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LOCAL HEAT FLUX MEASUREMENT TECHNIQUE FOR INTERNAL COMBUSTION ENGINES

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Abstract. The heat transfer from the combustion gases to the cylinder wall affects the efficiency, emissions and power output of an internal combustion engine. Measuring the heat transfer requires a heat flux sensor inside the combustion chamber that has a short response time and is able to withstand the harsh conditions during combustion. In this work, a suitable sensor is introduced and the measured wall temperature, heat flux and convection coefficient are compared to those measured with a commercial sensor. It was found that both sensors measure the same convection coefficient, but a different wall temperature and heat flux. This is because the presence of the sensor in the combustion chamber wall affects these quantities. A method is proposed to cancel this effect and calculate the actual heat flux through the cylinder wall.

Keywords: Internal Combustion Engine, Heat Transfer, Experimental, Sensor

1. INTRODUCTION

The transportation sector is responsible for 14% of the worldwide greenhouse gas emissions (IPCC, 2014). This has led to the introduction of legislation (Council of EU, 2009) that limits the maximum allowable CO_2 emissions of passenger cars. Complying with this legislation, and the increasingly stringent legislation for air pollutants (Council of EU, 2007), while aiming for a low fuel consumption and retaining the required power output is challenging. The simultaneous optimisation of all the engine settings for these three targets during the development of a new internal combustion engine, has become increasingly complex due to the large amount of engine technologies present in modern engines. For this reason, engine simulation software is being used to speed up the development process. However, the accuracy of the simulations depends on the quality of the used models.

An important model used in the simulation software is the heat transfer model. This model predicts the heat transfer from the bulk gases in the combustion chamber to the combustion chamber wall. The heat transfer needs to be calculated every time step to solve the equations of mass and energy and it also directly affects the emissions, fuel consumption and power output of the engine. An accurate prediction of the heat transfer is even more important for low-temperature combustion engines (Onishi *et al.*, 1979; Noehre *et al.*, 2006), where the combustion is mainly determined by chemical kinetics. However, the heat transfer models that are currently being used (Annand, 1963; Woschni, 1967) have been found to provide inaccurate results when using alternative fuels with different properties compared to fossil fuels (Demuynck *et al.*, 2011) and for low-temperature combustion engines (Broekaert *et al.*, 2015). Developing an improved heat transfer model that is also accurate under these conditions, requires new measurements of the in-cylinder heat transfer. This means measuring the instantaneous, or crank-angle resolved, heat flux through the combustion chamber wall.

A direct measurement of the heat flux requires a sensor that is mounted inside the combustion chamber, in direct contact with the combustion gases. The sensor needs to have a high enough frequency response (in the order of kHz) and be able to withstand the high pressure and temperature during the combustion. In this work, the wall temperature, heat flux and convection coefficient obtained by two different sensors simultaneously are compared. The effect of the sensor's presence in the combustion chamber wall on the measured heat flux is discussed. A post-processing method is introduced that cancels the effect of the sensor's presence and calculates the actual heat flux through the cylinder wall.

2. EXPERIMENTAL EQUIPMENT

2.1 Engine setup

The engine used in this work is a Waukesha Cooperative Fuel Research (CFR) engine. This is a single cylinder, four stroke engine operating at a constant speed of 900rpm. It is equipped with a programmable MoTeC M4 Pro Engine Control Unit to control the injection timing and duration. The engine is converted to operate in Homogeneous Charge Compression Ignition (HCCI) mode. This means a lean premixed fuel-air mixture is introduced into the combustion chamber, which auto-ignites due to the temperature rise during the compression stroke. As a consequence, the combustion occurs nearly simultaneously across the entire combustion chamber and the heat transfer is close to homogeneous (Broekaert *et al.*, 2016). HCCI-operation is achieved by heating the intake air with a 6kW Osram Sylvania inline heater which is controlled by a Gefran 600 Temperature Controller to keep the temperature at the inlet within 0.5°C of the set value. The engine is fueled with n-heptane which is injected 180mm before the intake valve to achieve a homogeneous air-fuel mixture in the cylinder. A cross section of the cylinder is displayed in Fig. 1, showing the sensor mounting positions. The engine's operating conditions are listed in Table 1.

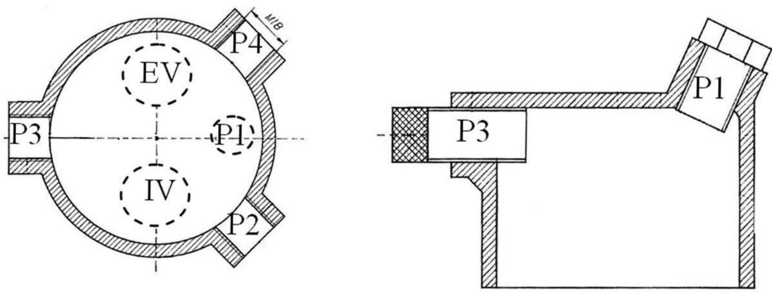

Figure 1. Cross section of the CFR engine, P1-4: possible sensor positions, IV: intake valve, EV: exhaust valve

Table 1. Operating conditions

Fuel	n-heptane
Compression ratio	9
Engine speed	900rpm
Inlet air temperature	100°C
Air-fuel equivalence ratio	3.25

The in-cylinder pressure is measured with a water-cooled Kistler 701A piezoelectric sensor (mounted in position P4). The inlet and outlet pressure are measured with two water-cooled Kistler 4075A10 piezoresistive pressure sensors. The in-cylinder pressure is referenced with the inlet pressure. The air flow is measured with a Bronkhorst F-106BZ flow sensor and the fuel mass flow rate is measured with a Bronkhorst mini Cori-Flow M13 coriolis mass flow meter. Finally, K-type thermocouples are used to measure coolant, oil and inlet and exhaust gas temperatures. A National Instruments PXI data acquisition system is used to sample the pressure and heat flux signals for 100 consecutive cycles. It is triggered by a crank angle encoder every 0.25°CA, resulting in a sampling rate of 21.6kHz. The other signals are averaged over time and acquired at a sampling rate of 1Hz.

2.2 Heat flux sensor

The wall heat flux and the wall temperature are measured with both a commercial thermopile sensor and an in-house built Thin Film Gauge (TFG) sensor mounted in the cylinder head. The thermopile sensor is mounted in position P1 (see Fig. 1) and the TFG sensor in position P3, in a way that both sensors' surfaces are flush with the combustion chamber wall and in direct contact with the combustion gases. The thermopile sensor is a Vatell HFM-7 sensor, that consists of a thermopile to measure the heat flux and a resistance temperature detector (RTD) to measure the wall temperature. It has a claimed response time of 17μs (Vatell Corporation, 2012). A Vatell AMP-6 amplifier is used as a current source for the RTD and as an amplifier for both output signals. The sensor outputs are directly correlated to their measured quantities. The heat flux and wall temperature are obtained by applying the calibration constants supplied by the manufacturer, without further signal processing.

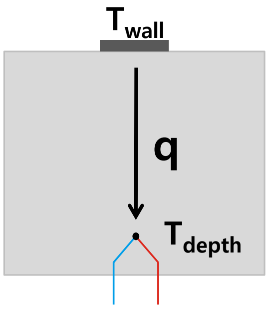

Figure 2. Schematic of the TFG sensor.

The TFG sensor is developed by the University of Oxford for heat flux measurements in gas turbines. It consists of a thin film RTD that measures the instantaneous wall temperature on an insulating Macor® substrate. A thermocouple is placed inside the substrate to measure the temperature at a known depth. A sketch of the construction is shown in Fig. 2. With the thin film RTD a response time of less than 10μs can be achieved (Piccini *et al.*, 2000). An HTA-3 amplifier is used as a current source for the RTD and as an amplifier for the output signal. The heat flux through the sensor is determined from the measured temperatures by solving the one-dimensional Fourier law (1) and Fourier's differential equation (2) (Welty *et al.*, 2001) using a Fourier transform. The one-dimensional heat transfer hypothesis was verified with a finite element analysis of the sensor.

$$q = -k\frac{dT}{dx} \tag{1}$$

$$\frac{\delta T(x,t)}{\delta t} = \alpha\frac{\delta^2 T(x,t)}{\delta x^2} \tag{2}$$

With q the instantaneous heat flux through the sensor, T the temperature of the substrate, x the distance from the sensor's surface, k the thermal conductivity of the substrate and α the thermal diffusivity of the substrate. The measured wall temperature and heat flux used in this work are both obtained by taking the ensemble average over 100 consecutive cycles to negate the effect of cyclic variation of the engine. The material properties of the substrate are obtained by the calibration process described in (De Cuyper *et al.*, 2015). The convection coefficient is determined according to Eq. 3.

$$h = \frac{q}{T_{gas} - T_{wall}} \tag{3}$$

With h the convection coefficient, T_{gas} the temperature of the combustion gases and T_{wall} the surface temperature. It is calculated for each engine cycle using the measured instantaneous heat flux, wall and gas temperature and subsequently averaged over 100 consecutive cycles. The instantaneous gas temperature is derived with the ideal gas law from the measured cylinder pressure, in-cylinder volume and flow rates of air and fuel. An error analysis was carried out according to the methods described in Taylor (1982). The worst case values of the relative errors are reported in Table 2. The experimental error of the convection coefficient is quite high, as it is comprises the error of the measured heat flux, wall temperature and gas temperature. The latter being determined by the error of the cylinder pressure and the mass flow.

Table 2. Experimental uncertainty

Variable	Relative error
Wall temperature	4%
Heat flux	5%
Convection coefficient	12%

3. EFFECT SENSOR PROPERTIES

Figure 3 shows the wall temperature and heat flux measured by both the HFM and the TFG sensor under the same engine operating conditions. It can be seen that the measured temperature and heat flux differ between the two sensors. The steady state temperature and the temperature swing at the surface are higher for the TFG sensor compared to the

HFM sensor. The heat flux measured by the TFG sensor is lower than the one measured by the HFM sensor. In previous work (Broekaert *et al.*, 2016) it was shown that the heat flux measured with the HFM sensor mounted at the different measurement locations is the same, due to the homogeneous nature of the heat transfer in the combustion chamber. This indicates that the presence of the sensors affects the measurements. Consequently, the measured temperature and heat flux are not the same as the actual temperature of the cylinder wall and the actual heat flux through the cylinder wall. This is because the sensors are constructed from another material than the engine block, with different thermal properties. When exposed to the same heat source, the sensors will take on a different thermal state than the cylinder wall, resulting in a different surface temperature and heat flux. The thermal properties that are responsible for this are listed in Table 3 for the engine block, made from cast iron, and the TFG sensor, made from Macor®. The thermal properties of the HFM sensor are unknown, but are expected to be closer to those of the engine block as the sensor is constructed largely from metal. The higher steady state temperature and temperature swing of TFG sensor can be attributed to the low thermal conductivity and thermal product of the ceramic substrate compared to metal. As a results of the higher wall temperature, the heat flux through the TFG sensor is lower than through the HFM sensor.

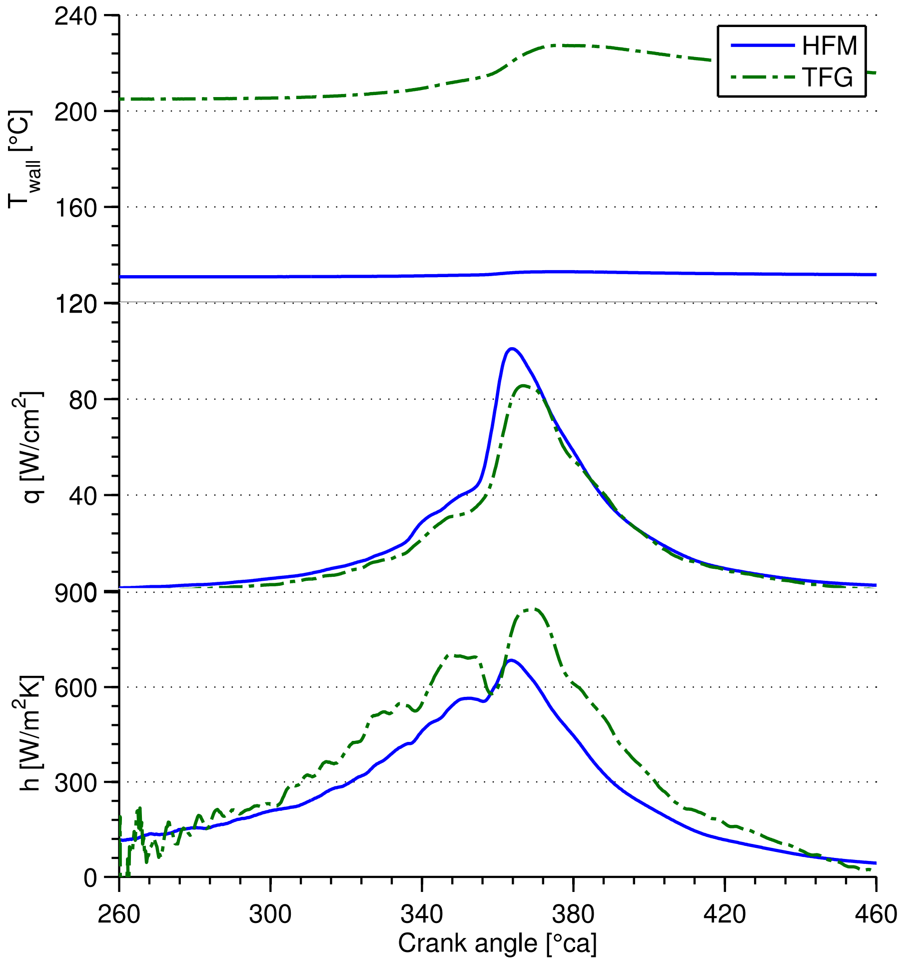

Figure 3. Comparison of the HFM and TFG measurements

Table 3. Thermal properties of the engine block and the TFG sensor

	Engine block	TFG sensor
material	cast iron	Macor
thermal conductivity k [W/m.K]	55	1.6
thermal diffusivity α [mm^2/s]	14.97	0.36

Contrary to the measured wall temperature and heat flux, the convection coefficient is not dependent on the sensor's thermal properties. This is because the convection coefficient is a property of the gas flow, only determined by the gas motion and the gas properties. This can be seen on Fig. 3, where the difference between the convection coefficient measured by the HFM and the TFG sensor is within the experimental uncertainty. This means that both sensors can be used for the evaluation and construction of heat transfer models, as these models predict the convection coefficient and not the heat flux. However, it is of interest to see how the measured temperature and heat flux compare to the actual heat flux and temperature of the cylinder wall. Mainly, to verify whether the measured heat flux can be used to calculate the total heat loss per cycle, but also because simulation software requires the actual wall temperature as input.

4. ACTUAL HEAT FLUX

In order to know the actual heat flux through the cylinder wall, a methodology is developed that reconstructs the heat flux through the cylinder wall. This is achieved by calculating the temperature distribution inside the cylinder wall, using the measured engine coolant temperature and the calculated gas temperature and convection coefficient. These quantities are chosen, as they are not affected by the sensors' material properties and because they represent the actual heat transfer. The temperature distribution inside the cylinder wall is calculated by solving the Fourier equation (Eq. 2). In order to solve this differential equation, two boundary conditions are needed. The first one (Eq. 4) states that the conductive heat transfer through the wall is equal to the convective heat transfer to the wall. The second boundary condition (Eq. 5) sets the temperature at the coolant side of the cylinder wall to the engine's coolant temperature. With these boundary conditions, it is not possible to solve the Fourier equation analytically, only numerically.

$$-k \cdot \frac{dT(x,t)}{dx}\bigg|_{x=0} = h \cdot (T_{gas} - T(0,t)) \tag{4}$$

$$T(L,t) = T_{coolant} \tag{5}$$

With $T_{coolant}$ the coolant's temperature and L the cylinder wall thickness. Fig. 4 shows a scheme of the heat transfer through the cylinder wall.

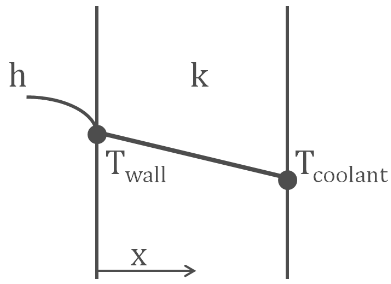

Figure 4. Scheme of the cylinder wall temperature

A linear temperature distribution from the wall to the coolant is chosen as an initial condition. The coolant temperature is known, but the initial value of the wall temperature needs to be guessed. By applying an iterative approach and calculating the temperature for multiple consecutive engine cycles, the solution converges to the actual temperature distribution. Consequently, the assumption of a linear temperature distribution and the guessed value for the initial wall temperature do not affect the final results. However, the gas temperature is only known during the closed part of the engine cycle, when the intake and exhaust valve are closed, and not during the intake and exhaust stroke of the engine. Because the convection coefficient calculation requires the gas temperature, it is also only known during the closed part of the engine cycle. This prevents calculating the temperature for multiple consecutive cycles. Therefore it is assumed that the convection coefficient is zero during the open part of the engine cycle. This is based on the fact that the measured heat flux is nearly zero during this part of the engine cycle. The heat transfer due to the gas exchange is several order of magnitude smaller than the heat transfer caused by the compression and combustion during the closed part of the engine cycle. Nevertheless, this assumption will be verified later.

The calculated (Wall) and measured (HFM and TFG) temperature and heat flux traces are compared in Figs. 5 and 6. The steady state temperature of the wall is lower, compared to both sensors, but the temperature swing is in between those measured by the HFM and TFG sensor. This is because the thermal conductivity of the cylinder wall is higher than both sensors. The calculated heat flux on the other hand, is higher than the heat flux measured by the TFG sensor, but within the measurement error of the heat flux measured by the HFM sensor. This indicates that the heat flux measured by the HFM sensors accurately represents the actual heat flux through the cylinder wall.

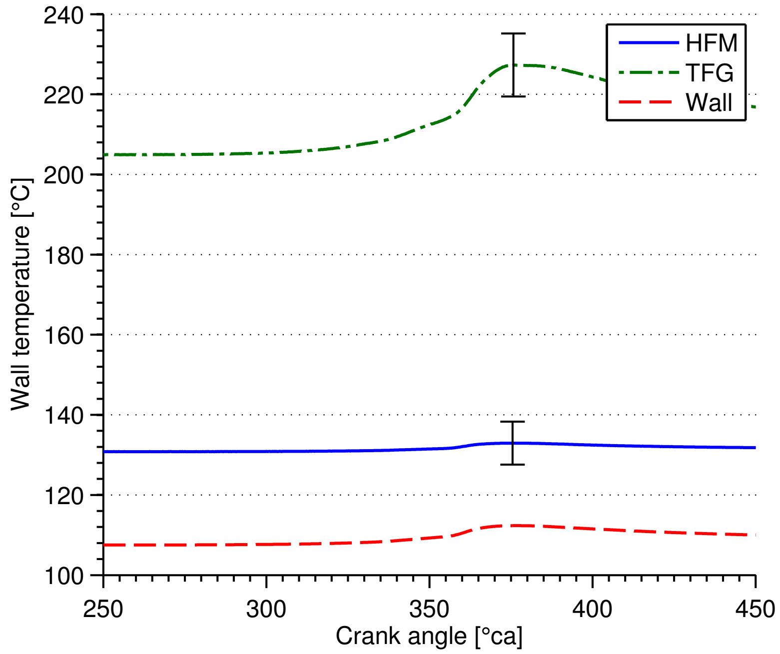

Figure 5. Measured and calculated cylinder wall temperature

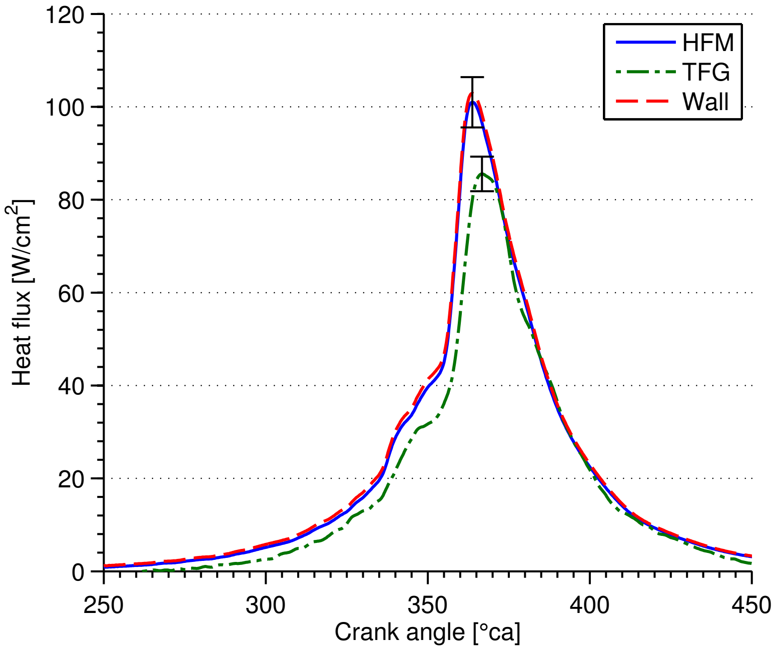

Figure 6. Measured and calculated cylinder wall heat flux

5. VALIDATION

The proposed method to calculate the actual heat flux is validated by reconstructing the temperature distribution of the TFG sensor. The calculated surface temperature and heat flux trace of the TFG sensor can be compared to the measured traces. In this case, the temperature measured with a thermocouple at a known depth in the ceramic substrate is used as the second boundary condition, instead of the coolant temperature. The resulting heat flux trace is shown in Fig. 7. It can be seen that the calculated heat flux corresponds well to the measured heat flux and that it lies within the experimental error. Additionally, the calculated wall temperature converges to the measured wall temperature, which validates the iterative approach. The wall temperature is underestimated by about 5% by neglecting the heat transfer during the open part of the engine cycle, but clearly this only has a small effect on the heat flux.

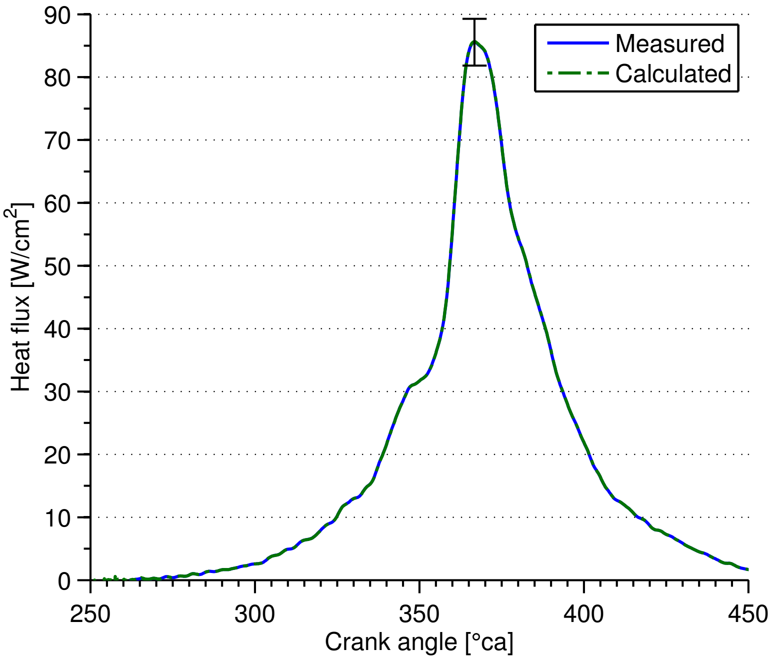

Figure 7. Measured and calculated heat flux through the TFG sensor

The assumption of no convective heat transfer during the open part of the engine cycle is verified with engine simulation software (GT-Power) under motored operation of the engine. The software uses a proprietary heat transfer model (WoschniGT) to simulate the convection coefficient during the complete engine cycle. The model is based on the heat transfer model of Woschni (Woschni, 1967), but takes into account the effect of the gas flow in and out of the cylinder during the gas exchange phase (Gamma Technologies, 2013). The simulated convection coefficient is scaled to match the measured convection coefficient during the closed part of the engine cycle, but is left default during the open part of the cycle. The obtained convection coefficient for the entire cycle is then used as an input to calculate the temperature with the proposed method. The wall temperature obtained with the simulated convection coefficient trace converges to the same temperature as with the assumption of no heat transfer during the open part of the engine cycle.

A different approach to solve the iteration problem, is to omit the gas exchange phase and only iterate over the closed part of the engine cycle. With this approach, the heat transfer to the wall and the wall temperature are overestimated, as there is less time to transfer heat to the coolant. The resulting wall temperature is about 10 % higher compared to the previous approach. Whereas the wall temperature might be underestimated when using the first approach, the second approach overestimates it. Consequently, two boundaries for the wall temperature are established, that encompass the actual wall temperature. Despite this large difference in wall temperature, the resulting heat flux traces differ only by 2%. Both heat flux traces lie within the measurement uncertainty for the heat flux. It can be concluded that the heat flux can be reconstructed accurately with the proposed method.

Finally, a sensitivity analysis was performed on the used engine parameters: the cylinder wall thickness, the thermal conductivity, the thermal diffusivity and the measured coolant temperature. Increasing or decreasing any of these parameters by 10%, changes the peak value of the temperature and heat flux by less than 2% and 1% respectively. Only a change in coolant temperature leads to an equally large change in wall temperature. Hence, the proposed method can be used to reconstruct the heat flux trace, even if the required engine properties are not exactly known.

6. CONCLUSION

In this work it was shown that the heat flux and wall temperature measured by different sensors in the combustion chamber wall are different. This is due to the material properties of the sensors. However, the convection coefficient determined with both sensors is the same. A method is proposed to calculate the actual heat flux through the cylinder wall by reconstructing the temperature distribution inside the wall. This requires the engine coolant temperature, the calculated gas temperature and the convection coefficient. By using an iterative approach, the imposed initial conditions do not affect the result. It is assumed that the convection coefficient and gas temperature are zero during the open part of the engine cycle. The validation shows that this method is accurate and the assumptions made when deriving the method are also valid. Furthermore, the method can be used, even if the required engine properties are not exactly known.

7. ACKNOWLEDGEMENTS

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