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Contribution of Ignition Timing Variation to the Greenhouse Gas Emission and Coolant Performance in Spark Ignition Engine

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Abstract. The ignition timing in a Spark Ignition Engine plays a prominent role in achieving high efficiency. However, ignition occurrence varies with the type of the engine and the type of the fuel in use. Ignition timing is different between engines, not only because of the thermal behavior of the combustion process, but also due to the morphology of the fuel. In the present study, the impact of ignition timing on greenhouse gas emission and the coolant performance are experimentally investigated. Four camshaft angles are selected and implemented for the study. Observation of the experiments’ outcome demonstrated that the larger the camshaft angle is, the lower the fuel consumption would be. In contrast, a substantial contribution to the enlargement of NOx and carbon monoxide production is observed. Data obtained from the coolant’s temperature sensors conclude that a slight improvement of the heat transfer process occurs when advancing the ignition timing.

1. Introduction
Internal combustion engines use fossil fuel as heat providers. However, the emission of by-product gases is one of the chief challenges of burning fossil fuels, mostly due to its negative effect on the environment, [1]. The development of the spark ignition engine has achieved a reasonable performance since the invention of the four-stork engine. SI engine manufacturers have admitted that the priority of the engine designer should be directed towards engine reliability, as the operation is viable and more efficient. Hence, controlling the production of such gases by enhancing the combustion efficiency becomes essential. To reach the manufacturers’ target, researchers have increased their attention on maximizing engine performance by using advanced combustion technology. Among numerous parameters that control the morphology of combustion products, ignition timing perches on the top of the most prominent factors. Recently, designers have paid more attention to the ignition timing of the four-stroke engine, both numerically and experimentally, in order to achieve the optimum engine operation reliability [2,3].

Over decades, many literatures have addressed the impact of ignition timing on ICE operation and numerous models were developed to map out the engine design strategy for optimal performance. Modeling and code simulation are usually conducted for the preliminary assessment of combustion and ignition timing design. However, extensive experimental tests are mandatory for optimizing the parameters' operation performance.

Computer simulation technique of ICE processes are useful tools to inexpensively achieve the optimum design of engine parameters. The technique could easily predict the pressure-time and
temperature-time variation in an engine for prescribed operating conditions. It is well known that precise modeling of flow dynamics and thermodynamics, intake manifold, ignition timing, and cylinder- water jacket heat transfer, is the key parameter for accurate prediction of engine performance [4].

For instance, Krieger and Borman [3], Blumberg and Kummer [5], and Heywood [6], employed a 1-D assumption in their calculation analysis of engine performance. However, Amsden [7], Kong et al [8], and Soylu [9], utilized a complex, multidimensional reaction code for the simulation of the combustion process. Engine knocking and exhaust gas emission can also be traced using the built-in thermodynamics equations. Hence, commercial codes such as the KIVA-CHEMKIN have recently been built to assist academic researchers and the R&D industry to understand and quantify the flow mobility mechanism, combustion and flame propagation, and the exhaust gas emission in transient status.

Accuracy obtained by employing multidimensional analysis in modeling all engine processes from the intake to the exhaust manifolds is at the expense of complexity, time consumption, and computer potential, [9-11]. To make the modeling more feasible and reasonable in terms of computer time cost, Rakopoulos [12, 13, 14] recommended to split the combustion process into a two-zone model.

Among many factors that impact the performance of SI engines, ignition timing is the most prominent. The ignition retard and its effect on the cylinder pressure distribution, stranded mass residual, and gas temperature are addressed by Chan and Zhu [15]. The authors calculated the thermodynamics parameters under different time ignition conditions.

The two-zone combustion scheme, adopted by Soylu and Van Gerpen, [2] was modeled to study the ignition timing influence on several engine parameters, including the combustion rate, cylinder pressure, equivalence ratio, and burning rate, [2]. Finally, Soylu and Gerpen [2] implemented a complete analysis of the burning rate to evaluate the flame inception and propagation speed at different operation conditions.

The objective of the present work is to examine the effect of the ignition timing on the gas emission performance of an SI engine. To achieve this goal, the ignition timing has been changed in the range of 7°CA ATDC to 20°CA BTDC, and the engine performance characteristics, such as the structure of exhaust gases, and the efficiency of cooling systems are obtained and discussed.

2. Modeling of Radiator Outlet Temperature
Models of the combustion process in SI engine may be categorized into three main groups:
1). Zero dimensional models
2). Quasi-dimensional models
3). Multi-dimensional models

The first model is the simplest and most feasible to investigate the impact of certain parameters on the rate of heat that transfers from the combustion zone to the coolant through the cylinder wall. We adopted this scheme in our analysis because the energy evolved during combustion can be simply obtained by applying the first law of thermodynamics to the system. Assuming that the heat lost to the surrounding is completely transferred to the water jacket and knowing that the heat released from fuel combustion in SI engine depends on the duration from ignition inception to the end of the combustion process, the following equation is derived by applying the first law of thermodynamics using the single zone model:

$$\frac{dQ_{ht}}{d\theta} = \frac{\partial Q_{ch}}{d\theta} + \frac{\partial W}{d\theta} + \frac{dU}{d\theta} - h_f \frac{dm_f}{d\theta} \quad (1)$$

Where: $\frac{\partial Q_{ch}}{d\theta}$: The apparent rate of heat produced from chemical reaction; $\frac{\partial Q_{ht}}{d\theta}$: The rate of heat transfer with surrounding; $\frac{\partial W}{d\theta}$: The rate of work transfer out of the system; $\frac{dm_f}{d\theta}h_f$: Rate of
enthalpy inflow with the fuel; $\frac{dU}{d\theta}$: Rate of change in internal energy of the system.

The following expression is obtained from ideal gas Equation:

$$\frac{dP}{P} + \frac{dV}{V} = \frac{dm}{m} + \frac{dR}{R} + \frac{dT}{T}$$

(2)

Considering the intake valve is closed just before the start of combustion, Eq (2) can be simplified to:

$$\frac{dP}{P} + \frac{dV}{V} = \frac{dT}{T}$$

(3)

Combining eqs. (1 and 3), the rate of heat transfers to the circulated water would be:

$$\frac{\partial Q_{ht}}{\partial \theta} = \frac{dQ_{ht}}{d\theta} - \left[\left(\frac{\gamma}{\gamma - 1}\right)\frac{dV}{d\theta} + \left(\frac{1}{\gamma - 1}\right)\frac{dP}{d\theta}\right]$$

(4)

Here, the volume change w.r.t. crank angle can be expressed as:

$$\frac{dV}{d\theta} = \frac{V}{2} \sin(\theta) \left[1 + \cos(\theta)\sqrt{1 - \sin^2(\theta)}\right]$$

(4-a)

By integrating eq.4 from the ignition inception to the end of combustion, the theoretical heat transfer to the radiator coolant is obtained. For the theoretical radiator outlet-water temperature:

$$T_{w, out} = T_{w, in} - \frac{Q_{ht}}{4.19 m_w}$$

(5)

The outlet temperature obtained from eq.5 is compared with its corresponding measured value to evaluate the discrepancy between the adopted model and the experimental outcome. It is convenient here to mention that the duration of combustion in the theoretical analysis for all cases is assumed to be ended at 5°ATDC.

3. **Experimental set-up**

The idea of the project is to compare and calculate the engine’s performance in terms of gas emission and environmental impact. Considering the capital cost and resource limitation, some compromises had to be done. For instance, the engine chosen for the experiments was bought from a used cars auction. The engine, illustrated in (Figure 1a) is a 2006 Mitsubishi Lancer, 4-cylinder, 1499 cc capacity, 75 mm bore with 0.88 bore/stroke ratio, and an estimated 80 kW horsepower at 6000 rpm, with maximum torque of 176.28 Nm. The engine is designed to emit up to 163 gr of CO$_2$ per km traveled. A steel stand, (Figure 1b), of 0.81x0.33 m-base and 0.84 m-height is used to mount the engine on top of it while a separate area is specified for the 42.0 liter fuel tank. The steel-constructed stand has an extra compartment for the battery, the wiring, and the engine computer. The stand is finished off with yellow paint for esthetic purposes. Temperature sensors are inbuilt in the radiator and the recorded data are used for calculating the rate of heat exchanged between the engine and the coolant. Dimension of the radiator is listed in Table 1.

| Table 1. Radiator Specification |
|---------------------------------|
| **Category** | **Description** |
| Weight | 4.830 kg |
| Depth | 16 mm |
| Width | 685 mm |
| height | 375 mm |
After assembling all parts and sensors, the camshaft timing (combustion timing) was adjusted for four values (figure 2), in order to observe the impact of ignition timing on the environment by measuring gas emission when conventional gasoline fuel is utilized.

Figure 1a. Mitsubishi Lancer Test Engine  
Figure 1b. The Test Stand

Figure 2. Camshaft Angle Adjustment Method

4. Results and Discussion

4.1 Performance of the Cooling System

The theoretical and experimental values of the water temperature leaving the radiator are plotted for the four-selected crank angles, as shown in Figure 3. The predicted temperature slightly deviates from the measured temperature. However, both show slight enhancement in the performance of the radiator as the ignition timing starts early.

Figure 3. Temperature leaving the radiator (both theoretical and measured) from four different angles

As illustrated in the figure, the optimum crank angle in terms of heat transfer between the coolant and the engine is achieved at 12°, since at this angle the temperature of the coolant leaving the radiator is the lowest. Advancing ignition further leads to a reduction in the performance of the radiator.
4.2 Gas emission

Figure 4 depicts the percentage of each gas emitted from the engine exhaust for the four different angles under study. Since carbon emission negatively impacts the ecosystem, measurement of carbon dioxide evolved can be considered as a benchmark of the pollution limit. Measurement illustrated in the figure indicates that the lowest percentage of carbon is released when the ignition angle is 12° BTDC, thus making it the most suitable angle. However, carbon monoxide is observed to be increased as well at this angle, apparently due to decreasing in the combustion efficiency.

![Figure 4. Ignition Angle vs. Gas Emissions Percentage](image)

4.3 NOx emission

NOx produced during the combustion process is measured at the engine exhaust pipe with the assistance of the BTU-900 gas analyzer. Figure 5 demonstrates the percentage of NOx emitted to the environment from the engine exhaust for the four crank angles. It is well known that the production tendency of NOx is proportionate to the in-cylinder peak temperature. Observing the trend of NOx variation, one can conclude that the ignition timing does affect the production of NOx, and the crank angle of 7° was found to be the lowest in comparison to others. This is attributed to the fact that the in-cylinder peak pressure increases with advancing the ignition timing, resulting in an increase in the peak temperature.

![Figure 5. NOx emission verses ignition timing](image)

5. Conclusion

The present study postulates that the angle of ignition inception is the dominating factor for maximizing the thermal performance of SI engine. In particular, the experiments conducted in this work focus on the exhaust emission of by-product gases and the performance of the cooling system. The former has been presented by measuring the contribution of CO\textsubscript{X} and NOx in the engine exhaust gases for different crank angles. The latter, however, is described by illustrating the coolant temperature exit from the radiator. The following summarize the corollaries of the study: 1) The 1-D theory of heat exchanger describes the cooling performance at moderately good agreement with the experiment. 2) The late ignition timing drops the radiator performance and decreases the combustion efficiency. 3) When the
ignition inception gets too advanced, more heat will be lost, resulting in dramatic reduction in the engine thermal efficiency and more CO was observed at the exhaust sensor. 4) NOx emission drops as the ignition timing is retard.

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