Combustion Modeling of a Common Rail multi-injection Diesel Engine Fueled with Diesel and Biodiesel*

Nguyen Khong Van, Trung Tran Anh, Vu Nguyen Hoang and Van Nguyen Cao

Abstract—**In recent years, the use of biofuels in the conventional diesel fuel injection system has been interested by the scientists, fuel producers as well as consumers. However, there has a few of using biodiesel research on Common Rail (CR) diesel engine. This paper descripts a combustion model of a CR multi-injection diesel engine using blends of palm oil derived biodiesel and diesel. The engine combustion parameters such as peak pressure, heat release rate and ignition delay were measured and computed, the mass fraction burn at 50% (MFB50) was also investigated in this paper.**

Keywords—**Combustion modeling, biodiesel, Wiebe function, CommonRail.**

I. INTRODUCTION

Conventional energy, such as fossil fuels, is being increasingly exploited due to the rapid development of industry, people's income and the growing number of vehicles. The use of fossil fuels will emit toxic substances which affect human health, the environment and greenhouse gases leading to climate change. Diversification of energy sources and reduction of dependence on fossil energy are the first and foremost objectives for the research and development of the internal combustion engine. Alternative fuels for internal combustion engines are seen as one of the most important solutions. Biodiesel is no exception.

Biodiesel is renewable biofuel derived from vegetable oils or animal fats based on the transesterification reactions with a monohydric alcohol (usually using methanol) and it is regarded as a potential alternative fuel to replace diesel fuel in future. Biodiesel consists of mono-alkyl esters of long chain fatty acids, so-called Fatty Acid Methyl Ester (FAME) [1].

The biodiesel properties can be changed in accordance with the different feedstocks. In comparison with petroleum diesel fuel, biodiesel has higher cetane number, density, cloud point, pour point, flash point and lower compressibility, sulfur content, volatility, net heating value, aromatic content. Also, it has several main advantages, such as renewable, biodegradable, nontoxic and several disadvantages, such as higher costs and solidifying temperature [1-3]. The difference in this attribute will affect combustion quality of diesel.

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There are many papers presented the characteristics of combustion in diesel engines [4,5,6]. It was reported that the heat release during the late combustion phase for biodiesel and their blends is marginally lower than that of diesel [5]. In addition, they are only applicable on regular diesel engines without building on Common Rail (CR) diesel engine. Which has high-pressure injection system (up to 2500 bar) and fuel injected can be split in up to 3 injections (Pilot, Main, Post) [6]. So the combustion process of CR engine must be different with conventional diesel engine. Therefore, the CR diesel engine combustion modelingis urgently needed to reduce the time and expense to run the optimally experimental data of the engine's controller. In this study, there has an investigation of the combustion characteristics for CR diesel engine with 2 injections (Pilot, Main) when using blends of the pure biodiesel fuel (B100) derived from Vietnam was manufactured by the previous work [7], and the petroleum diesel fuel (B0) is the commercial product in Vietnam from B10 to B100.

II. EXPERIMENTAL TECHNOLOGY FOR COMBUSTION STUDY

A. Experimental apparatus

The experiments were carried out with a engine AVL-5402 single-cylinder, four-stroke, direct injection diesel engine. Detailed specifications of the test engine are shown in Table 1. This engine uses the Common-Rail (CR) fuel injection system and open-ECU. The injection process is controlled by using AVL-INCA software, which allows setting and controlling the injection parameters such as injection timing, pressure, duration, etc. Figure 1 shows the schematic diagram of experimental apparatus setups.

B. Testing fuels

Biodiesel used in this work was derived from residues of a palm cooking oil production process using a methanol transesterification process with the aid of a high hardness solid ceramic metal catalyst [7]. It was found that the residues are still rich in fatty acid esters which have potential to manufacture biodiesel. In this paper, the fuel blends including B0, B10, B20, B50, B100 (corresponding with 0%, 10%, 20%, 50%, 100%) by volume of biodiesel in biodiesel–diesel mixtures, respectively) are experimentally tested in an engine and therefore this section briefly reports the important physicochemical properties of these fuel blends in comparison with those of pure biodiesel (B100) and fossil diesel.

Properties of the testing fuels are shown in Table 2.

C. Experimental methodology

Combustion parameters such as peak pressure, time of occurrence of peak pressure, heat release rate (HRR) and ignition delay were evaluated. The experiments were carried out by using blends of palm oil derived biodiesel (B10, B20, B50, B100) and diesel (B0) at n = 1400, 1600, 1800, 2000 rpm, BMEP = 5.3 [bar], pressure = 1000 bar.

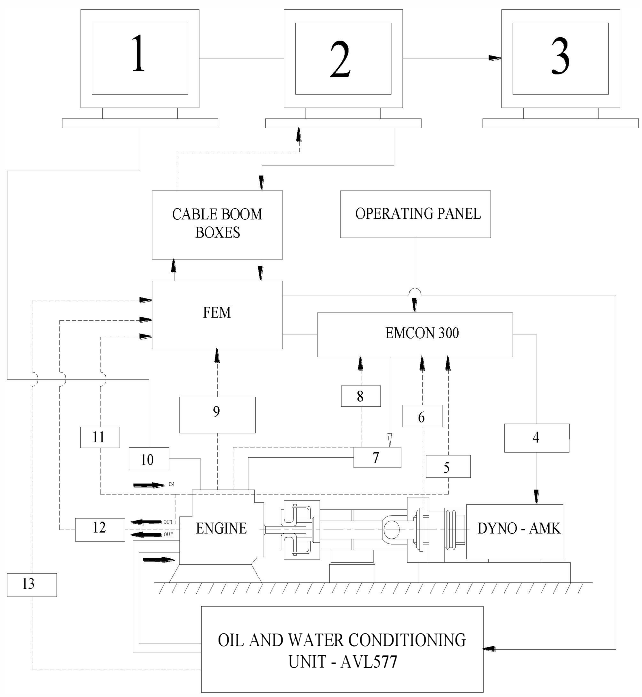

Figure 1. Schematic diagram of the experimental setups

1. INCA PC; 2. PUMA PC; 3. INDICATING PC; 5. Engine speed signal; 6. Engine torque; 7. Control throttle position signal; 8. Throttle position sensor; 9. Pressure and temperature sensor; 10. ECU; 11. Fuel balance; 12. Opacimater; 13. Temperature sensor

TABLE I. MAIN SPECIFICATIONS OF THE TESTING ENGINE

Parameter	Specification
Engine type	Diesel, 4 strokes, Naturally aspirated
Number of cylinders	1
Cylinder bore, [mm]	85
Stroke, [mm]	90
Compression ratio	17.3:1
Maximum power [kW]/Rated speed [rpm]	9/3200
Injection system	CommonRail (CP3)

TABLE 2. PROPERTIES OF B0 AND B100 FUEL, [7]

Parameter	Specification	
Fuel properties	B0	B20
Density at 15°, [g/cm^3]	0.836	0.866
Kinematics viscosity at 40°C, [mm^2/s]	3.14	4.1
Cetane number	52.4	66.9
Oxidation stability, at 110^0 C, [Hrs]	6.02	-
Acid number, [mgKOH/g]	0.06	-
Water content, [% kl]	0.20	-
Flash point, [^{0}C]	152	-

III. THERMODYNAMIC MODEL

Diesel combustion process is a complex, unsteady, heterogeneous, three-dimensional process. In this study, the objective is to build a very reduced model able to be used in real time application. An useful approach of solving combustion problem is to model the combustion as a heat release process, as originally proposed by Lyn [8]. For combustion simulations which predict engine performance, algebraic expressions can provide an adequate description of this heat release rate, provided that the constants in these expressions are chosen suitably to reflect the dependence of the actual fuel burning rate on engine type and particular operating conditions. Generally, these constants are calibrated from experimental data.

The global heat release rate equation may be written as:

$$\dot{Q} = m_{inj}.LHV.\frac{dx_b}{dt} \qquad (1)$$

where: LHV the lower fuel heating value (kJ/kg); xb is the mass fraction burned and minj (-> m_{inj}): total amount of fuel (kg/s).

Diesel combustion process proceeds in two phases: premixed combustion and diffusive combustion, Watson et al. [8] have developed a model where the apparent fuel burning rate is expressed as the sum of two components, one relating to premixed and the other to diffusion burning phase. The burned mass fraction is then given by the following relationship.

$$\frac{dx_b}{dt} = \beta\frac{dx_{bp}}{dt} + (1-\beta)\frac{dx_{bp}}{dt} \qquad (2)$$

where: β is the weight factor expressed as the ratio between the fuel burned by premixed phase and the total injected fuel.

In this study, the mass fraction of burned fuel has been modeled by the Wiebe function, which may be written as:

$$x_b = 1 - \exp\left[-a\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m+1}\right] \qquad (3)$$

The derivative of the above equation should be equal the rate of burned fuel

$$\frac{dx_b}{d\theta} = \frac{a(m+1)}{\Delta\theta}\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m}.\exp\left[-a\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m+1}\right] \qquad (4)$$

where: θ, θ_{SOC} , Δθ are the instantaneous crank angle, the crank angle of start of combustion and the combustion duration respectively. In this expression, a and m are tuning parameters.

Eq. (1) is possible to evaluate the rate of heat release through eq. (5).

$$\frac{dQ}{d\theta} = m_{\mathrm{inj}}.x_{fi}.LHV.\frac{a_i(m_i+1)}{\Delta\theta}\left(\frac{\theta-\theta_{SOCi}}{\Delta\theta_i}\right).$$
$$\exp\left[-a_i\left(\frac{\theta-\theta_{SOC_i}}{\Delta\theta}\right)^{m_i+1}\right] \quad (5)$$

where: LHV – Lower heating value, (kJ/kg); m_{inj}: total amount of fuel, (kg/s); x_{fi} : mass fraction of burned; Q: the rate of heat release (kJ).

To model the process of heating the cylinder motor Wiebe function has been fixed. The study [9] pointed out the Wiebe function depends on the engine working conditions, because fuel combustion method depends on the state of pressure in the air cylinder and the A/F ratio. In addition, the Wiebe function depends on the nozzle parameters such as injection timing and injection duration. In short, if N is the number to call Wiebe function it is neccesary, to use the Rate of Heat Release (ROHR) in the cylinder that can be calculated according to the equation (4).

$$\frac{dQ_n}{d\theta} = \sum_{i=1}^{N}\left(m_{\mathrm{inj}}.x_{fi}.LHV.\frac{dx_{bi}}{d\theta}\right) \quad (6)$$

The heat transfer to the walls is calculated with the assumption that this mechanism is due only to forced convection. The model used in this study is a phenomenological single zone model, where the radiative heat transfer is only due to the in cylinder burned gases temperature. In this case, the radiative heat transfer is small in comparison with the convective heat transfer [9]. Consequently, this term is written as:

$$dQ_w = A.h_c(T-T_w).d\theta \quad (7)$$

where: Tw is the wall temperature; A is the heat transfer surface area (cylinder, piston and liner) function of time; hc is the heat transfer coefficient (calculated from the well-known Woschni correlation [9])

Pressure in cylinder is calculated from in Eq. (8)

$$\frac{dp}{d\theta} = \gamma\frac{p}{V}\frac{dV}{d\theta} + (\gamma-1)\frac{1}{V}\left(\frac{dQ_n}{d\theta} - \frac{dQ_w}{d\theta}\right) \quad (8)$$

Named R = 287 [J/Kg.K] the in-cylinder gas constant, $\gamma = C_p / C_v$ has been evaluated as a function of the estimated in-cylinder temperature as follows [5].

$$C_p = 1403.06 - 360.72\left(\frac{10^3}{T}\right) + 182.24\left(\frac{10^3}{T}\right)^2 - 10.72\left(\frac{10^3}{T}\right)^3$$
$$C_v = R - C_p \quad (9)$$

Figure 2 shows the various stages of combustion in CR diesel engine. This figure was plotted for each of the engine operating condition analyzed to determine the values of the pilot and main injection ignition delay (ID). Determination of the pilot injection and main injection ID: The fuel injection signal has two distinct parts: one corresponding to pilot injection and the second corresponding to the main injection. The positions (in crank angles) where these two parts attain their maximum for the first time are noted as the beginning of the pilot and the main injection. Similarly, the normalized heat release rate diagram has two parts: one corresponding to the pilot burning and the second corresponding to the main burning. The position (in crank angles) for the peak of the first part is taken as the apparent pilot injection combustion starting point. The difference (in crank angles) between the pilot injection beginning point and the pilot injection combustion beginning point is the pilot injection ignition delay in crank angles. The same procedure was repeated to find the main injection ignition delay in crank angle. In the normalized heat release plot, it can be seen that the pilot burning curve shows a sudden rise and after being flat for some crank angles begins to go down. The point at which it begins to go down is noted as the end-point of pilot combustion. The same procedure is adopted to find the end-point of main combustion

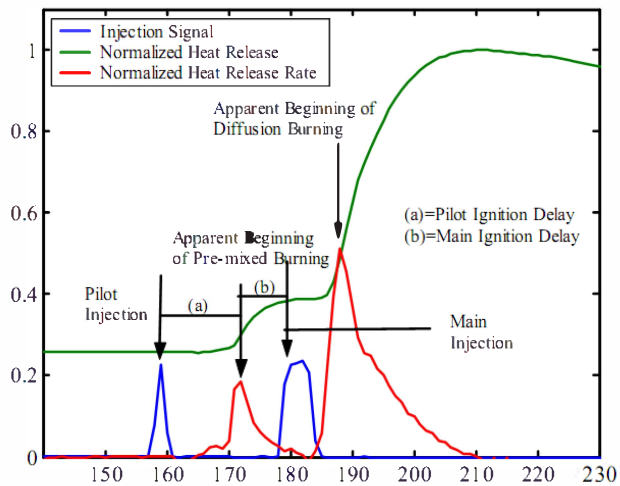

Figure 2. Various Stages of Combustion in CR diesel Engines, [8]

IV. MODEL PARAMETERS IDENTIFICATION

The development in cylinder model presented in the above section is mainly oriented to real time application. The recourse to a simplified formulation with calibration factors allows to reach both goals of short computational time and model accuracy in terms of in-cylinder profiles. The chosen combustion model based on double Wiebe equations is able to describe the two combustion processes: a premixed combustion taking place in the early stage which is followed by a diffusive combustion. The different parameters introduced in this model vary significantly for each injection and each engine operation, and have been identified and calibrated from measured pressure cycles. In the present study, focusing on single injection cycle, the engine operating points include only high engine speed (from 1400 rpm up to 2000 rpm) at varying BMEP (brake mean effective pressure) 5,3 bar . The engine specifications are presented in Table 1

From the measurement result of pressure changes in the cylinder it is possible to calculate the rate of heat release in the engine cylinder [8]. Figure 3 shows the presence of 2 peaks in ROHR waveform, that correspond to the premixed combustions associated to pilot injection and main injection respectively, followed by a slower heat release, corresponding to the diffusive part of the combustion process. This result is due to the fact that, in this approach, net heat release is taken into account, i.e. the difference between total heat release and the effects of blow-by and heat transfer through cylinder walls.

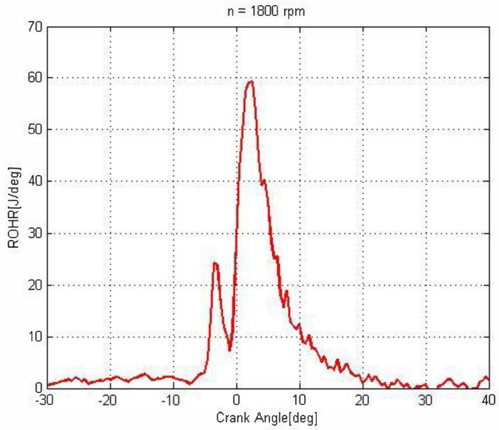

Figure 3. ROHR evaluated for an engine cycle performed

Cumulated heat release (CHR), i.e. Rate of heat release cumulated sum, provides information about the total amount of energy produced during the combustion process, which, is strongly dependent on the mass of fuel burned. Once CHR has been evaluated, MFB50 can be determined as the angular position corresponding to 50% of CHR itself [4]. Figure 4 shows how MFB50 has been evaluated for the above considered engine cycle performed.

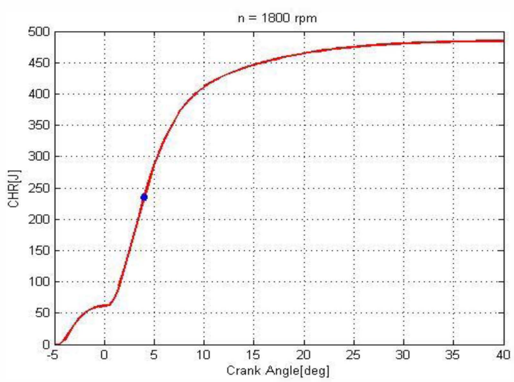

Figure 4. CHR evaluated for the engine cycle performed

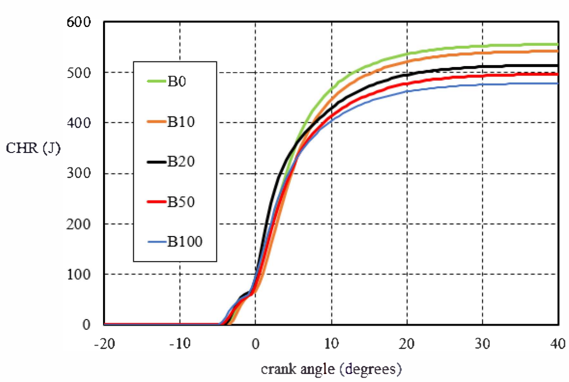

Figure 5. CHR evaluated at 75% load and n = 1800 rpm

Rate of heat release and MFB50 have been evaluated, for all the tests performed on the engine under study, starting from in-cylinder pressure measurement. The obtained results are reference values, that can be used to set up and validate the combustion model developed throughout this work. Figure 5 shows how CHR evaluated at 2000 rpm and BMEP = 5.3 bar with biodiesel.

V. Results and Discussions

As mentioned above, the geometric parameters mi said rules of each phase heat of fire. Its value shows that heat can speed concentrated within a short or prolonged angle in the entire combustion: m = 1.2 ÷ 1.6 can be used for the premix burning process and m = 0.5 ÷ 0.8 for diffusion combustion. Selecting m = 1.4 describes combustion mix and m = 0.65 describe diffusion combustion process [10]. We will select 3 Wiebe function to model the speed of heat: Wiebe for combustion spray primer, Wiebe for the main injection combustion, Wiebe for diffusion combustion. Figure 6 is the results of modeling of heat release rate diesel engines CR at 75% load and n = 1800 rpm.

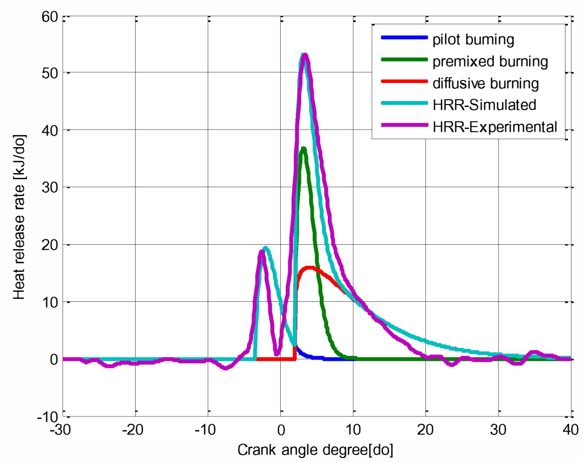

Figure 6. Simulated various stages of combustion in CR diesel engines

Figure 6 reproduces the comparison of the rate of burned fuel simulated by the model (Eqs. (4) and (5)) and estimated from experimental measurements when the engine speed is equal to 1600 rpm, the BMEP sets at 5.3 bar. On the same figure the rates of burned fuel by premixed combustion and diffusive combustion are plotted for more details. We observe a good accuracy between the simulated and the measured profiles in terms of amplitude and ignition delay time.

With the engine operating at the same BMEP equal to 5.3 bar and respectively at 1400 rpm and 2000 rpm, the comparison of in cylinder pressure profiles are plotted in Figure 10 to Figure 14 versus crank angle degrees. For these engine operating conditions the other parameters that vary are:

- Figure 7 (engine speed equal to 1600 rpm): SOI pilot 13 CAD BTDC, pilot duration 95 µs, SOI main 8 CAD BTDC main duration 470 µs.

- Figure 8 (engine speed equal to 1800 rpm): SOI pilot 13 CAD BTDC, pilot duration 95 μs, SOI main 9 CAD BTDC main duration 490 μs.
- Figure 9 (engine speed equal to 2000 rpm): SOI pilot 13 CAD BTDC, pilot duration 95 μs, SOI main 9 CAD BTDC main duration 530 μs.

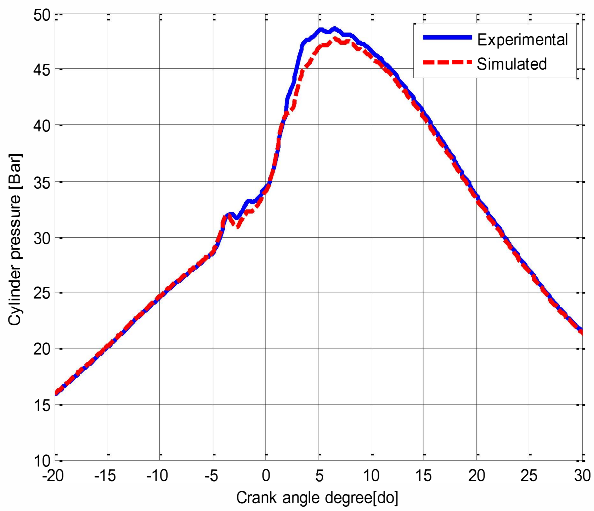

Figure 7. Comparison between simulated and measured in cylinder pressure at 1600 rpm and BMEP = 5.3 bar with B20

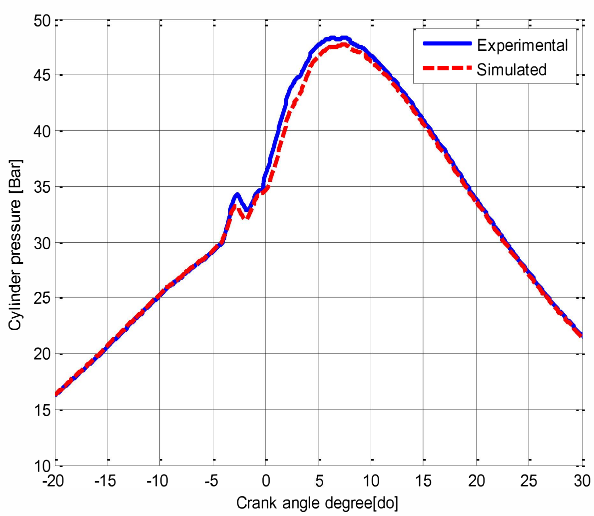

Figure 8. Comparison between simulated and measured in cylinder pressure at 1800 rpm and BMEP = 5.3 bar with B20

The comparison of in-cylinder traces reproduced in Figure. 7 to Figure. 10 shows that the developed model closely follows data in both magnitude (within 3%) and trend. For all the profiles, the start of combustion (SOC) occurs after the TDC (top dead center at 360 CAD) and the start of injection begins at 5 CAD BTDC (before top dead center). The accuracy between the experimental profile with the simulated profile is very good, while the simulated profile obtained with experimental profile correlation presents a very slight increase of the maximum peak pressure (lower than 2 bar).

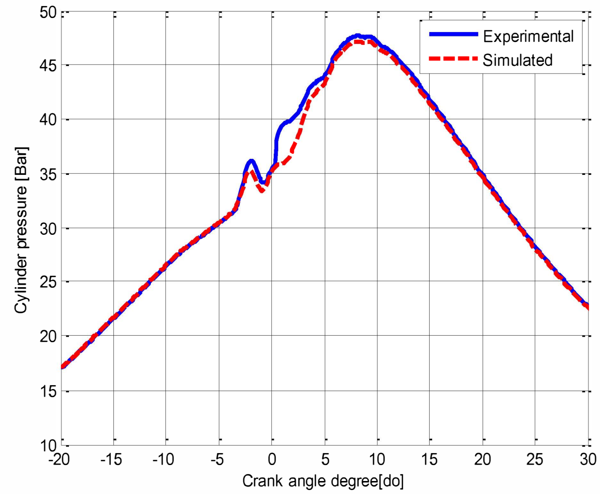

Figure 9. Comparison between simulated and measured in cylinder pressure at 2000 rpm and BMEP = 5.3 bar with B20

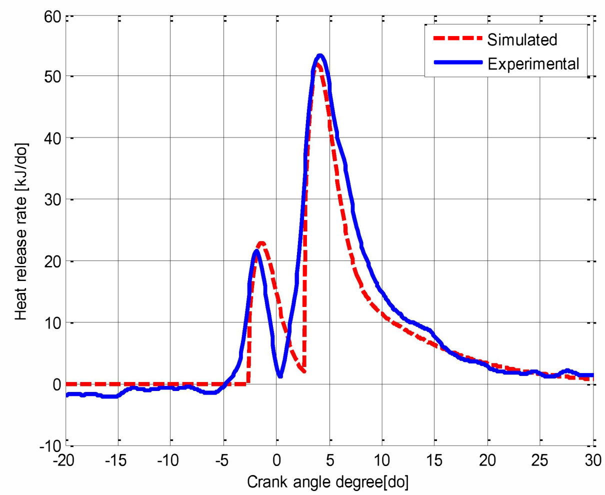

Figure 10. Comparison between simulated and measured heat release rate at 1600 rpm and BMEP = 5.3 bar with B20

The heat release rate profiles derived from in-cylinder pressure measurement and predicted by the simulation are plotted respectively in Figure. 10 to Figure. 12 for the same engine operations. At 1600 rpm engine speed (Figure 10), where pre- injection occurs through two pulses, it can be observed that for the experimental pre-combustion phase, the injected fuel is still burning until main combustion, while the model predicts an end of the pre combustion before the main combustion period. The model behavior is linked to the correlations of Wiebe factors, especially the duration of premixed combustion phase, fixed here constant equal to 13 CAD, to the combustion characteristic exponent and to the combustion duration of pre-combustion phase. When the engine speed is increased up to 2000 rpm (Figure 12), the same trend is observed for the pre-combustion period. However, the amplitude difference between the model and the experimental data is lower than the one observed at 1600 rpm

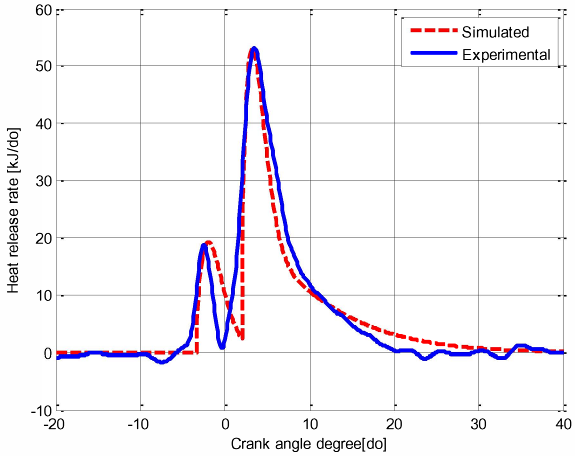

Figure 11. Comparison between simulated and measured heat release rate at 1800 rpm and BMEP = 5.3 bar with B20

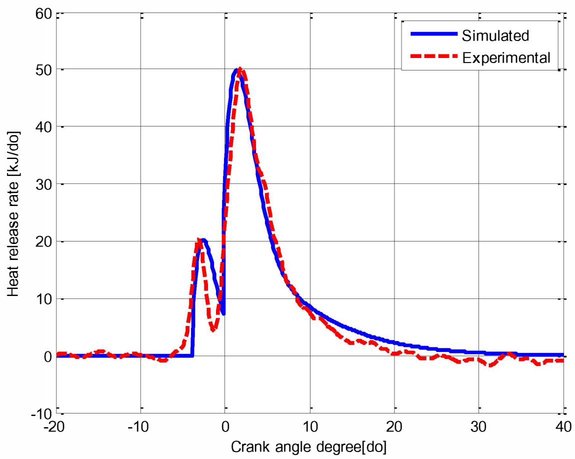

Figure 12. Comparison between simulated and measured heat release rate at 2000 rpm and BMEP = 5.3 bar with B20

VI. CONCLUSION

Combustion modeling of CR diesel engine fueled with palm oil derived biodiesel and diesel blends which has biodiesel rate from B20 to B100 has been studied, and drawn some following conclusions. The following conclusions can be drawn:

(1) Results is acceptable and fully reflects the biodiesel combustion characteristics on CR diesel engine.

(2) The accuracy between the experimental profile with the simulated profile is very good, while the simulated profile obtained with experimental profile correlation presents a very slight increase of the maximum peak pressure (lower than 2 bar).

ACKNOWLEDGMENT

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DEFINITIONS/ABBREVIATIONS

MFB: Mass Fraction Burned

CHR: Cumulated heat release

ROHR: Rate of Heat Release

IMEP: Indicated Mean Effective Pressure

BMEP: brake mean effective pressure

SOI_Pilot: Start Of Pilot Injection

SOI_Main: Start Of Main Injection SOC: Start Of Combustion

ID: ignition delay

θ: Crankshaft Angle

p: In-cylinder pressure

V: In-cylinder volume

Qnet : Net Heat Release