Effect of Ignition Timing and Combustion Duration on the Performance Characteristics of a Diesel Engine Using Vibe 2-Zone Model

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Efficient energy exploitation is a necessary issue because it helps reduce fuel consumption and environmental pollution. Finding the optimal ignition timing (IT) for diesel engines to create high power and efficiency deserves attention. This study utilizes AVL BOOST simulation software with the Vibe 2-Zone combustion model to investigate the effect of IT and combustion duration on engine characteristics such as power, torque, and brake-specific fuel consumption (BSFC) at different engine loads and speeds. Then, the prediction models of the optimal ITs versus combustion durations for maximum power and minimum BSPFC were computed. The results show that ITs strongly affect engine performance characteristics. The optimal ITs that the engine produces maximum power at different combustion durations are unaffected by engine load. In contrast, they are considerably influenced by engine load when considering the engine-generated BSFC. The correlations of optimal IT versus combustion duration are linear functions. The prediction models can be utilized to predict the optimal ignition timings of the engine since the experimental time can be reduced when applied to the actual engine.

Keywords: *ignition timing, combustion duration, diesel engine, performance characteristics, AVL BOOST, Vibe 2-Zone.*

1. INTRODUCTION

Diesel engines are frequently utilized in power generators and heavy transportation applications because of their excellent economy [1] and trustworthiness [2]. They significantly advance economic growth and industrial development. However, several pollutants, such as unburned hydrocarbons (HC) emissions, carbon monoxide (CO) emissions, particulate matter (PM) emissions, and nitrogen oxide (NOx) emissions, etc. are released during the diesel engine combustion process [3]. Thus, diesel engine exhaust adversely degrades the environment and endangers the general public's health [4]. There is a trade-off between PM and NOx, the two primary contaminants found in diesel engines, that is consistent with previous studies [5-8]. Many experiments and simulations were conducted on diesel engines to investigate their characteristics [9-13] and find ways to optimize engines to increase performance or reduce engine emissions. One efficient solution to decrease emissions and enhance engine performance in direct injection diesel engines is to control the ignition timing (IT), one of the variables influencing the delay period of combustion in the engine that needs to be investigated and optimized to understand better how it affects the compression ignition engines' performance characteristics.

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There have been many studies to optimize engine injection timing. Nagesh et al. [14] examined the impact of injection timing on engine performance using acid oil methyl ester-fueled diesel. The study's findings show that the optimal IT at 27° before top dead center (bTDC) achieves the lowest emissions (NOx, CO, and HC) and the highest brake thermal efficiency (BTE) when using acid oil methyl ester fuel. Ganapathy et al. [15] used a test method to examine the effect of injection timing on the Jatropha biodiesel engine's specificities. The findings demonstrate that the optimal period for injecting Jatropha biodiesel working with minimal smoke, HC, CO, and brake specific fuel consumption (BSFC) and with maximal peak cylinder pressure, brake thermal efficiency, and net heat release rate was determined to be 340 crank angle degree. Khatri et al. [16] investigated direct injection CI engine fuel injection timing using preheated Karanj-Diesel Blend. Based on the findings, the optimum period for injection for the Karanj-Diesel mixture is established and is discovered to be 19° bTDC. Parlak et al. [17] studied minimal heat rejection, indirect injection of diesel engines' nitrogen oxide emissions, and the impacts of injection timing. Those are the ideal settings for the initial engine, and they increase NOx emissions by roughly 15% when the low heat rejection engine has an injection timing of 38° crank angle bTDC (CA bTDC). Bora et al. [18] investigated the best times to input fuel for a dual-fuel diesel engine running on raw biogas. According to the findings, this specific diesel engine's pilot fuel injection timing of 29° bTDC produces the highest possible BTE of 25.44% and the highest possible fuel-liquid alternative rate of 82.11%. Indudhar et al. [19] studied the impact of injection timing on the efficiency of the biodiesel ester in a CRDI engine that burns hongeoil. According to the findings, the ideal injection timing was 10° bTDC, which achieved maximum BTE and minimal levels of CO, HC, and smoke emissions; however, the fuels utilized in that work had higher emissions of NOx. Sudarmanta et al. [20] developed a prototype DI 20C biodiesel engine and tested it by optimizing the injection timing for fuel spritze efficiency and engine emissions. With an improvement in engine power and BTE of 3.9% and 13.9%, respectively, the findings were that the engine performance and emissions could be optimized with the injection timing of 16° bTDC. Raheman et al. [21] experimented to see how a biodiesel-powered diesel engine performs with different ITs and compression ratios. The results demonstrated that, in terms of both the BTE and BSFC, an injection advance of 40° bTDC combined with a compression ratio of 20:1 was the most effective. Bakhshi Mehul et al. [22] simulated twin-spark and single-spark engines by adjusting the spark plug location and ignition timing. The results showed that the best performance of the single-spark engines occurred when the spark plug was located at the center of the cylinder and the ignition timing was early. In contrast, twin-spark engines function best when there is a significant differential in ignition timing among the two spark plugs.

Besides articles about engine experiments, there are also many articles about using software to simulate engines, and one of the popular engine simulation software is AVL BOOST [23-28]. AVL BOOST is a thermodynamic modeling program that computes the engine's performance. With the help of AVL BOOST, an engine period, and the gas transfer emulation program, you may create a model of the whole research engine by choosing components from a toolbox and connecting them with pipes. These components include improved junction models, turbochargers, intercoolers, air cleaners, catalysts, cylinders, etc. One-dimensional gas dynamics are considered inside the pipes. A modified Godunov-Schema provides the solution's excellent precision. BOOST can replicate both engine transients and steady-state operating points. A modified Godunov-Schema provides the solution's excellent precision. BOOST can replicate both engine transients and steady-state operating points.

Diesel engine combustion duration is defined by the crank angle degrees between the points of start of com–bustion (SOC) and end of combustion (EOC). The appa–rent heat release rate (AHRR) depicts the heat transfer pattern of the gases inside the cylinder, which may also be produced using these pressure data in conjunction with the appropriate cylinder volumes. In contrast to the EOC, which occurs at the crank angle (CA) when the AHRR achieves a flat slope just before the engine's exhaust stroke, The SOC is the crank angle when the AHRR value is lowest and then quickly increases; contrast, the EOC is identified where AHRR decreases and achieves a constant just before the engine's exhaust stroke [29].

The previous studies mentioned in the literature are primarily experimental to investigate the effect of the different injection timing on the engine and find out the optimal injection timing in case of maximum power or smallest value of BSFC at a specific operating condition; however, there are no studies utilize the simulation software to find out the optimal ignition timing and establish the predictive model for the optimal ignition timing to predict this value in a wide range of operating condition (load and combustion duration). Experimentation has the advantage that the results are more reliable than simulation. However, experiments have the disadvantage of consuming time, money, fuel, and limited human resources, so engine experiments are usually only performed under a few specific operating conditions. Engine simulation has the advantage of requiring few experiments and can investigate engine characteristics in many cases to predict engine characteristics in real conditions.

Consequently, in this work, an emulation model of a single-cylinder diesel engine was developed using AVL BOOST, and model calibration was carried out. Four input parameters, a start of combustion, a combustion duration, the shape factor "m" and Vibe parameter "a" were chosen for the simulation research. Then, the impact of ITs under different combustion durations on efficiency at 50%, 70%, and 85% engine load were investigated, and the optimal ITs at specific combustion durations were determined based on maximum power and minimum BSFC. Finally, a prediction model of the connection between combustion durations and optimal ITs at 50%, 70%, and 85% engine load was established.

This paper aims to (1) investigate the effect of different ignition timing on the performance of the diesel engine at different loads and (2) find the optimal ITs according to the maximum power and the minimum BSFC under different combustion durations and engine loads. The prediction models were created based on the optimal ITs and combustion durations at different loads. With specific engine loads, the researchers or manufacturer only experiment to find the combustion duration, then use the predictive model to predict the range of the optimum ignition timing. The injection timing can be determined if the engine's ignition delay is obtained. This work helps reduce the experiment time and save money when they need to improve this diesel engine's fuel economy or performance by changing the injection timing method.

2. SIMULATION SETUP

Figure 1 illustrates the modeled Vikyno RV125-2 diesel engine, a single-cylinder, four-stroke engine with a direct injection fuel system. Table 1 specifically lists and names the components of the simulated engine. The whole specifications of the Vikyno RV125-2 engine are revealed in Table 2.

The simulation model kept the general specifications of the original diesel engine. The engine ran at 85, 70, and 50% of the load, and the speed of the engine was set from 900 to 2400 rpm with a step of 300 rpm, while ignition timings (ITs) were investigated between 5 and 17°CA bTDC and combustion duration was surveyed from 80 to 120°CA by a step of 10°CA. The shape parameter m = 0.6 for all cases of the simulation. The Vibe parameter a = 6.9 for complete combustion. In this study, the Vibe parameter a = 6.9 was determined because the research assumes this combustion process is complete combustion (99.9% of the fuel is burned as the combustion process ends) [30-38]. In reference [33], the coefficient a was calculated when the assumption of 99.9% fuel burn is a = - Ln (l - x) = - Ln 0.001 = 6.9 with x = 99.9%. To evaluate the influence of various ITs under different combustion durations on engine performance, torque, power, and brake-specific fuel consumption (BSFC) are simulated and analyzed by changes in ITs at specific combustion durations.

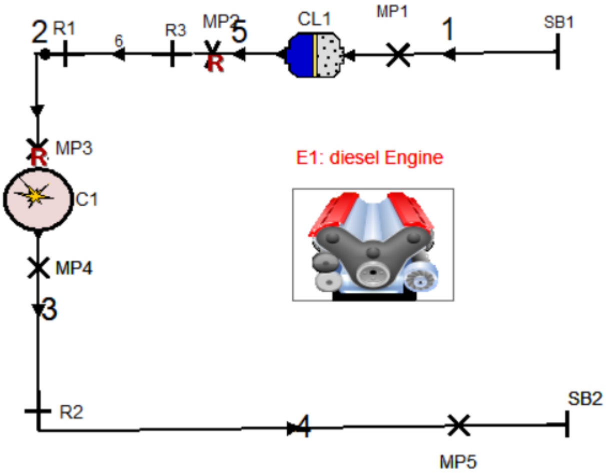


Figure 1. Simulation engine model in ALV BOOST

Table 1. The elements of the simulation engine model

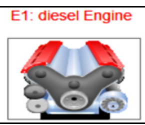
Elements	Symbols	Quantity
Boundary conditions (temperature, pressure, etc., input SB1, output SB2)		02
Piping (connecting elements)		06
Air cleaner (cleaning of intake air, determination of intake air volume)		01
Flow resistance (replaces pressure losses in the system)		03
Cylinder (combustion chamber parameters)		01
Measuring point (determine the flow parameters and gas conditions in the pipe)		05
Engine (set the number of crank revolutions, ignition timing, etc.)		01

Table 2. Typical specifications of Vikyno RV125-2 engine

Parameter	Unit	Value
Number of cylinders	---	1
Compression ratio	---	18:1
Displacement	cm^3	624
Power	HP/rpm	12.5/2400
Torque	Nm/rpm	4.04/1800
Bore	mm	94
Stroke	mm	90

2.1 Combustion model

An identical input for the single Vibe function is needed for the Vibe 2-Zone combustion model. However, 2 temperatures (burned and unburned regions) are deter–mined as opposed to one mass-averaged temperature. If the supplied Vibe function accurately captures the actual rate of heat escape, this combustion model can anticipate the engine's knocking characteristics [39]. This paper used the straightforward and frequently used Vibe 2-Zone combustion model to determine the HRR. The burned and unburned zones are separated into two zero-dimensional regions in the combustion chamber by the Vibe 2-Zone combustion model, based on the zero-dimensional model, considering the cylinder temperature and the disparate allocation of different molecules during the combustion process in the actual engine. The combustion process occurs in the burnt zone, which includes the air-fuel mixture introduced along with combustion byproducts. Some combustion byproducts remain in the burnt zone, while others enter the unburned zone. Air will enter the burning zone to assist and increase the combustion reaction in the burning zone, while the non-burning zone includes natural gas, air, and combustion byproducts from the burning zone.

Additionally, the aggregate of the capacities of the two zones is identical to the entire capacities of the cylinder, and the variation in the combined volume in the two regions corresponds to the variation in the capacity of the cylinder. Another crucial justification for using this model is that it can forecast the engine's knocking characteristics. The following is an explanation of the model formulation [40]:

$$\begin{aligned}\frac{\mathrm{d}\left(m_{\mathrm{b}}u_{\mathrm{b}}\right)}{\mathrm{d}\alpha} &= -p_{\mathrm{c}}\frac{\mathrm{d}V_{\mathrm{b}}}{\mathrm{d}\alpha}+\frac{\mathrm{d}Q_{\mathrm{F}}}{\mathrm{d}\alpha}-\sum\frac{\mathrm{d}Q_{\mathrm{Wb}}}{\mathrm{d}\alpha}\\ &+h_{\mathrm{u}}\frac{\mathrm{d}m_{\mathrm{b}}}{\mathrm{d}\alpha}-h_{\mathrm{BB,b}}\frac{\mathrm{d}m_{\mathrm{BB,u}}}{\mathrm{d}\alpha}\end{aligned} \tag{1}$$

$$\begin{aligned}\frac{\mathrm{d}\left(m_{\mathrm{u}}u_{\mathrm{u}}\right)}{\mathrm{d}\alpha} &= -p_{\mathrm{c}}\frac{\mathrm{d}V_{\mathrm{u}}}{\mathrm{d}\alpha}-\sum\frac{\mathrm{d}Q_{\mathrm{Wu}}}{\mathrm{d}\alpha}\\ &-h_{\mathrm{u}}\frac{\mathrm{d}m_{\mathrm{b}}}{\mathrm{d}\alpha}-h_{\mathrm{BB,u}}\frac{\mathrm{d}m_{\mathrm{BB,u}}}{\mathrm{d}\alpha}\end{aligned} \tag{2}$$

$$\frac{\mathrm{d}V_{\mathrm{b}}}{\mathrm{d}\alpha}+\frac{\mathrm{d}V_{\mathrm{u}}}{\mathrm{d}\alpha}=\frac{\mathrm{d}V}{\mathrm{d}\alpha} \tag{3}$$

$$V_{\mathrm{b}}+V_{\mathrm{u}}=V \tag{4}$$

where Q_F and α represent the total amount of heat created during fuel combustion and the crank angle, respectively. $h_u(\mathrm{d}m_b/\mathrm{d}\alpha)$ depict the transfer of enthalpy between the unburned and burned regions, respectively. m_b and m_u are burned and unburned masses, respectively. h_{BB} is the enthalpy of blow-by.

2.2 Heat transfer model

In cylinder of diesel engines, the heat transfer process is exceedingly complicated. The impact of heat convective conduction in cylinders is primarily examined during the heat transfer process [41]. The Woschni 1978 model [42] is typically employed for calculating high-pressure flow transfer of heat. It can be summed up as follows:

$$Q_W = A_i \alpha_w \left(T_c - T_{wi}\right) \quad (5)$$

where Q_W, A_i, α_w, T_c and T_w are heat loss through walls, surface area, temperature-transfer efficiency, cylinder temperature, and temperature of the wall, respectively.

$$\alpha_w = 130.D^{-0.2}.p_c^{0.8}.T_c^{-0.53}. \left[C_1.C_m + C_2.\frac{V_D.T_{c,1}}{p_{c,1}.V_{c,1}}.\left(p_c - p_{c,0}\right)\right]^{0.8} \quad (6)$$

where p_c is the pressure of the cylinder. C_1 and C_2 are the coefficients of gas velocity and the model constant, respectively. C_1 = 2.28 + 0.308C_u/C_m, and C_2 = 0.00342 for direct injection engine, C_m and C_u depict mean and circumferential piston velocities, respectively. V_D and D are cylinder displacement and diameter, respectively. $T_{c,1}$ and $p_{c,1}$ are the volume and pressure of the cylinder with the inlet valve closed, respectively. V and $p_{c,0}$ represent the realistic cylinder volume and the reversed cylinder pressure, respectively.

2.3 Calculate the friction mean effective pressure (FMEP)

In this study, intake airflow rate, fuel consumption rate, and torque were tested experimentally on a Vikyno RV-125 engine at two engine loads of 85, 70, and 50% in investigated speeds from 900 to 2400 rpm. The mode–ling simulation employed experimental parameters such as fuel consumption and intake air flow rates to establish the indicated mean effective pressure (IMEP), friction mean effective pressure (FMEP), and braking mean effective pressure (BMEP). The simulated torque of the engine can be determined from the BMEP following equation [43]:

$$Torque_{sim} = \left(BMEP \cdot V_D\right)/4\pi \quad (7)$$

Simulation power is obtained using the following formula [43]:

$$P_{sim} = 2\pi Torque_{sim} N \quad (8)$$

where N is engine speed (rpm)

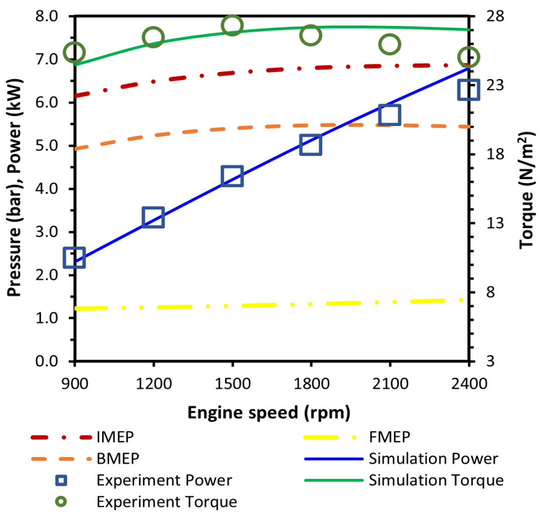
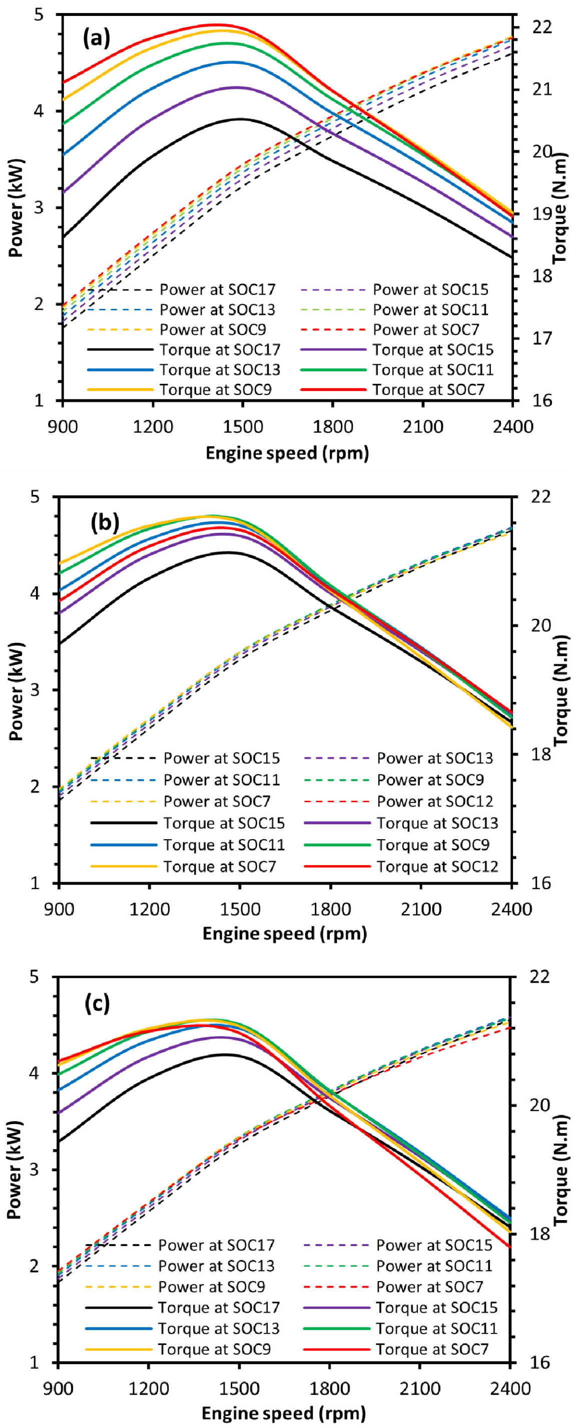


Figure 2. Experiment torque and power, simulation torque and power, BMEP, FMEP, and IMEP versus engine speed at 70% load

To maintain the reliability of the AVL BOOST model, torque and power measurements from the simu–lation are compared to the relevant experimental results. Figure 2 depicts the experiment torque and power, simulation torque and power, BMEP, FMEP, and IMEP as a function of engine speed at 70% load. This result signifies a good agreement between experiment data and simulation results in the investigating region. The simulated AVL model has a maximum error of less than 10%, indicating that the numerical model is reliable and accurate compared to actual engine performance.

3. RESULTS AND DISCUSSIONS

The effect of IT on power and torque at 85, 70, and 50% of engine loads



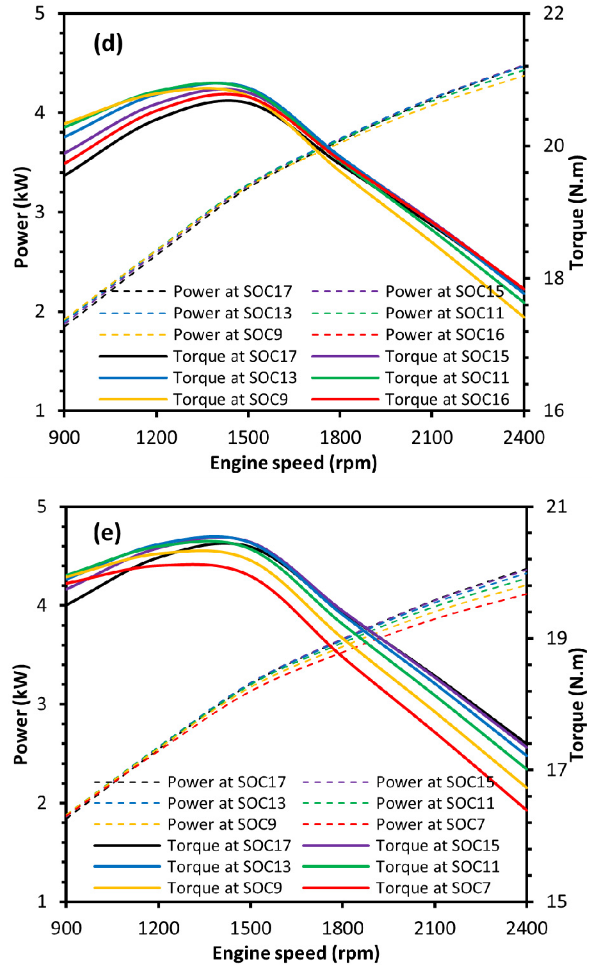
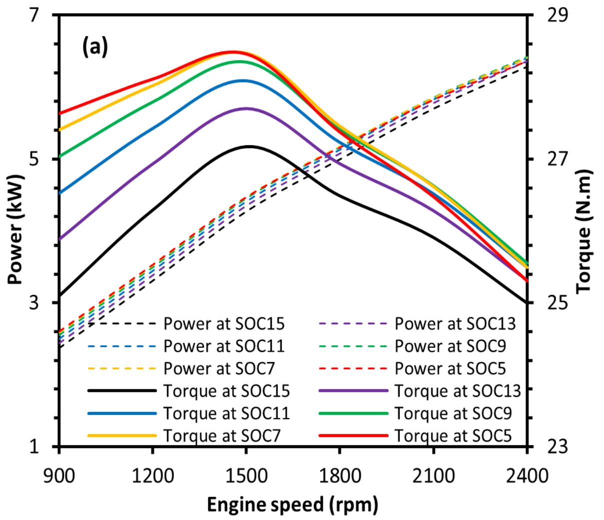


Figure 3. Power and torque as a function of engine speed at 85% load under different combustion durations of (a) 80, (b) 90, (c) 100, (d) 110, and (e) 120°CA.

The impact of ITs on the engine performance cha–racteristics at 85, 70, and 50% engine loads and com–bustion durations of 80, 90, 100, 110, and 120°CA are analyzed and shown in Figures 3, 4, and 5. As observed, the optimal IT, according to the maximum power, is unaffected by the engine load. When ITs are advanced from about 5 to 17°CA bTDC, the power reaches its peak value at IT of 9, 11, 13, 15, and 17°CA bTDC under combustion durations 80, 90, 100, 110, 120°CA, respectively, and they all occurred at 2400 rpm.



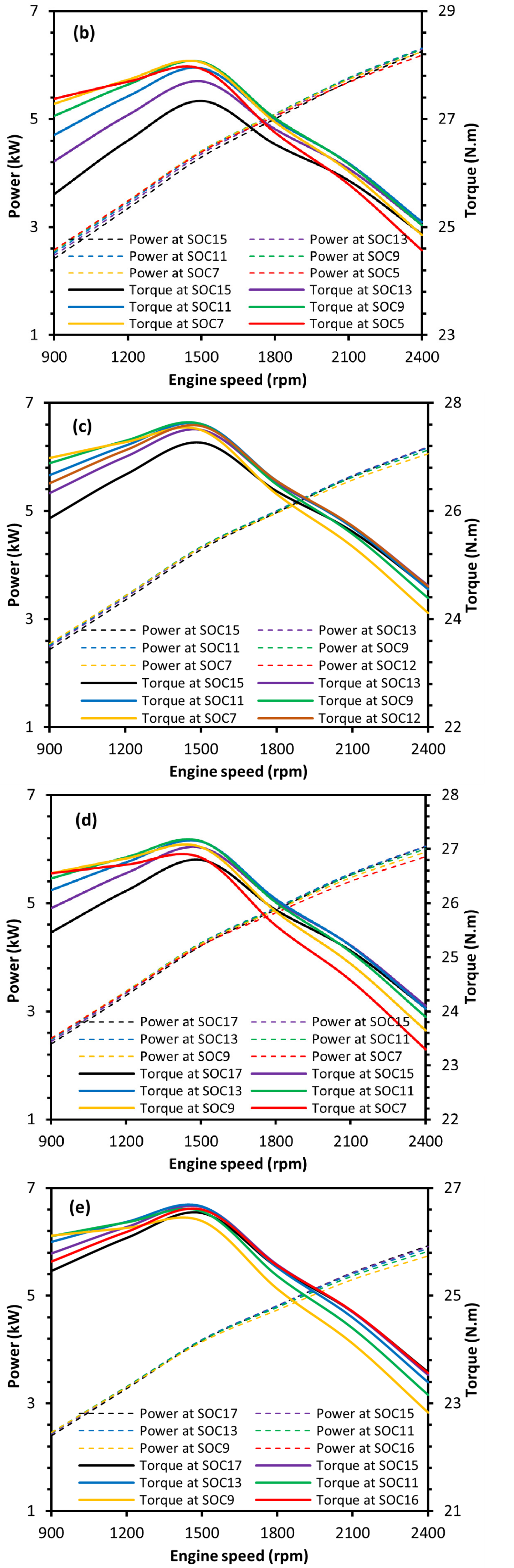
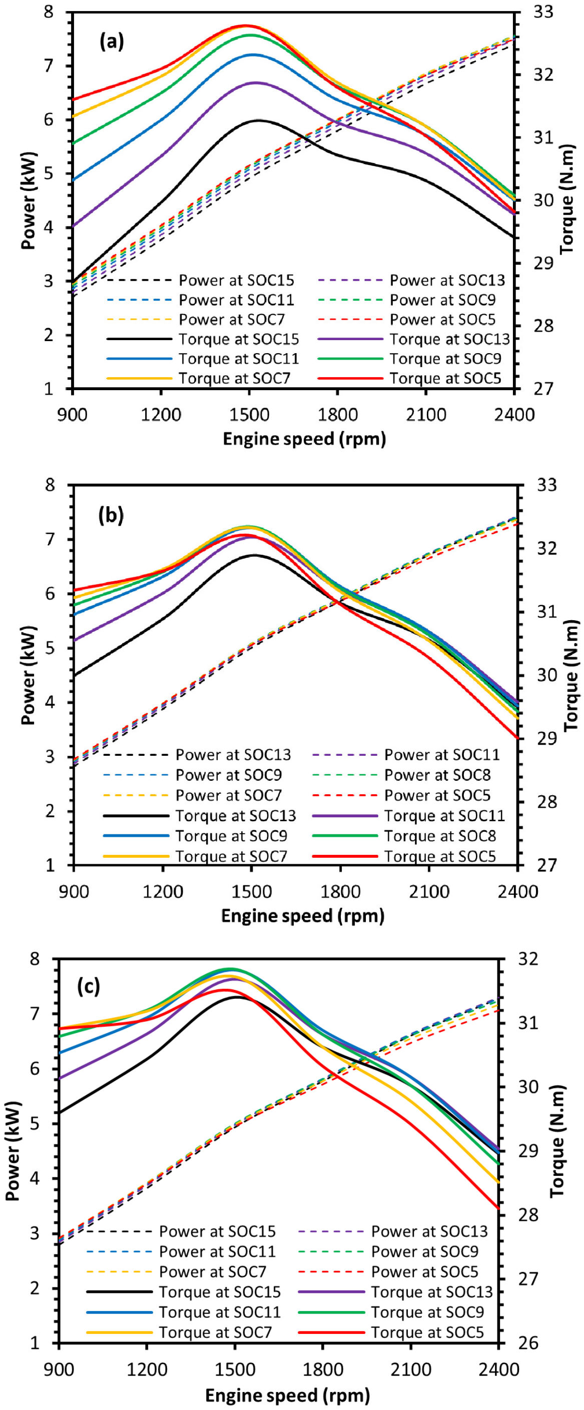


Figure 4. Power and torque as a function of engine speed at 70% load under different combustion durations of (a) 80, (b) 90, (c) 100, (d) 110, and (e) 120°CA

For a combustion duration of 80°CA, the optimal IT is 9°CA bTDC. When the IT is advanced above 9°CA bTDC, the combustion occurs soon, which causes the combustion chamber's pressure and temperature to rise quickly, significantly when the engine piston is advancing approaches the TDC, leading to the combustion energy loss to increase, which reduces power and torque. On the other hand, more delayed ITs (below 9°CA bTDC) may cause combustion to begin later. The majority of the combustion process occurs when the piston moves down far from TDC. In this instance, the rapid rise in combustion chamber volume results in a drop in combustion temperature and pressure. It thus reduces power and torque. For combustion duration 90, 100, 110, and 120°CA, the optimal ITs are 11, 13, 15, and 17°CA bTDC, respectively.



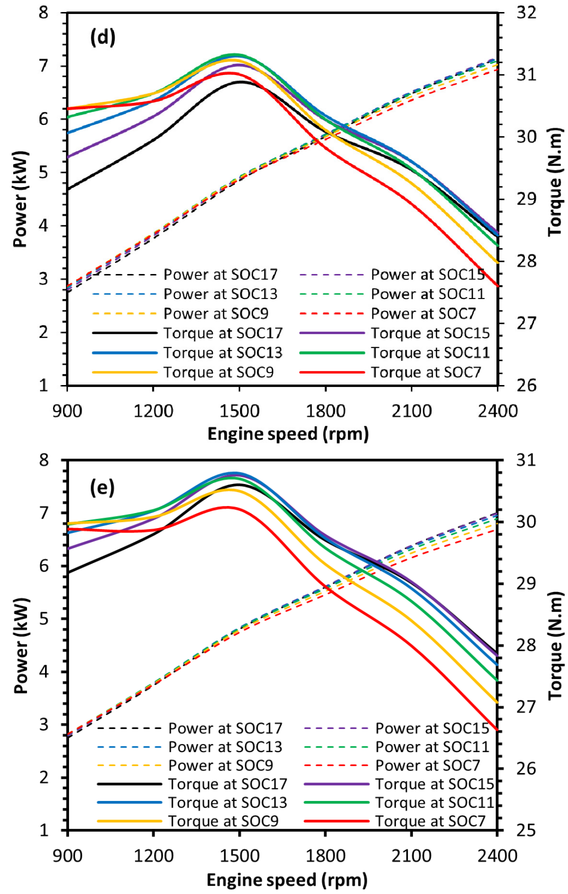


Figure 5. Power and torque as a function of engine speed at 50% load under different combustion durations of (a) 80, (b) 90, (c) 100, (d) 110, and (e) 120°CA

Prediction model of the optimal IT as a function of combustion duration at 85, 70, and 50% engine loads

The optimal ITs based on maximum power at 85, 70, and 50% engine load for each combustion duration 80, 90, 100, 110, and 120°CA have been shown in Figure 6. As observed, the optimal IT according to the maximum power is not affected by the engine load, and all reach the optimal value at 9, 11, 13, 15, and 17°CA bTDC for three different engine loads.

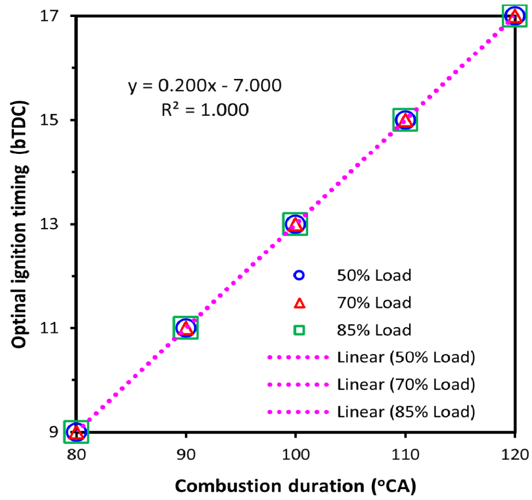


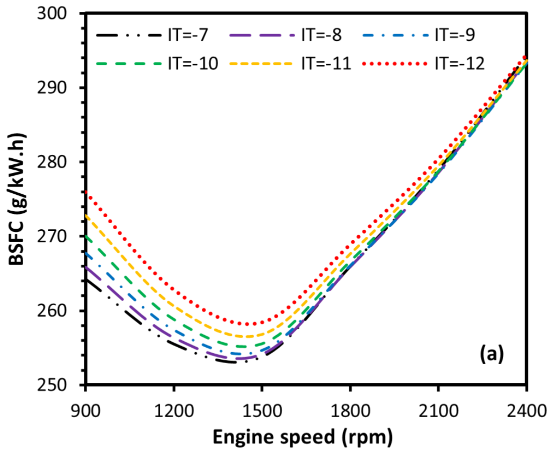
Figure 6. Correlation between optimal ITs and combustion durations for maximum power at 85, 70, and 50% engine loads

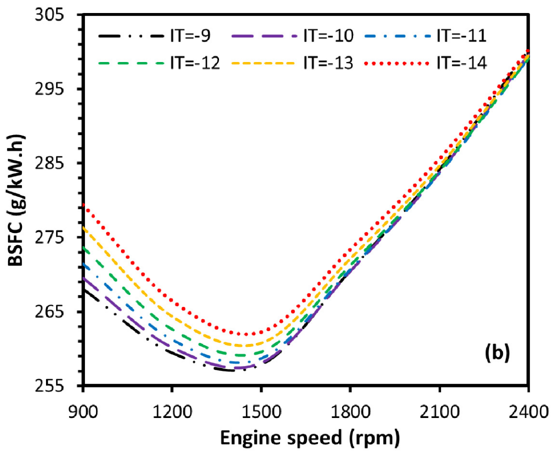
The prediction model of the connection between combustion duration and optimal IT at engine loads of 85%, 70%, and 50% is a linear function expressed as $y = 0.2x - 7$, with variable y being the optimal IT and variable x being the combustion duration.

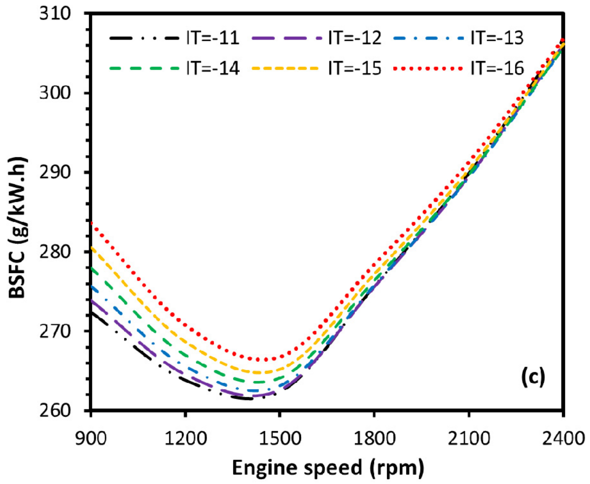
The effect of IT on BSFC at 85, 70, and 50% loads

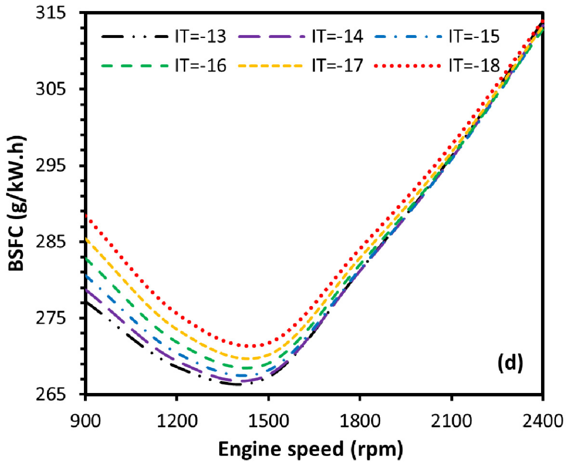
Figures 7, 8, and 9 show the BSFC as a function of engine speed at 85%, 70, and 50% loads under different combustion durations of 80, 90, 100, 110, and 120°CA.

The optimal IT based on minimum BSFC was determined in the engine stability zone from maximum torque to maximum power points. The BSFC area was calculated by integrating from 1500 rpm to 2400 rpm on each IT and selecting the smallest value.









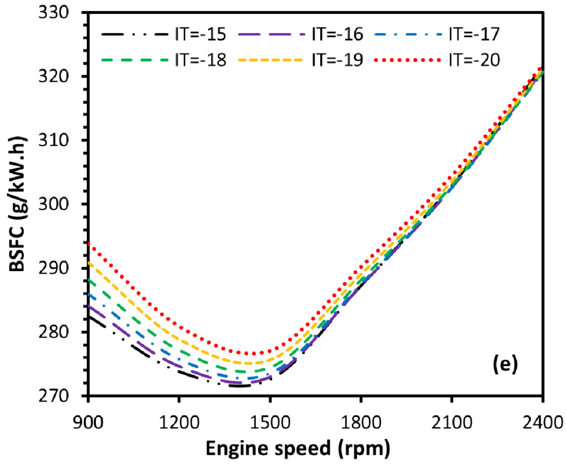
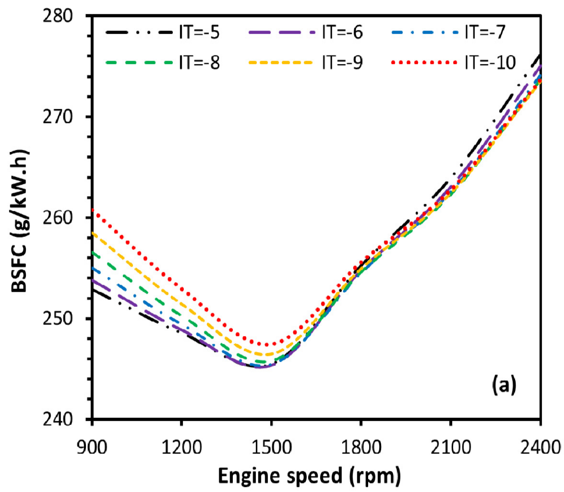


Figure 7. BSFC as a function of engine speed at 85% load under different combustion durations of (a) 80, (b) 90, (c) 100, (d) 110, and (e) 120°CA

Consequently, peak cylinder pressure and engine performance are decreased. As a result, fuel use per unit of output power will rise. As opposed to that, delaying injection timing results in retard combustion, which implies that pressure only increases when the cylinder capacity is increasing quickly, resulting in a lower useful pressure to perform work, leading decrease in BSFC [44]. The optimal ITs of 9, 11, 13, and 15°CA bTDC corresponding to combustion duration 90, 100, 110, and 120°CA give the minimum value of BSFC at 85% engine load, illustrated in Figures 7b-e.



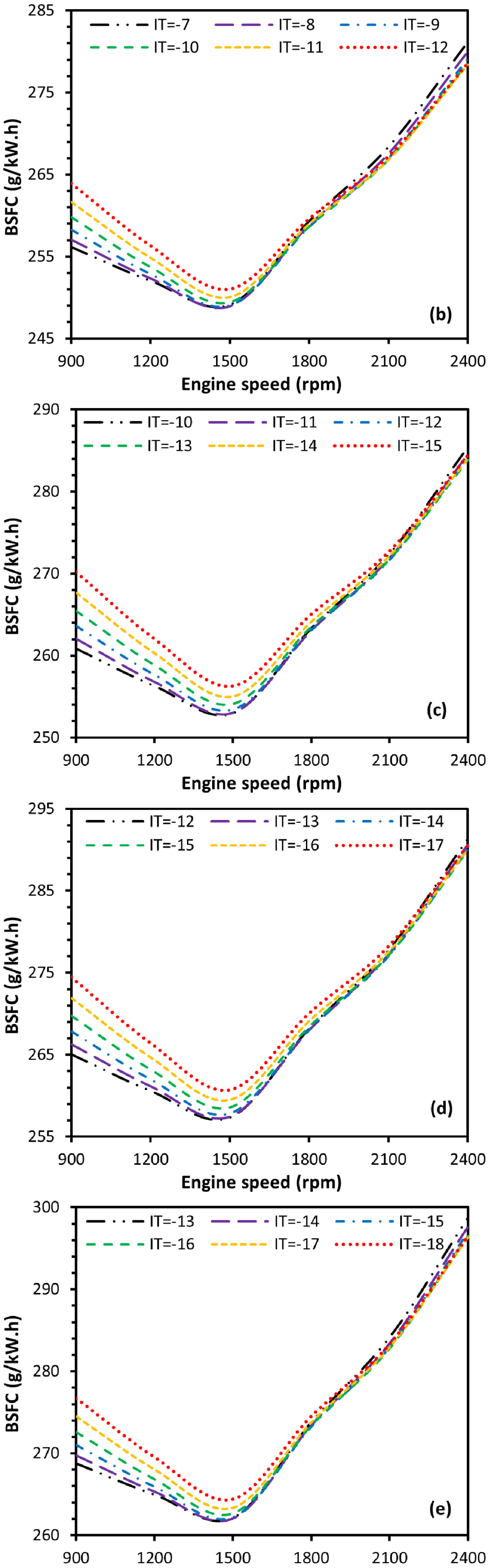
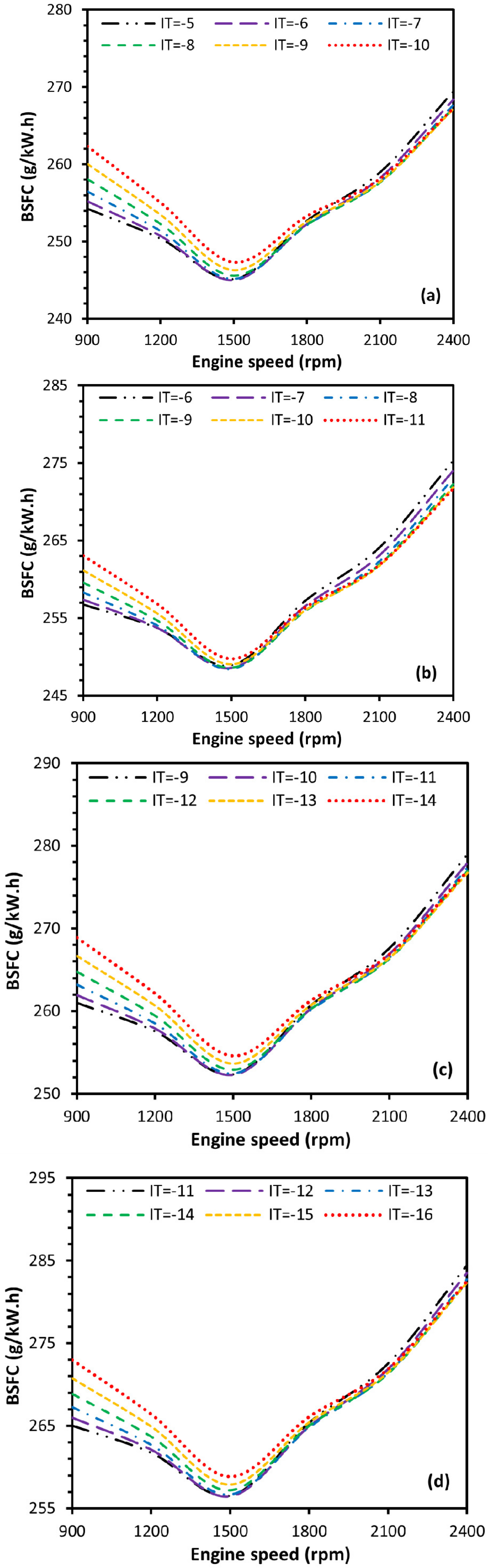


Figure 8. BSFC as a function of engine speed at 70% load under different combustion durations of (a) 80, (b) 90, (c) 100, (d) 110, and (e) 120°CA



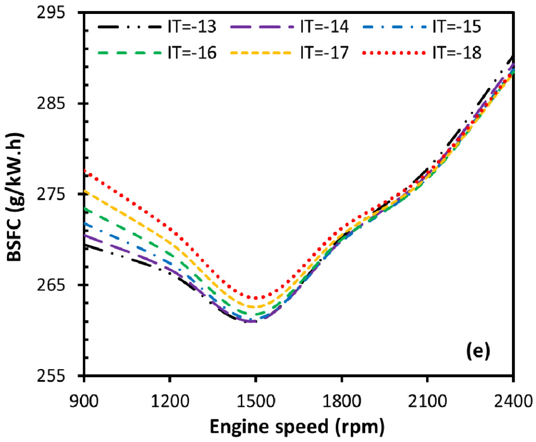


Figure 9. BSFC as a function of engine speed at 50% load under different combustion durations of (a) 80, (b) 90, (c) 100, (d) 110, and (e) 120°CA

Figure 8 revealed that at 70% engine load, the optimal ITs for minimum BSFC under combustion durations of 80, 90, 100, 110, and 120°CA are 8, 10, 12, 14, and 16°CA bTDC, respectively. While at 50% engine load, the optimal ITs of 9, 11, 13, 15, and 17°CA bTDC produce minimum BSFC values at combustion durations of 80, 90, 100, 110, and 120°CA, respectively, shown in Figure 9.

Prediction model of the correlation between combustion duration and optimal IT at 85%, 70%, and 50% engine loads

The optimal ITs based on minimum BSFC at 85, 70, and 50% engine loads for each combustion duration 80, 90, 100, 110, and 120°CA have been shown in Figure 10. The prediction models of the correlation between combustion durations and optimal ITs were established at three different engine loads. The linear equations were used to establish the model at 85%, 70%, and 50% engine loads, respectively are $y = 0.2x - 9$, $y = 0.2x - 8$, $y = 0.2x - 7$, with variable y being the optimal IT and variable x being the combustion duration. In Figure 10, the load has less influence on the optimal ignition ti–ming of minimum BSFC than combustion duration, and the optimal ignition angle is delayed as the load increases.

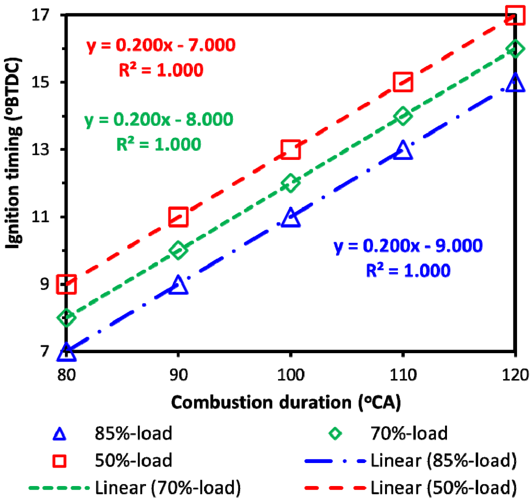


Figure 10. Correlation between optimal ITs and combustion durations for minimum BSFC at 85, 70, and 50% engine loads

The average differences in optimal timing between 50 and 70 % load or 70 and 85% load are around 1°CA. With a 10°CA increase in combustion duration, the optimal ignition timing will be 2° bTDC earlier. The loads increase, making the cylinder temperature decrease because there is more fuel injected into the combustion chamber, so make the ignition timing was retarded.

4. CONCLUSIONS

This work investigated the effects of optimal ignition timing (IT) and combustion duration on the engine brake-specific fuel consumption (BSFC), power, and torque. Then, the prediction models based on the combustion durations and optimal ITs for maximum power and the minimum BSFC were built. The primary conclusions are as follows:

- It significantly influences engine power and torque. The optimal ITs to maximum power have 9, 11, 13, 15, and 17° CA bTDC values, corresponding to combustion durations of 80, 90, 100, 110, and 120° CA at all three investigated engine loads.
- The relationship between optimal IT and combustion duration to produce maximum power is a linear equation. They are not dependent on the engine load.
- IT also has a strong influence on engine BSFC. Based on minimum BSFC, the optimal ITs at 50% engine load are 9, 11, 13, 15, and 17° CA bTDC, corresponding to combustion durations of 80, 90, 100, 110, and 120° CA, respectively, at 70% engine load, the optimal ITs are 8, 10, 12, 14, and 16° CA bTDC. At 85% engine load, the optimal ITs are 7, 9, 11, 13, and 15° CA bTDC, corresponding to combustion durations of 80, 90, 100, 110, and 120° CA bTDC, respectively.
- The functions of optimal IT versus combustion duration to achieve minimum BSFC at different engine loads are also linear. They are pretty influenced by engine load and tend to decrease as the engine load increases.

The optimal IT prediction models established in this study can reduce the time for experimental studies on the engine to determine the optimal fuel injection timing. This work is also helpful for reference when finding a new fuel injection timing while changing the engine operating load mode.

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NOMENCLATURE

A_i	surface area, m^2
C_1	gas velocity coefficient
C_2	model constant
C_m	mean piston velocity, m·s^{-1}
D	cylinder bore, m
h	enthalpy
m_u	unburned mass, kg
m_b	burned mass, kg
N	engine speed, rpm
p_c	pressure in the cylinder, bar
$p_{c,1}$	cylinder pressure at IVC, bar
$p_{c,0}$	motorized engine cylinder pressure, bar
P_{sim}	simulation power
Q_F	the energy of fuel, W
Q_w	heat loss through the wall, W
R^2	correlation between two variables
T_c	cylinder temperature, K
$T_{c,1}$	cylinder temperature when the intake valve closes, K
T_{wi}	the temperature of the wall, K
u	specific internal energy
V	volume in the cylinder, m^3
V_D	displacement in each cylinder, m^3

Greek symbols

α	angle of crankshaft
α_w	temperature-transfer efficiency

Acronyms

EOC	end of combustion
BMEP	brake means effective pressure
BTE	brake thermal efficiency
bTDC	before the top dead-center

BSFC	brake-specific fuel consumption
CA	crank angle
FMEP	friction means effective pressure
IMEP	indicated mean effective pressure
IT	ignition timing
SOC	start of combustion

УТИЦАЈ ВРЕМЕНА ПАЉЕЊА И ТРАЈАЊА САГОРЕВАЊА НА КАРАКТЕРИСТИКЕ ПЕРФОРМАНСИ ДИЗЕЛ МОТОРА КОЈИ КОРИСТИ ВИБЕ 2-ЗОНСКИ МОДЕЛ

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Ефикасна експлоатација енергије је неопходно питање јер помаже у смањењу потрошње горива и загађења животне средине. Проналажење опти–малног времена паљења (ИТ) за дизел моторе за стварање велике снаге и ефикасности заслужује пажњу. Ова студија користи софтвер за симулацију АВЛ БООСТ са Вибе 2-Зоне моделом сагоревања да би истражио утицај ИТ-а и трајања сагоревања на карактеристике мотора као што су снага, обртни момент и потрошња горива специфична за кочнице (БСФЦ) при различитим оптерећењима и брзинама мотора. Затим су израчунати модели предвиђања оптималних ИТ у односу на трајање сагоревања за максималну снагу и минимални БСПФЦ. Резултати показују да ИТ-ови снажно утичу на карактеристике перформанси мотора. Оптималне ИТ вредности да мотор производи максималну снагу при различитим дужинама сагоревања не утичу на оптерећење мотора. Насупрот томе, на њих значајно утиче оптерећење мотора када се узме у обзир БСФЦ који генерише мотор. Корелације оптималног ИТ у односу на трајање сагоревања су линеарне функције. Модели предвиђања се могу користити за предвиђање оптималног времена паљења мотора пошто се експериментално време може смањити када се примени на стварни мотор.