Journal of
Renewable Energy Research and Applications (RERA)

Vol 1, No 1, 2020, 75-83 DOI: 10.22044/RERA.2020.9082.1014

Predicting Optimum Spark Timing of a Biofuel Four Stroke Spark Ignition Combustion Engine

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Receive Date 31 October 2019; Revised 10 December 2019; Accepted Date 20 October 2019
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Abstract
In the recent years, there has been an increasing interest in the enhancement of the efficiency and functionality of engines, particularly the petrol ones. In this work, the four stroke spark ignition internal combustion engine cycle is simulated based on the first law of thermodynamics. The second law analysis is also been conducted to analyse the effects of the ignition timings and combustion duration as well as the engine speed upon the engine efficiency and performance. The availability (exergy) balance equations of the engine cylinder are considered in details. Moreover, the total availability fractions and process irreversibilities are evaluated. By considering the results obtained for brake and indicated mean effective pressure, it is shown that they behave in an opposite way in terms of increasing the engine speed. After perusing the figures, a conclusion is made, revealing that the exergetic efficiency rises by increasing the engine speed, whereas the opposite is true for the brake thermal efficiency. Furthermore, the optimum point at which the total efficiency (both thermal and exergetic) occurs shows that the highest possible level happens at a speed of 2500 rpm.

Keywords: *Internal combustion engine, Spark ignition, Availability analysis, Best spark timing.*

1. Introduction
The thermodynamic analysis of real engines could be employed as an effective mean for the analysis of the engine performance and studying the sensitivity of various operating parameters. While it has long been perceived that the traditional first-law analysis, required for modelling the engine processes, often fails to provide the engineers with a deep insight into the engine's functionality, the second-law analysis is necessary to analyse the engine performance that evaluates the inefficiencies associated with various processes. In terms of the second-law analysis, the key concept is the availability (exergy). In fact, the availability content of a material represents its capability to work efficiently. Unlike energy, availability can be destroyed due to the phenomena including the combustion, friction, mixing, and throttling.

The destruction of availability, often named irreversibility, is the main reason for the defective exploitation of turning the thermal energy contents of fuel into useful mechanical work in a compression or spark ignition engine. Reduction of irreversibility can lead to a more proper engine performance through more efficient exploitation of fuel. In order to decrease the irreversibility, we need to measure it, which requires us to implement the second law analysis to evaluate the destruction of availability.

Among the numerous research works conducted regarding the first and second law application to the internal combustion engines, Abd Alla [1] have studied a preliminary simulation of a four stroke spark ignition engine by assuming a closed system. The heat transfer from the cylinder, friction, and pumping losses have also been taken into account to predict the brake mean effective pressure, brake thermal efficiency, and brake specific fuel consumption.

Farzaneh-Gord *et al*. [2] have developed a computer programme to simulate a four stroke spark ignition engine. This simulation has been done in terms of the intake and exhaust processes as well as the compression and combustion procedures. They studied the effects of different parameters on the engine efficiency and proposed the optimized conditions.

The recent studies have shown that almost 1/3 of the energy of a fossil fuel is destroyed during the combustion process in power generation [3]. This

issue has caused a renewed interest in the availability analyses since the effective management and optimization of thermal systems emerge as a major modern technical problem.
Rakopolus *et al*. [4] have carried out a method for the second law analysis of the internal combustion engine operation and employed it to analyse the operation with alternative fuels.
Ismet Sezer and Atilla Bilgin [11] have observed the exergy analysis of a spark ignition engine during the compression, combustion, and expansion processes of the engine cycle. For this purpose, a thermodynamic-based engine cycle model was developed regardless of the geometric features of the fluid motion. In this work, the effects of changing some design and operating parameters such as the compression ratio, fuel-air equivalence ratio, and spark timing upon the variation and destruction of exergy have been investigated. The results obtained demonstrated that the design and operation conditions played a considerable role in the variation of exergy and irreversibility during the investigated parts of the cycle.
Concerning the internal combustion engines, investigation of the evaluation of the global engine operation via the second-law techniques was followed by the availability and irreversibility calculations during the engine cycle fuelled with compressed natural gas [5]. The overall energy and availability balance in the course of an engine cycle were studied analytically in Ref. [6]. The second-law arguments were used to evaluate the optimum speed of an internal combustion engine, which is practical to investigate the effect of engine speed on efficiency [7].
To the authors' knowledge, there is still a lack of detailed information about the correlation between the spark ignition combustion engine efficiency and the second-law principle. Consequently, the objectives of the present work can be summarized as follow:

- Provision of the details regarding the equations employed for the second law application for internal combustion engines operation, i.e. state properties, basic first law equations, fuel chemical availability, availability equations for the engine cylinder, entropy balance equations, second law efficiency, and basic relations for application of the second law analysis during transient operation.
- Simulation of engine functionality based on the first law of thermodynamics, in which the combustion chamber has been considered an open system. The simulation has been carried out for a full cycle (720 degrees of crank angle) so the intake and exhaust stroke has been modelled.
- Analysis of energy and availability balances for various ignition timings as well as the effect of the ignition timing parameter on the second law performance of internal combustion engines.

2. In-cylinder Processes

For simulating spark ignition engines, ideal Otto cycle, single zone or usually two-zone modelling techniques have been implemented. Regarding the two-zone modelling, one of the zones is the burned one, which contains the equilibrium products of combustion, and the other is the unburned gas zone composed of a homogeneous mixture of air, fuel, and residual gas. In a single zone model, the working fluid in the engine is assumed to be a thermodynamic system that undergoes energy and mass exchange with the surroundings, and the energy released during the combustion process is obtained by applying the first law of thermodynamics to the system.
In this work, the cylinder processes were simulated using a single zone, and the cylinder thermal properties were measured via zero dimensional models.
The equation of state for an ideal gas is:

$$PV = mRT \tag{1}$$

Taking the logarithm of both sides and differentiating with respect to crank angle gives:

$$\frac{1}{R}(PdV + VdP) = mdT + Tdm \tag{2}$$

It can be assumed that the internal energy is only a function of the temperature, and the combustion process is modelled as a single zone. Furthermore, the first law of thermodynamics can be written in a differential form for an open thermodynamic system. If the changes in potential energy are neglected, then the equation is derived as:

$$m_C C_V(dT) + C_V T_C(dm) = \partial Q - PdV + (dm)h_i + m_i C_P(dT) - (dm)h_e + m_e C_P(dT) \tag{3}$$

By combining equation (2) with equation (4) and arranging it we obtain:

$$\left(1 + \frac{C_V}{R}\right)PdV = \partial Q - \frac{C_V}{R}VdP + (dm)h_i + m_i C_P(dT) - (dm)h_e + m_e C_P(dT) \tag{4}$$

If the terms $C_P - C_V = R$ and $C_P = C_V K$ are substituted into Eq. (4), it can be expressed as below:

$$\frac{dP}{d\theta} = \frac{K-1}{V}\frac{\partial Q}{\partial \theta} - \frac{K}{V}P\frac{dV}{d\theta} + \left(\frac{dm_i}{d\theta}\right)RKT_i + m_i RK\left(\frac{dT_i}{d\theta}\right) - \left(\frac{dm_e}{d\theta}\right)RKT_e - m_e RK(\frac{dT_e}{d\theta}) \quad (5)$$

This equation has to be solved iteratively. For this purpose, its terms should be calculated.

According to figure 1, the volume of the engine cylinder and the rate of its change can be described, respectively, as:

$$V = \frac{1}{2}s(\frac{\pi B^2}{4})\left[\left(\frac{2}{Cr-1}\right) + 1 - \cos\theta + \frac{a}{4r}(1-\cos 2\theta)\right] \quad (6)$$

$$\frac{dV}{d\theta} = \frac{1}{2}s(\frac{\pi B^2}{4})\left[\sin\theta + \frac{a}{2r}\sin 2\theta\right] \quad (7)$$

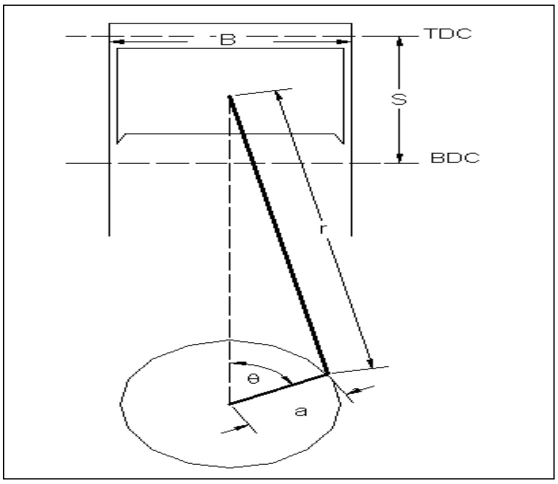

Figure 1. Engine geometric properties.

in which B is the cylinder internal diameter (Bore), and $S, Cr, a,$ and r are the engine stroke, engine compression ratio, crank radius of the engine, and length of the connecting rod, respectively [13].

Equation (5) is valid regardless of whether dQ is defined as heat addition caused by combustion or heat lost by the gases in the cylinder due to convection. To consider both possibilities, it can be written that:

$$dQ = Q_{in}dx - dQ_i \quad (8)$$

in which Q_{in} is the total value of the heat released from the combustion of the quantity m_f of fuel, which can be defined as follows:

$$Q_{in} = \eta_c m_f C_{fl} \quad (9)$$

where C_{fl} is the lower calorific value of the fuel and η_c is the combustion efficiency, which is considered to be constant with the value of 0.9 [8].

The heat addition for spark ignition engines may be a prescribed function of crank angle, and the function that is generally used for quantifying mass fraction burned during combustion process is weib function, which is defined as:

$$x_b(\theta) = 1 - e^{\left\{-y\left[\frac{\theta-\theta_0}{\Delta\theta}\right]^{n+1}\right\}} \quad (10)$$

where θ is the crank angle, and $\theta_0, \Delta\theta, y,$ and n are the start of combustion, total combustion duration ($x_b = 0\ to\ x_b = 1$), and adjustable parameters, respectively. It can be noted that the alteration of y and n changes the profile of the mass fraction burned. The actual mass fraction burned curve can match accurately with $y = 5\ and\ n = 2$ [1].

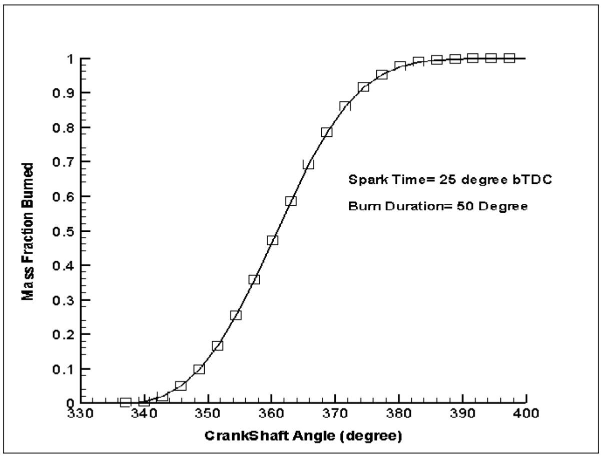

Figure 2. Mass fraction burned curve for a spark ignition engine.

The heat lost resulting from convection in the combustion chamber can be obtained as:

$$\partial Q_l = \frac{hA}{\omega}(T_g - T_w)\frac{\pi}{180} \quad (11)$$

wherein h stands for the heat transfer coefficient, A denotes the surface area in contact with the gases, T_g is the in cylinder gas temperature, and T_w shows the wall temperature.

The convective heat transfer coefficient can be derived using the woschni correlation.

$$h = 3.26B^{-0.2}P^{0.8}T^{-0.55}\upsilon^{0.8} \quad (12)$$

where P represents the instantaneous cylinder pressure, and T and v are the instantaneous gas temperature and the characteristic velocity of gases, respectively.
According to the woschni correlation [9], the characteristic velocity of in cylinder gas for a four stroke engine without swirl is:

$$v = c_1 S_P + c_2 \frac{V_d T_r}{P_r V_r}(P - P_m) \quad (13)$$

in which the constants are:

$$-180 \le \theta \le \theta_0,\ c_1 = 2.28,\ c_2 = 0 \quad (14)$$

$$\theta_0 \le \theta \le +180,\ c_1 = 2.28,\ c_2 = 3.24 \times 10^{-3} \quad (15)$$

and P_m is the motoring pressure that can be given as:

$$P_m = \frac{\left[(\frac{CrV_d}{Cr-1})^{\gamma} P_a\right]}{V^{\gamma}} \quad (16)$$

The mass flow rate through a poppet valve is usually described by the equation for the compressible flow through a flow restriction. This equation is derived from a 1D isentropic flow analysis, and the real gas flow effects are included by means of an experimentally determined discharge coefficient, C_D. The mass flow rate is a function of P_0, T_0, P_T, and A_R, denoting the upstream stagnation pressure, stagnation temperature, static pressure just at the downstream of the flow, and a reference area characteristic of the valve design, respectively.

$$\dot{m} = \frac{C_D A_R P_0}{(RT_0)^{0.5}} (\frac{P_T}{P_0})^{\frac{1}{\gamma}} \left\{\frac{2\gamma}{\gamma - 1}\left[1 - (\frac{P_T}{P_0})^{\frac{\gamma-1}{\gamma}}\right]\right\}^{0.5} \quad (17)$$

When the flow is choked, $\frac{P_T}{P_0} \le \left[\frac{2}{\gamma+1}\right]^{\frac{\gamma}{\gamma-1}}$, the appropriate equation is:

$$\dot{m} = \frac{C_D A_R P_0}{(RT_0)^{0.5}} (\gamma)^{\frac{1}{2}} \left\{\frac{2}{\gamma + 1}\right\}^{\frac{\gamma+1}{2(\gamma-1)}} \quad (18)$$

For the flow into the cylinder through an intake valve, P_0 is the intake system pressure (intake manifold) and P_T denotes the cylinder pressure. Furthermore, for the flow out of the cylinder through an exhaust valve, P_0, P_T, and A_R demonstrate the cylinder pressures of the exhaust system (exhaust manifold) pressure, respectively. The value of C_D for both the intake and exhaust valves was taken 0.7, as proposed by [8], and the reference area A_R was assumed the valve head area.

3. Application of Availability Balance

In the following sub-sections, the equations dealing with the availability balance concerning the engine cylinder and its sub-systems will be derived, which are indispensable for evaluating the various process irreversibilities.
For an open system that undergoes a mass exchange with the surrounding environment, the following equation holds for the total availability on a time basis

$$\frac{dA_{cv}}{dt} = \int_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_J - \left(\dot{W}_{cv} - P_0\frac{dV_{cv}}{dt}\right) + \sum_{in} \dot{m}_{in} b_{in} - \sum_{out} \dot{m}_{out} b_{out} - \dot{I} \quad (19)$$

In the above equation, $\frac{dA_{cv}}{dt}$ stands for the time rate of change in the availability of the control volume content, $\int_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_J$ shows the availability term of heat transfer, T_j is the temperature at the boundary of the system, and $\dot{Q}_J$ represents the time rate of heat transfer at the boundary of the control volume [9].
The second, third, and fourth terms on the right-hand side express the availability terms associated with the (mechanical or electrical) work transfer, and inflow and outflow of masses, respectively. It should be noted that the terms b_{in} and b_{out} in equation (19) refer to the flow or stream availability (or exergy) of the incoming and the outgoing cylinder mass flow rates, which can be defined as:

$$b = h - h_0 - T_0(s - s_0) \quad (20)$$

when the kinetic and potential energy contributions are neglected. The last term of Eq. (19), $\dot{I}$, denotes the rate of irreversibility production inside the control volume, caused by various factors including combustion, throttling, mixing, and heat transfer under a finite temperature difference to a cooler medium. It can be shown as $\dot{I} = T_0 \dot{S}_{irr}$ based on an entropy balance, in which S_{irr} denotes the rate of entropy creation due to irreversibilities.

3.1. Exergetic Efficiencies

We employ efficiency to analyse different engine size applications or evaluate various improvements effects, either from the first or the second law perspective. The second law (or exergy or availability) efficiency is also named as the effectiveness or exergetic efficiency, which measures how efficiently the input (fuel) is converted into the product. This term is usually written as:

$$\varepsilon = \frac{availability_{out}}{availability_{in}} = 1 - \frac{loss + destruction}{input} \tag{21}$$

For the cylinder, the following second law efficiency is often defined (four-stroke engine):

$$\varepsilon = \frac{W_{br}}{M_{fi} a_f} \tag{22}$$

where W_{br} and a_f represent the brake work production and the inlet flow availabilities, respectively. Having these relations in hand, co-efficiency ε can be compared with a first law one, as follows:

$$\eta_t = \frac{W_{br}}{Q_{in}} \tag{23}$$

4. Results and Discussion

Throughout the present work, a prototype single cylinder, four stroke engine, was applied to investigate all of the thermodynamic features to be considered. In this work, the compression ratio was 8, and our system had a bore of 100 mm, a stroke of 111.1 mm, and a swept volume of 872.5 cm^3 as well as an operating speed of 3000 rpm. The fuel was octane with a declared calorific value of 44.3 MJ/kg.

Figure 3 shows the distribution of different availability rate terms namely heat transfer, work, irreversibilities, and control volume for various crank angle ranging from 0 to 720 degree. It should be said that the calculations were done based on octane as fuel and the following consideration regarding the combustion parameters:

equivalence ratio $\phi = 1$, spark timing $\theta_s = 25$ degrees before TDC, and combustion duration $\theta_b = 45$.

During the compression stroke, the availability varies according to the work; A_W is negative. However, the thermo-mechanical availability, A_{th}, increases. It is noteworthy that the combustion process starts at 25 degree before TDC, where there is a corresponding rise in A_{th}, as the temperature and pressure increase by burning the fuel. Moreover, although the irreversibility curve traces the availability destruction due to the combustion, it does not change with crank angle after 380° until the exhaust valve opens. At the degree of 380°, the combustion process finishes.

The optimum operating conditions for engine are where appropriate torque (the most proper indicated mean effective pressure, IMEP) is achieved for the same geometric compression ratio, engine load as well as the engine speed. IMEP can be obtained as:

$$IMEP = \frac{\oint PdV}{V_d} \tag{24}$$

in which V_d and P show the displacement volume and pressure in the cylinder, respectively.

The friction mean effective pressure is expressed using the formula governed by Branes-Moss [12]:

$$FMEP(bar) = 0.97 + 0.15\left(\frac{rpm}{1000}\right) + 0.05\left(\frac{rpm}{1000}\right)^2 \tag{25}$$

It is worthy to note that the brake mean effective pressure (BMEP) can be evaluated by subtracting the friction mean effective pressure (FMEP) from the indicated mean effective pressure.

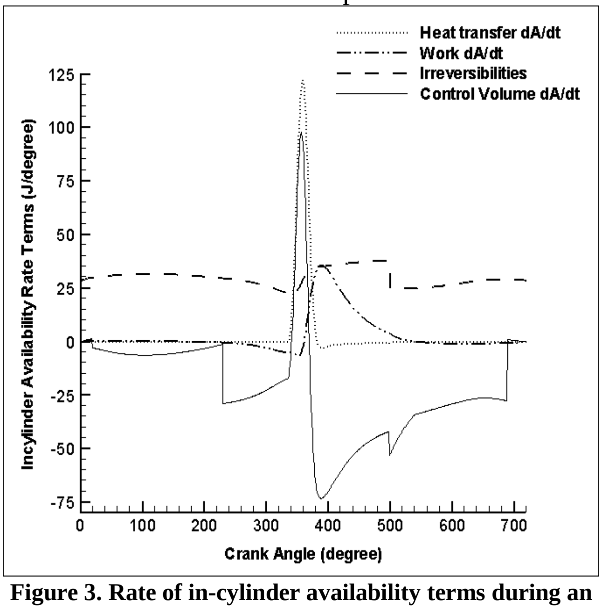

Figure 3. Rate of in-cylinder availability terms during an engine cycle.

After the analysis of various combustion parameters, we can find the optimum conditions for engine operation. For this purpose, the equivalence ratio is considered 1.1 and the engine speed varies from 1500 rpm to 4000 rpm. In addition, the variation in spark timing is between 40° before TDC and 5° before TDC.

Figure 4 illustrates the mean effective pressures of engine operation with respect to spark time for different speeds. It can be realized that increasing engine speed leads indicated mean effective pressure to increase; however, the brake mean effective pressure declines with engine speed increment, which can be seen in figure 5. The reason for this behaviour is that the friction lost is considerably high at high speeds. From figure 5, it is evident that the brake mean effective pressure initially increases by a delay in the spark time, and subsequently decreases. Indeed, sparking at 20 degrees before TDC leads to a higher mean effective pressure.

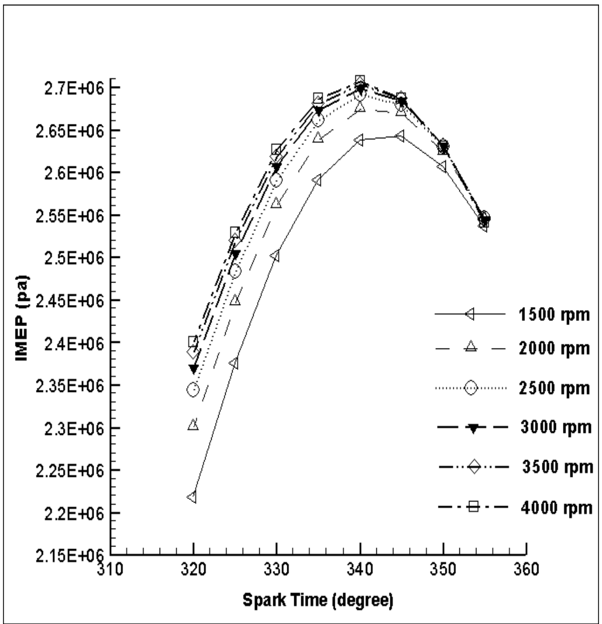

Figure 4. Variation in IMEP versus spark timing for various engine speeds.

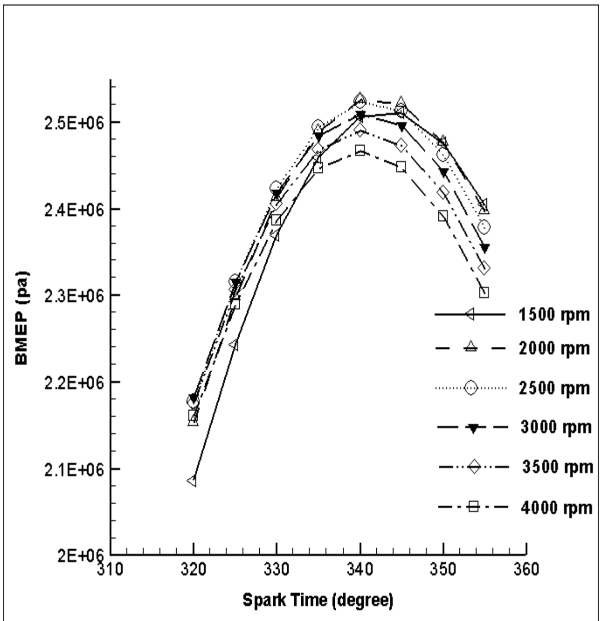

Figure 5. Variation in BMEP versus spark timing for various engine speeds.

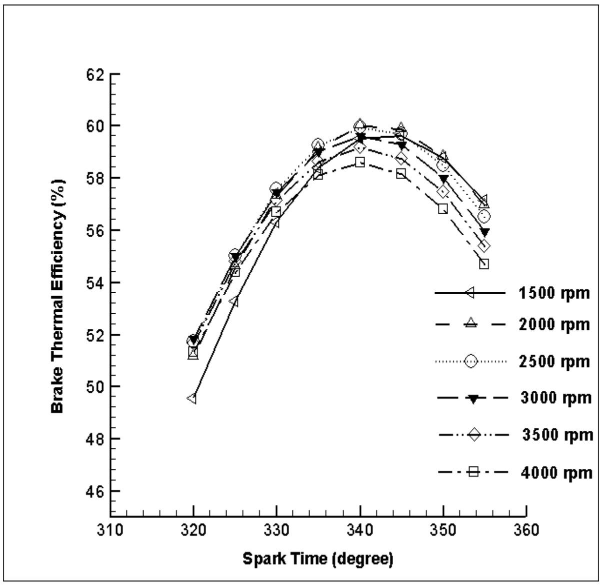

Figure 6. Variation in brake thermal efficiency versus spark timing for various engine speeds.

Figures 6 and 7 depict the effects of various engine speeds on the brake thermal efficiency and exergetic efficiency for different spark timings. It can be observed in figure 6 that the brake thermal efficiency has the same patterns as BMEP, shown in figure 5. It can be concluded that the highest amount of thermal efficiency occurs at a spark time of 340° for an engine speed between 2000 and 2500 rpm.

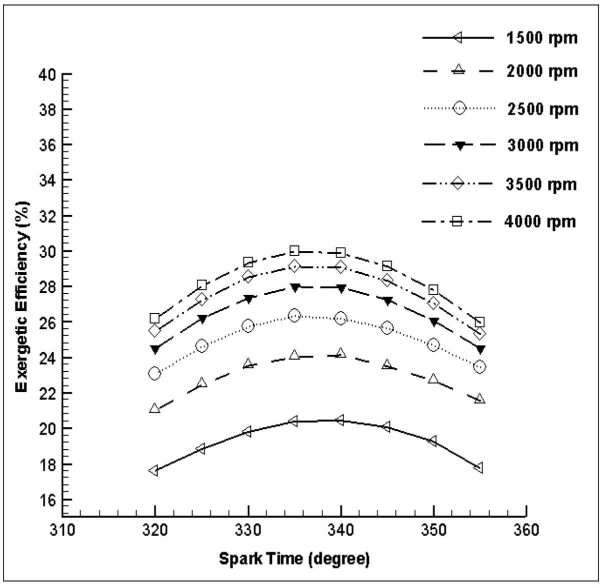

Figure 7. Variation in exergetic efficiency versus spark timing for various engine speeds.

The most notable feature of figure 7 is that while increasing the engine speed results in a rise in the exergetic efficiency, the optimum point for spark is still 20 degrees prior to TDC. Considering the spark time at 20 degree before TDC as well as simulating the engine efficiency through the first and second laws of thermodynamics for speeds ranging from 2000 to 2500 rpm, the optimum speed can be achieved.

Figure 8 demonstrates that the exergetic efficiency experiences an increase while the engine speed rises. However, owing to the friction lost, the brake thermal efficiency declinces in high speeds. Furthermore, figure 8 indicates that an engine speed of 2200 rpm results in the most proper condition for the brake thermal efficiency. It can be noted that at the reviewed analysis, the combustion duration is assumed to be constant with a value of 45 degrees, and the engine speed has no effect on the combustion velocity, whereas in reality, increasing the speed of engine pistons leads to the combustion speed augmentation of gases in cylinder.

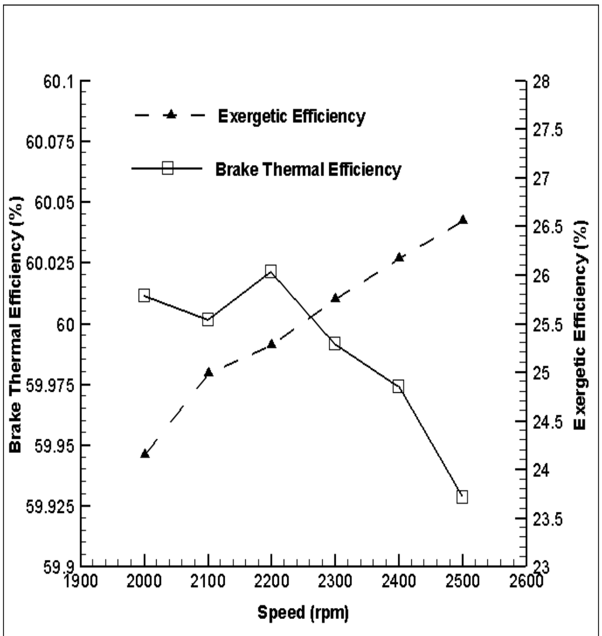

Figure 8. Exergetic and brake thermal efficiency for various engine speeds.

Figures 9 and 10 reveal the effect of raising combustion duration from 25 to 55 degrees upon the thermal and exergetic efficiencies. The inspection of these two types of efficiencies shows that when the combustion process happens for a long span, the engine is required to spark earlier. Therefore, the fuel has a sufficient time to burn. Comparing these figures, we can discover that the brake thermal and exergetic efficiency in a speed of 2500 rpm is higher than any other speeds, resulting in an optimum functionality of the engine at this speed. It should be noted that for each combustion duration, there is an optimum spark time in which fuel burns completely and effectively, bringing about a higher efficiency in the engine performance.

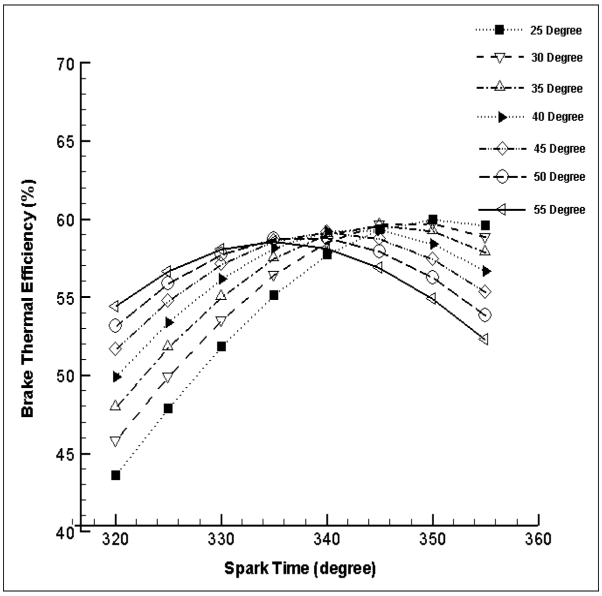

Figure 9. Variation in brake thermal efficiency versus spark timing for various combustion durations when engine speed is 2200 rpm.

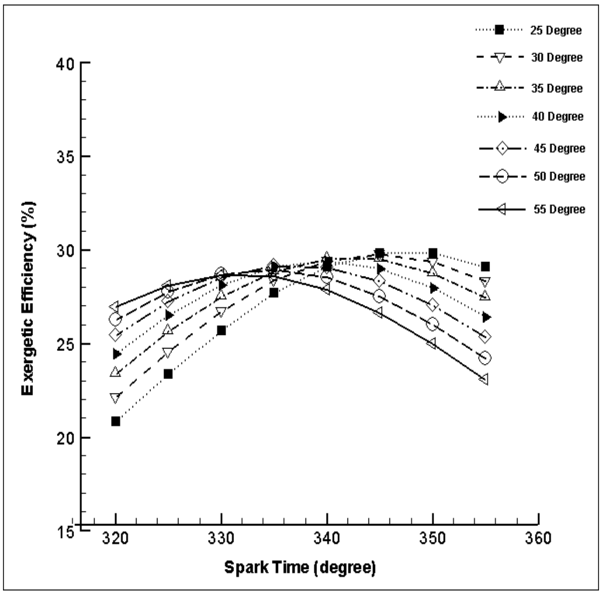

Figure 10. Variation in exergetic efficiency versus spark timing for various combustion durations when engine speed is 2200 rpm.

4. Conclusion

Simulation of the full cycle of four stroke spark ignition internal combustion engines was conducted based on the first law of thermodynamics. The second law analysis was also taken into account to peruse the effects of ignition timings and combustion duration as well as engine speed upon engine performance and efficiency. The total availability evolution during the cyclic process was studied as the outcome of the availability exchange and destruction through processes of heat transfer, work, and combustion. The trends and results associated with the availability loss, heat transfer, and destruction of the current work were in a close agreement with the outcomes and conclusions reported by other studies of similar nature for SI engines. The main results are summarized as follow:

i. Ignition timing and duration of combustion have a significant influence on the engine efficiency.
ii. An increase in the engine speed leads to a rise in IMEP, while the opposite behaviour is observed for the brake mean effective pressure; this is mainly due to a considerable amount of friction lost at higher speeds.
iii. The trends for BMEP in various engine speeds are approximately the same as the brake thermal efficiency.
iv. Regarding the engine speeds ranging from 2000 to 2500 rpm, the highest level of thermal efficiency can be achieved at the spark time of 340°.
v. Exergetic efficiency increases by augmenting the value of engine speed, whereas the brake thermal efficiency experiences an opposite trend, which can be attributed to the significant value of friction lost.
vi. The most appropriate condition concerning brake thermal efficiency occurs when the engine runs at a speed of 2200 rpm.
vii. The overall efficiency (thermal and exergetic) can be maximized at a speed of 2500 rpm.
viii. The longer combustion duration, the higher levels of total efficiency, which is because of a sufficient amount of time for fuel to burn completely.

Nomenclature

a	Crank radius (mm)
A	Surface area of the cylinder (m^2)
A_R	Characteristic area of the valve (m^2)
B	Bore diameter (mm)
BDC	Bottom dead centre
$bmep$	Break mean effective pressure (Kpa)
C_P , C_v	Constant pressure & volume specific heats $(\frac{kj}{kg.K})$
Cr	Engine compression ratio
C_D	Discharge coefficient
C_{fl}	Lower calorific value of the fuel $(\frac{Mj}{kg})$
h	Heat transfer coefficient $(\frac{KW}{m^2.K})$
$imep$	Indicated mean effective pressure (kPa)
h_i& h_e	Inlet and exhaust gas enthalpies $(\frac{kj}{kg})$
$\dot{m}$	Mass flow rate $(\frac{kg}{s})$
m	Mass $(\frac{kg}{s})$
n	Weib function adjustable parameter
P	Pressure
Q_{in}	Heat released from the combustion (kj)
Q_l	Heat lost due to convection (kj)
R	Gas constant $(\frac{kj}{kg.K})$
r	Length of the connecting rod (mm)
S	Engine stroke (mm)
T	Temperature
TDC	Top dead centre
V	In-cylinder volume (m^3)
V_d	Cylinder displacement volume (m^3)
V_c	Cylinder clearance volume
v	Characteristic velocity of gases $(\frac{m}{s})$
W	Work $(kj\ or\ joule)$
x	Mass fraction
y	Weib function adjustable parameter
θ	Crank angle
ω	Engine rotational speed $(\frac{rad}{s})$
θ_0	Start of combustion crank angle (rad)
$\Delta\theta$	Total combustion duration angle (rad)
η_c	Combustion efficiency
ϕ	Equivalence ratio

Subscripts

a	Air
b	Burned gas
C	Cylinder
e	Exit port
f	Fuel
i	Inlet port
W	Wall surface

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