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# Validation of a single zone diesel engine model for biodiesel combustion

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***Abstract*—The goal of the present work is to investigate the possibility of using eucalyptus biodiesel and its blends with diesel fuel as an alternative fuel for diesel engines. Innovative biodiesel development tests on the diesel engine require a lot of time and efforts. Here, mathematical model, which is based on the thermodynamic single zone model, is developed to analyze the combustion characteristics such as cylinder pressure and the performance characteristics of a DI diesel engine. The predicted results are validated by conducting experiments on a test engine under identical operating conditions on a DI diesel engine.**

***Keywords*— Biodiesel, Diesel engine, Performance, Eucalyptus**

## I. INTRODUCTION

In the wake of present energy–environment crises, it has become crucial to find renewable and alternative clean energy sources. One of the principal routes to undertake the problem of increasing prices and pollution problems of petroleum fuels is by using biomass sources, particularly vegetable oils. Several chemical properties of oils, among them are the high viscosity and high molecular weight, cause poor fuel atomization and low volatility, leading to incomplete combustion and severe engine deposits, injector coking and piston ring sticking [1-2]. To overcome these problems caused by the high viscosity of vegetable oils, several techniques have been used such as preheating the oil, blending or diluting the oil with other conventional fuel, oil micro emulsification, transesterification or thermal cracking/pyrolysis [3-5].

Previous researches showed that the most suitable technique to improve the properties of vegetable oils is the transesterification. It consists of a catalyzed chemical reaction involving vegetable oil and an alcohol which produces esters (biodiesel) and glycerol [6–8]. A lot of researchers have reported that biodiesels are of comparable performance to diesel fuel. Moreover, significant reduction in emissions of carbon monoxide, hydrocarbon and smoke were observed. However, a slight increase in NOx emissions and specific fuel consumption were depicted [9–12].

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 in order to develop new concepts. However the numerical simulation of biodiesel fuelling compression ignition engine still a restricted area of research and the relevant literature remains slight. Rakopoulos et al. [13] developed a thermodynamic multi-zone model evaluating both performance and pollutant emissions of a direct injection (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 [14] 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 under various speeds and compression ratios. They concluded that with 60 % blending the model gives higher performance in terms of brake power and brake thermal efficiency.

In the present work, a single zone thermodynamic model under various equivalence ratio values is developed in order to evaluate the performance of a DI diesel engine fuelled by eucalyptus biodiesel, diesel fuel and their blends (containing 75%, 50% and 25% 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 equations. The ignition delay is also taken into account in the combustion model. To demonstrate the reliability of the mathematical model, engine tests are carried out on a Lister- Petter DI diesel engine under identical operating condition. The simulated combustion and performance characteristics results are found satisfactory with the experimental value. Thus the developed model is highly compatible for simulation work with biodiesel as a suitable alternative fuel instead of diesel.

## II. DESCRIPTION OF THE MODEL

A single zone thermodynamic model has been developed to study the performance of single cylinder, four stroke, air cooled direct injection, compression ignition engine fuelled by

any hydrocarbon fuel viz. diesel, biodiesel and its blends. 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. As the combustion in diesel engine should always occur in lean air fuel mixture, this leads to temperature levels at which dissociation of products does not have much effect on engine performance [14].

*1) Energy equation*

Applying the thermodynamic first law, the general energy equation can be written as:

$$\frac{dU_{Cyl}}{d\theta} = \frac{dW}{d\theta} + \frac{dQ_T}{d\theta} + \sum \frac{dm_i}{d\theta} h_i \quad (1)$$

Where $\frac{dU_{Cyl}}{d\theta}$ is the rate of change of internal energy,

$\frac{dW}{d\theta}$ : is the rate of work transfer,

$\frac{dQ_T}{d\theta}$ *:* is the net heat release rate and is the difference between $\frac{dQ_C}{d\theta}$ and $\frac{dQ_h}{d\theta}$

$\frac{dQ_C}{d\theta}$ *:* Heat release rate due to combustion of fuel,

$\frac{dQ_h}{d\theta}$ : Heat transfer rate from in cylinder gases to the wall,

$\frac{dm_i}{d\theta}$ : Intake, exhaust and fuel mass flow rate,

$\theta$ : Crank angle position,

$h_i$ : Intake, exhaust and fuel enthalpy.

With rearrangement, Eq. (1) can be written as:

$$m_{cyl} C_v \frac{dT}{d\theta} = -p\frac{dV}{d\theta} + \frac{dQ_c}{d\theta} - \frac{dQ_h}{d\theta} - u\frac{dm_{cyl}}{d\theta} + h_{in}\frac{dm_{in}}{d\theta} + h_{ex}\frac{dm_{ex}}{d\theta} + h_f\frac{dm_f}{d\theta} \quad (2)$$

$m_{cyl}$,$C_v$,$T$, $P$ and $V$ are the mass, specific heat at constant volume, instantaneous temperature, instantaneous pressure and instantaneous volume of the cylinder content respectively. Instantaneous cylinder volume $V$ is given by,

$$V(\theta) = V_{clear} + \frac{\pi D^2}{4} L\left[1 + R_c(1-\cos\theta) - \sqrt{1-(R_c \sin\theta)^2}\right] \quad (3)$$

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.

*2) Heat transfer*

The rate of the heat transfer is expressed as:

$$\frac{dQ_h}{d\theta} = h\,A(T - T_w)\left(\frac{1}{\omega}\right) \quad (4)$$

where $T_w$ is the wall temperature, A the heat transfer area, ω the engine rotational speed, and $h$ the convective heat transfer coefficient. The latter is given by the Woschni model [15]:

$$h = 3.26\,D^{-0.2}P^{0.8}T^{-0.55}w^{0.8} \quad (5)$$

where $D$ is the cylinder bore, and $w$ is the velocity of the burnt gases. $w$ may be evaluated with the following formula [15]:

$$w = 2.28\overline{U_p} + C_1\frac{V_d\,T_r}{P_r\,V_r}(P - P_m) \quad (6)$$

where $T_r$, $V_r$ and $P_r$ are reference state properties. $P_m$ is the pressure at the same position allowing to obtain $P$ without combustion. The constant $C_1$ takes the value $C_1$ = 0 for the compression process and $C_1$ = 0.00324 for both combustion and expansion processes. $\overline{U_p} = (2NS/60)$ is the average piston velocity, $S$ is the engine stroke and $N$ the engine speed.

*3) Heat release analysis*

The combustion process can be described with varying complexity and accuracy. In the present modelling, the combustion process is taken to occur in two phases, namely premixed phase and diffusive phase. The two stage behaviour of combustion heat release rate curves are commonly identified as pre mixed combustion and diffusive combustion regardless of the operating conditions. Adequate curve fitting to this two stage combustion rate curves resulted in double Wiebe's function which is given by eq. (7). The heat release rate is calculated by using this equation [15].

$$\frac{dQ_c}{d\theta} = \sum_{i=1}^{2} 6.9\left(\frac{Q_i}{\theta_i}\right)(m_i+1)\left(\frac{\theta}{\theta_i}\right)^{m_i}\exp\left(-6.9\left(\frac{\theta}{\theta_i}\right)^{m_i+1}\right) \quad (7)$$

where $i$ refer to premixed and diffusion phases of combustion. $\theta_i$ represents the combustion duration of each phase, $Q_i$ the integrated energy release for each phase 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 I.

TABLE I
WIEBE'S CORRELATION PARAMETERS

| | ER | $\theta_1$ | $m_1$ | $Q_1$ | $\theta_2$ | $m_2$ | $Q_2$ |
|---|---|---|---|---|---|---|---|
| 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 |
| 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 |

*4) Ignition delay*

The ignition delay (*ID*) is the time (in m.s) between fuel injection start and combustion start. Ignition is initiated when the following integral becomes equal to one [16]:

$$\int_{t_{inj}}^{t_{inj}+ID} \frac{1}{ID}\,dt = \int_{t_{inj}}^{t_{inj}+ID} \frac{1}{A(ER)^B P^C \exp\left(\frac{D}{T}\right)}\,dt \qquad (8)$$

where A, B, C and D are constants to be identified by the least squares method to match experimental ignition delay of each fuel. For the current study, these values are reported in table II.

TABLE II
CONSTANTS OF THE IGNITION DELAY CORRELATION

| | A | B | C | D |
|---|---|---|---|---|
| **Biodiesel** | 5.24 | -0.09 | -0.45 | 1219 |
| **Diesel** | 3.57 | 0.09 | -0.35 | 1328 |

*5) 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 present in the energy equation, are calculated using Saint Venant equation:

$$\frac{dm}{d\theta} = S_m\, C_d\, P_{up} \sqrt{\frac{2\gamma}{(\gamma-1)RT_{up}}\left(R_p^{\frac{2}{\gamma}} - R_p^{\frac{\gamma-1}{\gamma}}\right)} \qquad (9)$$

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 [15]. $R_p$ 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 [15].

**For $0 < L_v < 0.004$**:

$$S_m = \pi\, L_v \cos\beta \left(D_v - 2\psi + \frac{L_v}{2}\sin 2\beta\right) \qquad (10)$$

where $L_v$ and $D_v$ are the instantaneous valve lift and valve head diameter respectively. $\beta$ the valve seat angle and $\psi$ the seat width.

**For $0.004 < L_v < 0.11$**:

$$S_m = \pi\left(D_v - \psi\right)\sqrt{\left(L_v - \psi\tan\beta\right)^2 + \psi^2} \qquad (11)$$

**For $L_v > 0.11$**:

$$S_m = \frac{\pi}{4}\left[\left(\frac{D_v}{2}\right)^2 - \left(\frac{D_v}{4}\right)^2\right] \qquad (2.12)$$

*6) Combustion reaction and gas properties calculation*

The number of moles of combustion products (exhaust gas constituents) is calculated from the equivalence ratio and molecular formula of the tested fuels. As the air fuel mixture is lean, dissociation of the combustion products is neglected in order to keep the analysis simple. The gas properties depend on temperature and the composition. Internal energy *u*, enthalpy *h*, specific heat at constant pressure $C_p$ and specific heat at constant volume $C_v$ of the gaseous mixture are calculated on the basis of charge composition and temperature. The above-mentioned gaseous mixture properties are calculated as follows [17]:

$$C_{p_i} = \frac{R}{M_i}\left(a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4\right) \qquad (13)$$

$$C_{v_i} = C_{p_i} - \frac{RT}{M_i} \qquad (14)$$

$$h_i = \frac{R}{M_i}T\left(a_1 + a_2\frac{T}{2} + a_3\frac{T^2}{3} + a_4\frac{T^3}{4} + a_5\frac{T^4}{5} + \frac{a_6}{T}\right) \qquad (15)$$

$$u_i = h_i - \frac{RT}{M_i} \qquad (16)$$

i: $O_2$, $N_2$, $H_2O$, Fuel…

*7) 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 off the crankshaft. Therefore, 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 [15]:

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

where $C_1 = 75\ kPa$.

## III. RESULTS AND DISCUSSION

The input parameters used in the theoretical model are the engine and operational specifications given in Table III, the stoichiometric air-fuel ratio for each fuel, and the equivalence ratio (ER). According to the engine loading conditions used in the experiments, the ER values are selected at various engine power output values (0.9, 2.25, 3.15 and 4.05 kW) [18].

TABLE III
SPECIFICATIONS OF THE TEST ENGINE

| **Make** | **LISTER-PETTER - TS 1** |
|---|---|
| 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 |

The predicted ignition delay of the different used fuels is presented in fig.1 in comparison with the experimental ones at various equivalence ratios. Analyzing these results, an average error less than 5% is noted. Also, it can be noticed that the ignition delay of the biodiesel is shorter than that for neat diesel fuel. This is due to the high cetane number and oxygen content. The new correlation expression gives a very good prediction of the ignition delay in the case of all influenced parameters such as the type of used fuel and the engine load.

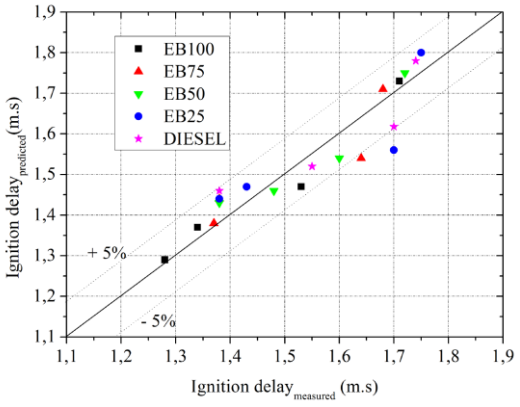


Fig. 1 Ignition delay comparison between simulation and experimentation

Fig.2 shows the comparison between the experimental heat release (the rate and its cumulative), obtained with analyzing the cylinder pressure signal, and the predicted ones achieved by the Wiebe correlation with dual functions. As it can be seen, the agreement between measured and predicted heat release laws traces is good for each used fuel. This is due to the good adjustment of the Wiebe's correlation parameters in both combustion stages. The same trend is observed for all tested fuels. It can also be observed that the combustion at low ER is more pronounced for the premixed phase. However, it is more pronounced at the diffusion phase for high ER. The engine load increasing leads to accumulate more fuel in the combustion chamber and to reduce the ignition delay making the quantity of fuel non-mixed with air in the ignition delay period increasing as well and continue to burn in the late combustion.

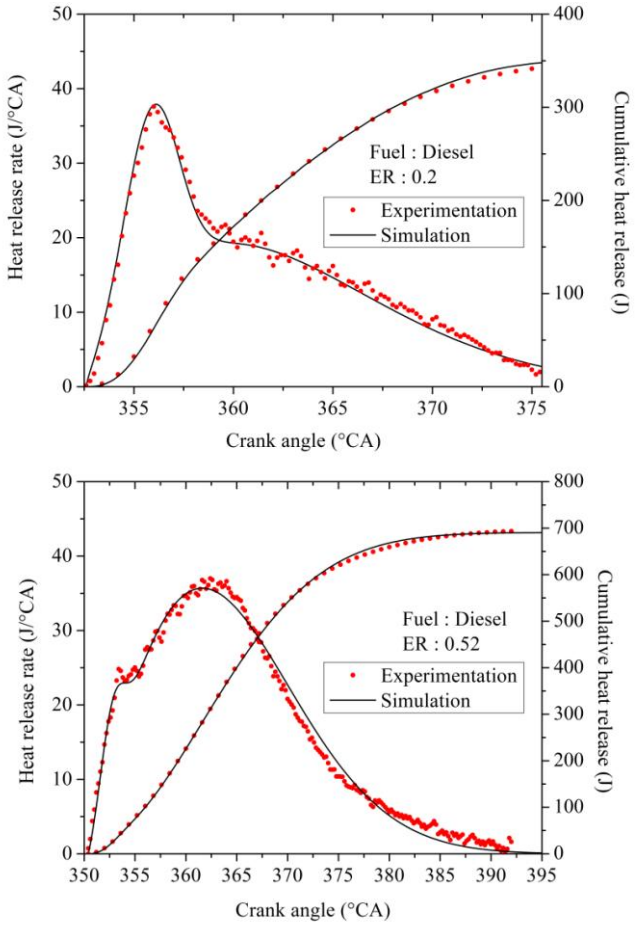


Fig. 2 Heat release rate and its cumulative

Fig. 3 shows the calculated and experimental cylinder pressure variations with respect to crank angle. The numerically computed values of instantaneous pressure and

experimentally measured values are compared for eucalyptus biodiesel (EB100) as well as for diesel fuel operation at the same operating conditions. The trends of the cylinder pressure characteristic curves for EB100 and diesel models and experimental are almost similar in nature. Also the predictions of the model are very close to the experimental one in the whole cycle. This is due to the better handling of the Wiebe's function parameters and in addition, to the adequate Woschni heat transfer coefficient for this model. Also, it can be seen that diesel fuel results in lower peak pressure as compared to the EB100 fuel. The 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 peak pressure values, respectively, for diesel and EB100 models, 87.12 bar, 90.23 bar occur at 4° and 3° crank angle after TDC. However, the model underestimates the value of the maximum pressure with an average error of 5% on the overall operating points, as it is shown in Fig. 4.

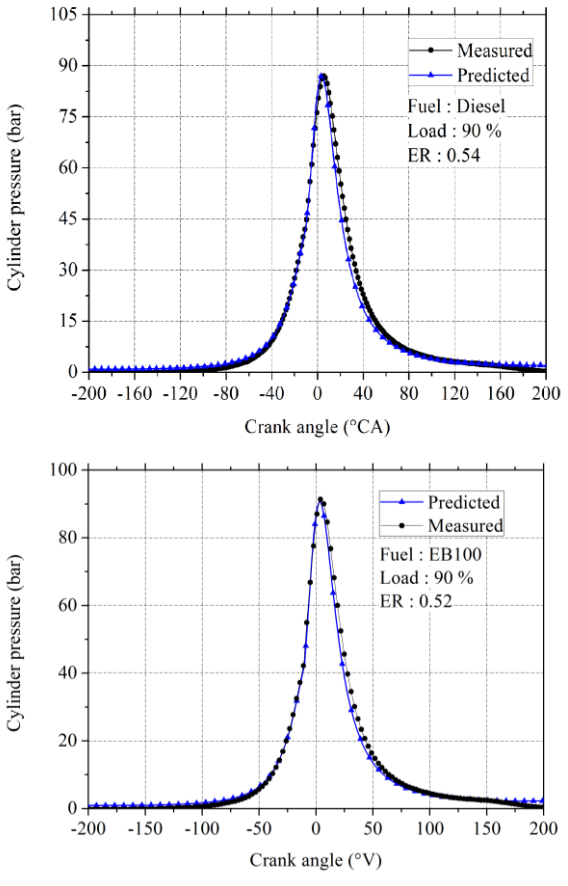


Fig. 3 Cylinder pressure comparison

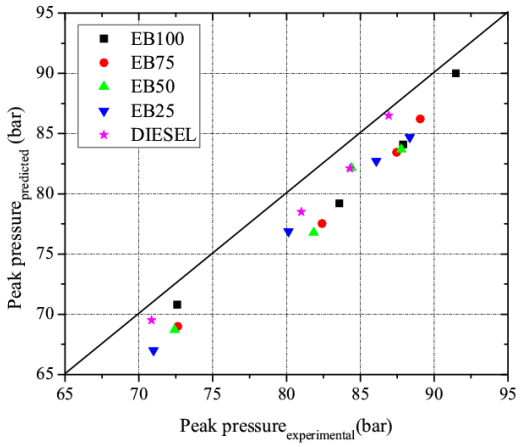


Fig. 4 Peak of cylinder pressure comparison

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.5 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.

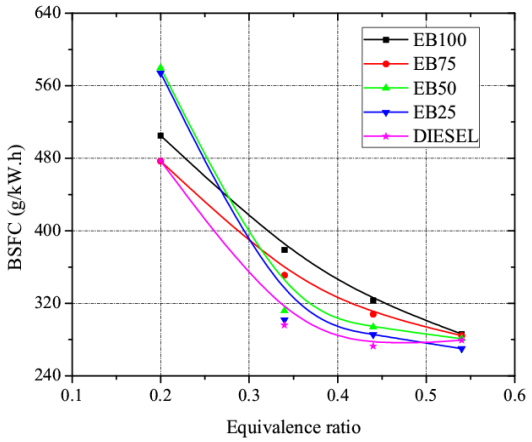


Fig. 5 Specific fuel consumption versus equivalence ratio

The relationship between brake thermal efficiency (BTE) and equivalence ratio is presented in Fig. 6 as regards the engine speed of 1500 rpm. 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 larger amount of Diesel fuel injected. As a result of this, the combustion process deteriorates beyond these operating conditions. But when the biodiesel or its blend is injected, there is no pronounced effect due to the presence of oxygen in the fuel composition. The maximum BTE of EB75, EB50 and EB25 blends is around 32% obtained at ER=0.54 against the 30% for diesel.

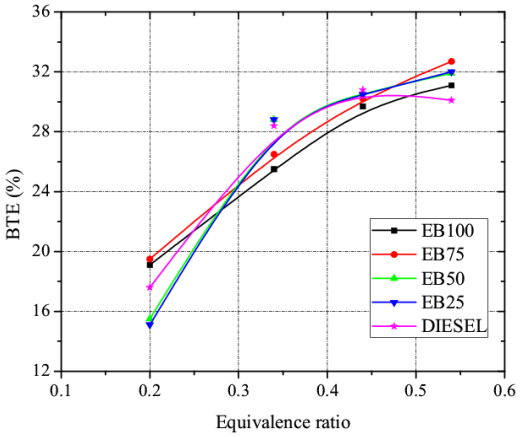


Fig. 6 Thermal efficiency versus equivalence ratio

## IV. CONCLUSION

In the present work numerical study was 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 eucalyptus 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, eucalyptus biodiesel and their blends fuelled 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. Therefore, it is concluded that in terms of performance characteristics, eucalyptus biodiesel blends at 50% and at 25% could be regarded as a potential substitute for diesel fuel.

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