Abstract

Engine optimization requires a good understanding of the in-cylinder heat transfer since it affects the power output, engine efficiency and emissions of the engine. However little is known about the convective heat transfer inside the combustion chamber due to its complexity. To aid the understanding of the heat transfer phenomena in a Spark Ignition (SI) engine, accurate measurements of the local instantaneous heat flux are wanted. An improved understanding will lead to better heat transfer modelling, which will improve the accuracy of current simulation software. In this research, prototype thin film gauge (TFG) heat flux sensors are used to capture the transient in-cylinder heat flux within a Cooperative Fuel Research (CFR) engine. A two-zone temperature model is linked with the heat flux data. This allows the distinction between the convection coefficient in the unburned and burned zone. The experimental time-resolved convection coefficient is then calculated by deriving the moment of flame arrival. The convection coefficient contains all the information of the driving force of the convective heat transfer. The work focuses on the effect of the flame passage on the convective heat transfer.

Introduction

A better understanding of the heat transfer phenomena inside the combustion chamber is key towards improving engine efficiency. The knowledge of the in-cylinder heat loss is gaining importance due to the trend of downsizing which increases the engine load and hence the heat loss. According to [1] up to 10% and according to [2] up to 15% of the fuel energy is lost due to in-cylinder heat transfer. Although an important process, recent heat transfer research is lacking. This can be attributed to the difficulty of performing accurate heat transfer measurements inside the combustion chamber. This environment is corrosive and characterized by high temperature and pressure oscillations demanding a robust and fast heat flux sensor.

The main mode of heat transfer in an SI engine is convection. Due to the absence of soot particles no radiative heat transfer takes place except in diesel engines[1]. This means that the heat transfer from the working gases to the cylinder walls is mainly driven by the in-cylinder gas motion and the gas properties of the fluid. The convective heat transfer can be expressed through the convection coefficient \( h \) if the heat transfer is assumed to be quasi-steady. This coefficient can be calculated using Equation (1).

\[
 h = \frac{q}{T_g - T_w}
\]  

(1)

Where \( q \) is the measured heat flux, \( T_w \) the measured wall temperature and \( T_g \) the gas temperature. The wall temperature and the heat flux are measured and calculated using thin film Resistance Temperature Detectors (RTDs). These heat flux sensors have also been used in [1, 3, 4, 5].

The heat transfer models used in current engine simulation software are mostly based on the models of Annand [6] or Woschni [7]. These models have been developed for a one-zone combustion model. This means the gas temperature is a bulk averaged value. The use of a bulk value implies that the effect of the flame propagation cannot be studied. However, other heat transfer models have been developed for a two-zone combustion model such as the model of Morel et al. [8]. Bargende [9] also calculated the burned and unburned gas temperatures. However he then took a weighted effect of the difference between both temperatures to model the influence of the combustion resulting in a spatially averaged model (one-zone). In this work GT-POWER simulation software [10] will be used to determine the gas temperature in the two zones: burned and unburned. The two gas temperatures and the heat flux will be used to calculate a local experimental convection coefficient. This can be used to study the effect of the flame passage on the convective heat transfer. The
influence of the flame propagation on the convective heat transfer has never been investigated. Only Heinle et al. [11] stated that the flame propagation influences the gas velocity, but did not go into detail. The flame propagation velocity might indeed affect the heat transfer through an effect on the local flow velocity.

Both the model of Annand and Woschni are based on the Polhausen equation which is based on the Reynolds analogy [12]. This analogy describes the analogous behavior of heat and momentum transfer. The Polhausen equation describes the forced convective heat transfer over a flat plate. Annand was the first to propose a dimensionless heat transfer equation based on the Polhausen equation, see Equation (2). The convective heat transfer is expressed in a dimensionless way using the Nusselt number (Nu=h·L/k, with k the thermal conductivity and L the characteristic length). The Nusselt number is written as a function of the Reynolds number (Re=V·L/ν, with V the characteristic velocity and ν the kinematic viscosity) and the Prandtl number (Pr=ν/α, with α the thermal diffusivity), as follows:

\[ \text{Nu} = a \text{Re}^b \text{Pr}^c \]  

\(2\)

Both Annand and Woschni lumped the Prandtl number into the coefficient a. In this work the gas properties are calculated using mixing rules described in [13] using the gas properties of the pure components, which were determined as a function of gas temperature with polynomials from the DIPPR database [14]. The gas properties will be influenced by the use of a one or two-zone gas temperature. In this work the Reynolds and Prandtl number will be used to investigate the effect of the flame passage on the convective heat transfer.

**Experimental Set-Up**

The engine used in this research is a Waukesha CFR engine. This is a single cylinder, port fuel injected (PFI), four stroke engine with a standardized engine head. A cross section of the head is shown in Figure 1. The figure shows the location of the inlet valve and exhaust valve. Four different mounting positions denoted by P1-P4 can be seen. The orifices allow an easy mounting of the in-cylinder pressure sensor and heat flux sensor. The engine has a variable compression ratio and can be operated at two fixed engine speeds, 600 and 900 rpm. For the measurements in this work the compression ratio CR is set to 8.7 and the engine speed is 900 rpm. The specifications are shown in Table 1. The engine is connected to an AC engine to provide the load. The engine control unit is a Motec M4 which can be programmed to vary the injection (timing and duration) and ignition timing (IT). In this work the CFR engine is run on stoichiometric methane gas (CH\(_4\)) and the ignition timing is 22 °BTDC. The engine load at these operating conditions is 7.16 bar IMEP.

**Table 1. CFR engine properties.**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement volume</td>
<td>618.8 cc</td>
</tr>
<tr>
<td>Bore</td>
<td>83.06 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>114.2 mm</td>
</tr>
<tr>
<td>Connecting rod</td>
<td>254 mm</td>
</tr>
<tr>
<td>Engine speed</td>
<td>600/900 rpm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>4:1 ~ 14.75:1 (8.7)</td>
</tr>
<tr>
<td>Number of valves</td>
<td>2</td>
</tr>
<tr>
<td>IVO</td>
<td>10 °C at TDC</td>
</tr>
<tr>
<td>IVC</td>
<td>208 °C at TDC</td>
</tr>
<tr>
<td>EVO</td>
<td>501 °C at TDC</td>
</tr>
<tr>
<td>EVC</td>
<td>12 °C at TDC</td>
</tr>
</tbody>
</table>

In-cylinder pressure was measured using a water-cooled Kistler 701A piezoelectric sensor (mounted in position P4, see Figure 1). The absolute inlet and outlet pressure were measured using Kistler 4075A10 piezoresistive pressure sensors. K-type thermocouples are used to measure the inlet air, exhaust, coolant and oil temperatures.

All signals are sampled and acquired using a National Instruments PXI system. The in-cylinder pressure and heat flux signal are sampled with a resolution of 0.5 °CA. 100 consecutive cycles are recorded and averaged to deal with cyclic variation. A schematic of the data acquisition system can be found in [15].

**Thin Film Gauge Heat Flux Sensor**

The heat flux sensor used in this research is a Thin Film Gauge (TFG) heat flux sensor. These sensors are developed at the University of Oxford [16]. Thin film platinum resistors are deposited onto a Macor® substrate. The platinum thin films act as Resistance Temperature Detectors (RTDs). The resulting thickness of the platinum films is only 0.1µm which results in a low thermal mass and hence a high frequency response of up to 100 kHz [17]. The TFG sensors are placed onto a bolt which facilitates the mounting into the CFR engine. Three TFG sensors are mounted. The bolt is placed in position P2 (see Figure 1). The bolt is shown in Figure 2. The spark plug is mounted in P1. The distance between the spark plug and the central TFG on the bolt is 30 mm. The central TFG sensor is indicated with an arrow.

**Working Principle**

The electrical resistance of the platinum films varies linearly with temperature. This relationship is shown in Equation (3).
\[ R(T) = R_0[1 + a_0(T - T_0)] \]  

\[ R_0 \] is the resistance at a reference temperature \( T_0 \) (atmospheric temperature). The temperature sensitivity of TFG sensor is expressed using the temperature coefficient \( a_0 \). This coefficient is calibrated in a water bath as discussed in [18]. The TFGs are connected to a current source and amplifier. A constant current \( I_0 \) is sent through the gauges transforming Eq. (3) into Equation (4). A change in temperature results in a voltage change.

\[ T - T_0 = \frac{\Delta V}{a_0 V_{I0}} \]  

The sensitivity depends on the value of \( V_{I0} \) which is determined by the amplitude of the current \( I_0 \) and the reference resistance \( R_0 \).

**Signal Processing**

The derived instantaneous wall temperature from the TFG sensor needs to be transformed into the instantaneous wall heat flux. A typical example of a measured wall temperature trace for the operating point on \( \text{CH}_4 \) is shown in Figure 3. We assume that the heat conduction into the substrate equals the convective heat transfer from the working gases to the wall. The signal processing method that is used in this work to calculate the heat flux using the measured wall temperature is called the Finite Impulse Response (FIR) method. This method is described in detail in [19]. The platinum film and Macor® are treated as a linear time invariant system. The discrete impulse response (imp[n]) is calculated analytically. The heat flux is then calculated using the discrete convolution of this impulse response and the sampled wall temperature data, see Equation (4).

\[ q[n] = \text{imp}[n] \ast r[n] \]  

In this work a Three Pressure Analysis is used in GT-POWER [10] to calculate the unburned and burned gas temperature traces. The TPA matches the measured in-cylinder pressure to determine the burn rate profile and also performs an energy balance over the entire engine cycle. The heat transfer model used in GT-POWER is based on the model of Morel et al. [20] for spark ignition engines. This model is based on a flow model tracking the in-cylinder fluid motions.

The following figures in this paragraph are based on a measurement of a stoichiometric methane mixture at an engine speed of 900 rpm. Wide open throttle and an ignition timing (IT) of 22 °BTDC are used. This operating condition is used as an illustration of the implementation of the two-zone temperature model and the related calculation of the two-zone convection coefficient.

**Two-Zone Temperature Model**

To capture the effect of the flame propagation and the influence of the flame passage on the convective heat transfer a two-zone gas temperature is used. This was also suggested in the work of Morel et al. [8] and Bargende [9]. They calculated the gas temperature of the unburned and the burned zone separately.
We notice that the peak value in the case of a two-zone model is clearly higher. This can be explained by the fact that a one-zone gas temperature is a spatially averaged value. The TFG sensor does not sense the average bulk gas temperature (one-zone model) but will see a sudden transition in gas temperature due to the passing of the flame. The two-zone gas temperature trace is now used further to determine the two-zone convection coefficient and investigate the physical phenomena when the flame passes.

Next we check if the instantaneous switch from the unburned to the burned gas temperature is justifiable. Two assumptions will be tested.

The first assumption is that we look at the TFG as a point sensor (infinitesimally small). Since the flame has a certain thickness and speed, it takes a certain amount of time for the flame front to pass over the TFG sensor. This time is calculated to check if an instantaneous switch between the unburned and burned gas temperature at the moment calculated previously is justified. This is in contrast with the infinitesimally thin flame front used in the work of Bargende [2] and Heinle et al. [11]. The unburned gas temperature at flame arrival is 780 K and the in-cylinder pressure is 20 bar. Using CHEM1D [21] software the laminar flame thickness and laminar flame speed was calculated. Next we calculate how much degree crank angle it takes for the flame front to pass over the TFG sensor. Thus, we divide the flame thickness with the laminar flame speed (assuming the flame to locally be a thin, laminar-like front). This gives us 0.04 ms. This results in a crank angle window of 0.22 °CA with an engine speed of 900 rpm (=5400 samples/second). The results are summarized in Table 2.

Table 2. CHEM1D results for the flame thickness and laminar flame speed for a stoichiometric methane mixture at 780 K (unburned temperature) and a pressure of 20 bar. The resulting crank angle window for the flame front to pass over the TFG sensor is also included.

<table>
<thead>
<tr>
<th>parameter</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH₄ - air mixture (λ=1)</td>
<td></td>
</tr>
<tr>
<td>flame thickness (m)</td>
<td>3.13 x 10⁻³</td>
</tr>
<tr>
<td>laminar flame speed (m/s)</td>
<td>0.76</td>
</tr>
<tr>
<td>delta °CA</td>
<td>0.22</td>
</tr>
</tbody>
</table>

An instantaneous switch from the unburned to burned gas temperature trace at the moment of flame arrival is therefore justified under this assumption since the sampling resolution of the measured heat flux and pressure trace is 0.5 °CA. However we should take into account that the dimensions of the TFG sensor are a couple of magnitudes greater than the flame thickness. This assumption is therefore unlikely. However it would be valid in the case the TFG senses the flame front (i.e. its resistance changes even from small contact with the hot flame) even though only a small portion is into contact with it.

Next the one and two-zone convection coefficient using the one and two-zone gas temperature trace from Figure 5 are calculated using Equation (1). Figure 6 shows the comparison between the one-zone and two-zone convection coefficient traces (in solid black and dash-dotted line respectively) and also plots the accompanying heat flux trace (in red).

We notice that during the compression stroke both convection coefficients have similar trend. Before the moment of flame arrival the one-zone coefficient starts to decrease which can be explained by the one-zone gas temperature that starts to increase earlier in comparison with the two-zone gas temperature (see Figure 5). At the moment of flame arrival the two-zone convection coefficient trace drops fast due to the jump in gas temperature and then recovers. The difference in maximum amplitude is 988 W/m²K. This is caused due to the peak gas temperature shift between the two models. Around 400 °CA in the expansion stroke the two traces converge (similar to the two gas temperature traces), corresponding to the end of combustion and thus the disappearance of an unburned zone.

Next these experimental (local) one and two-zone convection coefficient traces are compared with the spatially averaged convection coefficient trace extracted from the simulation software (h flow), see Figure 7. We clearly see that the flow convection coefficient increases before the one and two-zone trace. This can be explained by the fact that the experimental convection coefficient traces start to increase when the flame front has passed the measurement position (the moment the heat flux traces increases). This is not the case for the spatially averaged trace which starts to increase steeply at the moment of combustion. The spatially averaged convection coefficient is larger than the local traces. This means that the convective heat transfer at the cylinder liner is lower than the heat transfer at other combustion chamber surfaces.
Figure 7. Comparison of the experimental (local) one-zone and two-zone convection coefficient traces with the convection coefficient trace from the simulation software, h flow.

To investigate the effect of the flame on the convective heat transfer (q trace) we now take a look at the two-zone gas temperature and convection coefficient traces. The start of the steep rise in the heat flux trace is mainly driven by the temperature difference seen by the TFG sensor, since the convection coefficient momentarily drops. The drop in convection coefficient can be attributed to a change in gas properties due to the increase in gas temperature. Demuynck et al. [15] stated that the convection coefficient is inversely proportional to the Prandtl number. The Prandtl number trace is plotted in Figure 8.

Figure 8. Prandtl number for the two-zone temperature model.

We can confirm the results from Demuynck et al [15]. The drop in convection coefficient is partially explained by the increase in the Prandtl number. After the drop, the convection coefficient rises again. This can be partially attributed to the increase in gas velocity after the flame front has passed and the lower Prandtl number.

Demuynck et al. also states that the Nusselt number increases with the Reynolds number if the definitions for V and L are taken from Annand [6]. The characteristic speed (V) is taken to be the mean piston speed, which is constant throughout the cycle. The characteristic length (L) is the bore diameter, also constant. The Reynolds number trace for the two-zone model is calculated and shown in Figure 9.

Figure 9. Reynolds number for the two-zone temperature model.

This sudden drop is due to the influence of the gas properties on the Reynolds number, which are influenced by the gas temperature. The sudden increase in gas temperature increases the kinematic viscosity and therefore lowers the Reynolds number. An important remark is that the influence of the local gas velocity through the Reynolds number on the convection coefficient cannot be analyzed due to the definition of the characteristic speed that is used, i.e. which is constant.

The second assumption is the one also used in Bargende [9] and Heinle et al. [11] which is that of an infinitesimally thin flame front. However, here we will try to incorporate the dimensions of the sensor itself. The dimensions of the TFG are 0.5 mm x 2 mm. In the worst case the flame front needs to travel 2 mm approximately for the burned zone to fully cover the sensor. The accompanying crank angle window is shown in Table 3.

Table 3. CHEM1D results for the laminar flame speed for a stoichiometric methane mixture at 780 K (unburned temperature) and a pressure of 20 bar.

<table>
<thead>
<tr>
<th>characteristic length (L)</th>
<th>CH4 + air mixture (A=1)</th>
</tr>
</thead>
<tbody>
<tr>
<td>size TFG (m)</td>
<td>0.5-2 x 10^{-3}</td>
</tr>
<tr>
<td>laminar flame speed (m/s)</td>
<td>0.76</td>
</tr>
<tr>
<td>delta °CA</td>
<td>3.8-14.5</td>
</tr>
</tbody>
</table>

Using these assumptions, an instantaneous switch from the unburned to the burned gas temperature around flame arrival is not justified since it takes 14.5 °CA for the flame to pass over the TFG sensor. These assumptions are valid if the TFG sensor needs to be fully covered to sense the gas temperature. Of course, this is a worst case scenario, in reality a sensor response can be expected well before the whole sensor is in contact with the burned zone. Instead of an instantaneous switch a cubic spline is used. The cubic spline will bridge the crank angle window and connect the unburned and burned gas temperature. This way the effect of the TFG sensor size is taken into account in the two-zone gas temperature trace. The comparison with the previous assumption is shown in Figure 10.
We can clearly see the smoother transition from the unburned to the burned gas temperature due to the longer duration for the flame front to pass over the TFG sensor. This will affect the two-zone convection coefficient. Figure 11 shows the comparison between the one and two-zone model if the gas temperature with the cubic spline is used.

If we compare Figure 11 with Figure 6 the same trends can be observed. The steep decrease in convection coefficient for the two-zone model is less pronounced due to the smoother transition. The same conclusions about the flame propagation can be drawn.

Summary/Conclusions
In this work the effect of the flame passage on the convective heat transfer is analyzed. A two-zone temperature model is implemented to include the effect of the burned and unburned gas temperature and hence the propagating flame front. The measured heat flux trace is used to determine the moment of the flame arrival and hence the switch from the unburned to burned gas temperature. This two-zone and the one-zone gas temperature traces are used to determine the experimental and local one and two-zone convection coefficient traces. The two-zone convection coefficient trace gives insight in the effect of the flame passage on the convective heat transfer. The analysis uses the Reynolds and Prandtl number. The drop in convection coefficient around flame arrival can be explained due to an increase in Prandtl number and decrease of the Reynolds number. For the Reynolds number the definitions for the characteristic velocity and length are taken the same as the model of Annand.

The start of the steep increase in heat flux around flame arrival can be explained by the jump in gas temperature since the convection coefficient drops at the moment of flame arrival. This means that the steep rise in local experimental heat flux is mainly temperature driven.

References


Contact Information
Thomas De Cuyper, Research fellow
Department of Flow, Heat and Combustion Mechanics,
Ghent University
Sint-Pietersnieuwstraat 41, B-9000 Gent, Belgium
thomas.decuyper@ugent.be
Tel: + 32 9 264 34 53 | Fax + 32 9 264 35 90

Definitions/Abbreviations
AC - alternating current
ATDC - after top dead center
BTDC - before top dead center

°CA - degree crank angle
CFR - cooperative fuel research
EVC - exhaust valve closure
EVO - exhaust valve opening
FIR - finite impulse response
IMEP - indicated mean effective pressure (bar)
IT - ignition timing
IVC - inlet valve closure
IVO - inlet valve opening
PFI - port fuel injection
rpm - rounds per minute
RTD - resistance temperature detector
SI - spark ignition
TFG - thin film gauge
TPA - three pressure analysis
h - convection coefficient
imp - impulse response
k - thermal conductivity (W/mK)
L - characteristic length (m)
Nu - Nusselt number
Pr - Prandtl number
q - heat flux (W/cm²)
R - resistance (Ω)
Re - Reynolds number
T - temperature (K)
V - characteristic speed (m/s)
Vr - voltage (V)
0 - reference
g - gas
w - wall
α - thermal diffusivity (m²/s)
αR - temperature sensitivity coefficient (1/K)
λ - air-to-fuel equivalence ratio
ν - kinematic viscosity (m²/s)