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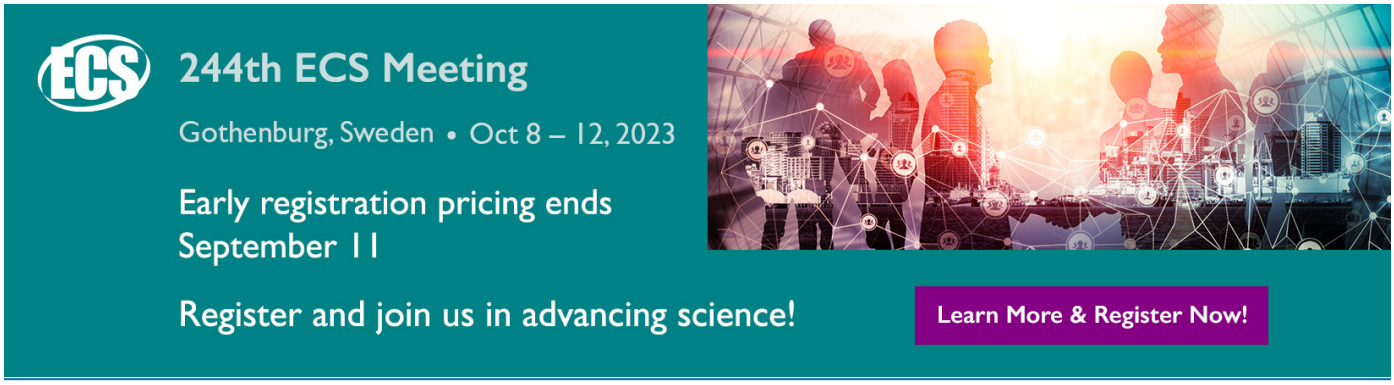
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Prediction of Exhaust Gas Temperature of a Diesel Engine Running with Diesel Fuel-biodiesel-1-pentanol Ternary Blends

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Abstract. Diesel engines are utilized in the transportation sector owing to their high efficiency. In recent years, biodiesel and higher alcohols have taken the attention of researchers as promising alternative fuels for diesel engines. In this article, diesel fuel is mixed with corn oil biodiesel at the ratio of 80:20 (v/v). 2%, 5% and 8% of 1-pentanol (v/v) are mixed into the diesel fuel-corn oil biodiesel binary blend for obtaining ternary blends. The impacts of ternary blends on some performance and combustion behaviors of a diesel engine are researched. Power and exponential models to predict exhaust gas temperature linking to maximum pressure rise rate, brake effective power, lower heating value, engine speed, equivalence ratio and latent heat of evaporation are derived through the least square error method. The use of ternary blends results in lower brake effective power (5.4246%-6.0066%), exhaust gas temperature (5.9504%-7.9459%,) and peak cylinder pressure (6.8502%-7.1629%), compared to diesel fuel. The average relative errors are specified as 1.9214% and 2.9749% for the power and exponential models, respectively.
Keywords: Internal combustion engines; Alternative fuels; Combustion; Exhaust gas temperature.

1. Introduction

Energy consumption per capita is one of the development indicators of a country. Today, the energy need is mostly met by fossil fuels. However, due to the increasing industrialization, urbanization, motorization and human population, energy demand is rising rapidly. This situation causes the socio-economic problems and the reduction in fossil fuel reserves. These troubles force researchers to concentrate on finding alternative fuels. Among alternative fuels, vegetable oils and biodiesel have received wide attention in the world for a long time. A number of researches on the utilization of vegetable oils have been continuing for a long time. However, it is not suitable for especially long-term operation since vegetable oils possess a higher viscosity than diesel fuel, which brings about poor atomization and incomplete combustion. Preheating oils, blending with other fuels, transesterification and thermal cracking methods can solve the high viscosity problem [1]. Among these methods, transesterification (alcoholysis or ester exchange) is the most commonly preferred one owing to its simplicity [1]. Chemically, biodiesel consists of methyl or ethyl alkyl esters instead of the alkanes and aromatic hydrocarbons of diesel fuel [2].

As a renewable alternative fuel, biodiesel has a higher flash point temperature (>120℃), and oxygen content (about 10% −12%, which decreases hydrocarbon and smoke emissions), compared to

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diesel fuel [2]. Biodiesel is non-toxic, non-aromatic and biodegradable fuel [2]. These superior fuel properties make biodiesel a more suitable fuel for diesel engines. Biodiesel owns also disadvantages that limit its more widespread use. These are higher viscosity, poor flow characteristics, and lower calorific value, relative to diesel fuel. The mixing of biodiesel with lower-viscosity fuels (ethers, alcohols, etc.) is one of the most popular methods in order to overcome these disadvantages.

Alcohols can be produced from natural and manufactured sources [3]. The main fuel properties of alcohols generally improve with increasing the carbon atom number in their chemical structures. In other words, in comparison with lower alcohols, higher alcohols (1-butanol and 1-pentanol) possess higher heating value, cetane number and flash point temperature. 1-butanol and 1-pentanol possess a lower hygroscopic nature. Therefore, 1-butanol and 1-pentanol are easily mixed up with diesel fuel [4-6].

Due to the properties of 1-pentanol mentioned above, it recently has been taken attention by many researchers as the promising alternative fuel (or additive) for diesel engines. The available literature presents many papers concerning the impacts of the usage of diesel fuel-biodiesel-1-pentanol ternary blends on engine performance, combustion and exhaust emissions. However, there are fewer studies in the literature pertaining to predicting the exhaust gas temperature (EGT) of diesel engines running with ternary blends (including diesel fuel, biodiesel and 1-pentanol) under different operating conditions. Hence, in this paper, two models (power and exponential) depending on latent heat of evaporation (LHE), brake effective power (BEP), engine speed, maximum pressure rise rate (MPRR), lower heating value (LHV) and equivalence ratio are obtained for the prediction of EGT when fuelling with diesel fuel-biodiesel-1-pentanol ternary blends.

2. Materials and methods

2.1. Properties of test fuels

Diesel fuel (DF), corn oil biodiesel (CBD), and 1-pentanol (Pn) are the base fuels for the preparation of ternary blends. Pn with 99% analytical grade purity is supplied by Merck company. DF is obtained from a local petrol station (Trabzon). CBD is obtained by means of transesterification under the following reaction conditions: 60 minutes reaction duration, 1:9 oil/methanol molar ratio, 0.90% sodium hydroxide amount and 50℃ reaction temperature [7]. For CBD, density at 15℃, kinematic viscosity at 40 ℃, flash point temperature, cold filter plugging point, higher heating value, stoichiometric fuel/air ratio $((F/A)_s)$ and cetane number were determined before as follows: 877.13 kg/m^3, 4.095 mm^2/s, 169℃, -5℃, 39930 kJ/kg, 1/12.6587, and 51.5935, respectively [7,8]. Important features of DF except for cetane number and latent heat of evaporation are determined at Prof. Dr. Saadettin GUNER Fuel Application and Research Center. Important features of DF are enlisted in Table 1. Moreover, important properties of Pn are taken from the cited literature [9-11]. Table 2 gives the properties of Pn. The latent heat of evaporation values of DF (274 kJ/kg), biodiesel (353 kJ/kg) and Pn (309 kJ/kg) are taken from the cited literature [12,13].

Table 1. Important features of DF.

Property	Test method	Limit		Value
		Min.	Max.	
Flash point temperature (℃)	TS EN ISO 2719	>55	-	58
Kinematic viscosity at 40℃ (mm^2/s)	TS EN ISO 3104	2	4.5	2.920
Higher heating value (MJ/kg)	DIN 51900-2	-	-	45.260

Copper strip corrosion at 3 h and 50℃	TS 2741 EN ISO 2160		1	1A
Stoichiometric fuel/air ratio $((F/A)_s)$	-	-	-	1/14.4649
Total contaminat. (mg/kg)	TS EN 12662	-	24	12.6
Cold filter plugging point temperature (℃)	TS EN 116	-	+5 (Sum.) -15 (Win.)	-20
Density at 15℃ (kg/m^3)	TS EN ISO 12185	820	845	835.6
Cetane index	TS EN ISO 4264	46	-	53.3
Cetane number[a]	EN ISO 5165	51	-	-

[a]: Cetane number is given by the supplier of DF.

Table 2. Important properties of Pn.

Molecular Formula	$C_5H_{11}OH$
Molecular weight (kg/kmol)	88.15
Oxygen (% wt.)	18.15
Carbon (% wt.)	68.13
Hydrogen (% wt.)	13.72
Cetane number	18.2-20
Auto-ignition temperature (℃)	300
Density (kg/m^3) at 15℃	814.8
Viscosity at 40℃ (mm^2/s)	2.89
Boiling point (℃)	138
Flash point (℃)	49
Solubility in water (g/L)	25
Gross heating value (MJ/kg)	37.60
Stoichiometric fuel/air ratio $((F/A)_s)$	1/11.8588

European Directive 2021/0218 (published on July 14, 2021) has promoted to use of more renewable energy for reducing the energy dependency of countries and protecting biodiversity and climate objectives [14]. Moreover, according to the directive, the current European Union target for the use of renewable energy in an integrated energy system is at least 32% for 2030 [14]. That is why, CBD and DF are blended on 20:80 (vol./vol.) ratio. Then, Pn (2%, 5% and 8% by vol.) is mixed into DF-CBD binary blend to obtain triple blends. These are named DFCBDPn2, DFCBDPn5 and DFCBDPn8, respectively. For example, DFCBDPn2 includes 78.4 vol.% DF, 19.6 vol.% CBD, and 2 vol.% Pn. Ternary blends are obtained by using a magnetic stirrer at room temperature prior to the experiments. At room temperature, there is no phase separation among ternary blends. It should be noted that higher volumetric ratios for CBD (20%) and Pn (2%, 5% and 8%) are not selected by considering the Pn and CBD's lower heating value, Pn's lower cetane number, and Pn and CBD's higher price. Some properties of ternary blends are computed from the following formulas [12,15]:

$$\rho_{blend} = \sum x_i \cdot \rho_i \quad (1)$$

$$H_{u,blend} = \frac{\sum x_i \cdot \rho_i \cdot H_{u,i}}{\sum x_i \cdot \rho_i} \quad (2)$$

$$(F/A)_{s,blend} = \frac{\sum x_i \cdot \rho_i \cdot (F/A)_{s,i}}{\sum x_i \cdot \rho_i} \quad (3)$$

$$In(\nu_{blend}) = \sum x_i \cdot In(\nu_i) \quad (4)$$

$$CN_{blend} = \sum x_i \cdot CN_i \quad (5)$$

$$LHE_{blend} = \sum x_i \cdot LHE_i \quad (6)$$

Here, ρ, x, H_u, $(F/A)_s$, ν, CN and LHE refer to density, blending components' volumetric ratio, lower heating value, stoichiometric fuel/air ratio, kinematic viscosity, cetane number and latent heat of evaporation. The subscript i means DF, CBD and Pn. The ternary blends' (DFCBDPn2, DFCBDPn5 and DFCBDPn8) density, kinematic viscosity, lower heating value, stoichiometric fuel/air ratio, latent heat of evaporation and cetane number are calculated as 843.32 kg/m^3, 842.45 kg/m^3, 841.58 kg/m^3 (using Kay's mixing rule); 3.1195 mm^2/s, 3.1122 mm^2/s, 3.1049 mm^2/s (using Arrhenius mixing rule); 41.2849 MJ/kg, 41.1044 MJ/kg, 40.9235 MJ/kg (using Kay's mixing rule); 0.0714, 0.0718, 0.0722; 290.184 kJ/kg, 290.76 kJ/kg, 291.336 kJ/kg, and 50.4963, 49.5628 and 48.6292, respectively. The lower heating value, density at 15℃, kinematic viscosity at 40℃, stoichiometric fuel/air ratio, latent heat of evaporation and cetane number of DF is 2.7730%, 3.1981%, 3.6241% higher, 0.9239%, 0.8198%, 0.7157% lower, 6.8322%, 6.5822%, 6.3322% lower, 3.3285%, 3.9074%, 4.4863% lower, 5.9066%, 6.1168%, 6.3270% lower, and 0.9876%, 2.8180% and 4.6486% higher than DFCBDPn2, DFCBDPn5 and DFCBDPn8, respectively.

2.2. Experimental setup and procedure

Engine tests are fulfilled at Karadeniz Technical University, Internal Combustion Engines Laboratory. There is a diesel engine, an electric dynamometer, a fuel tank, an airbox, an instrument console, and a control panel monitoring system in the experimental setup [16]. A four-stroke, air-cooled, single-cylinder and direct-injection engine is utilized. It has a basic mechanical fuel injection system. Modifications are not made for the fuel injection system and engine. For measuring the brake torque and the engine speed, a force sensor (strain-gauged electronic load cell) and an optical encoder (made by Hengstler) on the side of the electric dynamometer are used [16]. In order to collect the measured data (cylinder pressure, cylinder volume, engine speed, etc.) from the sensors for every test fuel, the engine cycle analyzer, the cylinder head pressure transducer and the optical crank angle (CA) encoder are utilized. For measuring the fuel consumption (mass flow rate of fuel), a volumetric fuel gauge including an optical sensor, a solenoid, and a calibrated burette is used. The mass flow rate of air is specified with an orifice and a pressure transducer [16]. Before the measurements, the engine fuelled with the test fuel is run for a while at the full-throttle position. Exhaust gas temperature (EGT) is determined using K-type thermocouples. The engine is run at between 1000 and 2500 rpm and under the full-throttle position. After the measurements, it is again run when fuelling with DF to drain out all blends in the fuel line. No difficulties are encountered during the engine tests for each test fuel. More information on the experimental setup and procedure are given in the author's study [17]. The brake effective power (BEP, N_e, kW) of the engine is computed as follows:

$$N_e = M_d \cdot \omega / 1000 \quad (7)$$

$$\omega = \pi \cdot n / 30 \quad (8)$$

where, M_d, ω and n are the brake torque (Nm), the angular speed of the crankshaft and engine speed (rpm), respectively. The pressure rise rate (PRR, bar/degree) is calculated using the first derivative of the cylinder pressure over CA degree as follows [18]:

$$PRR = \frac{dP}{d\theta} = \frac{P_i - P_{i-1}}{\theta_i - \theta_{i-1}} \quad (9)$$

where θ is the CA degree and P is the cylinder pressure (bar). Equivalence ratio (ϕ) is calculated by using the actual fuel/air ratio $(F/A)_{actual}$) and the stoichiometric fuel/air ratio $(F/A)_s$ as follows [3]:

$$\phi = (F/A)_{actual} / (F/A)_s \quad (10)$$

$$(F/A)_{actual} = \dot{m}_{fuel}/\dot{m}_{air} \quad (11)$$

where $\dot{m}_{air}$ and $\dot{m}_{fuel}$ are mass flow rates of air (kg/s) and fuel (kg/s).

2.3. Calculation of Uncertainties

A method by Holman [19] is utilized to determine uncertainties of measurements. The uncertainties in brake torque, engine speed, air box differential pressure and fuel consumption are 0.1 Nm, 1 rpm, 1 Pa, and 0.1 mL/min. The uncertainty values are computed between 0.4449% and 0.5244% for BEP, and 0.3953% and 1.1479% for equivalence ratio in the case of using all test fuels. To reduce uncertainties, all measurements are 20 times repeated and 50 consecutive engine cycles are taken on average. The optical CA encoder possesses a 1 CA degree resolution. Therefore, it can be said that combustion parameters computed from cylinder pressure data are reliable.

3. Results and discussion

3.1. Engine performance and combustion characteristics

Change of BEP with respect to engine speed is defined in Figure 1. As increasing engine speed, BEP rises. The maximum BEP values are obtained as 5.2595 kW, 5.3406 kW, 5.1182 kW, and 5.0044 kW at 2500 rpm for DF, DFCBDPn2, DFCBDPn5 and DFCBDPn8, respectively. In the case of using DFCBDPn2, DFCBDPn5 and DFCBDPn8 relative to DF, BEP decreases on average by 5.4246%, 5.6587%, and 6.0066%, respectively. The lower heating value, higher density, higher viscosity and lower cetane number of ternary blends cause the reduction in BEP, relative to DF.

Figure 2 and Figure 3 display the changes in peak cylinder pressure (PCP) and maximum pressure rise rate (MPRR) with engine speed. As increasing engine speed, PCP and MPRR generally decrease. The CA location of PCP and MPRR generally slightly goes away from the top dead center (TDC) as increasing engine speed. In comparison to DF, DFCBDPn2, DFCBDPn5 and DFCBDPn8 exhibit on average 2.0384%, 5.0321%, 8.9251% higher MPRR, and 6.8502%, 6.9436%, 7.1629% lower PCP. The increase in MPRR can arise from the ternary blends' lower cetane number. DF gives the highest MPRR (4.9817 bar/degree at 1300 rpm and 1 CA degree after the TDC) and the highest PCP (82.6740 bar at 1000 rpm and 6 CA degree after the TDC). The ternary blends' higher viscosity, higher density and lower heating value may cause the diminish of PCP. The location of PCP is between 6 and 11 CA degree after the TDC. The location of MPRR is between 4 CA degree before the TDC and 3 CA degree after the TDC.

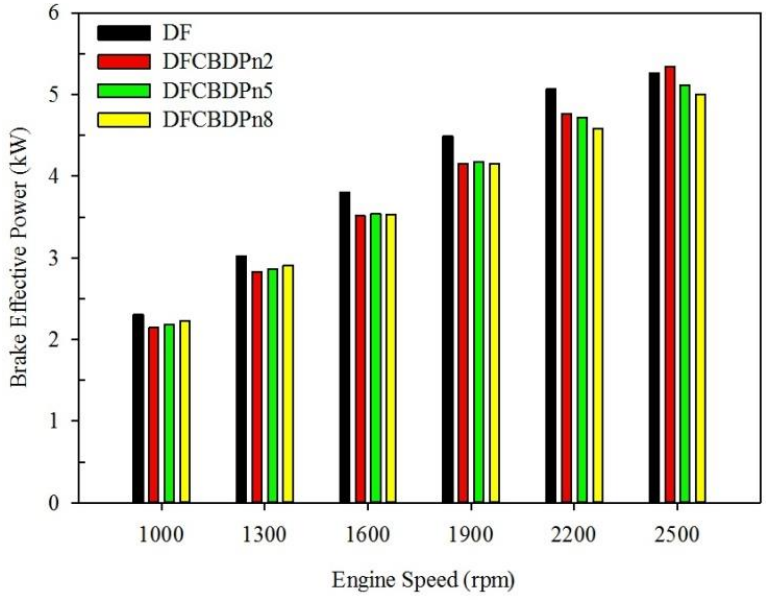


Figure 1. BEP variation for test fuels.

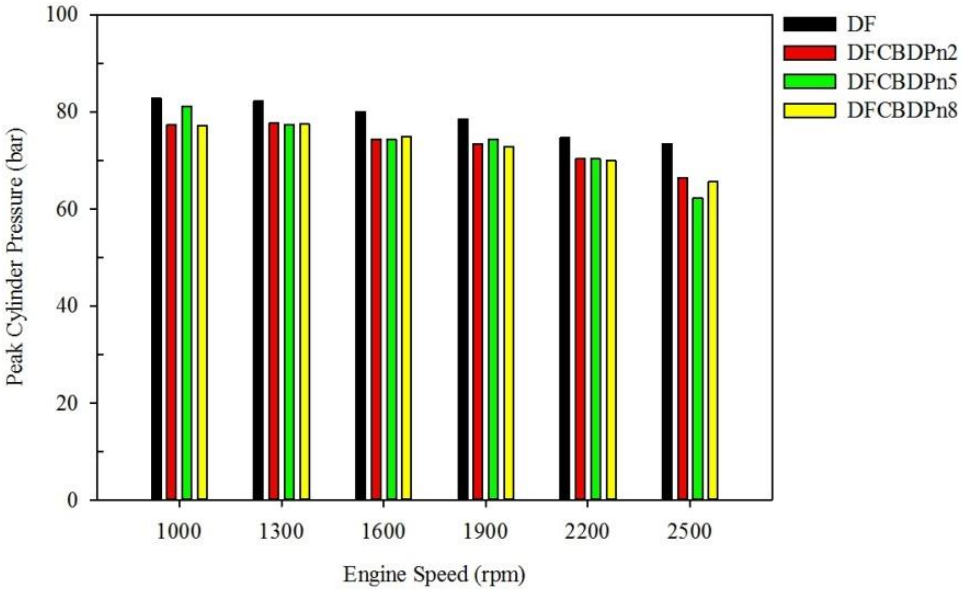


Figure 2. PCP variation for test fuels.

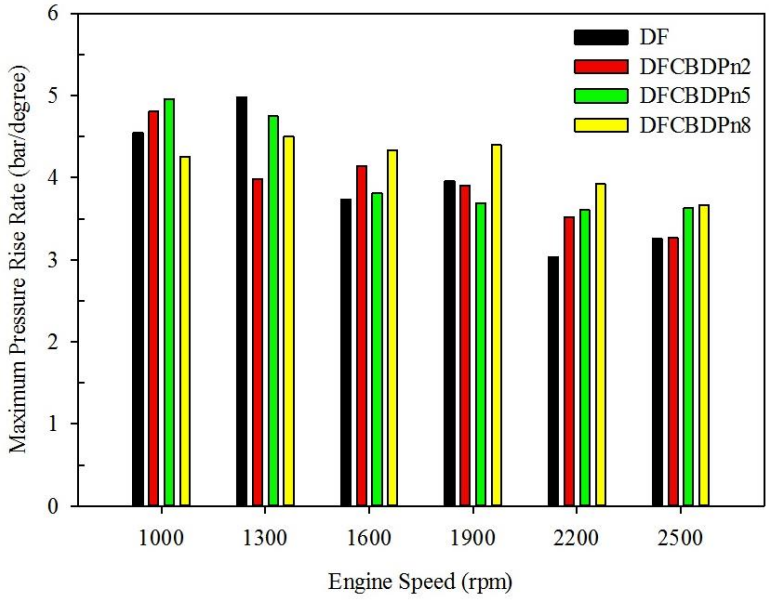


Figure 3. MPRR variation for test fuels.

Figure 4 illustrates the change in EGT depending on engine speed. The rise in engine speed induces an increase in EGT for each test fuel. The maximum EGT of 505℃, 483℃, 480℃ and 466℃ is measured at 2500 rpm for DF, DFCBDPn2, DFCBDPn5 and DFCBDPn8, respectively. EGT is lower at each engine speed when fuelling with the ternary blends. DFCBDPn2, DFCBDPn5 and DFCBDPn8 give averagely 5.9504%, 7.4286% and 7.9459% lower EGT, with reference to DF. Poor combustion characteristics of the ternary blends (owing to lesser calorific value and higher viscosity) as well as higher latent heat of evaporation of ternary blends can be reasons for the reduction in EGT.

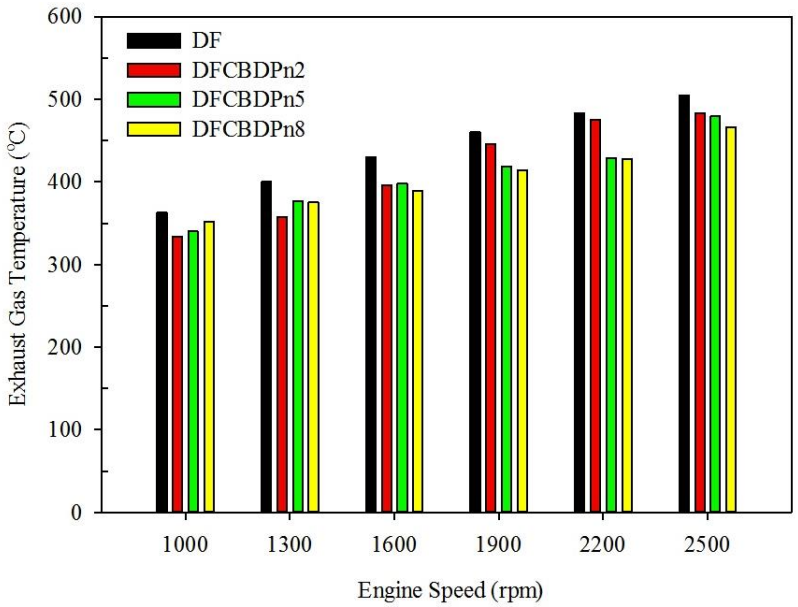


Figure 4. EGT variation for test fuels.

3.2. Prediction of EGT

EGT directly affects the performance of emission control systems [20], and it is an important quantity to interpret the performance, combustion and emission characteristics [3]. Therefore, the power and exponential models depending on BEP (N_e, kW), engine speed (n, rpm), equivalence ratio (ϕ), latent heat of evaporation (LHE, kJ/kg), MPRR (maximum pressure rise rate, kPa/degree) and lower heating value (LHV, H_u, kJ/kg) are derived by means of multiple linear regression and property of logarithm function to predict EGT (℃) as follows:

$$EGT = a_0 \cdot (N_e)^{a_1} \cdot (n)^{a_2} \cdot (\phi)^{a_3} \cdot (MPRR)^{a_4} \cdot (H_u)^{a_5} \cdot (LHE)^{a_6} \quad (12)$$

$$EGT = a_0 \cdot \exp[N_e \cdot a_1 + n \cdot a_2 + \phi \cdot a_3 + MPRR \cdot a_4 + H_u \cdot a_5 + LHE \cdot a_6] \quad (13)$$

where a_0, a_1, a_2, a_3, a_4, a_5 and a_6 are regression constants. The regression constants are computed using the Naive Gauss elimination method as 27.9963, -0.0021, 0.3385, 0.1627, -0.0680, 0.5341 and -0.8864 for the power model (Eq. (12)), respectively. Similarly, for the exponential model (Eq. (13)), the regression constants are computed using the Naive Gauss elimination method as 77.5560, -0.1034, 0.0004, 0.7127, -0.00004, 0.00005, and -0.0040, respectively. For the power and exponential models, the relative errors between the calculated values and measured data are computed within 0.2312% (DF at 1900 rpm)-4.9030% (DFCBDPn2 at 2200 rpm) and 0.0065% (DFCBDPn8 at 1300 rpm)-7.5807% (DFCBDPn2 at 2200 rpm), respectively. Considering the highest values of regression constants for Eq. (12) and Eq. (13), the lower heating value, the latent heat of evaporation and the equivalence ratio are thought to be more dominant on the variation of EGT. The average relative errors are specified as 1.9214% and 2.9749% for the power and exponential models, respectively.

4. Conclusions

Impacts of DF-CBD-Pn triple blends on some performance and combustion behaviors are investigated at varying engine speeds for a diesel engine. The ternary blends contain 2% (DFCBDPn2), 5% (DFCBDPn5) and 8% (DFCBDPn8) Pn by volume. Moreover, power (Eq. (12)) and exponential (Eq.(13)) models are proposed for predicting EGT. The primary outcomes can be listed as follows:

- For the use of DFCBDPn2, DFCBDPn5 and DFCBDPn8 in comparison to DF, there is an average reduction of 5.4246%, 5.6587%, and 6.0066% in BEP while there is an average increment of 2.0384%, 5.0321%, 8.9251% in MPRR.

- EGT is determined to decrease on average by 5.9504%, 7.4286% and 7.9459%, with reference to DF. DFCBDPn2, DFCBDPn5 and DFCBDPn8 give an average reduction of 6.8502%, 6.9436%, 7.1629% for PCP, relative to DF.
- The maximum relative errors arising from Eqs. (12) and (13) are 4.9030% and 7.5807% for predicting EGT. Moreover, Eq. (12) and Eq. (13) give 1.9214% and 2.9749% average relative error, respectively. Considering the regression constants in Eq. (12) and Eq. (13), the lower heating value, the latent heat of evaporation and the equivalence ratio seem to be more dominant for the change of EGT.

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