NUMERICAL EVALUATION BY MODELS OF LOAD AND SPARK TIMING EFFECTS ON THE IN-CYLINDER HEAT TRANSFER OF A SI ENGINE

A. Sanli1, C. Sayin2, M. Gumus2, I. Kilicaslan1, and M. Canakci1,3

1Department of Mechanical Education, Kocaeli University, Izmit, Turkey
2Department of Mechanical Education, Marmara University, Istanbul, Turkey
3Alternative Fuels R&D Center, Kocaeli University, Izmit, Turkey

The aim of this study is to examine numerically the effects of spark timing and load parameters on the in-cylinder heat transfer of a SI engine by using experimental engine test data. For the investigation, a four-stroke, air-cooled, single-cylinder SI engine was tested at different spark timings and loads at a single engine speed of 2000 rpm. Woschni, Hohenberg, and Han models were employed to estimate the in-cylinder heat transfer coefficient in the case of different test conditions because of being favorable models on the SI engine operations. The evaluations show that the in-cylinder heat transfer characteristics of the air-cooled SI engine strongly depend on the load while they slightly depend on the spark timing.

1. INTRODUCTION

In design of the internal combustion engines (ICEs), thermal behavior of the engine is an important topic because of its effect on indicated efficiency, power output, emissions, lubricant, and cooling capacity of the engine. Heat transfer to the combustion chamber walls influences the indicated efficiency because it reduces the cylinder pressure and temperature and thus the work done on the piston decreases per cycle. Heat loss to the cooling system of an ICE is approximately 30% of the total fuel energy supplied to the engine during the one working cycle. About half of this loss is due to the in-cylinder heat transfer and the rest is due to the cylinder head and to the exhaust port [1, 2].

From the in-cylinder points of view, the heat transfer changes locally instantaneous temperatures that have exponential dependence on controlling the formation rate of nitric oxide emissions (NOx). High temperatures lead to thermal

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Address correspondence to Mustafa Canakci, Department of Mechanical Education, Kocaeli University, Umuttepe Yerleskesi, Umuttepe, Izmit 41380, Turkey. E-mail: mustafacanakci@hotmail.com
stresses of material, and impact on fatigue failure limits of various engine components, thereby causing the cracks in cylinder head or deforming cylinder bore dimension and valve stems. Keeping below the wall temperature of the combustion chamber from the certain limits is necessary to prevent the oil film from deterioration or oxidizing and its viscosity from diminishing, so the oil film temperature should not exceed about 200°C. In order to avoid from pre-ignition and knock risks resulting from overheated spark-plug electrodes and exhaust valves in the spark ignition (SI) engines, the spark-plug and valves must be kept cool. In addition, the heat transfer to the inflowing air or charge reduces volumetric efficiency on account of reducing the inflowing fluid density [3].

A number of experimental and theoretical studies concerning the effects of various engine operating parameters on the heat transfer to the combustion chamber walls have been published since the last decades. Alkidas and Myers [4] investigated the effect of the air-fuel ratio (18, 16, 14, and 11.5 AFR) and volumetric efficiency (40, 50, 60%) on the heat flux and showed that the highest heat flux to the combustion chamber walls occurred at the near stoichiometric ratio (16 AFR) and the increasing volumetric efficiency resulted in an increasing in the peak heat fluxes. Alkidas [5] measured the heat flux at different engine speeds and loads and showed that the increasing engine speed and load increased the heat flux to the combustion chamber walls. He also compared the heat flux values obtained by the Woschni

<table>
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<tr>
<th>NOMENCLATURE</th>
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<tr>
<td>a, C₁, C₂, C₃</td>
<td>a air</td>
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<td>D</td>
<td>c combustion</td>
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<td>h</td>
<td>d displacement</td>
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<td>Uᵣ(2Sn/60) piston mean velocity, m/s</td>
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LOAD AND SPARK TIMING EFFECTS ON HEAT TRANSFER OF A SI ENGINE
model with the experimentally measured ones and observed that the one obtained by
the model was in good agreement with the experimental results. Karamangil et al [6],
investigated parametrically how the convective heat transfer coefficients (HTCs) for
the gas and coolant side vary with different engine operating parameters, such as the
engine speed, compression ratio, excess air coefficient, combustion duration, inlet
pressure, and temperature. Shojaefard et al [7], studied transient thermal analysis
of an exhaust valve by using the finite-element method. The temperature distribution
and resultant thermal stresses at each opening and closing time were obtained.
Rakopoulos and Mavropoulos [8] disclosed to be the significant differences between
the overall and local HTC variations of an air-cooled direct injection (DI) diesel
engine, and they also presented the effects of engine speed and load on the in-
cylinder heat transfer.

In a previous study [9], the effects of engine speed and load on the in-cylinder
heat transfer of a water-cooled, four-stroke diesel engine have been investigated. In
this study, in-cylinder heat transfer characteristics of an air-cooled SI engine were
investigated. However, this study is not presenting something new, but rather, it is
intended to raise the awareness and importance of the in-cylinder heat transfer in
the ICES, and to present how the in-cylinder heat transfer characteristics of a SI
engine are varying under different operating parameters by using some models in
the light of engine test data.

2. CALCULATION OF THE IN-CYLINDER HEAT TRANSFER

In the development of ICES, the heat transfer from the burning gases to the
in-cylinder walls is the main subject of numerous investigations. Historically, Nusselt
made pioneer attempts on the in-cylinder heat transfer based on the combustion
experiments in a spherical bomb in 1923. It was intended to predict the steady-state
or time-average heat flux, but it has often been used for the prediction of instan-
taneous heat flux because it was expressed in terms of instantaneous $P$ and $T$ values.
Then, Eichelberg [10] developed a model correlated in large-scale two-stroke and
four-stroke diesel engines. It is preferred by most investigators owing to being simple
relatively easy to use. However, when the unit analysis is made, it is shown that it
does not rely on the theoretical bases and yield dimensionally consistent results
[11]. It is believed that the fault is stemmed from the experiments with poor
technology in those years [12]. Oguri [13] used the Eichelberg model to estimate
the in-cylinder HTC of a water-cooled, single cylinder SI engine and observed that
the model’s results coincided well with the experimental ones at the expansion stroke,
but for the compression stroke it was not good. Therefore, he developed a new
in-cylinder heat transfer correlation by making some variations in the Elser formula,
including entropy change, specific heat under constant pressure change, and Nusselt
relation. In the following years, an approach, including Nusselt, Reynolds, and
Prandtl number relationships, has been suggested for the turbulent flow inside the
circular tubes, as follows.

$$\text{Nu} = a\text{Re}^m \text{Pr}^n$$

(1)
Pr can be omitted if the fluid is a gas. In this case, its effect might be included in the $a$ coefficient. $Nu$ includes the HTC. So, by expanding $Nu$ and Re numbers the equation becomes

$$
\frac{hl}{k} = a \left( \frac{pU_p l}{\mu} \right)^m
$$

The value of exponent $m$ generally ranges from 0.7 to 0.8 for forced convection with turbulent flow [14]. By emphasizing that the exponent $m$ is equal to 0.8, Woschni [15] developed the in-cylinder HTC below by using Eq. (2).

$$
h(0) = a P(0)^{0.8} T_r(0)^{-0.55} D^{-0.2} W(0)^{0.8}
$$

He assumed that the characteristic length $l$, piston mean velocity $U_p$, and fluid density $p$ can be replaced, respectively, with the cylinder bore diameter $D$, gas velocity $W$, and $P/RT$. Thermal conductivity $k$ and dynamic gas viscosity $\mu$ are proportional to $T^{0.75}$ and $T^{0.62}$, respectively [3]. Effective gas velocity $W$ was expressed by Woschni as follows.

$$
W(0) = \left[ C_1 U_p + C_2 \frac{V_d T_r}{P_m} (P(0) - P_m(0)) \right]
$$

where $V_d$ is the displacement volume, $T_r$, $P_r$, and $V_r$ are evaluated, respectively, as temperature, pressure, and volume at any reference state, such as inlet valve closing or combustion start. $(P(0) - P_m(0))$ is the pressure rise resulted from combustion. While $P(0)$ is the firing pressure, $P_m(0)$, calculated from the Watson-Janota model [16], is the motoring pressure at the same crank angle with $P(0)$. Values suggested for $C_1$ and $C_2$ are

$$
C_1 = 6.18 \quad C_2 = 0 \text{ for gas exchange period}
$$

$$
C_1 = 2.28 \quad C_2 = 0 \text{ for compression period}
$$

$$
C_1 = 2.28 \quad C_2 = 3.24 \times 10^{-3} \text{ for combustion and expansion period}
$$

Shayler et al. [17] investigated two methods of determining the rate of heat transfer from the combustion chamber of three SI engines, which are Ford’s 1.1 L Valencia, 1.6 L CVH, and 2.0 L DOHC at the different speed, load, spark timing, and AFR conditions. In the first method that is an analysis based on the application of the first law of thermodynamics. The instantaneous cylinder heat transfer rate was found to be subject to large errors stemmed from uncertainties in the gas properties. In the second method, the instantaneous cylinder heat transfer rates were calculated using the Woschni, Eichelberg, and Annand models and were integrated over the engine cycle to generate cycle-averaged heat transfer rates. The rates were compared with a cylinder heat transfer rate deduced from the difference between heat transfer to the coolant and heat transfer from the exhaust gases to the exhaust port surface.
The comparisons showed that the best agreement was obtained with the Woschni correlation. The researchers note that this is important information about the models’ comparison. From the results presented, it appears that the original coefficient values (for \(a\), \(C_1\), and \(C_2\)) proposed by Woschni can be applied without major adjustment. In addition, it should be noted that the Woschni formula not only includes the effect of convection but also includes radiation effect in a lumped form [2].

In the following years, Hohenberg [18] proposed an in-cylinder heat transfer equation by modifying the Woschni equation in the DI diesel engines with swirl. Equation (5) describes the in-cylinder HTC proposed by Hohenberg.

\[
h(\theta) = aP(\theta)^{0.8}T_g(\theta)^{-0.4}V(\theta)^{-0.06}(U_p + 1.4)^{0.8}
\]

Zeng et al. [19] improved a method adding the capability of predicting the effect of engine spark-timing on the cylinder pressure and applied it to a SI engine. The cylinder pressure was reconstructed including the third input of spark timing along with speed and load. Comparisons between measured and reconstructed cylinder pressures demonstrated that the method was applicable over a wide range of the SI engine operating. The reconstructed cylinder pressure was used to achieve the in-cylinder heat transfer and heat release analyses. In order to study the heat transfer, they used the Hohenberg model.

Han et al. [20] successfully established a new in-cylinder heat transfer model for the SI engines based on the turbulent flow. They used 0.75 for the value of the exponent \(m\) given in Eq. (2), taking into account the results of experiments carried out on 19 different ICEs by Taylor and Toong [21]. Consequently, the formula became as follows.

\[
h(\theta) = C_1P(\theta)^{0.75}T_g(\theta)^{-0.465}D^{-0.25}W(\theta)^{0.75}
\]

For the effective gas velocity \(W\), Han et al. assumed that an increase in the gas velocity during the combustion period was caused by the heat release revealed from the chemical reactions of gas mixture. Using this assumption, the gas velocity \(W\) can be expressed as follows.

\[
W = C_2U_p + C_3(\lambda P(\theta)dV + V(\theta)dP)
\]

The constants \(C_1\), \(C_2\), \(C_3\) and \(\lambda\) values are 687, 0.494, 0.73 \times 10^{-6}, and 1.35, respectively. Any comparison has not been coincided between the Han model and another model in the literature. More comprehensive information dealing with the in-cylinder heat transfer models can be found in references [1–3, 11, 22].

It is well known that the in-cylinder heat transfer is strongly influenced by the combustion gas pressure and temperature. The main data to calculate these models are the pressure, temperature, and volume in the cylinder. Once the pressure is acquired, the temperature of the gases in the cylinder can be calculated using the pressure, assuming the first law of thermodynamics. In the above equations, the instantaneous cylinder volume \(V(\theta)\) as a function of crank angle can be found using
the following well-known formulas.

\[ V(\theta) = V_c + \frac{\pi D^2}{4} x(\theta) \]  

(8)

\[ x(\theta) \] represents the distance from TDC, and thus according to the crank angle,

\[ x(\theta) = l + R - \left[ R \cos(\theta) + \sqrt{(l^2 - R^2 \sin^2(\theta))} \right] \]  

(9)

Applying the steady-state heat transfer assumption for the present study, the heat transfer on the inside surface of the cylinder is found from the Newton’s cooling law by multiplying the simultaneous HTC with the temperature difference, i.e.,

\[ q(\theta) = h(\theta)(T_g(\theta) - T_w) \]  

(10)

Figure 1 illustrates both the convective and radiative components of the heat transfer between gas and in-cylinder wall, and conduction through the combustion chamber wall, and convection to the air for air-cooled engines [23]. Mean gas temperature \( T_{mg} \) is depicted with dotted lines. Heat from the burning gases is also transferred to the cylinder wall indirectly by conduction through the piston rings and skirts. However, it is beyond the scope of this study. In the present study, it is focused on the heat transfer between gas and in-cylinder wall.

As mentioned previously, the temperature of the gases in the cylinder can be calculated using the cylinder pressure and volume, applying the first law of thermodynamics.

\[ T_g(\theta) = \frac{P(\theta)V(\theta)}{mR} \]  

(11)

Figure 1. Schematic view of the in-cylinder heat transfer process in an air-cooled engine.
where $P$ is the instantaneous in-cylinder pressure, $V$ is the instantaneous cylinder volume, $m$ is the air mass in the cylinder, and $R$ is the gas constant.

$T_w$, inside wall temperature of combustion chamber varies with the engine speed, load, equivalence ratio, start of combustion, charge motion, inlet temperature, wall material, and the coolant and combustion temperatures. It involves apparently too sophisticated to predict the wall temperature. Therefore, $T_w$ was replaced with spark-plug temperature $T_{sp}$ in this study, as it was assumed in a heat transfer study, which was performed by Wu et al. [24] for an air-cooled SI engine. $T_{sp}$ is formulated as follows.

$$T_{sp}(^\circ C) = a_{c1} - n \cdot a_{c2} - n^2 \cdot a_{c3} - P_{in} \cdot a_{c4} + P_{in}^2 \cdot a_{c5} + n \cdot P_{in} \cdot a_{c6}$$  \hspace{1cm} (12)$$

where $n$ is engine speed (rev/s), $P_{in}$ is the intake manifold pressure (bar), and $a_{ci}$ is the coefficients obtained from curve fitting by the reference.

3. EXPERIMENTAL APPARATUS AND PROCEDURE

The experiments were performed on a single cylinder, air-cooled, four-stroke SI engine. Its specifications are given in Table 1. The engine was loaded by an electrical dynamometer rated at 10 kW and 380 volts. The load on the dynamometer was measured by means of a strain gauge load sensor. An inductive pickup speed sensor was used to determine the engine speed. The in-cylinder pressure was measured using a Kistler Model 6052B air-cooled piezo-quartz pressure sensor mounted on the cylinder head. The pressure signals were then passed into a Kistler Model 5644A charge amplifier. Crankshaft position was obtained by a crankshaft angle sensor in order to determine cylinder gas pressure as a function of crank angle. The crank angle signal was attained through an angle-generating device mounted on the main shaft. The intervals of the cylinder pressure were 0.75°CA for all test cases and pressure average of the consecutive 100 cycles was received. The start of combustion was accepted 2°CA after the spark instant because of a period of time delay between the spark instant and start of combustion. The engine was fueled with the commercial grade gasoline of 95 research octane number.

<table>
<thead>
<tr>
<th>Table 1. Engine specifications</th>
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<tr>
<td><strong>Brand/model</strong></td>
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<tr>
<td><strong>Engine type</strong></td>
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<tr>
<td><strong>Bore/Stroke</strong></td>
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<tr>
<td><strong>Compression ratio</strong></td>
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<tr>
<td><strong>Connection rod length</strong></td>
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<tr>
<td><strong>Max. torque</strong></td>
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<tr>
<td><strong>Max. power</strong></td>
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<tr>
<td><strong>Inlet valve opening</strong></td>
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<td><strong>Inlet valve closing</strong></td>
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<tr>
<td><strong>Exhaust valve opening</strong></td>
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<tr>
<td><strong>Exhaust valve closing</strong></td>
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</tbody>
</table>
The test engine gives the maximum brake torque (MBT) at the optimum spark timing that is 20°CA BTDC. The electrical spark is provided by means of a magneto. By changing the magneto position, the spark timings were adjusted to 17.5 and 22.5°CA BTDC. All tests were conducted on a test-bench, as shown in Figure 2. Each of the tests was repeated three times and the average was taken. Before each test, the engine was carefully regulated according to the catalogue values and all data were collected after the engine was stabilized. The accuracy of the measurements and the uncertainty in the calculated results are given in Table 2.

4. RESULTS AND DISCUSSION

Heat transfer rates generally increase during the compression and the expansion strokes, and the peak heat flux mostly occurs after TDC for a typical engine.

Table 2. Accuracy of the measurements and the uncertainty in the calculated results

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Accuracy</th>
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<tbody>
<tr>
<td>Load</td>
<td>± 2 Nm</td>
</tr>
<tr>
<td>Speed</td>
<td>± 25 rpm</td>
</tr>
<tr>
<td>Crankshaft position</td>
<td>± 0.1°CA</td>
</tr>
<tr>
<td>Charge amplifier</td>
<td>± 1%</td>
</tr>
<tr>
<td>Temperatures</td>
<td>± 0.1 K</td>
</tr>
<tr>
<td>Calculated results</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>HTCT</td>
<td>± 2%</td>
</tr>
<tr>
<td>Heat flux</td>
<td>± 2.1%</td>
</tr>
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</table>
On the other hand, heat fluxes might nearly decrease too small negative values during the intake and exhaust strokes. Therefore, the data, obtained between the period of intake valve closing and exhaust valve opening, are presented in Figures 3–9. The figures include HTC and heat flux values of the Woschni, Hohenberg, and Han models as a function of crank angle at different spark timings and loads for 2000 rpm engine speed.

4.1. Evaluations of the Heat Transfer Models

The HTC and heat flux histories of three models examined are depicted in Figure 3 as a function of crank angle at the MBT spark timing of the full load for 2000 rpm engine speed. At first it was observed that HTC and heat flux amounts of the Han model were somewhat higher than those from the Woschni and Hohenberg models. The peak value of the Han model is 30% higher than the Woschni model, 37.6% higher than the Hohenberg model about the HTC; and the heat flux is 31% higher than the Woschni model, 37.1% higher than the Hohenberg model. A reason as to why the Han model is higher than the others may be because the \(a\) constant has been correlated to 128 in the Woschni and Hohenberg models; whereas, it has been correlated to 687 in the Han model, provided that pressure is in the bar unit. On the other hand, exponent \(m\) in the Woschni and Hohenberg models is higher than that in the Han model, i.e., 0.8 against to 0.75. However, it has been believed that the exponent \(m\) has not been as dominant as the constant \(a\) on the discrepancy. In view of the Hohenberg HTC and heat flux, when they were compared with those of Woschni, the Hohenberg model overestimates the HTC and heat flux of certain sections of the compression and expansion periods;

![Figure 3. Comparisons of Han, Woschni, and Hohenberg heat transfer correlations for the spark timing (20° CA BTDC) at full load—2000 rpm.](image-url)
however it underestimates at only peak regions. As pointed out by Soyhan et al. [25],
the Woschni model gives higher gas displacement velocity compared to the
Hohenberg model during the combustion phase; whereas, the Hohenberg model
gives a constant gas displacement velocity for the whole cycle. The heat transfer
results are comparatively seen in Figure 3, which are very close to each other.
Accordingly, in order to evaluate the spark timing and load effects on the in-cylinder
heat transfer, the results of each model were presented separately as a function of the
 crank angle.

Figure 4. Comparison of the Woschni HTCs and heat fluxes for different spark timings at full load.

Figure 5. Comparison of the Woschni HTCs and heat fluxes for different spark timings at 50% load.
4.2. Heat Transfer Behaviors at Different Spark Timings and Loads

Figures 4 and 5 show the histories of the HTCs and heat fluxes obtained from the Woschni model at different spark timings and load conditions. The peak HTC and heat flux values of Woschni at spark timings of 17.5, 20, and 22.5°CA for full load are 1577, 1606, and 1650 W/m²K, and 0.45, 0.47, and 0.49 MW/m², respectively; for the 50% load condition, they are 1421, 1428, and 1460 W/m²K, and 0.37, 0.38, and 0.40 MW/m², respectively.

Figure 6. Comparison of the Hohenberg HTCs and heat fluxes for different spark timings at full load.

Figure 7. Comparison of the Hohenberg HTCs and heat fluxes for different spark timings at 50% load.
The HTC and heat flux results of the Hohenberg model as a function of the crank angle at different spark timings and loads at 2000 rpm can be seen in Figures 6 and 7. At this speed, the peak HTC and heat flux magnitudes of the 17.5, 20, and 22.5°CA BTDC spark timings of the full load which are 1482, 1521, and 1551 W/m²K, and 0.43, 0.45, and 0.465 MW/m², respectively; whereas, at 50% load they are 1385, 1406, and 1441 W/m²K, and 0.37, 0.38, and 0.40 MW/m², respectively, which are almost the same as the magnitudes of Woschni at the same load.

Figure 8. Comparison of the Han HTCs and heat fluxes for different spark timings at full load.

Figure 9. Comparison of the Han HTCs and heat fluxes for different spark timings at 50% load.
Figures 8 and 9 show the histories of the HTCs and heat fluxes obtained from the Han model for the spark timing variations at the full load and 50% load at the 2000 rpm. At this speed, the peak values of the Han HTCs and heat fluxes for the spark timings of 17.5, 20, and 22.5°CA BTDC at full load are 2071, 2093, and 2108 W/m² K, and 0.60, 0.617, and 0.627 MW/m², respectively; whereas, at 50% load they are 1935, 1943, and 1985 W/m² K, and 0.522, 0.528, and 0.545 MW/m², respectively. In order to be seen easily, all peak values of the heat fluxes and HTCs are listed in Table 3.

As can be seen in the above figures, in-cylinder HTCs and heat fluxes of the SI air-cooled engine change with the variations of spark timing at the constant engine speed and load. Taking spark timing earlier or later with respect to the MBT spark timing at constant engine operating conditions causes the charge pressure and temperature to change near the TDC. Accordingly, in all models, earlier spark timing leads to a slight increase in the peak heat fluxes and HTCs, and it leads the peak values to happen at earlier CA than those of MBT spark timing. On the other hand, later spark timing leads to a decrease in the peak heat fluxes and HTCs, and it leads the peak values to happen at later CA than those of MBT spark timing. As for the load, the larger the load, the more the heat will be rejected to the combustion chamber walls. Since the increasing load means more fuel-rich mixture in the combustion chamber, the energy of the compressed mixture would increase and thereby increase the heat transfer to the cylinder walls. Furthermore, the histories of the HTCs and heat fluxes obtained from the Woschni, Hohenberg, and Han models are illustrated to be on the same line until the spark instant in the compression period because the engine operating parameters are constant until that point, but they would vary with the spark timings.

At the 50% load, the peak heat flux and HTC values nearly remain constant at the spark timing variations from 20 to 17.5°CA BTDC for each model, but at the full load the corresponding variations are comparatively noticeable.

In the middle of the expansion stroke, at the same crank angle, it can be said that the retarded spark timing gives higher HTC and heat flux compared with the advanced spark timing for all heat transfer models. However, from the spark instant until the peak HTC and heat flux points, it can be said that the advanced spark timing gives higher HTC and heat flux compared with the retarded spark timing for all the heat transfer models.

<table>
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<tr>
<th>Spark timing (°CA BTDC)</th>
<th>Full load</th>
<th>50% load</th>
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<tbody>
<tr>
<td>h (W/m² K)</td>
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<td></td>
</tr>
<tr>
<td>Han</td>
<td>2071</td>
<td>2093</td>
</tr>
<tr>
<td>Woschni</td>
<td>1577</td>
<td>1606</td>
</tr>
<tr>
<td>Hohenberg</td>
<td>1482</td>
<td>1521</td>
</tr>
<tr>
<td>q (MW/m²)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Han</td>
<td>0.60</td>
<td>0.617</td>
</tr>
<tr>
<td>Woschni</td>
<td>0.45</td>
<td>0.47</td>
</tr>
<tr>
<td>Hohenberg</td>
<td>0.43</td>
<td>0.45</td>
</tr>
</tbody>
</table>
5. CONCLUSION

In this study, the effect of spark timing and load variations on the heat transfer between gas and in-cylinder wall of an air-cooled, four-stroke, single cylinder SI engine run at 2000 rpm was investigated by using the Woschni, Hohenberg, and Han models. The conclusions given below are summarized from the present work.

When the spark timing is advanced from its original timing (spark timing for MBT), the in-cylinder HTC and heat flux increase slightly and their peak values occur at earlier crank angle than that of the original timing at constant load and speed conditions. In contrast to advanced timing, when the spark timing is retarded from its original timing, the in-cylinder HTC and heat flux decreases slightly and their peak values occur at later crank angle than that of the original timing.

The in-cylinder HTCs and heat fluxes at constant spark timing vary with the changing of engine load. With increasing the load, both the HTC and heat flux magnitudes increase more than magnitudes occurring with variations of spark timing.

The Han model gives higher HTC and heat flux compared with the Woschni and Hohenberg models under the same engine operating conditions.

Hohenberg model overestimates the HTC and heat flux at certain sections of compression and exhaust period, while it underestimates peak HTC and heat flux at only peak regions compared with the Woschni model.

From the beginning of spark timing toward the peak point, each model gives higher HTC and heat flux relative to the original timing at the advanced spark timing, but at the retarded spark timing corresponding HTC and heat fluxes have lower values. Toward the end of the compression stroke, each model gives lower HTC and heat flux values relative to the original timing at advanced spark timing, while at the retarding spark timing corresponding HTC and heat fluxes have higher values.

REFERENCES