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Research paper

Quasi-dimensional modeling of a fast-burn combustion dual-plug spark-ignition engine with complex combustion chamber geometries

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HIGHLIGHTS

- QD model was applied in dual plug engine with complex realistic combustion chamber.
- This method successfully modeled the combustion in the dual-plug Honda-Fit engine.
- The same combustion chamber is tested for various spark plug(s) locations.
- The centrally located single spark-plug results in the fastest combustion.

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ABSTRACT

This study builds on a previous parametric investigation using a thermodynamic-based quasi-dimensional (QD) cycle simulation of a spark-ignition (SI) engine with dual-spark plugs. The previous work examined the effects of plug-number and location on some performance parameters considering an engine with a simple cylindrical disc-shaped combustion chamber. In order to provide QD thermodynamic models applicable to complex combustion chamber geometries, a novel approach is considered here: flame-maps, which utilizes a computer aided design (CAD) software (SolidWorks). Flame maps are produced by the CAD software, which comprise all the possible flame radiuses with an increment of one-mm between them, according to the spark plug positions, spark timing, and piston position near the top dead center. The data are tabulated and stored as matrices. Then, these tabulated data are adapted to the previously reported cycle simulation. After testing for simple disc-shaped chamber geometries, the simulation is applied to a real production automobile (Honda-Fit) engine to perform the parametric study.

1. Introduction

The first four-stroke internal combustion engine was invented and produced in the late 19th century. Since then, internal combustion engines have undergone dramatic improvements in terms of specific weight, performance, fuel consumption, smoothness of operation, and environmental impact owing to significant theoretical and experimental research efforts worldwide. However, scientists believe that the optimum chamber design has not been achieved yet [1]. There are a large number of options available for the cylinder head, piston crown shape, intake port design, location of the spark plug(s) according to injection type (whether in-cylinder or port-injection), and size and number of valves. Debate revolves around issues such as chamber compactness, surface-to-volume ratio, flame traveling length, and the use of swirl-and-squish mixture types [1]. In the past several decades, the fast burning combustion chamber design for spark-ignition (SI) engines has been favored [1]. The concept of burning mixtures in a spark-ignition engine relatively fast is becoming increasingly applicable to engine designs worldwide [2]. The fast burning concept is actually not new. Examples of fast burning engines with higher efficiency and output that consume low-octane fuel have been reported since the 1920s [2]. However, the use of fast burning engines was curtailed during the 1970s because of the adverse noise and mechanical failures caused by their high pressure rates, particularly as compression ratios increased [2]. Following developments in engine, fuel, and emission control technologies, the interest in fast burning has

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Nomenclature

A	area, m^2
B	bore, m
C	blowby coefficient
C_p	specific heat value at constant pressure, kJ/kg/K
CAD	computer aided design
CSP	centrally located single plug
D_{vT}	partial derivative of specific volume with respect to temperature, $\frac{\partial \ln v}{\partial \ln T}$
D_{vp}	partial derivative of specific volume with respect to cylinder pressure, $\frac{\partial \ln v}{\partial \ln p}$
EGR	exhaust gas recirculation
EGT	exhaust gas temperature, K
h	specific enthalpy, kJ/kg
h_g	convective heat transfer coefficient, $W/m^2/K$
m	mass, kg
MBT	maximum brake torque, Nm
MDP	mid-radius located dual spark plugs
MSP	mid-radius located single plug
n	engine speed, rpm
p	cylinder pressure, bar
P_e	effective power, kW
Q	heat loss, J
QD	quasi-dimensional
r	radius, m
SI	spark ignition
S_T	turbulent flame speed, m/s
T	temperature, K
TDC	top dead center
v	specific volume, m^3/kg
V	volume (for cylinder and flame front, respectively), m^3
x	mass fraction burned

Subscripts

b	burned/burning
c	convective
d	dimensionless
f	flame
m	maximum
s	spark timing
u	unburned
w	wall

Greek letters

ϕ	equivalence ratio
θ	crank angle, °
ω	angular velocity, 1/s

renewed. The fast burning process has several benefits compared with normal burn-rate engines. (1) It directly increases efficiency because it approximates the Otto-cycle, which includes the theoretical thermal efficiency upper limit for SI engines. (2) Fast burning enables greater emission control in the combustion chamber by producing a robust and repeatable combustion pattern, which operates with a significantly large amount of exhaust gas recirculation (EGR) or with a very lean mixture, without deteriorating engine operation stability. Reduced gas temperatures with EGR also reduce the heat losses, resulting in higher thermal efficiency. (3) Robust and repeated combustion allows higher compression ratios without knocking combustion, improving fuel consumption. There are several techniques to achieve fast burning. The simplest and most efficient way to obtain fast combustion is by increasing the flame-front area and reducing the flame travel length. This is achieved by using more than one point to start ignition with a compact combustion chamber. In the present study, two ignition points provided by dual spark plugs are considered in the original Honda-Fit combustion chamber [3]. The dual spark plugs are located diagonally on a pent-roof cylinder head along a common diameter opposing nearly mid-radius points. Combustion chamber configurations with a single spark plug and centrally located single plug that are located at the same mid-radius points are compared. Combustion chamber configuration refers to the number and specific location of the spark plugs. Thus, three configurations are examined: MDP with a mid-radius located dual plug, MSP with a mid-radius located single plug, and CSP with a centrally located single plug (Fig. 1). The dual-zone quasi-dimensional thermodynamic model with flame maps data obtained by a CAD software (SolidWorks) is used for cycle simulations [8]. The results are compared in terms of the flame-fronts' geometrical features, cylinder pressure and temperature variations, exhaust gas temperatures, heat loss, combustion duration, and mass fraction burned during the cycle's combustion phase. The aim of this study is two-fold: to open a pathway for the application of quasi-dimensional thermodynamic models on real and complex combustion geometries and to investigate combustion in a real dual plug SI engine, the Honda-Fit engine [3].

2. Theoretical models

2.1. Thermodynamic model

The most important part of the engine-cycle model is the combustion phase. Generally, there are two groups of thermodynamic-based approximations for modeling combustion in SI engines. The first group is called the zero-dimensional (or finite heat release) models, wherein the heat-release rate is determined by functional relations on the basis of experimental data such as cosine or Wiebe functions [4,5]. The second group is called the quasi-dimensional (QD) or mass-fraction-burned models, wherein the burned mass fraction is calculated in conjunction with an appropriate flame-propagation model [5–7] on the basis of fundamental physical quantities. A thin flame-front radially propagating from the spark plug (usually assumed sections of the sphere's surface with a frontal area A_f) through the unburned mixture adds the space dimension to this group of models, making them quasi-dimensional. During combustion, this flame front divides the cylinder charge into a burned (behind the propagating spherical flame) and an unburned zone (in front of the propagating spherical flame) (Fig. 2) for a dual spark engine. During the flame propagation phase, each zone is assumed to have its own average uniform temperature and composition, while the pressure is assumed to be common for both zones. The variations in the thermodynamic states of both zones are determine by using the instantaneous conservation of the combustion chamber volume, time rates of the first law of thermodynamics, and ideal gas equations. The burned-gas zones are assumed to be in chemical equilibrium in order to determine the combustion products' composition. It is also assumed that there is no heat transfer throughout the flame front. Heat transfer only occurs between each