiesel helps for the complete combustion.

Fig. 6 points out the variation of Nitrogen oxides (NOx) emission with power output for the different fuels tested. At partial loads (20%, 50% and 70%), NOx emissions for neat biodiesel are slightly higher than those of the diesel fuel and their blends. Higher values of cylinder pressure and combustion temperature, caused by higher biodiesel proportion in the blend, result in an increase in NOx generation. At high loads (90%), neat biodiesel emits less NOx than other tested fuels. The maximum NOx emission is observed at 3.15kW power output for all tested fuels. However, biodiesel's lower sulfur content allows the use of NOx control technologies that cannot be otherwise used with conventional diesel. Hence biodiesel's fuel NOx emissions can be effectively managed and eliminated by engine optimization.

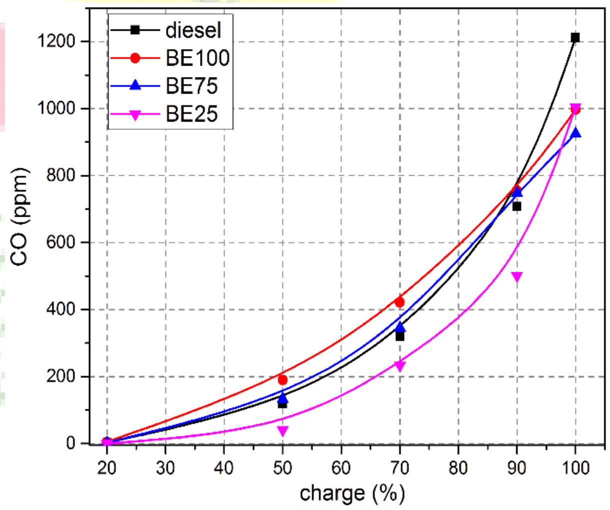

Fig. 5 Variation of carbon monoxide emission at various ER

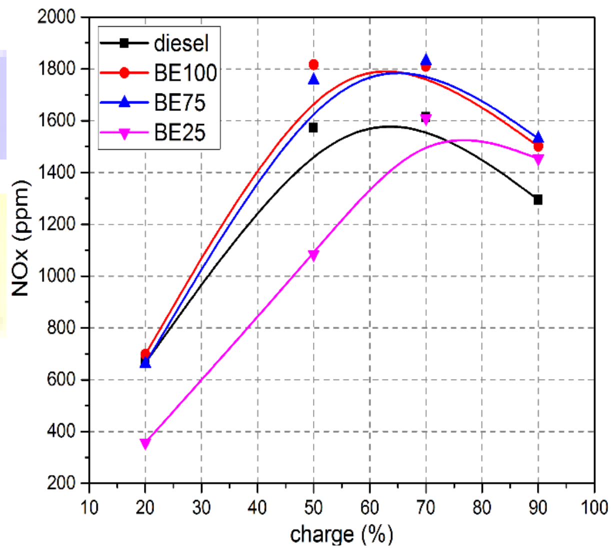

Fig. 6 Variation of Nitric oxide emission at various ER

IV. CONCLUSIONS

In the present work numerical study were carried out to evaluate the performance of a single cylinder, DI Diesel, Lister-Petter engine. The results obtained with Diesel fuel are taken as a reference for a comparison with a similar engine running with Pistacia Lentiscus biodiesel and its blends with diesel fuel in various proportions. A single zone thermodynamic model is developed and is applied for the design and operating data of this specific engine. The following conclusion can be drawn when varying the equivalence ratio for all tested fuels:

- The in-cylinder pressure histories computed by the model for the diesel, Pistacia Lentiscus biodiesel and their blends fueled diesel engine are closer to the experimental pressure data. No undesirable combustion features such as unacceptable high cylinder pressure rises are observed.
- Peak pressure and brake thermal efficiency are improved for all tested fuels whereas the brake specific fuel consumption decreases when increasing the equivalence ratio.
- CO emissions decrease, whereas NOx increases with the increase in percentage of biodiesel in the blend. However, the level of emissions increases with increase in the equivalence ratio for all fuels tested.

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