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Evaluation of empirical heat transfer models for HCCI combustion in a CFR engine

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HIGHLIGHTS

- Current heat transfer models require calibration.
- Calibration on the peak heat flux results in an overpredicted total heat loss.
- The Woschni heat transfer model requires calibration of both model coefficients.
- The effect of the engine settings is best predicted by the model of Bargende.
- Different coefficients for the compression and expansion results in a better fit.

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ABSTRACT

The heat transfer from the bulk gases to the combustion chamber walls has a strong effect on the combustion and emission formation process in an HCCI engine. In this work, the empirical heat transfer models of Annand, Woschni, Hohenberg, Bargende, Chang et al. and Hensel et al. are evaluated at various engine operating conditions. The modelled heat flux is compared to the measured heat flux in a CFR engine with a thermopile sensor. The shape of the heat flux trace, the maximum heat flux and the total heat loss are evaluated and different model calibration procedures are investigated. It is found that all models require calibration and need to be recalibrated if the fuel type and certain engine settings are changed. A better model fit can be obtained if different model coefficients are applied for the compression and the expansion phase.

1. Introduction

Various new operating modes are being developed that aim to replace traditional spark- and compression-ignition combustion in internal combustion engines. Of these combustion modes, Homogeneous Charge Compression Ignition (HCCI) [1,2] has received the most attention in the past decade, as it allows achieving both a high thermal efficiency and near-zero emissions of NO_x and soot [3]. Considering the widespread use of internal combustion engines for automotive, maritime and power generation applications, this technology has the potential to reduce the worldwide emissions of greenhouse gases and pollutants[4]. These favourable properties are attained by introducing a lean, premixed air-fuel mixture into the combustion chamber and having it auto-ignite by the compression heat. However, despite extensive research in recent years HCCI combustion still has several drawbacks, such as a limited operating range in which a stable combustion occurs [5,6] and a lack of control over the start of the combustion [7]. Moreover, the emissions of unburned hydrocarbons (UHC) and carbon monoxide (CO) can be excessively high when the mixture's temperature is too low for the late-cycle oxidation of these species.

Engine simulation software is commonly used to investigate and optimize the behaviour of engines at various operating conditions [8], to perform energy analyses [9,10] or for the control of the combustion [11]. Due to the complex interaction between fluid motion and combustion chemistry, full-cycle CFD-simulation with detailed chemical kinetics cannot be performed with acceptable computational times. This can be resolved by using simplified single- and multi-zone [12,13] simulation software, that require additional models to solve the equations of mass and energy. One of the required models is the wall heat transfer model, that predicts the heat transfer from the bulk gases to the combustion chamber walls. The heat transfer has a large impact on the simulation results, due to the strong temperature dependence of the auto-ignition process [14]. For example, overestimating the wall heat transfer will result in a reduced mixture temperature, which will predict the start of combustion happening too late. This will not only affect the shape of the pressure trace, but also the prediction of emissions such as

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Nomenclature

Abbreviations

ATDC	after top dead center
BBDC	before bottom dead center
CA	crank angle
CFR	Cooperative Fuel Research
CO	carbon monoxide
EV	exhaust valve
EVC	exhaust valve closing time
EVO	exhaust valve opening time
HCCI	homogeneous charge compression ignition
IV	intake valve
IVC	intake valve closing time
IVO	intake valve opening time
NVO	negative valve overlap
UHC	unburned hydrocarbons

Definitions

A	heat transfer surface area [m^2]
B	cyliner bore [m]
c_c	instantaneous piston speed [m/s]
c_m	mean piston speed [m/s]
h	convection coefficient [W/m^2 K]
h_{IV}	valve lift [m]
θ	crank angle [°ca]
k	thermal conductivity [W/m K]
k_{spec}	specific kinetic energy [m^2/s^2]
L	characteristic length [m]
L_c	instantaneous chamber height [m]
Nu	Nusselt number [–]
p	in-cylinder pressure [Pa]
Q_{tot}	total heat loss [J]
q	instantaneous heat flux [W/m^2]
Re	Reynolds number
T	temperature [K]
V	characteristic velocity [m/s]
V_{cyl}	instantaneous cylinder volume [m^3]
V_s	swept volume [m^3]
X	mass fraction burned [–]
μ	dynamic viscosity [N s/m^2]
ρ	density [kg/m^3]

NO_x, CO and UHC due to the temperature dependence of their formation process. Consequently, having an accurate prediction of the heat transfer is crucial when simulating the combustion in an HCCI engine.

The most commonly used heat transfer models, are empirical models that calculate the spatially averaged heat flux as a function of crank angle. Oftentimes, in simulation tools heat transfer models are used that were developed for spark- and compression ignition engines, e.g. the models proposed by Annand [15], Hohenberg [16] and Woschni [17]. However, these models are based on heat transfer measurements in spark- and compression ignition engines. It is expected that these heat transfer models are not applicable to HCCI engines, due to its different combustion process. However, little is known about the heat transfer in this type of engines, as almost no experimental heat transfer data is available. Chang et al. [18] and Hensel et al. [19,20] have tried to modify these models for HCCI operation, but their modified models have not yet been extensively validated. The implementation and calibration of the heat transfer models also differs from one simulation tool to the next. Sometimes the models are implemented with the coefficients proposed by the authors of each model, but often the scaling coefficient [21,22] or other model coefficients [23] are adjusted to calibrate the model. In some cases the model itself is modified. For example Olsson et al. [24] and Ogink et al. [25] omitted the pressure dependent term in the characteristic velocity of Woschni's heat transfer model. However, the calibration or modification of a heat transfer model using only pressure and emissions data is not straightforward because of the interaction between the heat transfer model and the other models. For this reason, an experimental study in which the predicted heat flux is compared to the measured heat flux is useful, as it eliminates the need for other models that affect the results. Nevertheless, only a limited number of such studies have been performed.

Chang et al. [18] measured the spatially averaged heat flux in a gasoline fueled single cylinder engine with exhaust rebreathing. The heat flux was measured with 7 thermocouples in the piston surface and 2 thermocouples in the cylinder head. Before constructing their own heat transfer model, they compared the measured heat flux to the heat flux calculated with the models proposed by Annand & Ma [26], Hohenberg and Woschni. Each model's scaling coefficient was adjusted to satisfy the energy balance in one engine operating point. The obtained scaling coefficients were not listed. With this approach, the heat flux trace calculated with Hohenberg's model was closest to the measured heat flux trace and Woschni's model was least accurate. A better prediction of the heat flux was obtained by reducing the coefficient of the pressure dependent term in the characteristic velocity of Woschni's model, to reduce the heat flux during the combustion. However, when changing the mass fuel rate, none of these models were able to predict the total heat loss accurately. This led the authors to suggest several modifications to adapt Woschni's model for HCCI operation.

Soyhan et al. [27] implemented the models of Hohenberg, Woschni and Chang et al. in a single-zone engine simulation code. They compared the shape of the heat flux traces by using a normalised form of each model and found that the shape of the heat flux traces predicted by Hohenberg and Chang et al. is very similar if the same length scale is used. When comparing the simulated pressure traces with the measured pressure trace for each model, they found that Woschni's model overestimates the heat transfer during the combustion and expansion stroke, Chang et al.'s model underestimates the heat transfer throughout the complete engine cycle and Hohenberg's model provides the best agreement. The model coefficients proposed by the author of each model were used. However, they recommended calibrating the scaling coefficients for each operating point to obtain more satisfactory results.

Hensel et al. [19,20] measured the heat flux in the cylinder head of two gasoline fueled single-cylinder engines, with Negative Valve Overlap (NVO). The heat flux was measured in a Rotax F650 engine with 8 thermocouples mounted in the cylinder head and in a Ricardo Hydra engine with 3 thermocouples in the cylinder head. A comparison between the heat transfer during SI and HCCI operation showed a similar heat loss during the combustion. The additional heat transfer during NVO-compression resulted in a higher total cycle heat loss. The measured heat flux was compared to the calculated heat flux trace with the models of Bargende [28], Chang et al., Hohenberg and Woschni. The model coefficients proposed by the author of each model were used. The authors concluded that both Bargende's and Woschni's model overestimated the heat flux during the combustion and that Hohenberg's model provided the best prediction. They also concluded that Chang et al.'s model predicted the maximum heat flux accurately for the Ricardo engine, but underpredicted it for the Rotax engine. Note that Chang et al.'s model underestimated the heat flux for the Ricardo Hydra engine used by Soyhan et al. This could be explained by the fact that both Chang et al. and Hensel et al. applied exhaust recirculation to