Investigation of Mixture Formation and Combustion in an Ethanol Direct Injection plus Gasoline Port Injection (EDI+GPI) Engine

By

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Certificate of Original Authorship

This thesis is the result of a research candidature conducted jointly with another university as part of a collaborative doctoral degree. I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as part of requirements for a degree except as part of the collaborative doctoral degree and/or fully acknowledged within the text.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

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List of Publications

Journal articles

[1] **Y. Huang**, G. Hong. Investigation of the effect of heated ethanol fuel on combustion and emissions of an ethanol direct injection plus gasoline port injection (EDI + GPI) engine. *Energy Conversion and Management* 2016; 123: 338-347.

[2] **Y. Huang**, G. Hong, R. Huang. Effect of injection timing on mixture formation and combustion in an ethanol direct injection plus gasoline port injection (EDI+GPI) engine. *Energy* 2016; 111: 92-103.

[3] **Y. Huang**, S. Huang, R. Huang, G. Hong. Spray and evaporation characteristics of ethanol and gasoline direct injection in non-evaporating, transition and flash-boiling conditions. *Energy Conversion and Management* 2016; 108: 68-77.

[4] **Y. Huang**, G. Hong, R. Huang. Investigation to charge cooling effect and combustion characteristics of ethanol direct injection in a gasoline port injection engine. *Applied Energy* 2015; 160: 244-254.

[5] **Y. Huang**, G. Hong, R. Huang. Numerical investigation to the dual-fuel spray combustion process in an ethanol direct injection plus gasoline port injection (EDI+GPI) engine. *Energy Conversion and Management* 2015; 92: 275-286.

[6] **Y. Huang**, S. Huang, P. Deng, R. Huang, G. Hong. The Effect of Fuel Temperature on the Ethanol Direct Injection Spray Characteristics of a Multi-hole Injector. *SAE Int. J. Fuels Lubr.* 2014; 7: 792-802.

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[8] **Y. Huang**, G. Hong, R. Huang. The Effect of Volume Ratio of Ethanol Directly Injected in a Gasoline Port Injection Spark Ignition Engine. *10th Asia-Pacific Conference on Combustion*, the Combustion Institute, Beijing China; 2015.

[9] **Y. Huang**, S. Huang, R. Huang, G. Hong. Macroscopic and Microscopic Characteristics of Ethanol and Gasoline Sprays. *19th Australasian Fluid Mechanics Conference*, Melbourne Australia; 2014.

[10] **Y. Huang**, G. Hong. Development of a Numerical Model for Investigating the EDI+GPI Engine. *19th Australasian Fluid Mechanics Conference*, Melbourne Australia; 2014.

[11] **Y. Huang**, G. Hong, R. Huang. Numerical Investigation to the Effect of Ethanol/Gasoline Ratio on Charge Cooling in an EDI+GPI Engine. *SAE paper 2014-01-2612*; 2014.

[12] **Y. Huang**, G. Hong, X. Cheng, R. Huang. Investigation to Charge Cooling Effect of Evaporation of Ethanol Fuel Directly Injected in a Gasoline Port Injection Engine. *SAE paper 2013-01-2610*; 2013.

Abstract

Ethanol direct injection plus gasoline port injection (EDI+GPI) is a new technology to utilise ethanol fuel in spark-ignition engines more effectively and efficiently than E10 or E85 fuels in the current market. It takes the advantages of ethanol's high octane number and great enthalpy of vaporisation which allow higher compression ratio and consequently increase the thermal efficiency. Primary experimental investigation showed that the engine performance was improved by EDI+GPI. The thermal efficiency was increased, the NO emission was decreased and the spark timing could be advanced without engine knock. However, the CO and HC emissions were increased when EDI was applied. To understand the mechanisms behind the experimental results, the mixture formation and combustion processes of an EDI+GPI engine were investigated using CFD simulation, and constant volume chamber and engine experiments.

To investigate the spray and evaporation characteristics of ethanol fuel and provide experimental data for CFD simulation, spray experiments were conducted in a constant volume chamber using high speed shadowgraphy imaging technique. The results showed that ethanol fuel evaporated slowly when fuel temperature was in the range of 275-325 K. However, the evaporation rate increased quickly when fuel temperature was higher than 350 K. The low evaporation rate of ethanol fuel in low temperature environment implied that EDI should be only applied in high temperature engine environment. When the excess temperature was smaller than 4 K, the spray behaved the same as the subcooled spray did. The spray collapsed when the excess temperature was 9 K. Flash-boiling did not occur until the excess temperature reached 14 K.

Numerical simulation of the EDI+GPI engine showed that the overall cooling effect of EDI was enhanced with the increase of ethanol ratio from 0% to 58%, but not with further increase of ethanol ratio. When the ethanol ratio was greater than 58%, the fuel impingement became severe and a large number of liquid ethanol droplets were left in the combustion chamber during combustion, leading to local over-cooling in the near-wall region and over-lean mixture at the spark plug gap. As a consequence, the CO and HC emissions increased due to incomplete combustion. Compared with GPI only condition, the faster flame speed of ethanol fuel in EDI+GPI condition resulted in shorter combustion initiation duration and major combustion duration, leading to the increase of IMEP and thermal efficiency when the ethanol ratio was 0-58%. However, the

combustion performance was deteriorated by over-cooling and fuel impingement when ethanol ratio was greater than 58%. Experimental results showed consistently that the combustion and emission performance of this engine could be the best in the ethanol ratio of 40-60% at the investigated engine condition (medium load, 4000 rpm and early EDI timing of 300 CAD BTDC). Numerical results showed that the best engine performance was resulted from effective charge cooling and combustion efficiency improved by avoiding the wall wetting, over-lean and local over-cooling issues. Numerical simulations were also carried out to investigate the effect of direct injection timing on the EDI+GPI. The results showed that when the EDI timing was retarded from 300 to 100 CAD BTDC, the mixture around the spark plug became leaner and the distribution of equivalence ratio became more uneven. Moreover, late EDI timing at 100 CAD BTDC resulted in severe fuel impingement and caused local over-cooling effect and over-rich mixture. Consequently, the combustion speed and temperature were decreased by retarded EDI timing, leading to the decreased NO emission and the increased HC and CO emissions. The fuel impingement and incomplete combustion of late EDI timing at 100 CAD BTDC could be addressed by reducing the ethanol ratio to an appropriate point.

Experiments on the EDI+GPI engine were conducted to verify the idea of EDI heating on improving the engine performance, which was developed based on the understanding gained from the numerical investigation. Results showed that EDI heating effectively reduced the CO and HC emissions at the original engine's spark timing of 15 CAD BTDC. Meanwhile, the NO emission was slightly increased, but still much smaller than that in GPI only condition. However, the IMEP and combustion speed were slightly reduced by EDI heating. To enhance the effect of EDI heating, experiments were conducted at varied spark timing. The results at the MBT timing (19 CAD BTDC) showed that the reduction of IMEP by EDI heating was less significant whilst the CO and HC emissions were effectively reduced. Therefore EDI heating was effective to address ethanol's low evaporation rate and over-cooling effect issues in the development of EDI+GPI engine in terms of minimizing the emissions.

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Definitions and Abbreviations

Acronyms

ABDC	After bottom dead center
ASOI	After the start of injection
ATDC	After top dead center
BBDC	Before bottom dead center
BDC	Bottom dead center
BTDC	Before top dead center
CAD	Crank angle degrees
CFD	Computational fluid dynamics
DI	Direct injection
ECFM	Extended Coherent Flame Model
EDI	Ethanol direct injection
EDI+GPI	Ethanol direct injection plus gasoline port injection
EVC	Exhaust valve close
EVO	Exhaust valve open
GDI	Gasoline direct injection
GPI	Gasoline port injection
IC	Internal combustion
IMEP	Indicated mean effective pressure
ITNFS	Intermediate turbulent net flame stretch
IVC	Intake valve close
IVO	Intake valve open
MBT	Minimum spark advance for best torque
MFB	Mass fraction burnt
PDF	Probability Density Function
PI	Port injection
RANS	Reynolds Averaged Navier-Stokes
SI	Spark ignition
TDC	Top dead center

Symbols

A_p	Particle surface area
C_D	Drag coefficient
Di	Diffusion coefficient in air
N_i	Molar flux of vapour
P	Pressure
D	Dissipation term of flame area
Di	Diffusion coefficient in air
P_1	Source term due to turbulence interaction
P_2	Source term due to dilatation in the flame
P_3	Source term due to expansion of burned gas
P_4	Source term due to normal propagation
T	Temperature
U_L	Laminar flame speed
V	Volume
X_i	Mole fraction of species i
Y_i	Mass fraction of species i
Z	Mixture fraction
B_m	Spalding mass number
Da	Damköhler number
Ka	Karlovitz number
Re	Reynolds number
We	Weber number
Φ	Fuel/air equivalence ratio
Σ	Flame area density
Γ_K	ITNFS term
c	Progress variable
c_p	Heat capacity
d	Diameter
h	Heat transfer coefficient
m	Mass

k	Turbulent kinetic energy
r	Radius
y	Distortion of the droplet
γ	Specific heat ratio
ε	Turbulent dissipation rate
σ	Surface tension
ρ	Density
μ	Dynamic viscosity
ν	Kinematic viscosity
t	Time
μ_t	Turbulent viscosity
u	Velocity
u'	Turbulent velocity fluctuation
l_t	Integral turbulent length scale
l_d	Diffusion thickness
l_r	Reaction zone thickness
τ_t	Turbulent time scale
τ_c	Chemical time scale
τ_k	Kolmogorov time scale
k_c	Mass transfer coefficient
δ_l	Flame thickness
$\varphi_{realized}$	Percentage of charge cooling realized
CA0-10%	Combustion initiation duration
CA10-90%	Major combustion duration
Pa/Ps	Ambient-to-saturation pressure ratio
E'X'	X% ethanol by volume. e.g. E46 is 46% ethanol via direct injection plus 54% gasoline via port injection
IT'XXX'	Injection timing of XXX CAD BTDC
ΔT	Spray excess temperature
ΔT_{actual}	Actual cooling effect
ΔT_{ideal}	Ideal cooling potential

Subscripts

d	Droplet phase
g	Gas phase
i	Species i
l	Liquid phase
p	Particle
rel	Relative
sat	Saturation
∞	Ambient bulk gas