PERFORMANCE INVESTIGATION OF COMPRESSION IGNITION ENGINE USING EMPIRICAL CORRELATION FOR BURNING DURATION

by

Ajeet KUMAR**, Jeevan Vachan TIRKEY, and Shailendra Kumar SHUKLA

Centre for Energy and Resources Development, Department of Mechanical Engineering,
Indian Institute of Technology (BHU), Varanasi, India

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Fuel burning rate plays a major role in optimizing the performance of internal combustion engine with reduced emission. In an attempt to optimize the performance of internal combustion engine, a novel empirical correlation is developed for fuel burning duration in tune with the methodology proposed by an earlier investigator for spark-ignition engine. The correlation was integrated with the quasi-dimensional mathematical model to analyse the combustion, performance and emission characteristics of engine. Engine speed and fuel injection timing were varied to assess the performance and corresponding exhaust emission of engine. Predictions relating to variation of burning duration with compression ratio at different equivalence ratios are in reasonable agreement with the published data on burning duration. The simulated results show that the optimum injection timing lies in the range of 23 °bTDC to 13 °bTDC for brake power and indicated power both, and the lowest brake specific fuel consumption and indicated specific fuel consumption were found close to 13 °bTDC. A sharp decrease in peak cylinder pressure was also observed with retarding injection timing, whereas both the retarding injection timing and increased engine speed accrue to reduced nitric oxide exhaust at exhaust valve open.

Key words: *compression ignition burning duration, validation, injection timing, speed, maximum pressure, engine performance, NO emission*

Introduction

Higher break power, reduced emission and low fuel consumption are the major concern for the design of engines operated by different fuels. Modelling and simulations are quite often used to minimize the experiments to be conducted on internal combustion (IC) engine to reduce time and cost. Among the zero dimensional model, phenomenological model, quasi-dimensional model, and multidimensional; multidimensional model takes more time to give solution, whereas, phenomenological and quasi-dimensional models provide better accuracy in predictive result in short time [1]. Computational mathematical modelling or numerical solutions can give more effective forecast results on given engine parameters [2]. Compression ignition (CI) engine using diesel fuel has become more popular for automotive applications due to its higher energy density, durability and efficiency. While CI engines used for blended fuel, optimization of operating parameters is required to enhance the conversion efficiency of fuel for producing higher indicated power with low pollutants [3]. Also alternative fuel, such as biodiesel can be used

* Corresponding author, e-mail: ajeetbhb@gmail.com

in existing CI engine with or without minor modification. The biodiesel is more attractive due their desirable attributes (such as sustainable, biodegradable, and carbon neutral) to environment conditions. It produces lower emission such as unburned hydrocarbon, carbon monoxide and better engine performance [4, 5]. Mikulski and Wierzbicki [6] were proposed zero-dimensional and two-phase combustion model for dual-fuel CI engine simulation. Rakopoulos *et al*. [7] have conducted the multi-zone modelling to analyse the development of fuel spray while diesel, biodiesel and crude vegetable oil as a fuel for Diesel engine. Sanjay [8] was used the MATLAB program to simulate the model to the analysis of a single cylinder 3.5 kW diesel engine using palm oil methyl ester, diesel and its blends as a fuel. The computational results on brake thermal efficiency (BTE) and IP were closer (2-3%) to experimental results. In addition, they have developed a single-zone thermodynamic model for Diesel engine and it integrated with triple-Wiebe function to simulate heat release and cylinder pressure. The prediction of cylinder pressure and heat release rate by simulation model were found to be closer (2.2% and 2.5%) to experimental results. Furthermore, in respect to modelling of blended fuel use in direct injection Diesel engine, ethanol-diesel blend [7], vegetable oil, biodiesel and diesel [9] fuelled engine study have been conducted through 2-D and multi-zone model. The burning duration is the specific parameter, which can be precisely estimated at starting of computational program to predict the engine performance easily. Several authors [11, 12] have used the Wiebe function to predict the burning duration and performance analysis of CI engine. Shipinski *et al*.[13] were used the single exponential Wiebe function to express the burning rate of DI and in-DI Diesel engine and they found that burning duration depends on equivalence ratio and engine speed. Watson e*t al*. [14] were used the combustion correlation for Diesel engine simulation describe by Wiebe functions. They reported that the parameters appear in Wiebe functions are depends on engine operating parameters (engine speed and load) and geometry of combustion chamber. On the other hand, two-Wiebe function can be used to describe the burning duration in Diesel engine [15]. This function includes premixed and diffusive combustion period and depend on fuel injection timing and crank shaft angle. However, they have not given any information regarding the calculation of burning duration. Therefore, this paper represents a novel empirical correlation for CI engine burning duration by following methodology developed by Bayraktar and Durgun [16] for spark ignition (SI) engine. The prepared empirical relation is function of compression ratio, engine speed, equivalence ratio and fuel injection timing. This paper provides inclusive performance analysis using burning duration in quasi-dimensional computational modelling. In engine combustion, performance and emission have been predicted with variation of fuel injection timing (43, 33, 23, 13, 3 °bTDC and 7 °aTDC) and engine speed (1500, 2000, 2500, 3000, and 4000 rpm).

Development of an empirical correlation for burning duration

Burning duration (θ_d), is an important measure to optimize performance and emission of an engine. It is a function of equivalence ratio (*ER*), compression ratio (*CR*), speed (*N*), and injection timing (θ_{inj}). The burning duration was taken as the crank angle interval between start of burning to end of burning (last part of the charge). Following equation is the modified empirical relation and developed for CI engine. It has developed on the basis of the function of operating parameters and represented by $f_1(CR)$, $f_2(N)$, $f_3(ER)$ and $f_4(\theta_{inj})$. Thus the general form of burning duration can be expressed as:

$$\theta_d(CR, N, ER, \theta_{inj}) = f_1(CR) \cdot f_2(N) \cdot f_3(ER) \cdot f(\theta_{inj}) \cdot \theta_{d1} \tag{1}$$

This equation is used to determine the function. At the reference condition all the variables (operating parameters) are divided by their initial values such as CR_{ref}, N_{ref}, ER_{ref} and

$\theta_{\mathrm{inj.ref}}$. The approximate functions $f_1(CR)$, $f_2(N)$, $f_3(ER)$, and $f_4(\theta_{\mathrm{inj}})$ can be evaluated by applying a curve fitting method with the used of numerical value of burning duration (θ_{d}) for different values of CR, N, ER, θ_{inj}. The following second degree polynomial equations have been obtained.

$$f_1(CR) = 3.46 - 3.42\left(\frac{CR}{CR_{\mathrm{ref}}}\right) + 0.98\left(\frac{CR}{CR_{\mathrm{ref}}}\right)^2 \tag{2}$$

$$f_2(N) = 0.76 + 0.30\left(\frac{N}{N_{\mathrm{ref}}}\right) - 0.03\left(\frac{N}{N_{\mathrm{ref}}}\right)^2 \tag{3}$$

$$f_3(ER) = 4.21 - 5.65\left(\frac{ER}{ER_{\mathrm{ref}}}\right) + 2.40\left(\frac{ER}{ER_{\mathrm{ref}}}\right)^2 \tag{4}$$

$$f_4(\theta_{\mathrm{inj}}) = 0.64 - 0.26\left(\frac{\theta_{\mathrm{inj}}}{\theta_{\mathrm{inj,ref}}}\right) + 0.10\left(\frac{\theta_{\mathrm{inj}}}{\theta_{\mathrm{inj,ref}}}\right)^2 \tag{5}$$

For the known value of reference operating parameters $CR_{\mathrm{ref}} = 12$, $N_{\mathrm{ref}} = 1000$ rpm, $ER_{\mathrm{ref}} = 1.0$, $\theta_{\mathrm{inj,ref}} = -30$ °CA, burning duration ($\theta_{\mathrm{d}I}$) is 45 °CA.

Figure 1 shows comparison of the computed burning duration by using the empirical correlation, eq. (1), and experimental value given by Kumar *et al.* [17] with CR. There is a good agreement between computed with experimental trend and value with 0.30% error. Similarly, fig. 2 shows the variation of burning duration calculated by using the empirical correlation, eq. (1), at different injection timing and compared with experimental results given by Agarwal *et al.* [18]. The figure shows the good agreement between burning duration results obtained from empirical equation and experimental value with 1.34% error.

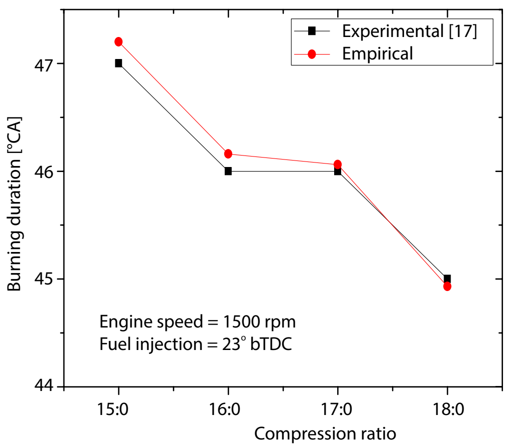

Figure 1. Burning duration *vs.* compression ratio

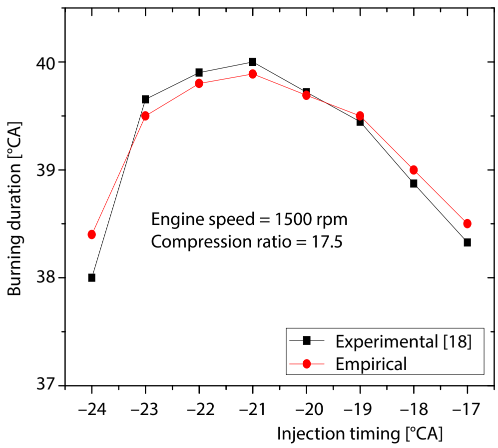

Figure 2. Burning duration *vs.* injection timing

Mathematical and computational modelling

In this paper, we have used the computational analytical approach for the prediction of combustion, engine performance, and emissions. The quasi-dimensional thermodynamic simulation model has been used to simulate closed power cycle of CI engine comprising, injection timing, delay period, combustion, and expansion. The models developed by Whitehouse and Sareen [19], and Kumar *et al.* [20], are being applied to simulate the Diesel engine working process and its spray mixing model. This model devides the mixture charge into burnt and unburnt zones; the burned zone corresponds to injected fuel and unburned zone corresponds to surrounding air. After injection, the ignition-delay period was calculated with respect to consuming 0.001 of the total volume of vaporized and mixed with air as spray nuclei [21]. The assumed spherical spray (burning) nuclei propagation in the auto flame region inside the cylinder are simulated with respect to crank angle and corresponding area of burned and unburned [22] till the end of combustion where the volume of fresh charge (Vm) is just negative. Once combustion is completed, the variables are organized for a single-zone calculation. Modified empirical relations have been used to estimate burning duration as a function of equivalence ratio, speed, CR and injection timing. Heat transfer between the burned and unburned zone was neglected and the heat transfer from cylinder content to the wall has been predicted with semi-empirical correlation developed by [23]. The FORTRAN based computational simulation used to calculate with respect to crank angle of unburned and burned temperature, pressure, burned and unburned mass, volume and 12 combustion species. The variables were integrated and solved with forth order Runge-Kutta method and numerical solution by Newton-Raphson technique. In the numerical model following assumption and approximation were considered:

– integrated area for the heat transfer from burned and unburned spray (burning) nuclei auto flame region inside the cylinder is assumed to propagate with Wiebe function,
– during fuel injection and combustion homogeneous conditions throughout the combustion chamber, and ideal gas behaviour,
– air entrainment due to shearing action of fuel spray,
– combustion does not produce additional air entrainment,
– at any time pressure throughout the cylinder is uniform,
– the negligible volume occupied by auto ignition nuclei in the flame reaction zone,
– except nitrogen for species, the burned gases are at complete thermodynamic equilibrium,
– the unburned gas is frozen at its original composition,
– both burned and unburned gases have uniform local specific heat, and
– there is no heat transfer between burned and unburned zone.

Compression modelling

Compression started from IVC and the trapped state includes fresh charge and residual charge of the previous cycle [24]. Computational modelling simulates 20 power cycles for the establishment of species fraction of the exhaust through Newton-Raphson method. During compression charges and residual species are assumed in freeze condition. Pressure variation with crank angle inside the combustion chamber during compression is govern by the following eq. (6):

$$\frac{\mathrm{d}p}{\mathrm{d}\theta}=\left[\frac{R}{C_v}\frac{\mathrm{d}Q}{\mathrm{d}\theta}-\frac{p\mathrm{d}V}{\mathrm{d}\theta}\left(\frac{R}{C_v}+1\right)\right]\frac{1}{V} \tag{6}$$

where V is the instantaneous volume of the cylinder:

$$V=\frac{\pi}{4}D^2r\left[\left(1-\cos\theta\right)+\frac{L}{r}-\sqrt{\frac{L^2}{r^2}-\sin^2\theta}\right] \tag{7}$$

At the same time temperature during compression is calculated:

$$\frac{\mathrm{d}T}{\mathrm{d}\theta}=T\left(\frac{1}{V}\frac{\mathrm{d}V}{\mathrm{d}\theta}+\frac{1}{p}\frac{\mathrm{d}p}{\mathrm{d}\theta}\right) \tag{8}$$

where Q is the heat transfer from the gas to the wall and (Q) is caculated by using Annand's equation [18]:

$$\frac{Q}{F} = \frac{a_c K_q}{D}\left(R_e\right)^{b_c}\left(T_m - T_w\right) + c_c\left(T_m^4 - T_w^4\right) \tag{9}$$

where a_c, b_c, and c_c are Annand's constant values, a_c = 0,4, b_c = 0,7, c_c = 4.3·10^{-9}, and F is the area of cylinder walls, D – the cylinder bore, K_q – the thermal conductivity, Re – the Reynolds number, T_m – the temperature of unburned charge/products, and T_w – the cylinder walls temperature.

Combustion modelling

After injection of fuel the delay period is calculated with empirical relation used by [7] In this stage, state of gas inside the cylinder is calculated applying First law of thermodynamic for whole closed system is:

$$\frac{\mathrm{d}Q}{\mathrm{d}\theta} = \frac{\mathrm{d}E}{\mathrm{d}\theta} + \frac{\mathrm{d}W}{\mathrm{d}\theta} \tag{10}$$

where E is total internal energy of the whole system include unburned and burned gases:

$$E = m_u e_u + m_b e_b \tag{11}$$

and the total internal energy variation corresponding to crank angle equation:

$$\frac{\mathrm{d}E}{\mathrm{d}\theta} = m_u \frac{\mathrm{d}e_u}{\mathrm{d}\theta} + e_u \frac{\mathrm{d}m_u}{\mathrm{d}\theta} + m_b \frac{\mathrm{d}e_b}{\mathrm{d}\theta} + e_b \frac{\mathrm{d}m_b}{\mathrm{d}\theta} \tag{12}$$

The $\mathrm{d}m_b/\mathrm{d}\theta$ is the burned mass fraction and $\mathrm{d}m_u/\mathrm{d}\theta$ is the unburned mass of a total mass ($m_t = m_u + m_b$). The rate of burning of fuel and corresponding area of total heat transfer $\mathrm{d}Q/\mathrm{d}\theta$ to cylinder wall from two zone (burned and unburned) are governed by Wiebe function:

$$\frac{\mathrm{d}m_b}{\mathrm{d}\theta} = m_t \frac{\mathrm{d}x_b}{\mathrm{d}\theta} \tag{13}$$

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_\mathrm{d}}\right)^n\right] \tag{14}$$

$$\frac{\mathrm{d}x_b}{\mathrm{d}\theta} = \left(1 - x_b\right)\frac{an}{\theta_\mathrm{d}}\left(\frac{\theta - \theta_s}{\theta_\mathrm{d}}\right)^{n-1} \tag{15}$$

where x_b is the mass fraction of fuel burnt at given crank angle θ, θ_s – the start of combustion, a – the Weibe efficiency factor (for $x_{\max}$ assumed value is 6.908), n – the Weibe form factor (for $x_{\max}$ assumed value is 3), and θ_d is the burning duration.

Applying thermodynamic equation for unburned and burned (product) gases temperature and pressure of the inside cylinder are calculated from:

$$\frac{\mathrm{d}T_u}{\mathrm{d}\theta} = \frac{1}{m_\mathrm{u} C_\mathrm{pu}}\left(V_u \frac{\mathrm{d}P}{\mathrm{d}\theta} + \frac{\mathrm{d}Q_u}{\mathrm{d}\theta}\right) \tag{16}$$

$$\frac{\mathrm{d}T_b}{\mathrm{d}\theta} = \frac{1}{m_\mathrm{u} C_\mathrm{pu}}\left[P\frac{\mathrm{d}V}{\mathrm{d}\theta} - \left(R_\mathrm{b} T_\mathrm{b} - R_\mathrm{u} T_\mathrm{u}\right)\frac{\mathrm{d}m_\mathrm{b}}{\mathrm{d}\theta} - \frac{R_\mathrm{u}}{C_\mathrm{pu}}\left(V_\mathrm{u}\frac{\mathrm{d}P}{\mathrm{d}\theta} + \frac{\mathrm{d}Q_u}{\mathrm{d}\theta}\right) + V\frac{\mathrm{d}P}{\mathrm{d}\theta}\right] \tag{17}$$

$$\frac{\mathrm{d}P}{\mathrm{d}\theta}=\frac{1}{\dfrac{C_{\mathrm{vb}}}{C_{\mathrm{pu}}}\dfrac{R_{\mathrm{u}}}{R_{\mathrm{b}}}V_{\mathrm{u}}-\dfrac{C_{\mathrm{vu}}}{C_{\mathrm{pu}}}V_{\mathrm{u}}-\dfrac{C_{\mathrm{vb}}}{R_{\mathrm{b}}}V}\left\{\begin{aligned}&\left(1+\frac{C_{\mathrm{vb}}}{R_{\mathrm{b}}}\right)P\frac{\mathrm{d}V}{\mathrm{d}\theta}-\frac{\mathrm{d}Q}{\mathrm{d}\theta}+\left[\left(u_{\mathrm{b}}-u_{\mathrm{u}}\right)-C_{\mathrm{vb}}\left(T_{\mathrm{b}}-\frac{R_{\mathrm{u}}}{R_{\mathrm{b}}}T_{\mathrm{u}}\right)\right]\frac{\mathrm{d}m_{\mathrm{b}}}{\mathrm{d}\theta}+\\&+\left(\frac{C_{\mathrm{vu}}}{C_{\mathrm{pu}}}+\frac{C_{\mathrm{vb}}}{R_{\mathrm{b}}}\frac{R_{\mathrm{u}}}{C_{\mathrm{pu}}}\right)\frac{\mathrm{d}Q_{\mathrm{u}}}{\mathrm{d}\theta}\end{aligned}\right\} \tag{18}$$

where subscript u is unburned and b shows burnt zone. The $\mathrm{d}Q/\mathrm{d}\theta$ heat transfer from the gas to cylinder coolant wall, which is estimated by Aannad equation [23] Runge-Kutta fourth order numerical procedure, was adopted to solve the differential equation, and the variables are incremented in order of any X variable:

$$X_{n+1}=X_n+\frac{\mathrm{d}X}{\mathrm{d}\theta}\Delta\theta \tag{19}$$

The specific heat of exhaust species was calculated using polynomial coefficients [25]:

$$\left(\frac{C_p}{\overline{R}}\right)=a_1+a_2T+a_3T^2+a_4T^3+a_5T^4 \tag{20}$$

$$C_V=(a_1-1)\overline{\mathrm{R}}+(a_2T+a_3T^2+a_4T^3+a_5T^4)\overline{\mathrm{R}} \tag{21}$$

Polynomial coefficients (a_1, a_2, a_3, a_4, a_5) for fuel of thermodynamic properties were taken from [26], and enthalpy calculated from eq. (22):

$$\overline{h}_f=A_{f1}T+\frac{A_{f2}T^2}{2}+\frac{A_{f3}T^3}{3}+\frac{A_{f4}T^4}{4}-\frac{A_{f5}}{T}+A_{f6}+A_{f8}\ [\mathrm{Jkg^{-1}mol^{-1}}] \tag{22}$$

where A_{f6} is the enthalpy constant at 298.15 K, $A_{f6}+A_{f8}$ the enthalpy constant at 0 K.

Internal energy for fuel $\bar{u}_f$ [$\mathrm{Jkg^{-1}mol^{-1}}$] is given by:

$$\bar{u}_f=(A_{f1}T-\overline{\mathrm{R}}T)+\frac{A_{f2}T^2}{2}+\frac{A_{f3}T^3}{3}+\frac{A_{f4}T^4}{4}-\frac{A_{f5}}{T}+A_{f6}+A_{f8} \tag{23}$$

Combustion and species formation

General combustion equation of the fuel ($C_mH_nO_p$) is given by:

$$\mathrm{C}_m\mathrm{H}_n\mathrm{O}_p+Y_{cc}(\mathrm{O}_2+3.76\mathrm{N}_2)\rightarrow m\mathrm{CO}_2+n\mathrm{H}_2\mathrm{O}+3.76Y_{cc}\mathrm{N}_2 \tag{24}$$

where m, n, and p are moles of C, H, and O. The Y_{cc} ($=m+n/4-p/2$) is the chemically correct moles of O_2 per mole of fuel, calculated by balancing the oxygen molecules both side of equation 24. Now the stoichiometric equation for combustion:

$$\mathrm{C}_m\mathrm{H}_n\mathrm{O}_p+\left(m+\frac{n}{4}-\frac{p}{2}\right)(O_2+3.76\mathrm{N}_2)\rightarrow m\mathrm{CO}_2+n\mathrm{H}_2\mathrm{O}+3.76\left(m+\frac{n}{4}-\frac{p}{2}\right)\mathrm{N}_2 \tag{25}$$

Average chemical formula for diesel fuel is $C_{12}H_{23}$, used for stoichiometric combustion equation. From previous reaction equation we can calculate the stoichiometric air-fuel ratio (α_s). Assuming 12 species (H_2O, H_2, OH, H, N_2, NO, N, CO_2, CO, O_2, O, and Ar) present in the combustion product in the cylinder and in the exhaust. The equilibrium combustion equation is:

$$\begin{aligned}\left\{\mathrm{C}_{12}\mathrm{H}_{23}\right\}+\left(m+\frac{n}{4}-\frac{p}{2}\right)\left(\mathrm{O}_2+3.76\mathrm{N}_2\right)\rightarrow \mathrm{X}_{\mathrm{H_2O}}\mathrm{H_2O}+\mathrm{X}_{\mathrm{H_2}}\mathrm{H_2}+\mathrm{X}_{\mathrm{OH}}\mathrm{OH}+\mathrm{X}_{\mathrm{H}}\mathrm{H}+\\ \mathrm{X}_{\mathrm{N_2}}\mathrm{N_2}+\mathrm{X}_{\mathrm{NO}}\mathrm{NO}+\mathrm{X}_{\mathrm{N}}\mathrm{N}+\mathrm{X}_{\mathrm{CO_2}}\mathrm{CO_2}+\mathrm{X}_{\mathrm{CO}}\mathrm{CO}+\mathrm{X}_{\mathrm{O_2}}\mathrm{O_2}+\mathrm{X}_{\mathrm{O}}\mathrm{O}+\mathrm{X}_{\mathrm{A_r}}\mathrm{A_r}\end{aligned} \tag{26}$$

These species could reach equilibrium condition if sufficient time is allowed for the reactions to take place under a certain state. Criterion for chemical equilibrium can be expressed by Gibbs function as:

$$(\mathrm{d}G)_{T,p} = 0 \tag{27}$$

where specific Gibbs function is expressed:

$$\frac{\mathrm{g}(T)}{\bar{\mathrm{R}}T} = a_\mathrm{g}(1-\ln T) - b_\mathrm{g}T - \frac{C_\mathrm{g}}{2}T^2 - \frac{d_\mathrm{g}}{3}T^3 - \frac{e_\mathrm{g}}{4}T^4 - k_\mathrm{g} \tag{28}$$

where a_g, b_g c_g, d_g, e_g, and k_g are constants. Considering the seven equilibrium equation of reaction used by [25], and the value of equilibrium constant K_p was obtained by universal equation:

$$\upsilon_a A + \upsilon_a B \rightleftharpoons \upsilon_c C + \upsilon_d D \qquad K_p = \frac{X_c^{\upsilon_c} X_\mathrm{d}^{\upsilon_d}}{X_a^{\upsilon_a} X_b^{\upsilon_b}} \times P^{\upsilon_c + \upsilon_\mathrm{d} - \upsilon_a - \upsilon_b} \tag{29}$$

$$\ln K_p = \left[\sum \left\{ \frac{\upsilon g(T)}{\bar{\mathrm{R}}T} \right\}_\mathrm{Reactant} - \sum \left\{ \frac{\upsilon g(T)}{\bar{\mathrm{R}}T} \right\}_\mathrm{Product} \right] - \frac{\Delta H_o}{\bar{\mathrm{R}}T} \tag{30}$$

where υ is the stoichiometric coefficient, X, – the molar fraction, and p – the total pressure.

To determine the 12 species concentration and relative equilibrium constants following 7 chemical equilibrium equations were used:

$$\text{(i)}\ \frac{1}{2}\mathrm{H_2} \Leftrightarrow \mathrm{H} \quad \text{(ii)}\ \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{O} \quad \text{(iii)}\ \frac{1}{2}\mathrm{N_2} \Leftrightarrow \mathrm{N} \quad \text{(iv)}\ \mathrm{2H_2O} \Leftrightarrow \mathrm{2H_2} + \mathrm{O_2}$$

$$\text{(v)}\ \mathrm{H_2O} \Leftrightarrow \mathrm{OH} + \frac{1}{2}\mathrm{H_2} \quad \text{(vi)}\ \mathrm{CO_2} + \mathrm{H_2} \Leftrightarrow \mathrm{H_2O} + \mathrm{CO} \quad \text{(vii)}\ \mathrm{H_2O} + \frac{1}{2}\mathrm{N_2} \Leftrightarrow \mathrm{H_2} + \mathrm{NO}$$

The formation of NO and CO during combustion in the cylinder is a non-equilibrium process. Present work comprises the rate kinetics of NO using rate of a kinematic model for NO formation developed by Lavoie *et al*. [26]. These are the seven governing equations for NO formation:

$$\text{(i)}\ \mathrm{N} + \mathrm{NO} \Leftrightarrow \mathrm{N_2} + \mathrm{O} \quad \text{(ii)}\ \mathrm{N} + \mathrm{O_2} \Leftrightarrow \mathrm{NO} + \mathrm{O} \quad \text{(iii)}\ \mathrm{N} + \mathrm{OH} \Leftrightarrow \mathrm{NO} + \mathrm{H}$$

$$\text{(iv)}\ \mathrm{H} + \mathrm{N_2O} \Leftrightarrow \mathrm{N_2} + \mathrm{OH} \quad \text{(v)}\ \mathrm{O} + \mathrm{N_2O} \Leftrightarrow \mathrm{N_2} + \mathrm{O_2} \quad \text{(vi)}\ \mathrm{O} + \mathrm{N_2O} \Leftrightarrow \mathrm{NO} + \mathrm{NO}$$

$$\text{(vii)}\ \mathrm{N_2O} + M \Leftrightarrow \mathrm{N_2} + \mathrm{O} + M$$

where M is the third body which may be any of the other species in the mixture and remains chemically unchanged during the reaction, K_{1f} – the forward reaction rate of constant, K_{1b} – the backward reaction rate of constant, using this relation find the value of K_{1b}:

$$K_{1f}[\mathrm{N}]_e[\mathrm{NO}]_e = K_{1b}[\mathrm{N_2}]_e[\mathrm{O}]_e = R_1 \tag{31}$$

for first equation and similar relation was adopted for R_2, R_3, R_4, R_5 , R_6, and R_7 .The detail is in [21]. The total rate equation for NO is:

$$\frac{1}{V}\frac{\mathrm{d}}{\mathrm{d}t}\Big[[\mathrm{NO}]V\Big] = 2\left(1-\alpha_e^2\right)\left[\frac{R_1}{1+\alpha_e \dfrac{R_1}{R_2+R_3}} + \frac{R_6}{1+\dfrac{R_6}{R_4+R_5+R_7}}\right] \tag{32}$$

where $\alpha_e = [\mathrm{NO}]/[\mathrm{NO}]_\mathrm{e}$ and e represents the equilibrium condition.

Validation, results and discussion

The computational program was run with use of new empirical correlation of burning duration. The experimental works have been conducted on VCR engine run with diesel. The engine details are given in tab. 1. Figure 3 shows the validation of P-θ for experimental and computational results when engine fuelled with pure diesel at compression ratio 16. The different experimental result of P-θ in experimental works, among then we have taken P_{avg} at 215 CAD as trap pressure (P_{trap}). The natures of P-θ for computational results are nearly same as experimental with 5.67% error. The figs. 4 and 5 shows comparison of computed results (BMEP and brake power) with experimental value obtained by Atul Dhar *et al.* [27] and Rajasekar *et al.* [28] for burning duration, and shows good agreement between computed with experimental trend and value with 0.43% and 1.81% error. The slight variation in results due to the taking some assumption during mathematical formulation for computational program. The following results (BP, IP, BSFC, ISFC, BMEP, IMEP, P_{max} and NO emission) have obtained by previous validated computational program.

Table1. Engine specification

Parameters	Specification
General details	4-stroke, single cylinder, water cooled, compression ignition, multi-fuel, VCR
Product code	240PE
Rated power	3.5kW at 1500 rpm
Compression ratio	17.5:1 (range 12:1 to 18:1)
Bore/Stroke/Capacity	87.5 mm/110 mm/661 cc
Injection timing	23 °bTDC (range 0-30 °bTDC)
IVO/IVC	4.5 °bTDC/35.5 °aTDC
EVO/EVC	35.5 °bTDC/4.5 °aTDC

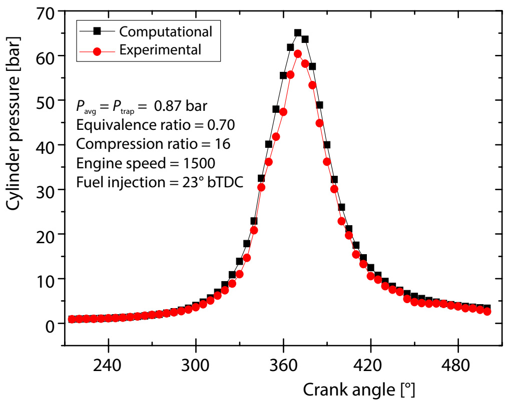

Figure 3. Validation of cylinder pressure and CA

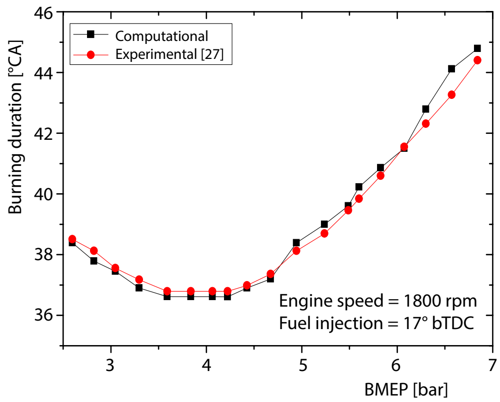

Figure 4. Burning duration *vs.* BMEP

Figure 6 shows the effect of injection timing (43, 33, 23, 13, 3 °bTDC and 7 °aTDC) on BP and IP at different engine speed (1500, 2000, 2500, 3000, and 4000 rpm). The value of BP and IP are increases as the injection timing is advanced or retarded from the rated value (23 °bTDC). It has the maximum value at 13 °bTDC and minimum at 7 °aTDC at all speed. BP and IP decreased by 0.97% and 0.71% for retarded the injection timing (from 43 °bTDC to 7 °aTDC). The figure also depicts that the BP and IP increase with engine speed.

Figure 7 depicts that the BSFC and ISFC are slight increases as the injection timing advanced from rated value. On the other hand, its value decreased with retarded condition. The minimum BSFC and ISFC are found at 13 °bTDC and maximum at 7 °aTDC for all speed ranges. Ganapathy *et al.* [29] reported that the value of BSFC are increased when the fuel injec-

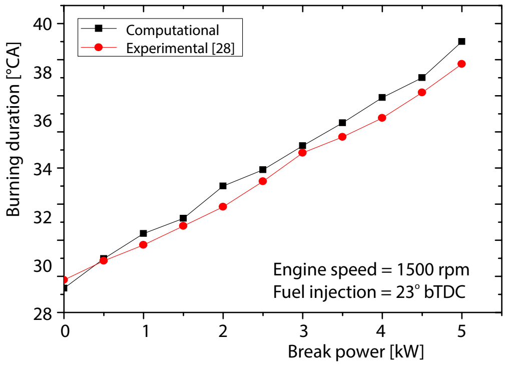

Figure 5. Burning duration *vs.* BP

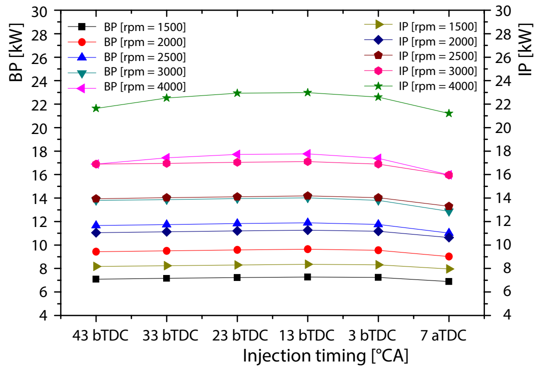

Figure 6. The BP and IP *vs.* injection timing

tion timing advanced or retarded from the rated crank angle degree (345) for different load and speed when engine run with diesel. As fuel injection timing increased from 340 to 350 CAD (crank angle degree) the value of BSFC continuously increased at given load and speed. It was also reported that the optimum injection timing is 340 CAD for minimum BSFC when engine fuelled with jatropha biodiesel. The figure also depicts that the increase in BSFC with increasing the engine speed. This is due to increase in frictional power at a rapid rate when the engine operates at higher speed consequently, power increasing slower rate than fuel consumption and hence increase the BSFC [30, 31]. On the other hand, reverse trends observed for ISFC. The maximum value of BSFC and ISFC are 0.259 and 0.195 kg/kWh at 4000 rpm with 7 °aTDC injection timing. The minimum value of BSFC and ISFC are 0.21345 and 0.18026 kg/kWh at 1500 and 4000 rpm with 13 °bTDC injection timing. In the literature [30] is also reported that the minimum value of BSFC is obtained at advanced fuel injection timing. The value of BSFC and ISFC are increased by 1.273% and 0.981% for retarded the injection timing from 43 °bTDC to 7 °aTDC.

Figure 8 shows the effect of injection timing on BMEP and IMEP and depicts that its value decreases as the injection timing advanced from rated value 23 °bTDC. On the other hand, opposite trends follow for retarded condition up to 13 °bTDC, then after its value decreases for all speed range. It produces minimum BMEP and IMEP at 7 °aTDC and maximum at 13 °bTDC. The value of BMEP and IMEP are decreased by 1.18% and 0.92% for retarded the injection timing from 43 °bTDC to 7 °aTDC. The figure also depicts that the decrease in BMEP

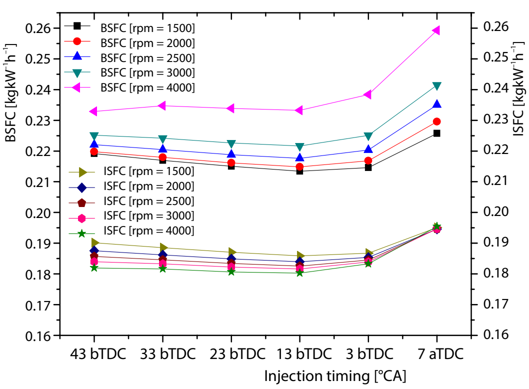

Figure 7. The BSFC and ISFC *vs.* injection timing

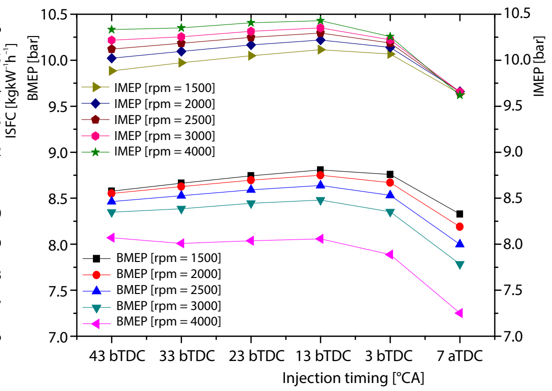

Figure 8. The BMEP and IPMEP *vs.* injection timing

and increase in IMEP while increasing the engine speed. This is primarily due to the increase in friction mean effective pressure (FMEP) with speed [25].

Figure 9 shows that the effect of injection timing on peak cylinder pressure (P_{max}), and depicts that the higher P_{max} at advanced (43 °bTDC) injection timing at all speed. Generally, cylinder pressure increases with advanced injection timing. With advancing the fuel injection timing the ignition delay increases, due to this the preparation of better air/fuel mixture and hence good combustion and higher cylinder pressure [33, 34]. However with retardation of injection timing shortens the ignition delay period, due to this reduced the cylinder pressure during the initial stage of combustion. The maximum value of P_{max} is 77.491 bars at 4000 rpm with injection timing 43 °bTDC. The Peak cylinder pressure 77.6 and 73.6 bar for diesel and pure biodiesel at 27 °bTDC [34]. In spite of these results, there is a significant decrease in peak pressures with speed increasing the speed at given injection timing; this is due to increase the ignition delay and decrease the combustion duration [29]. The peak cylinder pressure is decreased by 5.26% with retarded the injection timing from 43 °bTDC to 7 °aTDC.

The fig. 10 shows the effect of injection timing on NO emission at the different speed. It depicts that the NO decreases with retarded the injection timing from 43 °bTDC to 7 °aTDC. Ignition delay period increases with advancing the injection timing, due to this more burning of the air/fuel mixture in premixed combustion phase and hence higher combustion temperature. On the other hand, NO emission decreases for retarded injection timing. Similar variations have also been obtained by Gnanasekaran *et al*. [34] and Sayin *et al*. [32]. The NO formation decreased by 13.67% with retarded the injection timing from 43 °bTDC to7 °aTDC. Another variation, the NO emissions decreases with increasing the engine speeds. The minimum and maximum values of NO are 429.1 ppm (at 4000 rpm) and 1942.4 ppm (at 1500 rpm) for 7 °aTDC and 43 °bTDC, respectively.

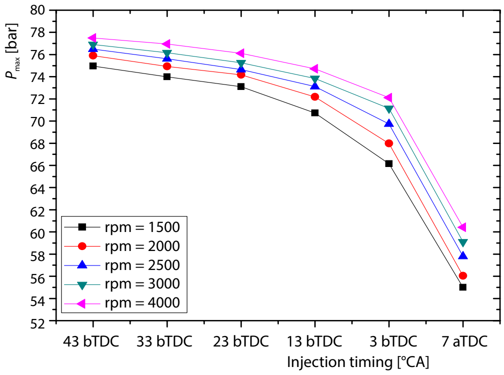

Figure 9. Peak cylinder pressure *vs.* injection timing

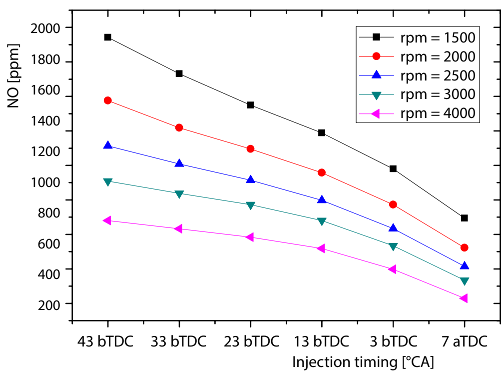

Figure 10. The NO formation *vs.* injection timing

Conclusions

The empirical correlation has been proposed for evaluation of burning duration in CI engine. This developed correlation comprises the effects of engine speed, compression ratio, fuel injection timing and equivalence ratio. The correlation was validated and by using of this improves the accuracy of computational result and reasonably agrees with experimental results. The following conclusions are drawn based on the mathematical modelling and computational program with new developed empirical correlations for burning duration.

- The brake power and indicated power are slightly increases with retarded condition (from 43 °bTDC to 13 °bTDC), after that its value slightly decreased. Both BP and IP are increases with increasing engine speed at any injection timing.
- Both BSFC and ISFC decrease with retarded injection timing from 43 °bTDC to 13 °bTDC after that its value increased. But, BSFC increases while ISFC decreases with respect to engine speed for all injection timing.
- The BMEP and IMEP are increased with retarded the injection timing from 43 °bTDC to 13 °bTDC, after that its value decreased. The decrease of BMEP and increased of IMEP with speed for all injection timing.
- The peak cylinder pressure decreases with retarding condition (43 °bTDC to 7 °aTDC) while its value increases with increasing the engine speed.
- The NO emission gradually decreases with with retarding condition (43 °bTDC to 7 °aTDC), but it value decreases with increasing the engine speed.

Nomenclature

aTDC – after top dead centre
bTDC – before top dead centre
°CA – degree of crank angle
CR – compression ratio
C_V/ C_P – specific heat of gas at constant volume/ pressure, [Jkg^{-1}K^{-1}]
E – total internal energy, [J]
e – specific internal energy, [Jkg^{-1}]
F – surface area of cylinder wall, [m^2]
h – specific enthalpy, [Jkg^{-1}]
K_q – thermal conductivity
L – connecting rod length, [m]
m – mass of fuel, [kg]
N – engine speed, [rpm]
P_{avg} – average pressure, [bar]
$P_{\max}$ – peak cylinder pressure
P_{trap} – trap pressure, [bar]
Q – heat, [W]
R – gas constant, [Jkg^{-1}K^{-1}]
$\bar{\mathrm{R}}$ – universal gas constant, [Jkmol^{-1}K^{-1}]
Re – Reynolds number
r – crank radius, [m]
T – temperature, [K]
V – cylinder volume, [m^3]
W – work, [W]

Greek symbols

θ_{inj} – injection timing, [°CA]
θ_{d} – burning duration, [°CA]
θ – crank angle, [deg]

Acronymes

BP – brake power
BMEP – brake mean effective pressure
BSFC – brake specific fuel consumption
CV – calorific value of fuel, [Jkg^{-1}]
ER – equivalence ratio
EVO – exhaust valve open
EVC – exhaust valve close
IMEP – indicated mean effective pressure
IP – indicated power
ISFC – indicated specific fuel consumption
IVO – inlet valve open
IVC – inlet valve close
NO – nitric oxide

Subscripts

u – unburned
b – burned
ref – reference
w – wall

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