

Applying Design of Experiments to Determine the Effect of Gas Properties on In-Cylinder Heat Flux in a Motored SI Engine

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ABSTRACT

Models for the convective heat transfer from the combustion gases to the walls inside a spark ignition engine are an important keystone in the simulation tools which are being developed to aid engine optimization. The existing models have, however, been cited to be inaccurate for hydrogen, one of the alternative fuels currently investigated. One possible explanation for this inaccuracy is that the models do not adequately capture the effect of the gas properties. These have never been varied in a wide range because air and 'classical' fossil fuels have similar values, but they are significantly different in the case of hydrogen. As a first step towards a fuel independent heat transfer model, we have investigated the effect of the gas properties on the heat flux in a spark ignition engine. The effect of the gas properties was decoupled from that of combustion, by injecting different inert gases (helium, argon, carbon dioxide) into the intake air flow of the engine under motored operation. This paper presents the results of the experiment, which was designed with DoE techniques. The paper shows that the three investigated effects (throttle position, compression ratio and gas) and the interaction between the throttle and the compression ratio are significant with a significance level of 1%. Both the individual and combined effects of the gas properties are investigated. The most remarkable effect observed in the data was that the dynamic viscosity influences the heat flux in two contrasting ways. At the one hand, it increases the heat flux by increasing the gas temperature, at the other hand, it reduces the heat flux through the convection coefficient. A preliminary test shows that modeling under motored operation could be based on classical concepts. However, some scatter occurs in the data which needs further investigation.

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INTRODUCTION

A lot of research efforts focus on further improving the efficiency, emissions and power output of internal combustion engines (ICE) to propel heavier vehicles with lower fuel consumption and emissions. To enhance this development, computer tools are being developed to simulate the engine cycle. An important part in these models is the prediction of the heat transfer inside the engine, since it has an effect on all the improvement targets given above. In particular, the convective and radiant heat transfer from the combustion gases to the inner cylinder walls is needed at every calculation step to solve the conservation equations of mass and energy. Therefore, in-cylinder heat transfer has been the subject of many research programs since the 1950s [1, 2].

In spark ignition engines, only convection is considered since radiation is negligible [1] and the heat transfer is assumed to be quasi-steady. As a consequence, the convective component of the heat flux (q) can be described by a convection coefficient (h), defined in equation 1.

$$h = \frac{q}{(T_g - T_w)} \quad (1)$$

Where (T_w) is the wall temperature and (T_g) the bulk gas temperature. Annand [3] was the first to propose a dimensionally consistent heat transfer model by gathering the research results until the early sixties. His model was based on the boundary layer theory, which describes the heat transfer caused by a flow over a flat plate. The heat transfer is characterized by the Nusselt number ($= mL/k$), being modeled

as a function of the Reynolds ($=V.L/\nu$) and Prandtl ($=\alpha/\nu$) number:

$$Nu = a \cdot Re^b \cdot Pr^c \quad (2)$$

A characteristic length (L) and velocity (V) have to be defined for the calculation of the Nusselt and Reynolds number. Annand used the cylinder bore (D) and the mean piston speed (c_m), respectively. He neglected the Prandtl number to reduce calculation efforts, since it is almost constant and around 0.7 for most of the gases. Annand fitted the model to the available heat transfer measurements at that time and concluded that the parameter a should have a value between 0.35 and 0.8 and that b should be equal to 0.7. Parameter a depends on the engine geometry and charge motion. Therefore, it can be used as a scaling factor to fit the model to a certain engine. Annand and Ma [4] tried to improve the first model of Annand by including a time derivative of the gas temperature, but could not provide enough evidence of the improvements. Consequently, the first model of Annand has been used ever since.

A second widely used model is that of Woschni [5], who followed Annand and based his model on equation 2, also with the Prandtl number lumped into the parameter a . Woschni's model mainly differs from that of Annand on three aspects. First, Woschni took parameter a from models which describe the heat transfer of flows in tubes ($a=0.045$), being an order of magnitude lower compared to that of Annand and suggested b to be 0.8. Second, he made assumptions on the gas properties, given below, to convert equation 1 and 2 into Equation 3 (as described in [6]) in order to calculate the heat flux as a function of only the cylinder bore, the characteristic velocity, the pressure and the temperature. Third, Woschni added the pressure difference between fired and motored operation to the characteristic velocity to represent the effect of the combustion on the heat transfer.

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

$$q = a_{Wo} \cdot D^{-0.2} \cdot p^{0.8} \cdot V^{0.8} \cdot T^{-0.53} \cdot (T_g - T_w) \quad (3)$$

Ever since, authors (e.g. references [7] and [8]) proposed new models based on that of Woschni, by adjusting the exponents of pressure and temperature. These models cannot be considered as a fundamental improvement and have no link anymore with equation 2, so extrapolation beyond the measurement range for which they were developed is expected to be even worse than for the models of Annand and Woschni. Although there has been a lot of research on the heat transfer in combustion engines, clearly the mechanisms are still not well characterized, especially not for new combustion types (e.g. HCCI [9]) and alternative fuels (e.g. hydrogen [10, 11]).

The most obvious possible reason for the models being inaccurate for an alternative fuel like hydrogen, is that the effect of the gas properties is not well captured. These gas properties have never been varied in a wide range when the models were developed, since only air and fossil fuels with comparable properties have been used. Next, both Annand and Woschni neglected the effect of the Prandtl number, because almost all the gases have a comparable value. However, water vapor has a Prandtl number around unity, being present in every engine exhaust gas and being the main exhaust gas component for hydrogen combustion. Furthermore, Woschni made assumptions about the temperature dependency of the gas properties which are only valid for air. The resulting error for fossil fuels could be small, but the thermal conductivity and heat capacity of hydrogen, on the other hand, differ significantly compared to air.

This paper investigates the effect of the gas properties (thermal conductivity, viscosity, heat capacity and density) on the heat flux in an ICE without the interference of combustion as a first step towards a fuel independent heat transfer model. Therefore, different inert gases (argon, helium and carbon dioxide) have been injected into a test engine under motored operation. Each gas is chosen because it has a certain property that differs significantly compared to air (see below). The experiment is designed and analyzed with Methods of Design of Experiments to be able to add a statistical significance to the conclusions.

EXPERIMENTAL METHOD

EQUIPMENT

The engine used in this research is a four-stroke single-cylinder spark ignition engine based on a CFR (Cooperative Fuel Research) engine operated at a constant speed of 600 rpm. A cross section is given in Figure 1, showing the possible sensor positions in the cylinder wall. It is equipped with PFI (port fuel injection) and has a variable compression ratio. The compression ratio has to be kept below 10, because the moving piston would otherwise damage the heat flux sensor. The details of the engine are given in Table 1. The engine has recently been revised, resulting in a larger bore diameter and different valve timings compared to previous publications. The port fuel injection system is used to inject the inert gases into the engine and to form different mixtures of air and inert gases. The injection of the inert gases is controlled by a MoTeC M4Pro electronic control unit.

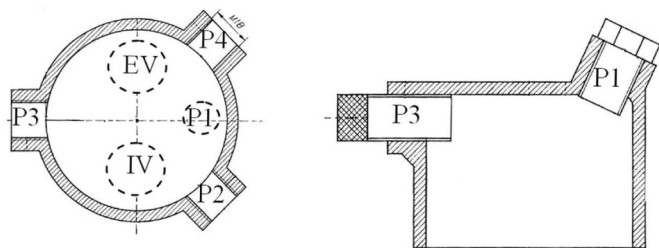


Figure 1. Cross section of the CFR engine, P1: spark plug position, P2-P4: possible sensor positions, IV: intake valve, EV: exhaust valve

Table 1. CFR-Engine properties

Bore	83.06 mm
Stroke	114.2 mm
Connecting rod length	254 mm
Swept Volume	618.8 cm ³
IVO	10 °CA ATDC
IVC	29 °CA ABDC
EVO	39 °CA BBDC
EVC	12 °CA ATDC

The wall temperature was measured with a thin film gauge (TFG), developed at the University of Oxford [12] and shown in Figure 2. The TFG is an RTD (resistance temperature detector), having a resistance that changes linearly with temperature. A total of three TFGs were deposited in parallel on a ceramic substrate (Macor©) that was inserted in the cylinder at position P2 (see Figure 1). The heat flux is only measured at P2, because a previous investigation did not show spatial variation under motored operation [13]. For the same reason, only the measurements of the middle TFG were used for this paper. The relationship between the resistance and the temperature of the TFG is given in equation 4, with R_0 being the reference resistance at a reference temperature T_0 . α_0 (in 1/K) is the coefficient which expresses the linear relationship between the resistance and the temperature of the sensor and has to be calibrated. The maximum observed wall temperature in the motored engine is 80°C. Therefore, the coefficient α_0 of the middle TFG was determined with a calibration experiment in an oven within a range of 25-80°C (see Figure 3) and was found to be 0.0016 for a T_0 of 22°C.



Figure 2. 3 thin film gauges (TFG) deposited on a ceramic substrate (Macor©)

$$R = R_0 \cdot [1 + \alpha_0 \cdot (T - T_0)] \quad (4)$$

A constant current was sent through the TFG to generate a change in the voltage over the sensor proportional to the change in its resistance. Multiplying the equation above with the current I results in the following equation which is used to derive the temperature increase out of the measured voltage increase (V_0 is the measured voltage at T_0):

$$\Delta T = \frac{\Delta V}{\alpha_0 \cdot V_0} \quad (5)$$

The measured wall temperature is converted into surface heat flux with an electrical circuit that models the one dimensional equation of Fourier [14]. Both the surface temperature and the heat flux were recorded for the analysis.

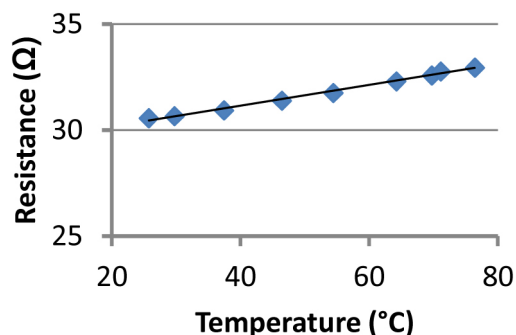


Figure 3. Calibrated resistance of TFG as a function of temperature

In-cylinder pressure was measured with a water-cooled Kistler 701A piezoelectric sensor (mounted in P4). Inlet and outlet pressure were measured with two Kistler 4075A10 piezoresistive pressure sensors. The inlet pressure was used to reference the in-cylinder pressure. Gas flows were measured with Bronkhorst Hi-Tec F-201AC (gas) and F-106BZ (air) flow sensors. Finally, type K thermocouples were used to measure coolant, oil and inlet and exhaust gas temperatures.

All the signals were acquired with a National Instruments PXI system, of which a scheme is shown in Figure 4. Crank angle resolved signals (TFG and pressure signals) were acquired synchronously with a PXI-6143 S-series card every 0.5 °CA (sample rate of 7.2 kHz) during 100 consecutive cycles. The other signals were averaged over time and acquired with a PXI-6224 M-series card at a sampling rate of 1 Hz.

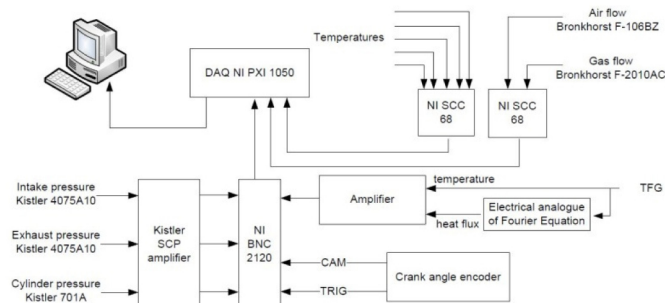


Figure 4. Overview of data acquisition system

EXPERIMENTAL DESIGN

The experiment was designed according to DoE methods (Design of Experiments) described in ref. [15]. The first step is to plan the experiment, with the identification of the dependent and independent variables. The dependent variable in this case is the heat flux (q) and a root cause analysis, of which the result is shown in Figure 5, was executed to determine the possible independent factors. The basic question to be answered is which factors have an influence on the in-cylinder heat flux in an engine under motored operation. Three main groups were identified to have an effect: in-cylinder flow, gas properties and temperature difference of which the first two are represented by the convection coefficient. Those three groups were further developed downwards to arrive at the factors that can actually be changed on an engine. Three categories of factors were put forward: unchangeable (fixed on the engine in the test setup), uncontrollable (atmospheric properties) and controllable factors. Further on in the analysis, the pathways from the factors to the top can be used to explain any observed trends in the heat flux. The atmospheric conditions did not change during the experiment, so these can be neglected. The analysis resulted in the definition of three controllable factors: throttle position (TP), compression ratio (CR) and injected gas (type and quantity).

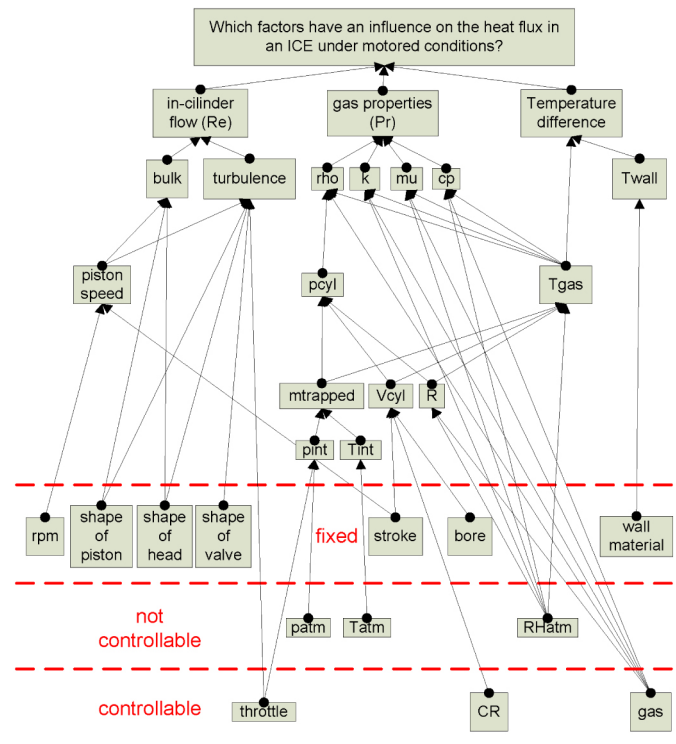


Figure 5. Root cause analysis to identify the factors to be investigated

Next, the number of levels for each factor had to be determined of which an overview is given in Table 2. Since the first two can vary continuously, they were given two levels to vary between their extreme values. For both factors, it would actually have been more accurate to include intermediate points to investigate non-linearities. For some gases, the heat flux has been measured for intermediate throttle positions (4), but not for all so this is not taken into consideration for the ANOVA analysis below. This will be improved in the design of future experiments under fired operation. The measurement that have been recorded can, however, be used to check the validity of modeling concepts for intermediate throttle positions. For the injected gas it was examined whether the gas properties itself could be used as the independent variables, since the ANOVA then would directly indicate which of them are most significant. It was concluded that this was not possible, since not enough inert gases could be identified to have a sufficient number of combinations of the factors to study their main effects. Therefore, the gas factor was defined as a categorical one and three inert gases were chosen, because they have a certain property that differs significantly compared to air: helium (high thermal conductivity k and heat capacity c_p), argon (high viscosity μ and low c_p) and CO_2 (high Prandtl number Pr at high temperatures). The gas properties of the different mixtures used (see below) can be compared to air at 298 and 900K in Table 3.

It is not possible to fill the engine entirely with a pure inert gas, because the amount of gas that can be injected is limited by the maximum flow rate through the injector (gas dependent) and the engine cycle duration. To increase the maximum flow rate and to increase the mixture homogeneity, the injection system was adopted so that an inert gas could be injected in the intake manifold during the entire engine cycle, instead of only during the intake stroke which is normally the case. Because of the long engine cycle time of 200ms, enough inert gases could be injected in the engine to change the gas properties significantly compared to air. For argon and CO₂, the maximum achievable volume percentages in atmospheric air for the wide-open-throttle (WOT) cases were 80vol% and 67vol%. In contrast, it was possible to obtain a 100% helium mixture in the engine, because of its low density compared to argon and CO₂. However, two intermediate mixtures were tested (20 and 50vol% of helium in atmospheric air), because the k of a helium-air mixture already significantly increases with a small volume percentage of helium (20vol%) in contrast to the c_p . As a consequence, the Prandtl number drops when the helium content is increased from the value of air (0.7 at 293K) over 0.56 for a 20vol% mixture to a minimum of 0.46 for a 50vol% mixture, making it two interesting mixtures to test. The gas level labels in Table 2 only contain the amount of inert gas or pressurized air, the rest of the mixture consists out of atmospheric air.

Since injecting a high amount of gas in the engine increases the trapped mass (especially with a closed throttle) compared to the natural aspiration of atmospheric air, an extra level of the gas factor was added by injecting an extra amount of air from a pressurized tank at the same pressure as the other gases (3 bar absolute) to result in a total of 6 levels for the factor 'gas'. For the extra air level, the maximum achievable value of 75vol% pressurized air in atmospheric air was used. Because the air from the pressurized tank is dry, the effect of the atmospheric humidity on the heat flux can be investigated as a consequence as well.

Table 2. Overview of the factors' levels

factor	level	level code
throttle	closed	-1
	wide open throttle (WOT)	1
CR	8	-1
	10	1
gas	20vol% helium	1
	50vol% helium	2
	80vol% argon	3
	67vol% CO ₂	4
	100vol% atmospheric air	5
	75vol% pressurized air	6

Table 3. Gas property comparison of the different mixtures

T (K)	gas level	k (W/m.K)	c_p (J/kg.K)	μ (Pa.s)	Pr
298	1	3.9e-2	1.1e3	1.9e-5	0.56
	2	6.5e-2	1.5e3	2.0e-5	0.46
	3	2.0e-2	0.6e3	2.2e-5	0.67
	4	2.0e-2	0.9e3	1.6e-5	0.70
	5	2.6e-2	1.0e3	1.9e-5	0.70
	6				
900	1	8.9e-2	1.3e3	4.1e-5	0.57
	2	14.4e-2	1.6e3	4.3e-5	0.47
	3	4.5e-2	0.6e3	4.9e-5	0.67
	4	6.2e-2	1.2e3	3.6e-5	0.72
	5	6.3e-2	1.1e3	3.9e-5	0.70
	6				

The second step is to design the experiment. The dependent variable for the analysis of variance (ANOVA) was chosen to be the maximum heat flux (q_{max}). All the combinations ($2^6=24$) of the factors could be run in two test days, so it was decided to run the full factorial matrix to be able to determine second and third order interaction between the factors. Information on the random error is needed to be able to conduct a statistical test on the significance of the factor's effects. This information is provided by replication of the same measurement points. It was the objective to include the cyclic variation in the random error and different approaches were tested. First, the 100 stored cycles were divided into two groups of 50 cycles and the maximum heat flux of the two groups was used (2 replications). Second, the maximum heat flux of a random number of cycles were chosen out of the 100 stored ones (2, 5 or 100 replications). Third, the mean plus or minus the standard deviation of the maximum heat flux of all the stored cycles was used (3 replications). The first approach did not capture the cyclic variation because there was very little difference in the q_{max} of the two groups. The last approach was abandoned as well, because the three values (mean $\pm \sigma$) resulted in a widening of the normal distribution of q_{max} compared to the actual normal distribution of the 100 cycles. Therefore, it was concluded that the second approach was the best one. Taking q_{max} from all the cycles did not result in meaningful conclusions because all the effects and interactions, even the three way interaction, were significant. This was caused by the fact that the degrees of freedom (df) of the error term were boosted too much, resulting in a very low error mean sums of squares value. A number of 5 replications was concluded to be the best amount of replication in order to have the same number of replications as the df of the gas factor (number of levels minus 1).

The next step, the statistical analysis of the results will be discussed in the results section of this paper after a qualitative discussion of the measurement set.