

Indian journal of Engineering

To Cite:

Al- Hasan M. Influence of ignition timing on heat transfer rate between the burned gases and the gas-side cylinder wall of a spark ignition engine. *Indian Journal of Engineering*, 2026, 23, e13ije1731
doi: <https://doi.org/10.54905/disssi.v23i60.e13ije1731>

Author Affiliation:

Department of Mechanical Engineering, Faculty of Engineering Technology, Al-Balqa' Applied University, Al-Salt, Jordan

* Corresponding author:

Mohammad Ibrahim Al-Hasan,
Department of Mechanical Engineering, Faculty of Engineering Technology, Al-Balqa' Applied University, Al-Salt, Jordan, Email: moh_05_alhassan@yahoo.com; m.alhasan@bau.edu.jo

Peer-Review History

Received: 21 November 2025

Reviewed & Revised: 18/December/2025 to 16/August/2026

Accepted: 30 August 2026

Published: 30 September 2026

Peer-Review Model

External peer-review was done through double-blind method.

Indian Journal of Engineering
pISSN 2319-7757; eISSN 2319-7765



© The Author(s) 2026. Open Access. This article is licensed under a [Creative Commons Attribution License 4.0 \(CC BY 4.0\)](https://creativecommons.org/licenses/by/4.0/), which permits use, sharing, adaptation, distribution and reproduction in any medium or format, as long as you give appropriate credit to the original author(s) and the source, provide a link to the Creative Commons license, and indicate if changes were made. To view a copy of this license, visit <http://creativecommons.org/licenses/by/4.0/>.

Influence of ignition timing on heat transfer rate between the burned gases and the gas-side cylinder wall of a spark ignition engine

Mohammad Al-Hasan*

ABSTRACT

Heat transfer is one of the most important issues that must be considered when studying internal combustion engines (ICEs), due to its influence on engine operating parameters such as temperature and pressure inside the cylinder, engine brake thermal efficiency, fuel consumption, and emissions. The objective of this study was to investigate the effect of engine Ignition Timing (IT) on the heat transfer rate between the gases burned in the cylinders, and the cylinder walls of a spark ignition engine (SI). For this purpose, the engines design geometric parameters were utilized to calculate the instantaneous cylinder volume (V_g), in-cylinder pressure (P_g) and temperature (T_g), gas-side cylinder wall temperature, and the convective heat transfer coefficient (h_g) for different ignition timing of 30° , 25° , 20° , 15° , and 10° before Top Dead Center (bTDC). The results show that, when the combustion process starts early during the compression stroke i.e. at an ignition timing of 30° bTDC, most of the air-fuel mixture burns before the piston reaches the TDC. Consequently the P_g , T_g , h_g and the heat transfer rate, $Q(\theta)$, reaches their maximum value at 30° bTDC. However, as the ignition timing decreases up to 15° and 10° bTDC the values of the stated above parameters increase and reaches their maximum values after Top Dead Center (aTDC). Furthermore, the obtained results were compared with finding from other researcher's, indicating that the present results are largely consistent with previously published data. Additionally, cylinder wall temperature distribution results obtained via FlexPDE software conformed to those from the analytical method, as demonstrated by the coefficient of determination, R^2 , of 0.934, 0.926, and 0.894 for ignition timing of 10° , 20° , and 30° bTDC, respectively. Thus, taking into account the aforementioned results, the best ignition timing for the engine used in this study is found to be 10° CA bTDC based on the obtained maximum values of 7.5 (MPa), 3188 (K), 576 ($W/m^2 \cdot K$), 625 (K) and 8316 (W), for P_g , T_g , h_g , T_w and $Q(\theta)$, respectively.

Keywords: heat transfer, ignition timing, cylinder pressure and temperature and FlexPDE

1. INTRODUCTION

The study of heat transfer effects on ICE performance plays an important role in engine design and development. This phenomenon is difficult to module and analysis, because of its complexity. It involves different systems such as: the intake and exhaust systems, liquid cooling system (water jacket around the cylinder walls and head) and lubrication system. Furthermore, the different of heat transfer mechanisms (heat transfer by radiation and convection from the burned gases to the cylinder walls and head, conduction through the cylinder walls and by convection from the wall cylinder to cooling water). In addition, heat transfer is non-uniform and unsteady due to changes in the combustion process inside the cylinder as a result of the variations in the charge composition and engine operating conditions.

The heat loss from exhaust gases transferred to the cooling water in ICEs represents about 25 to 30 % of the total input heat released from the combustion of the air-fuel mixture during the combustion process (Heywood, 1988). It depends mainly on the engine type and operating conditions. Decreasing these losses will be led to enhancing engine operating parameters, such as fuel economy, effective power and effective thermal efficiency, as well as reducing engine exhaust emissions.

According to the above – mentioned context, many experimental and theoretical investigations have been performed in the recent years. Several authors have investigated the influence of the engine cylinder assembly's temperature on exhaust gas emissions and fuel consumptions: (Torregrosa et al., 2006; Ladommatos et al., 2005; Cortona, 2000; Ogawa et al., 1999; Reksowardojo et al., 1996; Krause & Spies, 1996; Asou et al., 1992).

Shojaeefard et al., (2017) experimentally investigated the valve cover heat transfer and thermal balance of a gasoline engine at various speeds. They found that about one-third of the total fuel energy might be converted to effective power. In addition, heat transfer losses at higher engine speeds were low, and it was impossible to prevent increase in wasted energy. Khajorntraidet & Ito, (2016) studied the application of in-cylinder pressure to estimate heat transfer during the compression stroke. Based on the thermodynamic assumptions and the first law of thermodynamics, they derived a state-space model for the compression process, including both in-cylinder pressure and temperature. Their results showed that the proposed method is capable of quantifying the magnitude and the direction of heat transfer. Przybyla et al., (2013) proposed an algorithm to calculate the heat-transfer coefficient in the combustion chamber of an ICE based on in - cylinder pressure data. They determined the average values of the heat-transfer coefficient from the burned gases to the chamber walls during the compression stroke.

Fonseca et al., (2020) presented a comprehensive overview of engine heat transfer and wall temperature modelling. They reviewed the most important correlations for three engine heat transfer approaches: gas to wall (convection), wall-to-wall (conduction), and wall to liquid (convection) heat transfer models. They concluded that, to obtain a reliable prediction of engine heat transfer and wall temperature, these three approaches must be simultaneously applied. Karamangil et al., (2006) investigated the convective heat transfer coefficient on both the in-cylinder wall and water jacket sides of an SI engine. They calculated the cylinder temperature and pressure based on the first law of thermodynamics and evaluated the in-cylinder heat transfer coefficient using Annand's (1963) and Woschni's (1966) correlations. Shayler et al., (1993) studied the heat transfer process from the combustion chambers of spark-ignition engines and found that using the Eichelberg model led to more precise results than the first law of thermodynamics for calculating the instantaneous heat flux in the combustion chamber.

Olmeda et al., (2015) developed a heat transfer model that account for the tumble motion. They performed a sensitivity study to evaluate the model's robustness, and their findings substantiate the average root mean square error reduction of 70% compared to the Woschni model, as evidenced by instantaneous heat transfer evolution. Enomoto et al., (2014) performed an experimental study to measure the instantaneous temperature, thereby calculating the instantaneous heat flux, the mass-averaged gas temperature, and the local and overall heat transfer coefficients of the combustion chamber wall surfaces in a naturally aspirated, compression ignition engine. The experimental overall heat transfer coefficients were compared with those obtained from Eichelberg's and Woschni's empirical equations at various ignition timings and engine speeds. Results showed that the experimental average coefficients exhibited different characteristics compared to those computed via the conventional empirical correlations

Van et al., (2025) investigated the effects of ignition timing angle changes on the in- cylinder pressure and temperature of an SI engine and found that delaying the IT reduces the cylinder pressure and increases the temperature. The effect of ignition timing on engine parameters, such as power, torque, in-cylinder pressure variation, heat generation and burn rate of an SI engine, was studied experimentally by Tunka and Polcar (2017). The experiments were performed at 8 different ignition timings, ranging from 18° to 32° CA bTDC, at wide open throttle and at a constant engine speed of 2500 rpm with a stoichiometric AFR. Their results showed that as the

IT angle increased, both engine power and torque increased, and combustion became more stable. Furthermore, no significant difference was found in the combustion duration. Deng et al., (2013) conducted experiments on an SI engine fueled with 30 and 35% by volume butanol/gasoline blends which were compared with pure gasoline. They found that butanol provided higher knocking resistance by allowing advanced the ignition timing in SI engines, leading to more efficient combustion. Kakaee et al., (2011) conducted an experimental and theoretical study on an SI engines performance under different ignition timings. They found that the optimal power and torque occurred at an ignition timing of 31° CA bTDC, and the maximum thermal efficiency was achieved as when the peak in- cylinder pressure lay between 5 and 15°CA aTDC.

Heywood (1988) stated that the combustion process should be initiates at an optimal location relative to the TDC to obtain maximum torque. The combined duration of the flame-development and flame-propagation stages typically ranges from 30 to 90 degrees of crankshaft rotation. Combustion initiates before the piston reaches top dead center (TDC) during the compression stroke and continues past TDC into the early stage of the power stroke, remaining active well after peak cylinder pressure is achieved. If the ignition timing is excessively advanced, the resistance of the cylinder gases pressure against the upward moving pistons increases significantly (negative work). Conversely, if ignition timing is excessively retarded, the end of combustion is delayed, causing peak pressure to decrease and occur further after TDC during the power stroke (positive work). Consequently, either conditions advance or retard of ignition timing reduce the power developed by the engine. Optimal ignition timing depends on the air-fuel mixture, engine operating conditions, engine design, and the combustion products.

The combustion process begins with an electrical discharge between the spark plug electrodes anywhere between 10 and 30 degrees before top dead center (TDC). The typical combustion angle, which is the angle at which the crankshaft rotates during combustion process, is about 35 degrees for most engines. For example, if the combustion process is to be completed 15 degrees aTDC, ignition should occur about 20 degrees bTDC (Pulkrabek, 2014). Currently, the most commonly used correlations for calculating in- cylinder heat transfer in ICEs are listed in Table 1.

Table 1. The most commonly used heat transfer correlations

Authors	Equation
First group form $h(t) = f[V_p, P_g(t), T_g(t)]$ (1)	
Nusselt (1928)	$h(t) = 0.0278(1 + 0.38 V_p)[P_g(t) T_g(t)]^{0.333}$ (2)
Eichelberg (1939)	$h(t) = 0.0565 V_p^{\frac{1}{3}} [P_g(t) T_g(t)]^{0.5}$ (3)
Pflaum (1960)	$h(t) = .0399 [P_g(t) T_g(t)]^{\frac{1}{2}} (6.2 - (5.7)^{-(0.0305 V_p)^2} 0.00762 V_p)[1.175(P_g)^{0.25}]$ (4)
Second group form $N_u(t) = f[R_e(t), G_p(t)]$ (5)	
Woschni (1966)	$h(t) = \text{Constant} * [k(t)/D] [R_e(t)]^{0.7}$ (6)
	Where $R_e(t) = \rho(t) (2.28 V_p) D/\mu(t)$ (7)
Annand (1963)	$h(t) = \text{Constant} * [k(t)/D] [R_e(t)]^{0.8}$ (8)
	Where $R_e(t) = \rho(t) (V_p) D/\mu(t)$ (9)
Sitkei (1962)	$h(t) = \text{Constant} * [k(t)/D_e(t)] [R_e(t)]^{0.7}$ (10)
	Where $R_e(t) = \rho(t) (V_p) D_e(t)/\mu(t)$ (11)
	And $D_e(t) = 2DS(t) + D$ (12)
LeFeuvre (1969)	$h(t) = \text{Constant} * [k(t)/D] [R_e(t)]^{0.8} [P_r(t)]^{0.333}$ (13)
	Where $R_e(t) = \rho(t) (D/2)^2 * \omega g/\mu(t)$ (14)

The above - mentioned correlations include constants that must be determined experimentally by measuring the average heat transfer in an ICE and can be divided into two groups: the first group consists of empirical correlations, while the second group comprises semi-empirical models. In addition, there is a lack of investigations dealing with the effect of ignition timing on the in-cylinder heat transfer rate of an SI engine, which represents the main objective of this study. For this purpose, the heat transfer by

convection from the burned gases to gas- side cylinder wall and by conduction through the cylinder sides will be calculated in the terms of the heat transfer coefficient. Accordingly, the instantaneous cylinder volume, pressure, and temperature will be determined based on the engine's geometric parameters (i.e., cylinder clearance volume, cylinder bore, connecting rod length and crank offset) at different ignition timings: 30°, 25°, 20°, 15°, and 10° CA bTDC. The in-cylinder heat transfer coefficient will be calculated by using the Woschni correlation (1966). Consequently, the in-cylinder gas-side wall temperature will be calculated based on the principle of energy conservation from the equality of conductive heat transfer to the convective heat transfer through the cylinder walls, as well as by using Finite Element Model Builder for Partial Differential Equations (FlexPDE, Version 7).

2. METHODOLOGY

The engine heat transfer rate depends on the engine design parameters (i.e., cylinder volume and compression ratio) as well as operating parameters, such as air-fuel ratio, engine speed, and instantaneous in-cylinder pressure and temperature. Therefore, to simulate the heat transfer process inside the cylinder of a four-stroke, four-cylinder SI engine, the parameters listed in table 2 were selected. Consequently, the following steps were performed to calculate the parameter required for predicting the instantaneous engine heat transfer between the burned gases and the gas- side cylinder wall.

Table 2. Spark ignition engine parameters

Parameters	Quantity
Number of Cylinderes	4
Engine speed, N	3000 rpm
Fuel used (Isooctane)	(C ₈ H ₁₈)
Air/Fuel Ratio	15.14:1
Cylinder bore, B	85 mm
Piston stroke, S	90 mm
Engine displacement volume	2040 cc
Compression ratio, r _c	9.00
Connecting rod length–to-crank offset	3.5
Crank offset	45mm
Connecting rod length	157.5 mm
Cooling system type	Water cooling
Ignition system type	Electronic controlled
Volumetric efficiency, η _v	85 %

Instantaneous Cylinder Volume, V_g

$$V_g = V_c [1 + 0.5(r_c - 1)\{R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{0.5}\}] \quad (15)$$

$$\text{Where: } R = l/a \quad \text{and} \quad a = S/2$$

$$V_c = V_s/(r_c - 1) \quad \text{and} \quad V_s = (\pi/4) B^2 * S$$

R is the rod-to-stroke ratio; [R = 3 to 4] and θ is the crankshaft angle, crank offset a

Instantaneous Cylinder Pressure, P_g

The pressure variation during the combustion period is related to the pressure at the end of the compression process, P_o (i.e., the beginning of the combustion process) through the following equation:

$$P = P_o [1 - X_b) / (1 - Y_b)]^n \quad (16)$$

The mass fraction burned (X_b) at any crankshaft angle can be calculated by using the Wiebe function (Stone 2012)

$$X_b = 1 - \exp[-a((\theta - \theta_o)/\Delta\theta)^{m+1}] \quad (17)$$

$$Y_b = 4 X_b / (1 + 3X_b) \quad (18)$$

Where, Y_b are the volume fraction burned, θ_o is the crankshaft angle at the start of combustion, θ is the instantaneous crankshaft angle, $\Delta\theta$ is the total combustion duration, and **a** and **m** are adjustable constant parameters (**a** = 5.0 and **m** = 2.0), and **n** is the polytropic index (**n** = 1.25 to 1.35), as suggested by Heywood (1988)

Air and Fuel Mass Flow, AFR

The cylinder air mass flow can be computed using the following equation

$$m_a = \rho_a \eta_v V_s \quad (19)$$

Where, ρ_a is the inlet air density; ($\rho_a = 1.18 \text{ kg / m}^3$) and η_v is the volumetric efficiency. The mass of fuel entering the cylinder per cycle can be determined from the air-fuel ratio and the mass of air.

Considering a complete combustion of a mixture of isooctane fuel (C_8H_{18}) and air the stoichiometric air- fuel ratio can be determined by the following equation:

$$C_a H_b + (a + b/4)(O_2 * 3.76 N_2) = aCO_2 + b/2 H_2O + (a + b/4)3.76N_2 \quad (20)$$

For Isooctane fuel C_8H_{18} the stoichiometric air fuel ratio is 15.14 : 1

Heat Transfer Rate $Q(\theta)$

The heat transfer rate from the burned gases to the engine cylinder wall can be estimated using Newton's convective heat transfer equation, as follows:

$$Q(\theta) = h_g(\theta) A_p (T_g(\theta) - T_w) \quad (21)$$

According to the literature review, many correlations have been proposed by several researchers to calculate the in-cylinder heat transfer coefficient. Each correlation usually yields different results, even when applied to the same engine under the same engine operating conditions. One of the most commonly used correlations to determine the heat transfer coefficient during the combustion process in SI engines is the Woschni (1966) correlation

$$h = 129.9 (B)^{-0.2} (P_g)^{0.8} (T_g)^{-0.53} [C_1 V_p + C_2 (V_s T_r / P_r V_r) (P_g - P_o)]^{0.8} \quad (22)$$

Where: P_g and T_g are instantaneous gas pressure and temperature, respectively, P_r , T_r , and V_r are known values corresponding the beginning of combustion and $C_1 = 2.28$, $C_2 = 3.24 \times 10^{-3}$ for the combustion process. The motored pressure, which occurs without combustion, can be calculated by assuming that the compression process is isentropic, as proposed by Watson and Janota (1982)

$$P_o = P_r (V_r / V_g)^\gamma \quad (23)$$

$$\text{And } \gamma = 1.4 - 0.16(\phi)$$

V_g - is the instantaneous volume corresponding to the reference conditions

The instantaneous gas temperature was calculated by using the equation of state as:

$$T_g(\theta) = \frac{P_g V_g}{m_m R} \quad (24)$$

The gas-side cylinder wall temperature (T_w) is affected by many factors including engine speed and load, equivalence ratio, start of combustion, charge motion, inlet temperature, cylinder wall material, liquid coolant, and combustion temperatures. In this study, (T_w) is calculated based on the principles of energy conservation, equating the conductive heat transfer to the convective heat transfer through the cylinder wall:

$$(K/t) (T_w - T_{cw}) = h_g(T_g - T_w) \quad (25)$$

Solving for T_w , we get :

$$T_w = (h_g T_g + (K/t) T_{cw}) / (h_g + (K/t)) \quad (26)$$

Where $T_{cw} = 350$ (K) is the cooling water temperature, t is the cylinder wall thickness (m) and K is the thermal conductivity of the cylinder wall material (for cast iron; $K = 54$ W/m.K), (Heywood 1988)

Substituting h_g , T_g and T_w in equation (21), the rate of heat transfer to the cylinder wall is calculated.

3. RESULTS AND DISCUSSION

Ignition timing

The results obtained by using the relations stated above were graphically illustrated in the figures 1 to 5 for different ignition timing (10, 15, 20, 25 and 30) bTDC and ignition duration of 40 degrees of crank angle. Figure 1 shows the Variation of the instantaneous cylinder volume as a function of the crankshaft rotation angle. As shown from the figure, the maximum value of the cylinder volume is 544 cm^3 (displacement volume) at a crank angle of -180° (at BDC). When the piston travels from BDC toward TDC, the volume decreases until it reaches its minimum value at TDC (clearance volume).

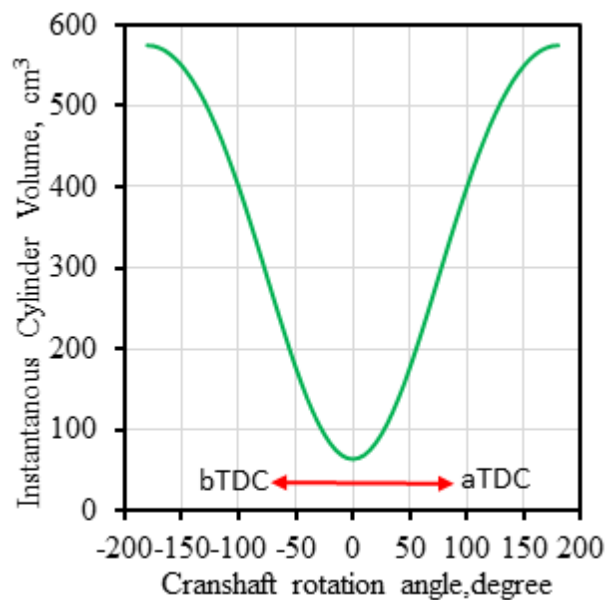


Figure 1. Variation of the instantaneous cylinder volume as a function of the crankshaft rotation angle.

In general, the combustion process in an SI engine is divided into three stages: (1) Spark ignition and flame development, (2) flame propagation, and (3) flame termination.

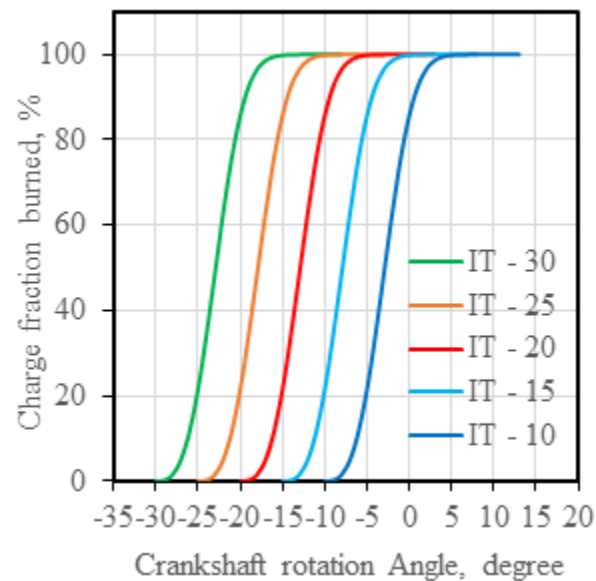


Figure 2. Influence of ignition timing on the charge mass fraction burned.

Figure 2 shows the effect of ignition timing on the charge mass fraction burned. During the first stage, the mass fraction burned was minimal, equaling 5% for all ignitions timing. Whereas, during the flame propagation stage, the mass fraction burned increased up to 100% for ignition timing angles of 30°, 25°, and 20° bTDC, and at TDC and slightly aTDC for 15° and 10° ignition timing, respectively. This increase can be attributed to the decrease of the cylinder volume and the cosequantly rise in-cylinder pressure and temperature, which increase the flame speed and enhance the combustion process.

The variation of the in-cylinder pressure and volume at the start of ignition as a function of the crankshaft rotation angle for different ignition timing is shown in figure 3. As shown in the figure, the in-cylinder pressure increased due to the decrease in cylinder volume as the piston travels from BDC to TDC, as well as due to advancement of ignition timing. These parameters were used in equations 23 and 24 as a reference value of the pressure and temperature at the start of combustion (P_r and V_r) for different ignitions timings.

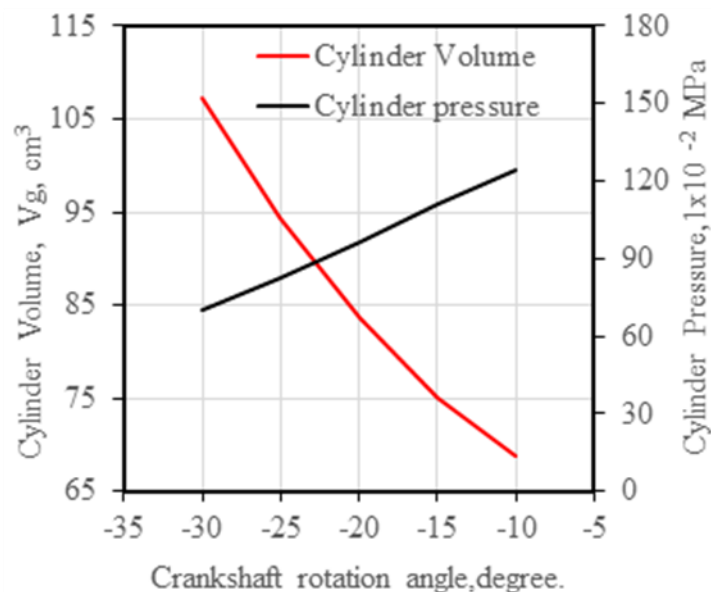


Figure 3. Variation of in-cylinder pressure and volume at the start of ignition as a function of the crankshaft rotation angle for different ignition timing.

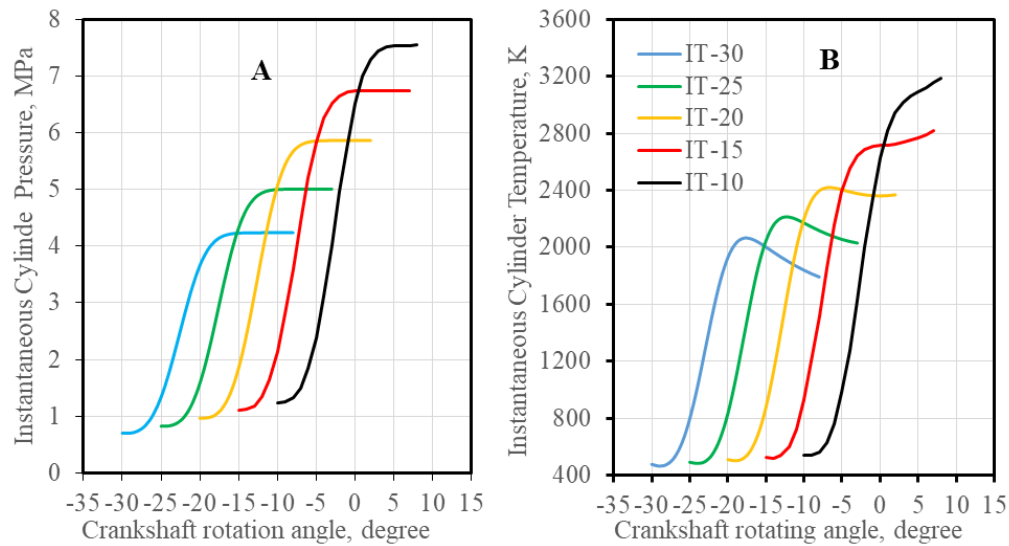


Figure 4(A–B). Instantaneous cylinder pressure and temperature versus CA for different IT, respectively.

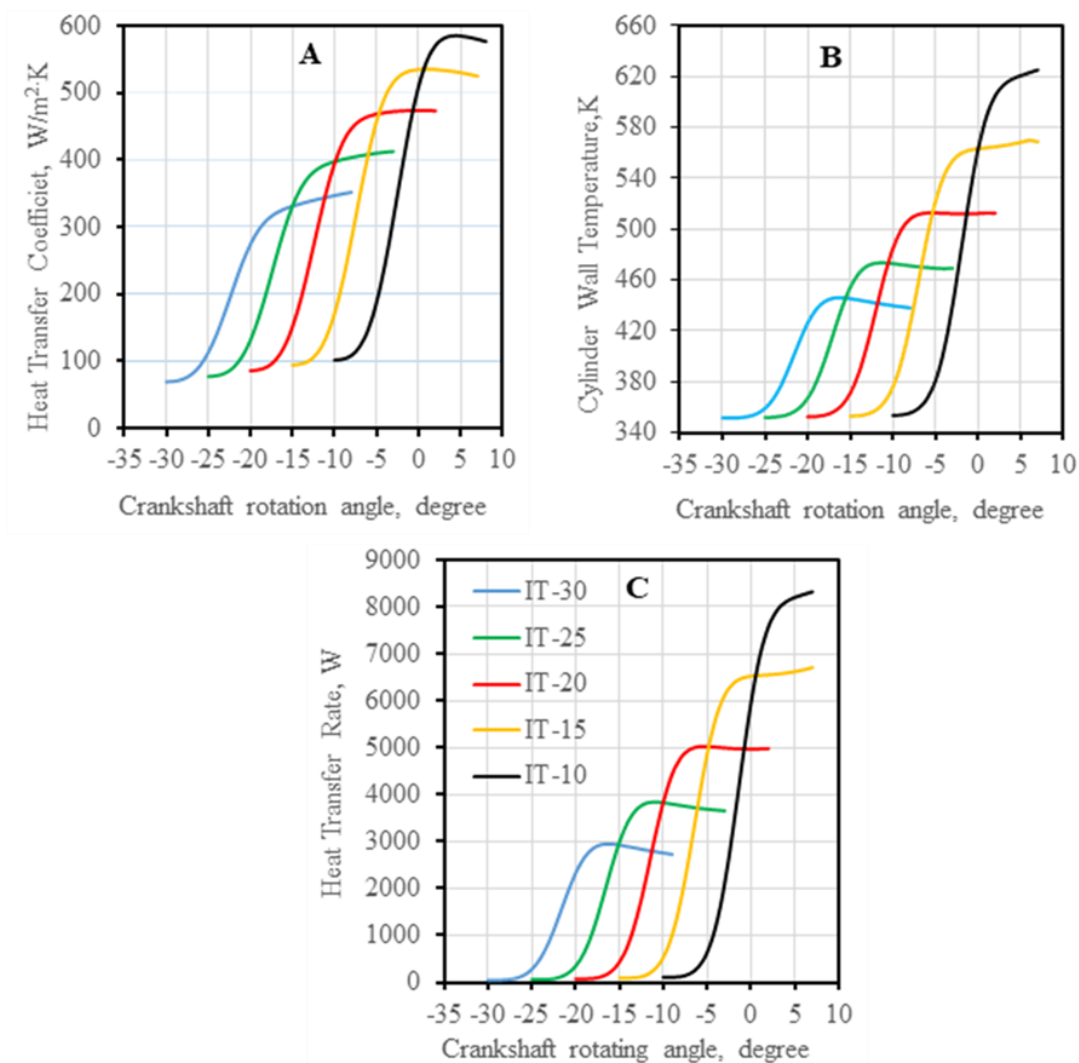


Figure 5(A - C). Effect of ignition timing angle on convective heat transfer coefficient, gas- side cylinder wall temperature, and heat transfer rate, respectively.

As a previously mentioned, the combustion process in a SI engine consists of three stages: During the first stage, the combustion process initiates following the spark ignition event; however, the energy released from the developing flame kernel is negligible due to the small mass fraction of the charge burned during this stage. Consequently, a little rise in pressure and temperature occurs, as shown in Figures 4A and B respectively, for all ignition timings. On the other hand, during the second stage, the in-cylinder pressure and temperature sharply rise and reaches their maximum value before TDC for ignition timings of 30°, 25°, and 20°, and after TDC for ignition timings of 15° and 10°. This increase primarily attributed to the decrease of cylinder volume (figure 1), release of fuel chemical energy, and flame propagation through the combustion chamber.

Furthermore, when comparing the ignition timing angles from the viewpoint of in-cylinder pressure and temperature, the maximum values of 7.45MPa and 3188K respectively, occur at an angle of 8° after TDC, which corresponds to an ignition timing of 10° before TDC.

The effect of the ignition timing on the convective heat transfer between the burned gases and the cylinder wall, characterized by the (h_g), is shown in Figure 5A. From the figure, it can be observed that the behaviour of the (h_g) closely matches that of the instantaneous burned gas and the gas - side cylinder wall temperatures, as shown in Figure 4B and 5B respectively. This can be attributed to the fact that the convective heat transfer coefficient h_g is directly proportional to the temperature difference between the burned gases and the cylinder wall. The maximum value of the (h_g) coefficient is 582 W/m² K occurs at an ignition timing of 10° bTDC instantaneous. The effect of ignitions timing angle on the gas- side cylinder wall temperature (T_w) is shown in figure 5B. The behavior of T_w is affected by the instantaneous cylinder temperature (T_g). When T_g increases, heat transfer from the burned gases to the cylinder wall increases too, accordingly T_w increases, and vice versa. The maximum value of the T_w is 625 K, which occurs at an ignition timing of 10° bTDC.

Figure 5C illustrates the variation of the heat transfer rate, $Q(\theta)$, from the burned gases to the engine cylinder wall as a function of the ignition timing. The behavior of the $Q(\theta)$ follows the same trend as the burned gas temperature, $T_g(\theta)$, convective heat transfer coefficient, $h_g(\theta)$, and gas - side cylinder wall temperatures, T_w , for all ignitions timing angles, as shown in figure 4B, 5A, and 5B, respectively. Therefore, the heat transfer rate increases as $h_g(\theta)$ and the temperature difference between $T_g(\theta)$ and T_w increase. This is due to the direct proportionality between $Q(\theta)$, and $h_g(\theta)$, and ($T_g(\theta) - T_w$). The maximum value of $Q(\theta)$ is 8.32 kW, which occurs at an ignition timing of 10° bTDC.

Temperature Distribution

Figure 6(A - C). Temperature distribution through the cylinder wall at ignition timing of 10°, 20°, and 30° bTDC, respectively. From the figures, it can be observed that the gas- side cylinder wall temperature for ignition timing of 10° bTDC is higher than that of 20° and 30° bTDC. This complies with the results obtained for mass fraction burned gases. During the heat transfer process the maximum cylinder gas temperature, $T_g = 3200$ K, as shown in Figure 4B decreases to the cylinder wall temperature, $T_w = 635$ K (Figure 6A), both for ignition timing of 10° bTDC. This decrease can be attributed to the following: In general, the in-cylinder moving gases adjacent to the wall cylinder forms a gas layer that acts as a thermal insulator. Furthermore, the heat transfer mechanism from the cylinder gases to the cylinder wall occurs through this layer, which consequently results in a large temperature decrease herein. The comparison between ignition timing from the viewpoint of the maximum value of P_g , T_g , h_g , T_w and (θ) was presented in Table 3. As shown in table, the best ignition timing is the 10° bTDC.

Table 3. Comparison of the maximum calculated cylinder parameters under different ignition timing

Parameter	Ignition Timing, CA degree				
	-30	-25	-20	-15	-10
P_g , (MPa)	4.2	5.0	5.8	6.7	7.5
T_g , (K)	1791	2029	2368	2820	3188
h_g , (W/m ² . K)	351	412	472	524	576
T_w , (K)	438	479	512	568	625
$Q(\theta)$, (W)	2728	3654	4979	6705	8316

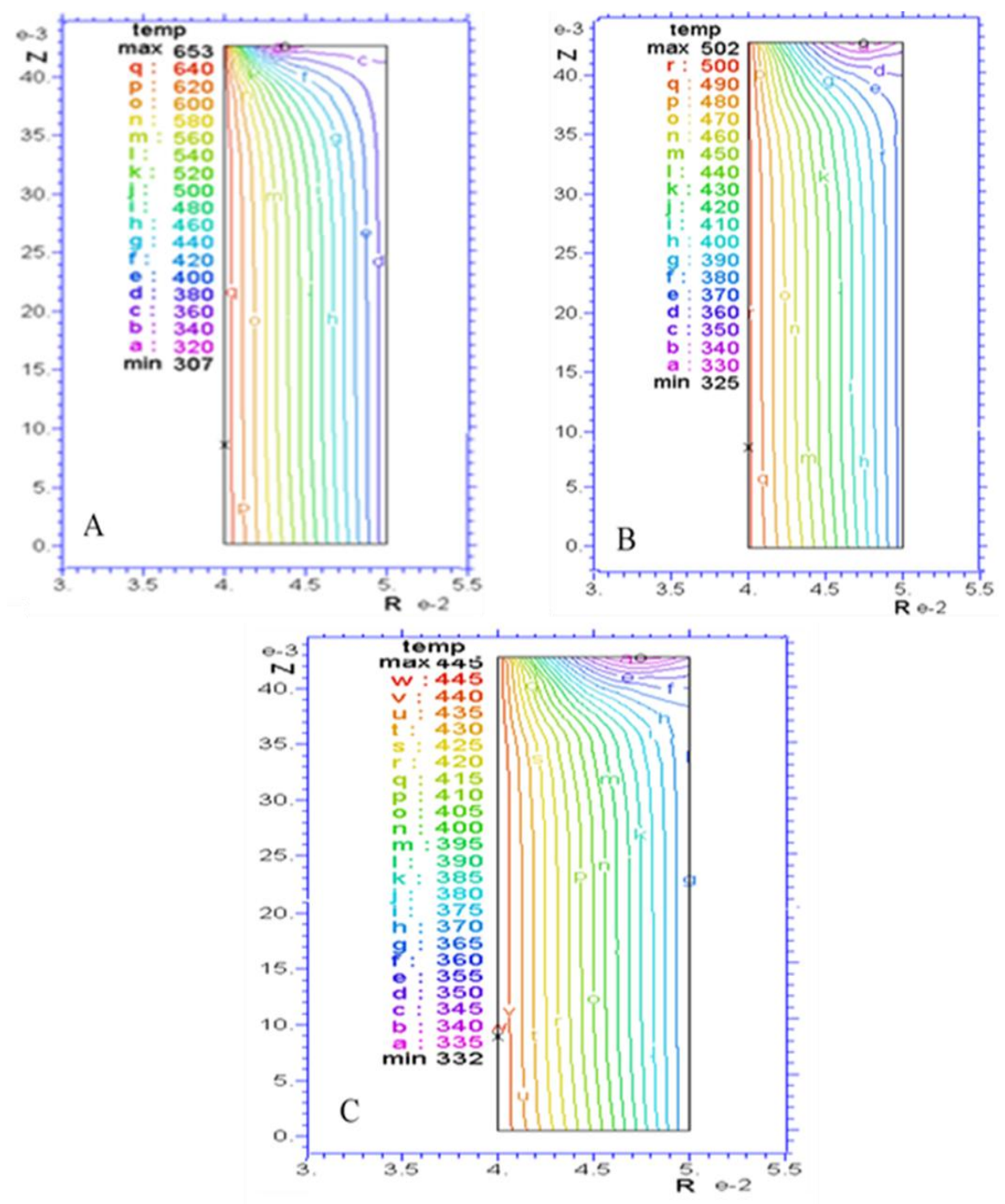


Figure 6(A - C). Temperature distribution through the cylinder wall at ignition timing of 10°, 20°, and 30° bTDC, respectively.

Table 4. Comparison of optimum ignition timing obtained in this study with results from other researchers

Angle of Ignition timing	Ending of combustion duration corresponding to:		Reference
	Maximum in-cylinder pressure	Maximum brake torque	
10° bTDC	8° aTDC		This study
10° to 40° bTDC	10° to 15° aTDC		Heywood (1988)

10° to 15° bTDC	closer to TDC		Guzzella & Onder (2010)
	5° to 20° aTDC		Pulkrabek (2014)
	5° to 20° aTDC	10° to 15° aTDC	Stone (2012)
		10°, 14°, 18° bTDC	Kale et al. (2015)

Table 4 shows the comparison of the obtained optimum ignition timing finding from other researchers. The advance of ignition timing and ending of combustion duration observed in the current study align closely to the results reported by Heywood (1988), Guzzella and Onder (2010), Pulkrabek (2014), Stone (2012), and Kale et al., (2015).

4. CONCLUSION

The objective of this study was to investigate the effect of engine ignition timing on the heat transfer rate between the in - cylinders burned gases and the gas - side cylinder wall temperature of an SI engine. From the results obtained in this study, the following conclusions can be made:

The instantaneous in-cylinder heat transfer, pressure, and gas temperature depend upon the amount of mass fraction burned. On other words on the heat released from the combustion of the air- fuel mixture, which is further affected upon the ignition timing and combustion duration. The combustion characteristics were described using a Wiebe function with different combustion durations. An increase in the combustion duration, i.e. the combustion process starts early during the compression stroke - for instance, by advancing the ignition timing 30° bTDC, results in most of the air-fuel mixture burning before the piston reaches the TDC. Consequently, the mass fraction burned, instantaneous cylinder temperature and pressure, heat transfer rate, and convective heat transfer coefficient increase and reaches their maximum values bTDC. However, if the ignition timing is gradually retarded (decrease in combustion duration), i.e., from 30 up to 10° bTDC the values of the P_g , T_g , h_g , and $Q(\theta)$ increases and reaches their maximum values after TDC. In addition, the results of cylinder wall temperature obtained by FlexPDE software conformed to those obtained by analytical method.

Based on the proceeding discussions, the heat transfer rate of a SI engine highly depends on ignition timing, therefore, its optimum value must be determined to achieve optimal engine operating parameters. For the engine used in this study the best ignition timing was 10° bTDC.

While conventional engine literature dictates that mid-speed (3000 rpm) operations necessitate an advanced spark timing (25°–35° bTDC) to mitigate shrinking cycle time, this study demonstrates that, under high in-cylinder pressure, high temperature, and stoichiometric air-to-fuel ratio conditions, an 18° CA combustion duration allows a retarded timing of 10° CA bTDC to effectively achieve near-ideal, constant-volume combustion around TDC. Consequently, this energy release generates a high peak cylinder pressure of 7.5 MPa.

NOMENCLATURE

A- Area, m²
a- crank offset
AF Air fuel ratio
B- Cylinder bore, m
BD Bottom Dead Center
CI- Compression ignition
h- heat transfer coefficient, W/m² K
IVC Inlet valve close
IVO Inlet valve open
CA- Crankshaft rotating angle, degree
IT- Ignition timing, degree
l- Connection rod length, m
m- Mass flow, kg

TD Top Dead Center
V_p- peripheral gas velocity, m/s
V - Volume, m³
Up - mean piston speed, m/s
w- effective gas velocity (m/s)
x- piston travel distance (m)

GREEK SYMBOLS

q- Density of cylinder gases, kg/m³
μ- Gas dynamic viscosity, kg/ms
γ - Specific heat ratio, [γ = 1.3]
R- Universal gas constant, 287 J/kg.K
CI- Compression ignition
P- Density of cylinder gases, kg/m³

m-	mass flow rate, kg/s	SUBSCRIPT	
N-	Engine speed, rpm	a-	air or after
P-	Cylinder pressure, N/m ²	b-	befor
P _i -	Pressure at the begining of combustion, N/m ²	g-	gas
Q(g)	Heat transfer rate, kW	c-	clearance or compression
Re-	Reynolds number	m-	mean or motoring
R-	Universal gas constant, 287, J/kg.K	r-	reference
r _c -	Compression ratio	rpm	revolutions per minute
S-	Piston stroke, m	t-	total
SI-	Spark ignition	s	swept or surface
T-	Fluid temperautre, K	w	wall
t _w -	Cylinder wall thickness, m		

Acknowledgement

The author gratefully acknowledge the support of the Al-Balqa Applied University / Mechanical Engineering Department for providing the facilities and resources required to complete this study.

Author Contributions

Formulating the research problem and its objectives, building the modeling framework, applying mathematical techniques to solve the research problem, structuring and organizing the sets of calculations and preparing the manuscript.

Ethical issues

Not applicable. This study does not involve any experiments on humans or animals. Hence, ethical approval was not required.

Informed consent

Not applicable.

Funding

This research did not receive any external funding like specific grant from funding agencies in the public, commercial, or nonprofit sectors.

Conflict of Interest

The author declare that he has no conflicts of interest, competing financial interests or personal relationships that could have influenced the work reported in this paper.

Data and materials availability

Data that support the findings of this study are embedded within the manuscript.

REFERENCES

1. Annand W. Heat transfer in the cylinders of reciprocating internal combustion engines. *Proc Inst Mech Eng.* 1963; 177(1):973–96. doi: 10.1243/PIME_PROC_1963_177_077_02
2. Asou Y, Tsurutani K, Fujimoto H, Senda J, Nagae M. Combustion in a small Diesel engine at starting. *SAE Technical Paper.* 1992; Paper No. 920697. doi: 10.4271/920697.
3. Cortona E. Engine thermo management for fuel consumption reduction [doctoral thesis]. Zürich: ETH Zürich; 2000. Diss. ETH No. 13862.
4. Deng B, Zhang F, Jianqin F, Yang J, Feng R, Liu J, et al. The heat release analysis of bio-butanol/gasoline blends on a high-speed SI engine. *Energy.* 2013; 60:230–41. doi: 10.1016/j.energy.2013.07.055
5. Eichelberg G. Some new investigations on old combustion engine problems. *Engineering.* 1939; 148:463–75.
6. Enomoto Y, Aoki Y, Emi M, Kimura S. Heat transfer coefficient on the combustion chamber wall surfaces in a naturally aspirated direct-injection diesel engine. *Int J Engine Res.*

- 2014;15(5):562–74. doi: 10.1177/1468087413500060
7. FlexPDE. Multiphysics software for partial differential equations. Version 7. PDE Solutions Inc. Url: <http://www.pdesolutions.com>
 8. Fonseca L, Novella R, Olmeda P, Valle RM. ICE heat transfer and wall temperature modeling: An overview. *Arch Comput Methods Eng*. 2020; 27(5):1661–79. doi: 10.1007/s11831-019-093 61-9
 9. Guzzella L, Onder CH. *Introduction to Modeling and Control of Internal Combustion Engine Systems*. Berlin: Springer Science & Business Media; 2010. doi:10.1007/978-3-642-10775-7
 10. Heywood JB. *Internal Combustion Engine Fundamentals*. New York: McGraw-Hill; 1988.
 11. Kakaee AH, Shojaeefard MH, Zareei J. Sensitivity and effect of ignition timing on the performance of a SI engine: An experimental and modeling study. *J Combust*. 2011; 2011:678719. doi: 10.1155/2011/678719
 12. Kale AB, Patil VR, Dhande DY. Sensitivity and effect of variable ignition timing and engine speed on the performance of a spark ignition engine. *IJSDR*. 2015; 3(10): 115-17
 13. Karamangil MI, Kaynakli O, Surmen A. Parametric investigation of cylinder and jacket side convective heat transfer coefficients of gasoline engines. *Energy Convers Manag*. 2006;47(6):601–15. doi: 10.1016/j.enconman.2005.05.018
 14. Khajomtraidet C, Ito K. An application of in-cylinder pressure for compression heat transfer estimation. *IFAC-Papers Online*. 2016; 49(11):769–74. doi: 10.1016/j.ifacol.2016.08.112
 15. Krause W, Spies W. Dynamic control of the coolant temperature for a reduction of fuel consumption and hydrocarbon emissions. *SAE Technical Paper*. 1996; Paper No. 960271.
 16. Ladommatos N, Xiao Z, Zhao H. The effect of piston bowl temperature on diesel exhaust emissions. *Proc Inst Mech Eng D J Automob Eng*. 2005; 219(3):371–88. doi: 10.1243/09544070 5X6550
 17. LeFeuvre T. *Instantaneous metal temperatures and heat fluxes in a diesel engine [doctoral thesis]*. Madison: University of Wisconsin; 1969.
 18. Nusselt W. Der wärmeübergang zwischen arbeitsmedium und zylinderwand in kolbenmaschinen. *Forschungsarb Geb Ingenieurwes*. 1928; 300:1–32.
 19. Ogawa H, Raihan KA, Iizuka K, Miyamoto N. Cycle to cycle transient characteristics of Diesel emissions during starting. *SAE Technical Paper*. 1999; Paper No. 1999-01-3495. doi: 10.4271/1999-01-3495
 20. Olmeda P, Martín J, Novella R, Carreño R. An adapted heat transfer model for engines with tumble motion. *Appl Energy*. 2015; 158:190–202. doi: 10.1016/j.apenergy.2015.08.051
 21. Pflaum W. Der Wärmeübergang bei Dieselmotoren mit und ohne Aufladung. *Jahrb Schiffbautech Ges*. 1960; 54:39–55.
 22. Przybyla G, Postrzednik S, Zmudka Z. The heat transfer coefficient calculation in the ICE cylinder based on in-cylinder pressure. *J KONES*. 2013; 20(4):381–8. doi: 10.5604/12314005.11 37850
 23. Pulkrabek WW. *Engineering Fundamentals of the Internal Combustion Engine*. 2nd ed. Pearson New International Edition. Harlow, Essex: Pearson Education Limited; 2014
 24. Reksowardojo KI, Ogawa H, Miyamoto N, Enomoto Y, Kitamura T. Time resolved nature of exhaust gas emissions and piston wall temperature under transient operation in a small Diesel engine. *SAE Technical Paper*. 1996; Paper No. 960031. doi: 10.4271/960031
 25. Shayler PJ, May SA, Ma T. The determination of heat transfer from the combustion chambers of SI engines. *SAE Technical Paper*. 1993; Paper No. 931131. doi: 10.4271/931131
 26. Shojaeefard M, Azarikhah P, Qasemian A. Experimental investigation of thermal balance and valve cover heat transfer in a small ICE. *Int J Automobile Eng*. 2017; 7(2):2421–32. doi: 10.22068/ijae.7.2.2423.
 27. Sitkei G. *Beitrag zur Theorie des Wärmeüberganges im Motor. Konstruktion*. 1962; 14:67–71.
 28. Stone R. *Introduction to Internal Combustion Engines*. 4th ed. London: Palgrave Macmillan; 2012.
 29. Torregrosa A, Olmeda P, Martín J, Degrauwe B. Experiments on the influence of inlet charge and coolant temperature on performance and emissions of a DI Diesel engine. *Exp Therm Fluid Sci*. 2006; 30(7):633–41. doi: 10.1016/j.expthermflsci. 2005.12.002.
 30. Tunka L, Polcar A. Effect of various ignition timings on combustion process and performance of gasoline engine. *Acta Univ Agric Silviculturae Mendelianae Brun*. 2017; 65(2):545–54. doi: 10.11118/actaun201765020545
 31. Van TP, Vinh ND, Long HD, Thang NM. A simulation study of the effect of ignition timing on the pressure and temperature profile of a 4-stroke gasoline engine. In: *Proceedings of the 2nd International Conference on Sustainability and Emerging Technologies for Smart Manufacturing (SETSM 2025)*. Singapore: Springer; 2025. doi: 10.1007/978-981-95-1750
 32. Watson N, Janota M. *Turbocharging the Internal Combustion Engine*. London: Macmillan; 1982.
 33. Woschni G. Computer programs to determine the relationship between pressure flow, heat release, and thermal load in diesel engines. *SAE Technical Paper*. 1966; Paper No. 650450. doi: 10.4271/650450