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Theoretical investigation of engine performance and exhaust emissions in ethanol-fueled dual-plug SI engine

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ABSTRACT

Most of the on-road vehicles with two and four wheels have spark-ignition (SI) engines. Investigations on SI engine performance and exhaust emission characteristics are still valuable due to environmental reasons. This study, therefore, investigated the effect of equivalence ratio, spark timing, and spark-plug location on engine performance and exhaust emission characteristics in an ethanol-fuelled dual-plug SI engines by using a theoretical model. Findings showed that dual-spark plug configuration (SpL@d) in an SI engine is the convenient solution to continue superior engine performance and exhaust emission characteristics if there are some design constraints in contrast to the centrally located single plug (SpL@c) configuration giving the best engine performance and fuel economy.

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1. INTRODUCTION

Operation with lower exhaust emissions in SI engines has become a critical challenge for the automotive industry [1]. Numerous researches have focused on the performance and the exhaust emission characteristics of spark-ignition (SI) engines [2-4]. Also, the improved engine performance is an everlasting requirement. The exploitation of dual-plug in SI engines has the potential to achieve all expectations on performance and emissions [1, 5]. Dual-plug configuration with an SI engine ensures robust and stable combustions [6]. Alternative fuels associated with engine configuration are of great significance in SI engines [7]. Ethanol (C_2H_5OH) is of great importance among biofuels [5]. It is obtained from any fermentable material [8] and has a high-octane number and flame speed [7]. Also, ethanol fuel can be used as pure or mixed with gasoline in SI engines without any modification [8]. The disadvantage of ethanol is high production cost relative to gasoline. In literature, engine performance characteristics in an ethanol-fueled dual-plug SI engine were experimentally investigated by Almeida [9]. It was found that there was an improvement in the performance parameters of all test conditions [9]. Nakayama et al. investigated a new engine concept (gasoline-fueled 1.3L 2-plug SI engine-L13A) which achieved both low fuel consumption and low emissions [10]. Wada et al. investigated fuel economy, power, and low emission technology of the i-DSI 2-plug engine [11]. They found improvement in fuel consumption and maximum engine torque-speed. Nakata et al. studied the effects of high RON fuels (ethanol and ethanol blends) on the engine thermal efficiency of the dual-plug SI engine [12]. The use of ethanol resulted in an improvement in the thermal efficiency of the SI engine and mitigated exhaust emissions (HC, NO_x, and CO_2) [12]. Raja et al. considered various gasoline-ethanol blends for twin spark-ignition engines in the study [13]. High ethanol percentage increased brake specific fuel consumption and volumetric efficiency while it decreased exhaust emissions [13]. Yontar numerically investigated the effects of ethanol and blending fuels on engine performance characteristics in a sequential ignition dual-plug SI engine [14]. The use of E85 fuel increased engine performance value when compared to gasoline.

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The literature survey showed variety of studies investigated the engine performance and exhaust emissions of dual-plug SI engines. However, a comprehensive analysis could not be found, including the effect of equivalence ratio, spark timings, and spark plug locations on these characteristics.

This study aims to reveal the effect of equivalence ratio, spark timing, and spark plug location on the performance and the exhaust emissions of ethanol-fueled dual-plug SI engines.

2. THEORETICAL MODEL

There are two theoretical study methods to study of the combustion and performance characteristics of internal combustion engines: (1) thermodynamic modelling and (2) CFD based modelling. The thermodynamic modelling method is composed of a zero and quasi-dimensional model (QD). The zero-dimensional model uses finite heat release rate, i.e., Cosine or Wiebe functions [15]. Quasi-dimensional models consider the flame propagation model for governing the combustion process in an SI engine [1, 3, 5-6, 8]. In this study, a two-zone quasi-dimensional thermodynamic cycle simulation model was used to investigate ethanol-fueled dual-plug SI engine performance and exhaust emission characteristics. The model uses the flame propagation model approach to meet the quasi-dimensional concept. An infinitesimally thin spherical flame front divides the enclosed combustion chamber into two regions called burned and unburned zones. The schematic representation of the thermodynamic model is shown in Figure 1. The thermodynamic model's governing equations (Equations 1-4) are differential form of the energy conservation equation and obtained by applying an open thermodynamic system approach to the SI engine combustion chamber. Further details of the QD thermodynamic model can be found in elsewhere [6]. Calculation of the performance parameters was carried out by Equations (10-14). Specifications for dual-plug SI engine and fuel properties are shown in Table 1.

Table 1. Engine specifications and fuel properties [4, 5].

	Parameter	*Value*
Engine	Bore x Stroke (mm)	73 x 80
	Connecting rod length (mm)	135
	Compression ratio (-)	10.8:1
	Ignition system	DPSI
	Maximum power (kW@rpm)	63@5700
	Maximum torque (Nm@rpm)	119@2800
Fuel	Chemical formula	C_2H_5OH
	Molecular weight (kg/kmol)	46.07
	Stoichiometric AFR by mass (-)	8.94
	Lower heating value (MJ/kg)	≈27
	Research octane number (-)	111
	Laminar flame speed (cm/s)	≈ 39

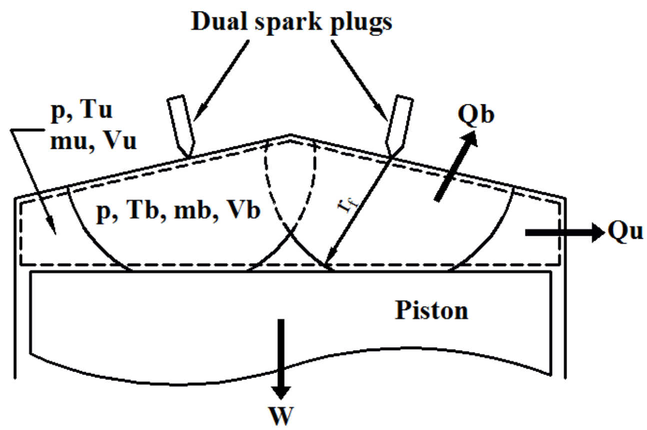


Fig. 1. Schematic of the two-zone quasi-dimensional thermodynamic model [6].

Three spark plug locations (diagonally located two (SpL@d), centrally located single (SpL@c), and side located single spark plug (SpL@s) on cylinder head), equivalence ratios (from 0.8 to 1.1 by 0.1), spark timings (from -35 CA to -20 CA by 5 CA) and nominal engine speed (5700 rpm) [6] were considered to represent of the operating conditions in the simulation studies. The equation set of the QD thermodynamic model was simultaneously solved by computer code.

Governing equations of the QD model:

$$\dot{Q} - p\dot{V} = m\dot{u} + u\dot{m} + \dot{m}_L \dot{h}_L/\omega \tag{1}$$

$$\dot{p} = \frac{A+B+C}{D+E} \tag{2}$$

$$\dot{T}_b = \frac{-h_g A_b (T_b - T_w)}{\omega m C_{Pb} x_b} + \frac{v_b}{C_{Pb}}\frac{\partial \ln v_b}{\partial \ln T_b}\dot{p} + \frac{(h_u - h_b)}{x_b C_{Pb}}\left[\dot{x}_b - \left(x_b - x_b^2\right)\frac{C_L}{\omega}\right] \tag{3}$$

$$\dot{T}_u = \frac{-h_g A_u (T_u - T_w)}{\omega m C_{Pu}(1-x_b)} + \frac{v_u}{C_{Pu}}\frac{\partial \ln v_u}{\partial \ln T_u}\dot{p} \tag{4}$$

where,

$$A = \frac{1}{m}\left(\dot{V} + \frac{VC}{\omega}\right) \tag{5}$$

$$B = \frac{h_g}{\omega m}\left[\frac{v_b}{C_{Pb}}\frac{\partial \ln v_b}{\partial \ln T_b}\left(1 - \frac{T_w}{T_b}\right)A_b + \frac{v_u}{C_{Pu}}\frac{\partial \ln v_u}{\partial \ln T_u}\left(1 - \frac{T_w}{T_u}\right)A_u\right] \tag{6}$$

$$C = -(v_b - v_u)\dot{x}_b - v_b\frac{\partial \ln v_b}{\partial \ln T_b}\frac{(h_b - h_u)}{C_{Pb}T_b}\left[\dot{x}_b - \frac{(x_b - x_b^2)C_L}{\omega}\right] \tag{7}$$

$$D = x_b\left[\frac{v_b^2}{C_{Pb}T_b}\left(\frac{\partial \ln v_b}{\partial \ln T_b}\right)^2 + \frac{v_b}{p}\frac{\partial \ln v_b}{\partial \ln p}\right] \tag{8}$$

$$E = (1 - x_b)\left[\frac{v_u^2}{C_{Pu}T_u}\left(\frac{\partial \ln v_u}{\partial \ln T_u}\right)^2 + \frac{v_u}{p}\frac{\partial \ln v_u}{\partial \ln p}\right] \tag{9}$$

Performance parameters are given below equations.
Mean indicated pressure:

$$p_{mi} = \frac{W_i}{V_h} \tag{10}$$

Mean effective pressure:

$$p_{me} = p_{mi} - p_{m,m} \tag{11}$$

Effective (brake) power:

$$P_e = \frac{p_{me} V_h z n}{60k} \tag{12}$$

Effective efficiency:

$$\eta_e = \frac{p_{me} R T_o}{F_s \phi H_u p_o \eta_v} \tag{13}$$

Specific fuel consumption:

$$b_e = \frac{3600}{H_u \eta_e} \tag{14}$$

2.1. Validation of the Theoretical Model

Two methods were used to validate the theoretical model: (1) engine performance data, including brake power and engine torque, and (2) mean absolute percentage error (MAPE) [5]. MAPE was calculated by Equation (15).

$$MAPE = \frac{1}{n}\sum_{i=1}^{n}\left|\frac{t_i - c_i}{t_i}\right| \times 100 \tag{15}$$

Where *t* is the experimental data, *c* is the model data, and *n* is the number of data points. Figure 2 shows validation of the presented theoretical model by using performance data. The numerical results of the validation tests are also given in Table 2. Validation tests (Figure 2 and MAPE) show that the presented model complies with the existing literature [4].

Table 2. Numerical results of the validation tests

Performance parameters	*Test 1 (Deviation from the exp. data)*	*Test 2 (MAPE)*
Engine torque	11.3% @ 2800 rpm	7.69%
Engine power	3.84% @ 5700 rpm	8.51%

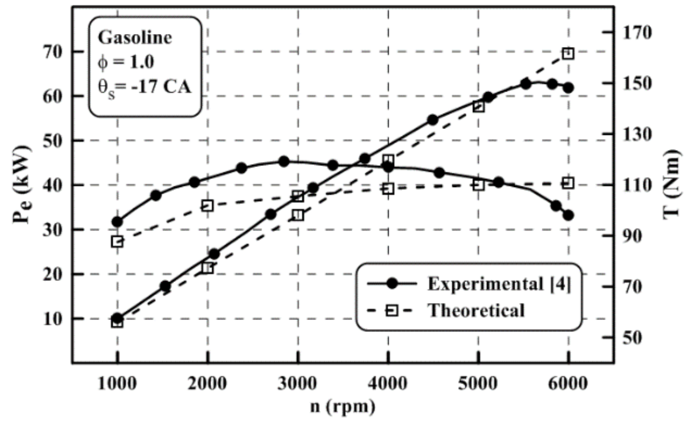


Fig. 2. Validation of the QD thermodynamic model.

3. RESULTS AND DISCUSSION

3.1. Evaluation of Performance Parameters

Figures 3(a), (b), and (c) show the effect of equivalence ratio on brake power, brake specific fuel consumption, and maximum burned gas temperature of ethanol-fueled dual-plug SI engine, respectively. The highest brake power in the simulation was obtained at the SpL@c configuration, having the shortest flame travel distance throughout all equivalence ratios, as shown in Figure 3(a). The data of brake power at SpL@d case were close to SpL@c case because dual ignition accelerated the combustion process by increased flame front area [3]. Enhanced combustion led to expanding the brake power of the SI engine. Maximum brake power was obtained in the vicinity of the stoichiometric ratio (ϕ=1) at all three cases. 2.2% higher brake power at SpL@c was obtained for the stoichiometric ratio compared to the SpL@d case. Operating at a higher equivalence ratio (in other words, rich mixture) did not affect the brake power. Brake specific fuel consumptions (bsfc) were indicated in Figure 3(b). bsfc at SpL@c case was minimum throughout all equivalence ratios. There were significant differences among brake specific fuel consumptions at lower equivalence ratios. It was determined that the equivalence ratio did not dominate at higher equivalence (more than stoichiometric) ratios. Equivalence ratio ϕ=0.9 led to minimum bsfc in the ethanol-fueled dual-plug engine.

Figures 4 (a), (b), and (c) show the effect of spark timing on brake power, brake specific fuel consumption, and maximum burned gas temperature of ethanol-fueled dual-plug SI engine, respectively. Spark timing close to the top dead centre resulted in higher brake power and lower brake specific fuel consumption. SpL@c case can be reported here as the best configuration, as well. Maximum burned gas temperature is another critical parameter in SI engines since the higher burned gas temperature can result in higher NO_x emissions. The maximum burned gas temperature was obtained at SpL@c and SpL@d cases, as shown in Figures 3 (c) and Figures 4 (c).

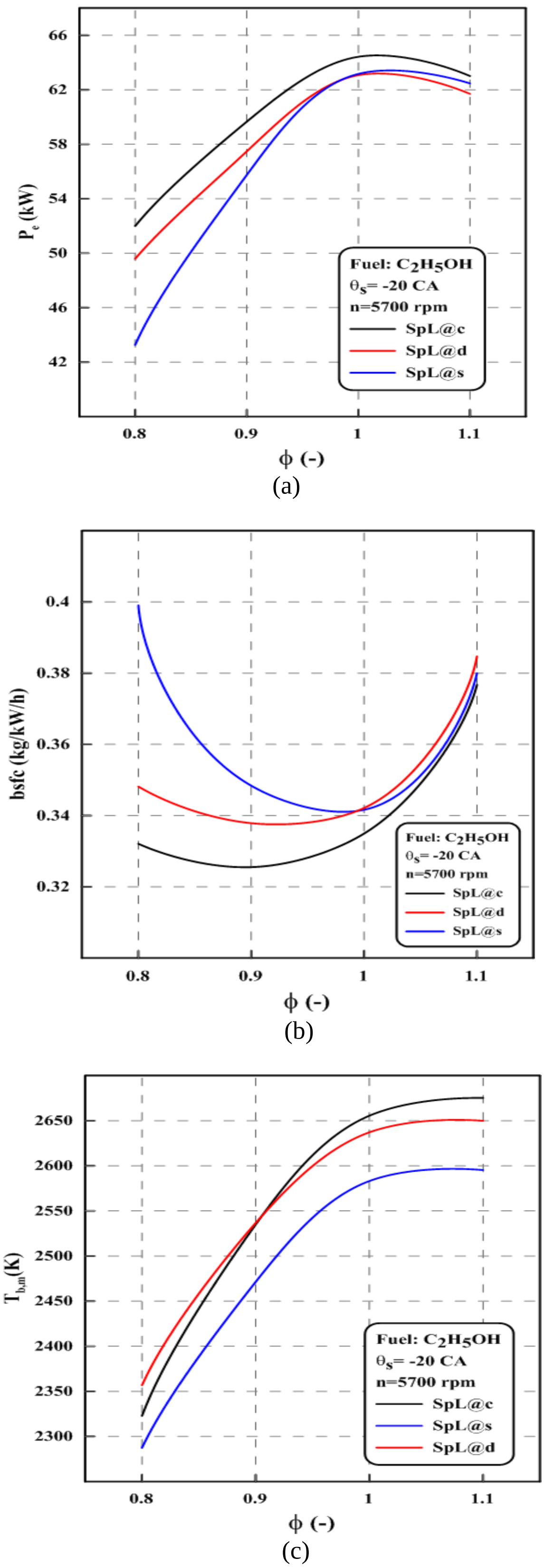


Fig. 3. Effect of equivalence ratio on the performance parameters: (a) brake power, (b) brake specific fuel consumption, and (c) maximum burned gas temperature.

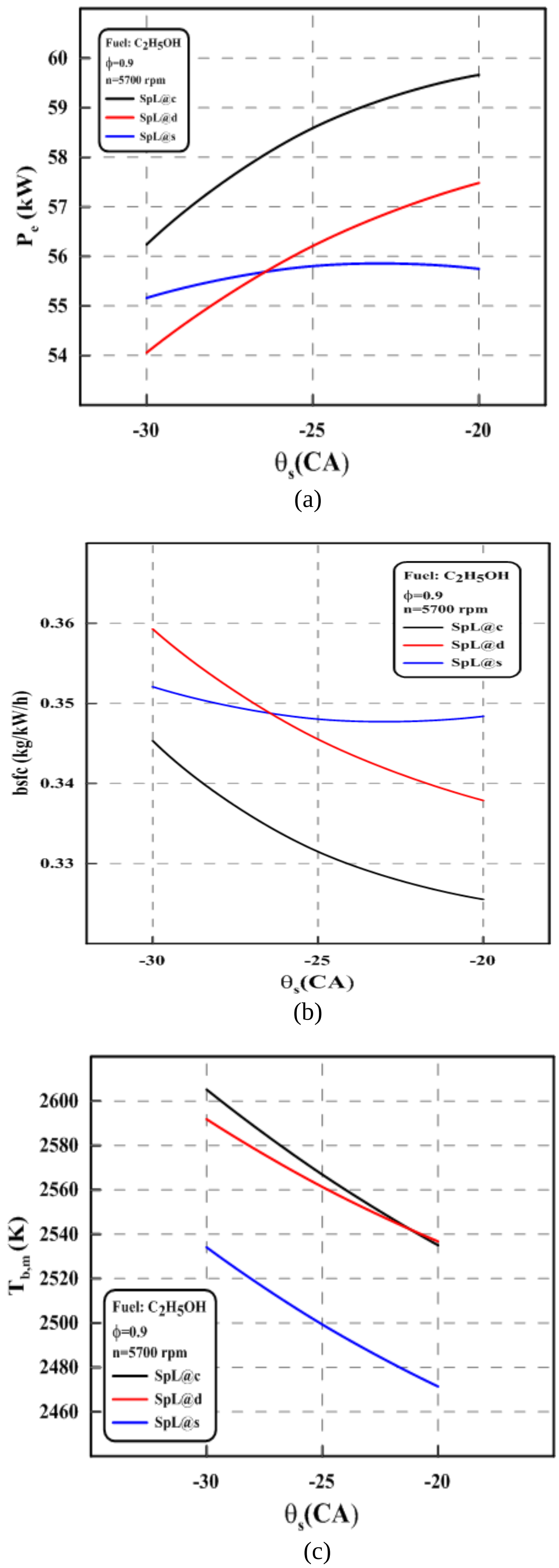
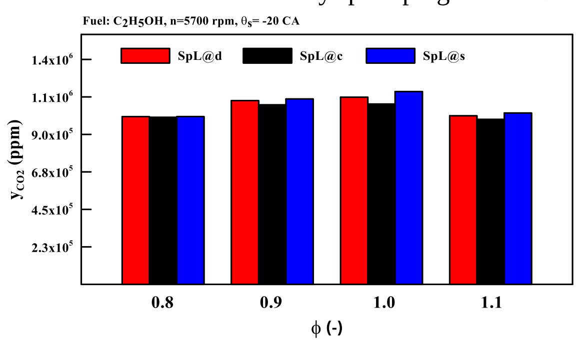


Fig. 4. Effect of spark timings on the performance parameters: (a) brake power, (b) brake specific fuel consumption, and (c) maximum burned gas temperature.

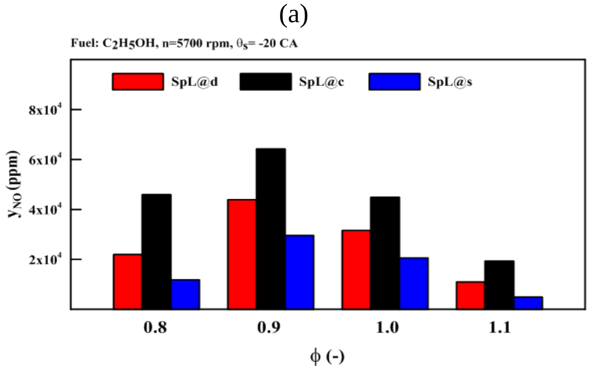
3.2. Evaluation of Exhaust Emissions

Figure 5 shows the effects of the equivalence ratio on exhaust emissions (CO_2, CO, and NO). CO_2 emission indicates combustion efficiency. A higher value of CO_2 emission leads to more global warming. Figure 5(a) relates to CO_2 emission. The highest CO_2 emission was at a stoichiometric ratio with various spark plug configurations. Improvement in the combustion process with dual-plug can be indicated according to CO_2 emission levels in Figure 5 (a). CO emission increased as the equivalence ratio increased because of the lack of oxygen in the combustion chamber. NO emission was the highest at ϕ=0.9 (a bit lean mixture). This finding complies with the literature [16].

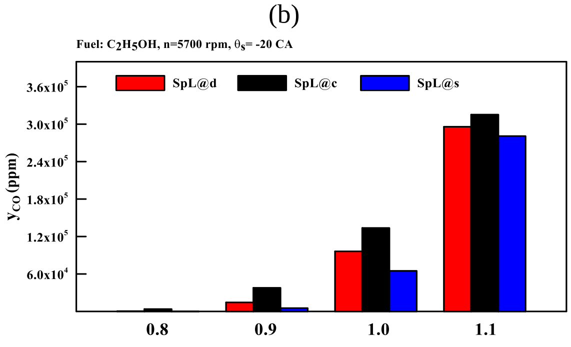
The effect of spark timing on exhaust emissions was presented in Figure 6. Minimum CO_2 emission was obtained from SpL@c configuration due to fast combustion, as shown in Figure 6 (a). CO emission results have the same trend at each spark timing and are independent from spark timings as expressed in Figure 6 (b). Figure 6 (c) shows NO emission results. The most effective parameter on NO emission is clearly spark plug location.



(a)

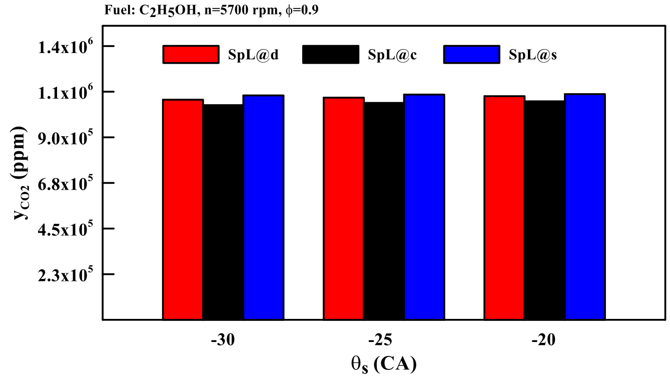


(b)

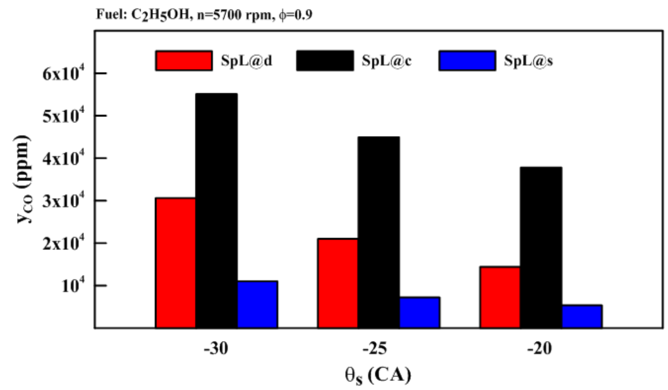


(c)

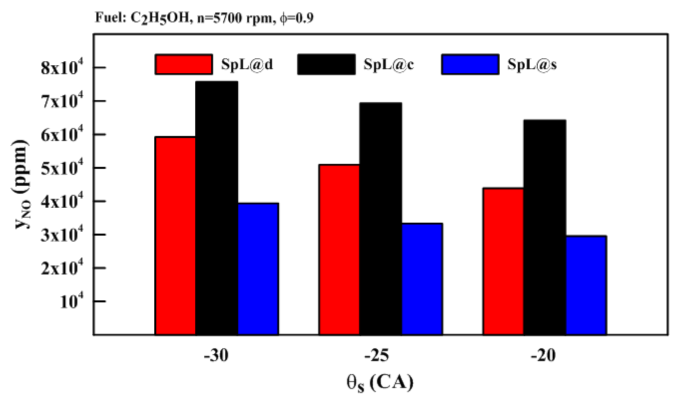
Fig. 5. Effect of equivalence ratio on exhaust emissions:(a) CO_2, (b) CO, and (c) NO (cont'd).



(a)



(b)



(c)

Fig. 6. Effect of spark timing on exhaust emissions: (a) CO_2, (b) CO, and (c) NO (cont'd).

4. CONCLUSIONS

This study theoretically investigated the effect of equivalence ratio, spark timing, and spark plug location on engine performance and exhaust emission characteristics in ethanol-fueled dual-plug SI engines. The findings of the present study led to the following conclusions:

- The centrally located single plug configuration (SpL@c) gives the best engine performance and fuel economy.
- Dual-spark plug configuration (SpL@d) in an SI engine is more convenient solution to continue improved engine performance characteristics if there are some design constraints.

– Equivalence ratio ϕ=0.9 led to minimum bsfc in the ethanol-fueled dual-plug engine.
– SpL@c case has a maximum burned gas temperature.
– The highest CO_2 emissions were at the stoichiometric ratio with various spark plug configurations.
– The highest NO emission was obtained at ϕ=0.9 (a bit lean mixture).
– CO emission decreased at advanced spark timing.

Nomenclature

-	Dimensionless parameter
be	Brake specific fuel consumption
CA	Crank angle (°)
CO	Carbon monoxide
CO_2	Carbon dioxide
F_s	Stoichiometric fuel/air ratio (-)
h	Specific enthalpy (kJ kg^{-1})
H_u	Lower heating value (kJ kg^{-1})
k	Constant (k=2 for 4-stroke engines)
m	Mass (kg)
MAPE	Mean absolute percentage error (%)
n	Engine speed (rpm)
NO_x	Nitrogen oxide
p	Cylinder pressure (bar)
p_o	Ambient pressure (bar)
P_e	Brake power (kW)
P_{me}	Mean effective pressure (bar)
P_{mi}	Mean indicated pressure (bar)
$P_{m,m}$	Mean pressure of mechanical losses (bar)
Q	Heat loss (J)
QD	Quasi-dimensional
r_f	Flame radius (mm)
R	Ideal gas constant (J g^{-1} K^{-1})
SpL@c	Centrally-located single plug
SpL@d	Diagonally located dual-plugs
T	Temperature (K)
T	Engine torque (Nm)
T_o	Ambient temperature (K)
U	Specific internal energy (kJ kg^{-1})
V	Instantaneous cylinder volume (m^3)
V_h	Displacement volume (m^3)
W	Work (J)
x	Mass fraction burned (-)
y_{CO}	Volumetric ratio of CO in combustion products (ppm)
y_{CO2}	Volumetric ratio of CO_2 in combustion products (ppm)
y_{NO}	Volumetric ratio of NO in combustion products (ppm)
z	Cylinder number (-)
ϕ	Equivalence ratio (-)
η_e	Brake thermal efficiency (-)
η_v	Volumetric efficiency (-)
θ	Crank angle (°)
ω	Angular velocity (s^{-1})
b	Burned
i	Indicated
u	Unburned
L	Loss

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Biographies

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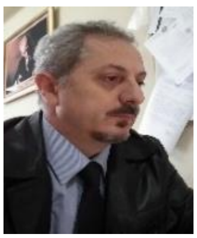

Atilla Bilgin was born in 1963 and passed away in 2021. He completed his Ph.D. study in the Department of Mechanical Engineering at the Karadeniz Technical University in 1994. He was a full professor at the Karadeniz Technical University. His research interests were internal combustion engines, alternative fuels and combustion, motor vehicles, CFD analysis, turbulence modelling, thermodynamic cycle analysis and exergy analysis.

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