Evaluation of empirical heat transfer models using TFG heat flux sensors

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Abstract

Thermodynamic engine cycle models are used to support the development of the internal combustion engine (ICE) in a cost and time effective manner. The sub model which describes the in-cylinder heat transfer from the working gases to the combustion chamber walls plays an important role in the accuracy of these simulation tools. The heat transfer affects the power output, engine efficiency and emissions of the engine. The most common heat transfer models in engine research are the models of Annand and Woschni. These models provide an instantaneous spatial averaged heat flux. In this research, prototype thin film gauge (TFG) heat flux sensors are used to capture the transient in-cylinder heat flux behavior within a production spark ignition (SI) engine as they are small, robust and able to capture the highly transient temperature swings. An inlet valve and two different zones of the cylinder head are instrumented with multiple TFG sensors. The heat flux traces are used to calculate the convection coefficient which includes all information of the convective heat transfer phenomena inside the combustion chamber. The implementation of TFG sensors inside the combustion chamber and the signal processing technique are discussed. The heat transfer measurements are used to analyze the spatial variation in heat flux under motored and fired operation. Spatial variation in peak heat flux was observed even under motored operation. Under fired operation the observed spatial variation is mainly driven by the flame propagation. Next, the paper evaluates the models of Annand and Woschni. These models fail to predict the total heat loss even with calibration of the models coefficients using a reference motored operating condition. The effect of engine speed and inlet pressure is analyzed under motored operation after calibration of the models. The models are able to predict the trend in peak heat flux value for a varying engine speed and inlet pressure. Next, the accuracy of the models are tested for a fired operating condition. The calibrated coefficient using a motored operating conditions are inadequate to predict the heat loss under a fired operating condition.

Keywords: internal combustion engine, heat transfer, experimental, model, spark ignition.

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Nomenclature

Abbreviations

°CA degree crank angle

NO\textsubscript{x} oxides of nitrogen

ABDC after bottom dead centre

AE absolute error

ATDC after top dead centre

BBDC before bottom dead centre

BTDC before top dead centre

CFR Cooperative Fuel Research

EVC exhaust valve closure

EVO exhaust valve opening

FS full scale

HCCI homogeneous charge compression ignition

HFM heat flux microsensor

IMEP indicated mean effective pressure, [bar]

IT ignition timing

IVC intake valve closure

IVO intake valve opening

LTI linear time invariant

PFI port fuel injection

RTD resistance temperature detector

SI spark ignition

TDC top dead center

TFG thin film gauge

S sample
Greek symbols
\begin{itemize}
\item \( \alpha \) thermal diffusivity, \([m^2/s]\)
\item \( \lambda \) air-to-fuel equivalence ratio
\item \( \nu \) kinematic viscosity, \([m^2/s]\)
\item \( \rho \) density, \([kg/m^3]\)
\end{itemize}

Roman Symbols
\begin{itemize}
\item \( c_m \) average piston speed, \([m/s]\)
\item \( B \) cylinder bore, \([m]\)
\item \( c \) heat capacity, \([J/(kg \cdot K)]\)
\item \( h \) convection coefficient, \([W/(m^2 \cdot K)]\)
\item \( h_{imp} \) impulse response
\item \( k \) thermal conductivity, \([W/(m \cdot K)]\)
\item \( L \) characteristic length, \([m]\)
\item \( \text{Nu} \) Nusselt number
\item \( \text{Pr} \) Prandtl number
\item \( \dot{q}_s \) heat flux, \([W/cm^2]\)
\item \( q_{max} \) maximum heat flux
\item \( Q_l \) total cycle heat loss, \([J]\)
\item \( \text{Re} \) Reynolds number
\item \( T(x,t) \) temperature field, \([\degree C]\)
\item \( T_g \) gas temperature, \([\degree C]\)
\item \( T_{wall} \) wall temperature, \([\degree C]\)
\item \( V \) characteristic velocity, \([m/s]\)
\item \( V_c \) in-cylinder volume, \([m^3]\)
\end{itemize}
1. Introduction

The understanding of the heat loss occurring in the combustion chamber is key in improving engine efficiency. The heat loss affects the power output, the efficiency and engine out emissions. Up to 10% of the fuel energy is lost due to in-cylinder heat transfer according to [1]. The heat transfer phenomena are complex. The rapid changes in gas temperature, pressure and velocity field contribute to its complexity resulting in a highly transient and spatial nature of the heat transfer.

Heat transfer measurements inside the combustion chamber pose a challenge in instrumentation due to the harsh environment. In this work, thin film gauge heat flux sensors are implemented at different locations in the combustion chamber. The spatially measured heat transfer database is a valuable contribution to existing literature.

The heat loss in SI engines is mainly driven by convective heat transfer. No radiative heat transfer takes place due to the absence of soot particles which can radiate heat [1]. The propagating flame and highly turbulent flow characterize the convective heat transfer under fired operation [2]. Under motored operation the heat transfer is mainly driven by the bulk gas flow and temperature.

Simulations tools are used to optimize internal combustion engines. It is clear that the sub model describing the convective heat transfer will greatly affect the accuracy of the engine model. The sub model is used to calculate and predict the total amount of heat that is lost to the cylinder walls during each calculation step of the engine cycle simulation. For modelling purposes it is assumed that the convective heat transfer is quasi steady. Most heat transfer models used in simulation software are based on the models of Annand [3] or Woschni [4] which are based on the Reynolds analogy [5]. Other heat transfer models are mostly derived from these models. These models predict the spatially averaged instantaneous heat flux. These models have been proven to fail for alternative fuels like hydrogen [6] and for alternative combustion modes such as HCCI operation [7]. Annand and Woschni suggest that a single calibration for every engine is sufficient to predict the heat transfer for varying engine conditions. This hypothesis will be checked for heat transfer measurements in a production engine by calibrating the models under motored operation, checking their accuracy for varying engine settings and checking their accuracy for fired operation. The implications of using a spatially averaged heat transfer model will be investigated by analyzing the spatial variation in heat flux under motored and fired operation.

2. Experimental method

2.1. Engine setup

The internal combustion engine used in this research is a Volvo B4184S SI engine. This is a four in-line cylinder 1.8l engine. The engine is equipped with a modified port fuel injection (PFI) system, using two fuel rails and two sets of port fuel injectors capable of injecting both liquid and gaseous fuels. A MoTeC M800 engine control unit (ECU) can be used to control the engine parameters. The characteristics of this engine are shown in table 1. The set-up is illustrated in Figure 1.

Crank angle measurements were performed with a Kistler COM 93218 crank angle encoder. Pressure measurements were performed in both the cylinder and the inlet
Table 1: Geometrical properties and valve timing of the engine

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>83 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>83 mm</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>152 mm</td>
</tr>
<tr>
<td>Swept volume</td>
<td>1796 cm³</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>10.3:1</td>
</tr>
<tr>
<td>Number of valves per cylinder</td>
<td>4</td>
</tr>
<tr>
<td>IVO</td>
<td>20 °CA BTDC</td>
</tr>
<tr>
<td>IVC</td>
<td>70 °CA BTDC</td>
</tr>
<tr>
<td>EVO</td>
<td>60 °CA BBDC</td>
</tr>
<tr>
<td>EVC</td>
<td>90 °CA ATDC</td>
</tr>
</tbody>
</table>

Figure 1: Volvo engine setup

The in-cylinder pressure was measured using an instrumented spark plug, equipped with a Kistler type 6118AFD13 piezoelectric pressure transducer. The inlet manifold absolute pressure is measured using a Kistler type 4075A10 piezo-resistive pressure transducer. Temperature measurements were performed on different locations of the engine. Thermocouples are installed in the inlet and outlet port of the instrumented cylinder. The air-to-fuel equivalence ratio (λ) is measured by means of a Bosch LSU 4.2 lambda sensor situated in the exhaust and can also be read by an Innovate LM-2 air-fuel reading unit. The air flow through the inlet manifold is measured by a Bronkhorst F-106BZ mass air flow sensor. All sensor signals are read with an NI CompactDAQ data acquisition system. Data pre-processing was done by a LabVIEW program and final data processing is performed by MATLAB scripts. Table 2 gives an overview of the measurement errors of the equipment.
Table 2: The accuracy of the measurement equipment

<table>
<thead>
<tr>
<th>Variable</th>
<th>Device</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>In-cylinder pressure</td>
<td>Kistler 6118AFD13</td>
<td>±1%</td>
</tr>
<tr>
<td>Intake manifold pressure</td>
<td>Kistler 4075A10</td>
<td>±0.03 bar</td>
</tr>
<tr>
<td>Air flow rate</td>
<td>Bronkhorst F-106BZ</td>
<td>±0.4%FS</td>
</tr>
<tr>
<td>Atmospheric temperature</td>
<td>ATAL TRP232-102D</td>
<td>±0.4 ºC</td>
</tr>
<tr>
<td>Atmospheric pressure</td>
<td>ATAL TRP232-102D</td>
<td>±130 Pa</td>
</tr>
</tbody>
</table>

2.2. Thin Film Gauge heat flux sensor

The thin film gauge (TFG) heat flux sensor was constructed at the Osney Thermo-Fluid laboratory of the University of Oxford. The TFG sensor is a resistance temperature detector (RTD) type sensor. The sensor is used to measure the instantaneous wall temperature. The sensor consists out of a thin film of platinum which is painted and baked onto an insulating substrate. Platinum is stable in oxidizing environments which makes it a suitable choice for this application. The resulting platinum thin film has a thickness in the order of 0.1 µm which gives the sensor a low thermal mass and hence a high frequency response of the TFG sensor of upto 100 kHz. The effect of the sensor on the in-cylinder gas flow is negligible. The insulating substrate used is the machinable glass ceramic Macor®. A schematic section of the construction of the sensor can be seen in Fig. 2 where \( \dot{q}_s \) indicates the wall heat transfer which is calculated using the wall temperature \( T_{wall} \) measured with the TFG sensor.

\[
x = 3.648 \cdot \sqrt{\alpha \cdot t}
\]  

(1)

Figure 2: Schematic drawing of a TFG heat flux sensor
In Fig. 3, the implementation of the TFGs can be seen. Three different zones (squish zone surface, end gas zone and inlet valve surface) of the combustion chamber are instrumented to capture the spatial variation in heat transfer. A pocket is machined at each location to mount a Macor® insert with painted platinum thin films. Each zone is instrumented with 5 TFG sensors. The squish zone is used to induce turbulence at the end of the compression stroke to enhance fast combustion and improve engine efficiency. Air is pushed radially inwards when a part of the piston face approaches the cylinder head closely [10]. The end zone is located furthest away from the spark plug (at the cylinder liner). The inlet valve surfaces represent a large part of the cylinder head surface making it a interesting surface for instrumentation.

Figure 3: Implementation of the thin film gauge heat flux sensors, showing 3 instrumented zones in the combustion chamber

The resistance of the platinum TFG has a linear relationship with temperature and is expressed through the temperature sensitivity coefficient $\alpha_0$. The linear relationship is given in equation 2. The suffix 0 represents a reference condition which is selected to be at atmospheric temperature.

$$R = R_0[1 + \alpha_0(T - T_0)] \quad (2)$$

The TFGs are connected to a current source and amplifier which sends a constant current $I_0$ through the gauges transforming eq. 2 into eq. 3. It can be seen that the sensitivity is linearly proportional to the reference $V_0$. This reference voltage is however limited due to ohmic heat dissipation which could result in a temperature offset error. $V_0$ is taken to be 250 mV after performing an ohmic heating test [11]. The voltages are read with an NI 9220 cDAQ module of National Instruments. This module allows a simultaneous sample rate of 100 kS/s over 16 differential voltage channels.

$$\frac{\Delta V}{V_0} = \alpha_0 \cdot \Delta T \quad (3)$$
The TFGs measure the instantaneous wall temperature of the Macor® substrate. This temperature trace is then used to calculate the instantaneous surface heat flux $\dot{q}_s$. The signal processing method used in this paper is the Finite Impulse Response (FIR) method described in detail in [12]. This method is shown to be advantageous over the Fourier method, described in [13]. The FIR method is more accurate and faster. The platinum thin film and Macor® are treated as a Linear Time Invariant system (LTI). The impulse response $h_{imp}$ of the LTI is calculated analytically by solving the one dimensional heat conduction equation 4. The impulse response $h_{imp}$ is a function of the material property and the thermal product $\sqrt{\rho ck}$ where $\rho$ is the density, $c$ is the heat capacity and $k$ is the thermal conductivity of the substrate. $T(x,t)$ is the temperature of the Macor® substrate at a given depth $x$ (see Fig. 2) and time $t$.

$$\frac{\partial T(x,t)}{\partial t} = \alpha \cdot \frac{\partial^2 T(x,t)}{\partial x^2}$$

(4)

The transient part of the surface heat transfer is then calculated by calculating the discrete convolution of the impulse response $h_{imp}$ with the sampled instantaneous wall temperature $T_{wall}$ using the MATLAB fftfilt command, see equation 5. An extra boundary condition is needed to calculate the steady state part of the surface heat flux. We assume that there is no convective heat transfer from the working gases to the wall when the gas temperature ($T_g$) is equal to $T_{wall}$. This way the total surface heat flux $\dot{q}_s$ is calculated.

$$\dot{q}_{trans} = fftfilt(h_{imp}, T_{wall})$$

(5)

To perform accurate measurements the temperature sensitivity $\alpha_0$ and the thermal product $\sqrt{\rho ck}$ need to be calibrated. A static calibration method is used to calibrate $\alpha_0$ and a dynamic one for the calibration of $\sqrt{\rho ck}$, the calibration process is described in detail in [11].

We assume that convective heat transfer from the working gasses to the substrate equals the conductive heat transfer into the substrate calculated using eq. 5.

A thorough error analysis is performed using the methods described in [14] to assess the experimental results. The analysis starts with the determination of the errors on the measured variables. Then, the propagation of these errors will be investigated to obtain the experimental uncertainty on the calculated variables. The following general equation is used to calculate the propagation of the errors of variables $a$, $b$ and $c$, $X$ being a random function of $a$, $b$ and $c$:

$$AE_x = \sqrt{(\frac{\partial f}{\partial a} AE_a)^2 + (\frac{\partial f}{\partial b} AE_b)^2 + (\frac{\partial f}{\partial c} AE_c)^2}$$

(6)

The maximum measurement uncertainties are shown in Table 3 (see Table 4).
Table 3: The maximum relative errors on the measured and calculated variables of measurement 1, see Table 4

<table>
<thead>
<tr>
<th>Variable</th>
<th>Symbol</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat flux</td>
<td>( q_s )</td>
<td>±4%</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>( T_{wall} )</td>
<td>±5%</td>
</tr>
<tr>
<td>Gas temperature</td>
<td>( T_g )</td>
<td>±5%</td>
</tr>
<tr>
<td>Air flow rate</td>
<td>-</td>
<td>±5%</td>
</tr>
<tr>
<td>Total cycle heat loss</td>
<td>( Q_l )</td>
<td>±5%</td>
</tr>
<tr>
<td>Convection coefficient</td>
<td>( h )</td>
<td>±12%</td>
</tr>
</tbody>
</table>

3. Empirical heat transfer models

In this section a short overview of the heat transfer models used in simulation software for Spark Ignition (SI) engines is given. In the next paragraph these heat transfer models will be evaluated for heat flux measurements performed in the Volvo engine. The heat transfer models discussed here assume that the heat transfer is quasi-steady. Then the convective heat transfer can be described by a convection coefficient \( h \), see equation 7. \( T_g \) is the bulk gas temperature and \( T_{wall} \) is the wall temperature.

\[
q = h \cdot (T_g - T_{wall}) \tag{7}
\]

The heat transfer model needs to predict the total amount of heat loss and peak heat flux during the engine cycle. A correct prediction of the total amount of heat loss is needed to solve the energy equation. The peak heat flux influences the peak gas temperature which in turn has a great effect on emissions formation such as thermal \( NO_x \) [1].

The discussed heat transfer correlations are based on the Polhausen equation which is based on the Reynolds analogy. This analogy describes the analogous behavior of heat and momentum transfer. The Polhausen equations describes the forced convective heat transfer over a flat plate [5]. Annand [3] proposed a dimensionless consistent equation based on the Polhausen equation by keeping its form and finding the appropriate coefficients \( a \), \( b \) and \( c \), see eq. 8. The heat transfer is represented by the Nusselt number \( (Nu = h \cdot L/k) \) as a function of the Reynolds \( (Re = V \cdot L/\nu, \nu \) is the kinematic viscosity) and Prandtl \( (Pr = \nu/\alpha) \) number.

\[
Nu = a \cdot Re^b \cdot Pr^c \tag{8}
\]

The Prandtl number is equal to 0.7 for most gases. Annand therefore lumped the Prandtl number into the coefficient \( a \) which can be used to scale the correlation to different engine setups. Annand suggested to use the bore diameter \( (B) \) and the mean piston speed \( (\epsilon_m) \) as the characteristic length \( (L) \) and speed \( (V) \) respectively. Equation 8 can then be rearranged into equation 9. Annand suggested a value of 0.7 for the coefficient \( b \) and a value between 0.35 and 0.8 for coefficient \( a \). The value of \( a \) varies widely with the intensity of the charge motion. Annand got these values by fitting equation 9 to the data of Elser [15].
A second widely used model is the model of Woschni [4] which is also based on equation 8. Woschni lumped the Prandtl number into parameter $a$ like Annand. Woschni made assumptions on the gas properties which are listed below.

- $\rho \sim \frac{p}{T}$
- $k \sim T^{0.75}$
- $\mu \sim T^{0.62}$

Equation 8 is transformed into equation 10 using the above assumptions. This gives an equation for the convection coefficient as a function of the cylinder pressure, temperature and the characteristic length and velocity. Coefficient $a_{wo}$ is equal to 0.013 and coefficient $b$ is equal to 0.8 these values are based on heat transfer correlations which describe the heat loss of internal flows in tubes.

$$h = a_{wo} \cdot B^{-0.2} \cdot \rho^{0.8} \cdot V^{0.8} \cdot T^{-0.53}$$ (10)

The same characteristic length, the cylinder bore $B$, is used as by Annand. Woschni however adapted the characteristic speed to account for the effect of combustion on the in-cylinder heat transfer by adding an extra term as a function of the pressure difference between a fired and a motored cycle. The characteristic speed is shown in equation 11.

$$V = c_{1} \cdot c_{m} + c_{2} \cdot \frac{V_{s} \cdot T_{r}}{p_{r} \cdot V_{r}} \cdot (p - p_{0})$$ (11)

With the following values for the coefficients:

- $c_{1} = 6.18$ during the scavenging period and $c_{1} = 2.28$ during the compression, combustion and expansion period
- $c_{2} = 0$ during the scavenging and compression period and $c_{2} = 3.24 \cdot 10^{-3}$ during the combustion and expansion period, [$m/s^\circ C$]
- subscript $r$ denotes a reference state where volume, pressure and temperature are known
- $p_{0}$ is the in-cylinder pressure under motored conditions

The experimental heat flux traces at the three different zones (squish, end, valve zone) will be compared with predictions using the correlations of Annand and Woschni. It was chosen to only evaluate these two correlations for several reasons. The first reason is that these models are widely used in commercial simulation software to predict heat flux traces in SI engines. Second, other models (e.g. [16, 17, 18]) have tuned the exponent of the pressure and temperature in equation 10 for a particular measurement set. These models are therefore no longer based on equation 8. Most models that are developed later use the models of Annand and Woschni as a basis.
4. Results

Table 4 shows the operational conditions that are used to evaluate the models. Measurement 1 in bold is used to calibrate the heat transfer model coefficients. This calibration is needed to tune the models for the engine. After calibration, the effect of the engine speed and inlet pressure on the heat loss will be investigated for motored operation. Next the models are tested for a fired operation condition with the same model coefficients.

Table 4: Overview of the measurements used for the evaluation

<table>
<thead>
<tr>
<th>measurement</th>
<th>operation</th>
<th>fuel</th>
<th>rpm</th>
<th>$p_{inlet}$[hPa]</th>
<th>$\lambda$</th>
<th>IMEP[bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>motored</td>
<td>air</td>
<td>1900</td>
<td>650</td>
<td>-</td>
<td>-0.63</td>
</tr>
<tr>
<td>2</td>
<td>motored</td>
<td>air</td>
<td>1900</td>
<td>350</td>
<td>-</td>
<td>-0.8</td>
</tr>
<tr>
<td>3</td>
<td>motored</td>
<td>air</td>
<td>1900</td>
<td>950</td>
<td>-</td>
<td>-0.4</td>
</tr>
<tr>
<td>4</td>
<td>motored</td>
<td>air</td>
<td>1200</td>
<td>650</td>
<td>-</td>
<td>-0.6</td>
</tr>
<tr>
<td>5</td>
<td>motored</td>
<td>air</td>
<td>2600</td>
<td>650</td>
<td>-</td>
<td>-0.6</td>
</tr>
<tr>
<td>6</td>
<td>fired</td>
<td>gasoline</td>
<td>1900</td>
<td>650</td>
<td>1</td>
<td>5.1</td>
</tr>
</tbody>
</table>

The reproducibility and reliability of the TFG heat flux sensors is tested first by comparing two different prototypes of instrumented inlet valves. The two designs can be seen in Fig. 4. Design 1 has been improved to withstand higher temperatures as the design failed during fired operation.

![TFG inlet valve design 1](image1.png) ![TFG inlet valve design 2](image2.png)

(a) TFG inlet valve design 1  (b) TFG inlet valve design 2

Figure 4: Different prototypes of instrumented inlet valves

Both designs are tested for the same operating condition (measurement 1 from Table 4). The heat flux traces derived from the two inlet valves are shown in Fig. 5. The traces are measured with a TFG from the center of the valve to ensure equal conditions. Since the error bars overlap, no significant difference in heat flux trace is measured with the two different valves confirming the reliability and repeatability of TFG heat flux measurements.
4.1. Motored operation

4.1.1. Spatial variation

The spatial variation between the three different measurement zones is first compared for measurement 1 (see Table 4). Figure 6 shows the heat flux traces for the closed part of the engine cycle measured with one of the TFG sensors mounted in each zone of the combustion chamber (end TFG 5, squish TFG 3, valve TFG 3). No significant difference is observed between the heat loss in the end and valve zone. However the heat flux trace measured in the squish zone differs significantly. First, the peak heat flux is lower compared to the other two zones. Second, a phase lag is present with the peak and decrease in heat flux occurring earlier for the squish zone. The peak squish velocity occurs slightly before Top Dead Center (TDC) [10]. Air is pushed radially inwards when the piston approaches TDC increasing the in-cylinder turbulence leading to a peak in heat flux slightly after TDC due to the gas flow inertia for the valve and end zone TFGs.

The total heat loss is calculated by multiplying the instantaneous heat flux with the instantaneous combustion chamber surface and integrating over the closed part of the engine cycle. The total heat loss using the experimental trace of valve TFG 3 is 12.8 J (see Table 5). The total heat loss calculated using squish TFG 3 (see Fig. 6) is 12.0 J. Even though a significant difference in heat flux (between the zones) is observed around TDC (Fig. 6), the total amount of heat loss does not differ significantly since it is within measurement uncertainty. This is mainly due to the low instantaneous combustion chamber surface around TDC. We can conclude that under motored operation a spatial difference in peak heat flux can be observed, this difference however does not result in a different calculated total heat loss.
4.1.2. Effect of engine settings

The models of Annand and Woschni are calibrated for the peak heat flux measured with one of the valve TFGs (valve TFG 3) for the reference measurement, measurement 1 in bold in Table 4. If the model predictions approach the shape of the heat flux trace then a good agreement in total heat loss between the experiment and model will be observed. These models are then evaluated for a variation in inlet pressure and engine speed. The parameters that are calibrated are the coefficient $a$ in the model of Annand and coefficient $c_1$ in the model of Woschni. Since no combustion takes place parameter $c_2$ is 0 and does not need to be calibrated. The tuned coefficient $a$ is 0.21 which is lower than the suggested minimum of 0.35. The calibrated $c_1$ value is 2.44 compared to the suggested value of 2.28 which is in the same order of magnitude.

Table 5 below shows the simulation results of Annand and Woschni for $q_{max}$ and $h_{max}$ and compares the total heat loss with (modified) and without (standard) calibration of the model coefficients for measurement 1. Even with the minimum value of 0.35 coefficient $a$, the model of Annand overestimates the peak values and total heat loss significantly. Woschni slightly underpredicts the $q_{max}$ but predicts the $h_{max}$ within the measurement uncertainty. The total heat loss even for the tuned models is a bad estimate since it is outside the measurement uncertainty. This is due to the fact that the shape of the simulated heat flux traces does not compare with the experimental trace.

<table>
<thead>
<tr>
<th></th>
<th>$q_{max}$ [W/cm²]</th>
<th>$h_{max}$ [W/m²K]</th>
<th>$Q_{l,\text{standard}}$ [J]</th>
<th>$Q_{l,\text{modified}}$ [J]</th>
</tr>
</thead>
<tbody>
<tr>
<td>exp.</td>
<td>19.7</td>
<td>306.8</td>
<td>12.8 J</td>
<td>12.8 J</td>
</tr>
<tr>
<td>Annand</td>
<td>64.1 %</td>
<td>63.1 %</td>
<td>121.2 %</td>
<td>34.8 %</td>
</tr>
<tr>
<td>Woschni</td>
<td>−5.3 %</td>
<td>−5.9 %</td>
<td>22.4 %</td>
<td>29.3 %</td>
</tr>
</tbody>
</table>
The tuned coefficients are now used for the simulations of the other motored measurements to check for the simulation accuracy in predicting the effect of inlet pressure and engine speed. The numerical values of the peak convection coefficient $h_{\text{max}}$ and total heat loss $Q_l$ are shown in Table 6, the measurement numbers are the same as in Table 4.

### Table 6: Overview of the simulations’ accuracy for motored operation

<table>
<thead>
<tr>
<th>meas.</th>
<th>$h_{\text{max}} [W/m^2K]$</th>
<th>$h_{\text{max,Annand}}$</th>
<th>$h_{\text{max,Woschni}}$</th>
<th>$Q_{\text{L,exp}} [J]$</th>
<th>$Q_{\text{L,Annand}}$</th>
<th>$Q_{\text{L,Woschni}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>306.8</td>
<td>0 %</td>
<td>0 %</td>
<td>12.8</td>
<td>34.8 %</td>
<td>29.3 %</td>
</tr>
<tr>
<td>2</td>
<td>142.6</td>
<td>30.1 %</td>
<td>18.3 %</td>
<td>8.4</td>
<td>89.7 %</td>
<td>64.9 %</td>
</tr>
<tr>
<td>3</td>
<td>462.1</td>
<td>-8.7 %</td>
<td>-3.3 %</td>
<td>17.8</td>
<td>7.4 %</td>
<td>9.3 %</td>
</tr>
<tr>
<td>4</td>
<td>193.6</td>
<td>7.0 %</td>
<td>0.7 %</td>
<td>9.5</td>
<td>39.6 %</td>
<td>25.8 %</td>
</tr>
<tr>
<td>5</td>
<td>402.7</td>
<td>-0.7 %</td>
<td>3.6 %</td>
<td>16.4</td>
<td>24.0 %</td>
<td>24.4 %</td>
</tr>
</tbody>
</table>

First, the effect of the inlet pressure (measurements 1-2-3) on the convection coefficient $h$ is shown in Fig. 7. The solid lines are the experimental convection coefficient traces. The solid blue line represents measurement 1. The simulation results of Annand and Woschni are plotted in a dotted and dash dotted line respectively. A higher inlet pressure results in a higher convection coefficient. A higher inlet manifold pressure leads to a higher pressure drop over the inlet valves during the intake stroke resulting in an increase in turbulence and hence an increase in convection coefficient. A higher inlet pressure results in a higher peak pressure around TDC due to an increase of trapped air. The higher convection coefficient results in a higher heat flux $\dot{q}_t$. Overbye et al. [19] and Dao et al. [20] confirmed this result. We see that the models can predict the trend in convection coefficient with a varying inlet pressure. The simulations of Annand and Woschni however overpredict the convection coefficient during the compression and expansion stroke. The simulations overpredict the peak convection coefficient for 350 hPa.
Second, the effect of the engine speed is analyzed (measurement 1-4-5). Figure 8 shows the experimental and simulation traces for the convection coefficient. A higher engine speeds leads to a higher in-cylinder gas flow and hence a higher convection coefficient. This is confirmed by [21, 20, 22]. The peak values are predicted by both simulations since their relative error is smaller than the measurement error on the peak convection coefficient. The models of Annand and Woschni are able to predict the peak heat flux $q_{\text{max}}$ for different motored operating conditions after calibration. However the total amount of heat loss can not be predicted accurately due to the overprediction during compression and expansion.

Figure 7: The effect of inlet pressure on the simulation results under motored operation (measurements 1-2-3)

Figure 8: The effect of engine speed on the simulation results under motored operation (measurements 1-4-5)
4.2. Fired operation

Next, we will test if the previous calibration of coefficient \( a \) of the model of Annand results in a good simulation for fired operation. The heat loss under fired operation is analyzed for the operating condition 6 in Table 4 which represents a part load operating point. The ignition timing (IT) is 22°CA BTDC. These engine settings result in a load of 5.1 bar IMEP. Figure 9 compares the simulation results with experimental traces from the squish and valve area using the tuned coefficients (from section 4.1) and the proposed \( c_2 \) value from the work of Woschni \( (c_2 = 3.24 \cdot 10^{-3}) \). The TFGs mounted in the end gas zone region unfortunately did not function anymore due to failure. Two traces are plotted for the squish (black) and the valve zone (red).

![Figure 9: The simulations of Annand and Woschni in comparison with the experimental traces from the squish and valve zone for fired operation (measurement 6) with the calibrated coefficient for measurement 1.](image)

We can clearly see the effect of flame propagation. Figure 3 shows that the valve is closer to the spark plug than the squish TFG sensors. The flame arrives earlier at the valve than the squish zone resulting in an earlier rise in heat flux. This is clearly visible in Figure 9 and confirms the results observed in [2, 23]. The cited authors ascribe this difference in heat flux rise to the sudden temperature increase. The peak heat flux is a magnitude larger than under motored operation \( (179.7 \text{ W/cm}^2 \text{ compared to 19.7 W/cm}^2) \). There is clear spatial variation in peak heat flux when comparing both zones. This was also observed in [23, 24] and must be due to local differences in gas velocity, turbulence and gas temperature. No spatial variation can be observed between the TFG sensors in the same zone. This means that one TFG trace represents the heat flux in its zone.

Both simulations overpredict the experimental heat flux with the tuned coefficients for measurement 1. The model of Annand performs badly even though its coefficient \( a \) should only be calibrated once for each engine. Woschni has a second parameter that can be tuned, coefficient \( c_2 \) which determines the effect of the combustion process on the characteristic velocity \( V \). Next, coefficient \( a \) of the model of Annand is recalibrated.
together with coefficient $c_2$ of the model of Woschni. The coefficients are tuned to predict the peak heat flux of valve TFG sensor 3. The resulting values for $a$ and $c_2$ are 0.12 and $9.89 \cdot 10^{-4}$. Coefficient $a$ is again smaller than the minimum suggested by Annand and coefficient $c_2$ is an order of magnitude smaller than the value used in Woschni’s work. This means that the effect of combustion on the characteristic speed was overestimated significantly. Figure 10 shows the results after calibration of $a$ and $c_2$. The peak heat flux of valve is predicted for valve TFG 3 but simulations overpredict the heat loss during the compression and expansion. The measurements in the squish and end zone are not representative for the global heat transfer.

The numerical values are shown in Table 7 for the valve TFG 3 sensor. $Q_{\text{standard}}$ is the total heat loss using the coefficients of section 4.1 and $Q_{\text{modified}}$ using the tuned coefficient for fired operation. All simulations are outside the measurement uncertainty. The total heat loss is best predicted with the simulation of Woschni after calibration (relative error of 20.6 %). Both models are unable to capture the total amount of heat loss when calibrated for the peak heat flux. This is due to the underlying assumptions. The model of Woschni can predict the heat flux trace after passing of the flame front accurately for both zones. However the steep rise in heat flux due to the propagating flame front is not captured leading to a significant difference in the calculated total heat loss. The experimental total heat loss using squish TFG 2 is equal to 83.8 J and is not within measurement uncertainty of the value calculated with valve TFG 3 (93.9 J). This is an expected result because of the spatial variation (see Figure 9).

![Figure 10: The simulations of Annand and Woschni in comparison with the experimental traces from the squish and valve zone for fired operation (measurement 6) with retuned coefficients](image-url)
Table 7: Overview of the simulations’ accuracy for measurement 6 for valve TFG 3, fired operation

<table>
<thead>
<tr>
<th></th>
<th>$q_{\text{max}} [W/cm^2]$</th>
<th>$h_{\text{max}} [W/m^2K]$</th>
<th>$Q_{\text{1,standard}} [J]$</th>
<th>$Q_{\text{modified}} [J]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>exp.</td>
<td>179.7</td>
<td>590.8</td>
<td>93.9</td>
<td>93.9</td>
</tr>
<tr>
<td>Annand</td>
<td>75.4 %</td>
<td>80.8 %</td>
<td>172.3 %</td>
<td>55.4 %</td>
</tr>
<tr>
<td>Woschni</td>
<td>93.9 %</td>
<td>105.4 %</td>
<td>98.0 %</td>
<td>20.6 %</td>
</tr>
</tbody>
</table>

5. Conclusions

Prototype TFG heat flux sensors were mounted on several surfaces of the combustion chamber of a production engine. The spatial variation in heat flux inside the combustion chamber is analyzed by comparing the instantaneous heat flux traces in the three different zones of the chamber. It was shown that even under motored operation a significant difference in heat flux is observed between different measurement zones however this did not lead to a significant difference in the total amount of heat loss. The spatial variation under fired operation is mainly driven by the propagating flame front and thus by the distance relative to the spark plug. A difference in rise in heat flux and peak value is observed.

The models of Annand and Woschni were then evaluated. The models fail to predict the peak heat flux and total amount of heat loss without calibration of the models coefficients, which is expected. Next, the models were calibrated for the peak heat flux under a specific motored operation condition. Even after calibration the total amount of heat loss is not predicted accurately with relative errors up to 34.8 % for the model of Annand. The models overpredict the heat flux trace during the compression and expansion stroke. The trend in peak heat flux value for a variation in inlet pressure and engine speed could be predicted. However, the predictions on the total amount of heat loss were significantly different than the calculated value using the experimental traces. For the lowest inlet pressure (350 hPa) the predictions are worse than for the highest inlet pressure (950 hPa).

As expected both models fail to capture the effect of the flame passage. The predictions using the model of Woschni are better especially for fired operation due to the fact that the steep decrease, after the flame front has passed the sensor, is predicted more accurately. The overall heat loss calculated using Woschni still differs 20.6 % from the experimental value. This clearly suggest that a two-zone temperature model is needed to capture the effect of flame passage. This must lead to a better predictions of the total heat loss.

Appendix A

This appendix describes the calculation of the variables that have to be introduced into equations 9 and 10. First, the difference between the bulk gas temperature and wall temperature has to be known in both heat transfer models. The wall temperature is measured using the TFG sensor. The combustion gases are assumed to behave like ideal
gases. Therefore, the bulk gas temperature is calculated with the following equation of state: \( T_g = \frac{p \cdot V_c}{m \cdot R} \).

- The in-cylinder pressure \( (p) \) is measured and the volume \( (V_c) \) can be calculated out of the crank position.
- The mass \( (m) \) can only be determined during the closed part of the combustion cycle, being the sum of the measured incoming mass (air and fuel) and the residuals.
- The specific gas constant \( (R) \) at IVC can be calculated out of the mass average of the specific gas constants of the air, the fuel and the residual gases. This value is used until the beginning of the combustion. At the end of the combustion, \( R \) is equal to that of the combustion products. In between, the specific gas constant is calculated with a linear interpolation. The instant where the combustion begins and ends is determined with a rate of heat release analysis.

The thermal conductivity, kinematic viscosity and Prandtl number of the gas mixture have to be calculated at each instant for the model of Annand. The heat capacity and the dynamic viscosity are calculated on top of the thermal conductivity to determine the Prandtl number. These variables are all calculated as a function of the gas temperature in the same way as the specific gas constant (three zones: between IVC and beginning of the combustion, during the combustion and during the expansion period), using the mixing rules described in [25].

Woschni has converted the equation of the boundary layer theory so that it is only a function of pressure and temperature (besides the characteristic length and velocity). Consequently, it needs less data input. The measured cylinder pressure for the fired and motored case have to be filled in. IVC is taken as the reference state in the calculation of the characteristic velocity.

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**References**


