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A new quasi-dimensional flame tracking combustion model for spark ignition engines

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ABSTRACT

A new quasi-dimensional combustion model based on the flame tracking approach is described and presented in the paper. The new quasi-dimensional flame tracking model is able to simulate the turbulent combustion process in premixed fuel/air/residual gas mixtures. A new method for the description of the geometry of the combustion chamber and flame front was developed enabling the visualisation of flame front movement across the combustion chamber. The control of local turbulence quantities in the flame front near the wall enables that the developed flame tracking model can predict the entire turbulent combustion process after the flame kernel development in spark-ignition engines without case-dependent calibration requirement. The developed quasi-dimensional combustion model was validated with the experimental and results of multidimensional model of a single cylinder spark ignition engine on averaged cycles. The model was integrated with the previously developed ignition, mixture stratification and cyclic variability sub-model that enable the simulation of cyclic combustion variability triggered by the stochastic variations of flow angle at the spark plug, mixture stratification and in-cylinder turbulence level. Due to the novelty of the model which includes the control of local integral length scale and turbulent kinetic energy in the flame segments the predictive capability of quasi-dimensional model is achieved with the application of single set of parameters related to average cycle and cyclic combustion variability. Flame tracking model with the low computational time represents a promising tool to calculate the turbulent combustion process including cyclic combustion variability in modern spark ignition engines.

1. Introduction

The majority of transport worldwide will depend on fossil fuels for a number of years to come. The transport is recognized as one of the major sources of pollution and therefore governments around the world have imposed a series of regulations that aim at reducing the harmful emissions from internal combustion (IC) engines. As a result of these regulations and general industry development, the research of IC engines has been directed towards the optimization of the combustion process so that more effective and cleaner processes can be achieved [1].

As the computer power is getting more and more increased over the last three decades, numerical simulations became very powerful engineering tools used in variety of applications during the entire engine development process from the conceptual phase to the calibration of the engines. In the field of modelling of the IC engine working process, the engine cycle-simulations, usually called 1-D/0-D models, offer a good trade-off between the simulation model accuracy and calculation time. Modelling of the combustion process in these models can be performed by the application of empirically based combustion models (e.g. Vibe function [2]) or by the application of predictive quasi-dimensional combustion models. If the predictive quasi-dimensional combustion models are properly calibrated, in the combination with 1-D flow model across the intake and exhaust pipes, it can be used as a predictive simulation tool for estimation of engine performance and emissions. In the study presented in [3] the calibrated quasi-dimensional combustion model was applied to predict the maximum engine torque and power over the different engine speeds when the engine was fuelled by different fuels and the maximum achieved relative difference of measured and simulated engine torque was ± 3%. The predictive capabilities of quasi-dimensional combustion model were demonstrated in [4] where the simulated emission formation of CO and NO from SI engine fuelled by gasoline and natural gas matched well the experimental data.

The most commonly used quasi-dimensional combustion models for spark-ignition (SI) engines are the fractal combustion model and the turbulent entrainment model. The fractal combustion model is based on

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Nomenclature

A	surface area (m^2)
B	bore (m)
F	reduction factor (–)
H_s	lower heating value of mixture (J/kg)
k	turbulent kinetic energy per unit mass (m^2/s^2)
L_i	integral length scale of *i*-th flame point (m)
m	mass (kg)
R_{BZ}	individual gas constant of the burned zone (J/kg K)
R_i	radius of *i*-th flame point (m)
$T_{i,flame}$	turbulence intensity in the flame (–)
t	time (s)
u	flame speed (m/s)
u'	turbulent velocity pulsation (m/s)
V	volume (m^3)

Greek symbols

λ	excess air ratio (–)
υ	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
Ω	solid angle (sterad)

Subscripts

act	active
af	anode fall
bd	breakdown
BZ	burned zone
cf	cathode fall
cg	column of gas
f	flame
max	maximum
N	normal
opp	opposite
p	piston
q	quenching
red	reduction
s	secondary
SP	spark plug
t	turbulent
tria	triangle

Abbreviations

AKI	anti-knock index
AKTIM	arc and kernel tracking ignition model
BDC	bottom dead center
BTDC	before top dead center
CA	crank angle (°)
CAD	computer-aided design
CCV	cycle-to-cycle variations
CFD	computational fluid dynamics
CFR	cooperative fuel research
CoV	coefficient of variation (%)
CO	carbon monoxide
CR	compression ratio (–)
DNS	direct numerical simulation
ECFM	extended coherent flame model
EVO	exhaust valve open
FTM	flame tracking model
FTPM	flame tracking particle model
HC	hydrocarbon
IC	internal combustion
IVC	intake valve closure
IMEP	indicated mean effective pressure (bar)
LES	large eddy simulation
NO_x	nitric oxides
Q-D	quasi-dimensional
QDIM	quasi-dimensional ignition model
ROHR	rate of heat release (J/°CA)
SI	spark ignition
TDC	top dead center
0-D, 3-D	zero, three dimensional

the fractal theory considering the highly wrinkled thin flame front that propagates across the combustion chamber by the laminar flame speed. In the fractal combustion model integral length scales of turbulent flow field are assumed to be larger than the thickness of reactive flame front which means that the flame is considered in the wrinkled flamelet regime [5]. The concept of the fractal combustion model is applicable to a fully developed and freely expanding turbulent flame that is not the case during the entire combustion process in SI engines. The application of such combustion model and its predictive feature was shown in [6] where SI engine was fuelled by hydrogen enriched compressed natural gas. Although the equation of fractal dimension was improved accounting for the hydrogen mole fraction in the mixture, the simulation results demonstrated that the early flame stage and the late combustion stage cannot be captured in a good agreement with the experimental data without additional modifications of fractal dimension during these two stages of combustion process. The transition from laminar to fully developed turbulent flame that occurs during the early flame stage can be described so that the progressive increase function is imposed on the definition of maximum fractal dimension. Due to decrease of flame wrinkling near the walls the late combustion stage is characterized as the laminar one and another modification of the burning rate has to be defined. Hence, the required additional modifications of the fractal combustion model increase the number of calibration constants that have to be calibrated. The extended fractal approach applied together with the previously mentioned additional modifications that account for the early and late combustion stage has shown slightly improved prediction capability of the fractal combustion model over the different engine speed compared to the standard fractal model, as presented in [7]. Although the total number of calibration constants remains unchanged, the potential solution to eliminate the necessity of artificial wall combustion mode and to better predict the late combustion stage is the application of two zone turbulence model where the unburned turbulence quantities are used for the definition of flame wrinkling [8].

The turbulent entrainment model, firstly developed by Blizard and Keck [9], assume the combustion process to take place in two steps. The unburned mixture is firstly entrained by the smooth spherical flame front that propagates by the entrainment velocity. After that, the entrained turbulent eddies in the fresh mixture burn in a specific time that is proportional to the characteristic eddy size and laminar flame speed. There are many versions of the turbulent flame entrainment model based on the previously mentioned two steps of the combustion process, with differences primarily arising from the choice of the characteristic turbulent eddy size (integral length scale) and the entrainment velocity which governs the entrainment of fresh mixture by the flame front. The validation of entrainment turbulent combustion model using the experimental data of in-cylinder pressure and flame imaging was presented in [10]. It was shown that two combustion coefficients have to be calibrated for each operating condition of the engine, especially the characteristic integral length coefficient which indicates the significant influence of operating parameters on the scale at which burn-up occurs

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