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The Effects of Port Water Injection on Spark Ignition Engine Performance and Emissions Fueled by Pure Gasoline, E5 and E10

Farhad Salek [1], Meisam Babaie [2,*], Maria Dolores Redel-Macias [3], Ali Ghodsi [1], Seyed Vahid Hosseini [1], Amir Nourian [2], Martin L Burby [2] and Ali Zare [4]

[1] Faculty of Mechanical and Mechatronic Engineering, Shahrood University of Technology, Shahrood 3619995161, Iran; farhadsalek@shahroodut.ac.ir (F.S.); alighodsi89@gmail.com (A.G.); v_hosseini@shahroodut.ac.ir (S.V.H.)

[2] School of Science, Engineering and Environment, University of Salford, Manchester M5 4WT, UK; a.nourian@salford.ac.uk (A.N.); m.burby@salford.ac.uk (M.L.B.)

[3] Rural Engineering Department, Ed Leonardo da Vinci, Campus de Rabanales, Universidad de Córdoba, 14071 Córdoba, Spain; ig1remam@uco.es

[4] Flow, Aerosols & Thermal Energy (FATE) Group, School of Engineering, Deakin University, Geelong VIC 3216, Australia; ali.zare@deakin.edu.au

* Correspondence: m.babaie@salford.ac.uk; Tel.: +44-(0)161-295-5547

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Abstract: It has been proven that vehicle emissions such as oxides of nitrogen (NOx) are negatively affecting the health of human beings as well as the environment. In addition, it was recently highlighted that air pollution may result in people being more vulnerable to the deadly COVID-19 virus. The use of biofuels such as E5 and E10 as alternatives of gasoline fuel have been recommended by different researchers. In this paper, the impacts of port injection of water to a spark ignition engine fueled by gasoline, E5 and E10 on its performance and NOx production have been investigated. The experimental work was undertaken using a KIA Cerato engine and the results were used to validate an AVL BOOST model. To develop the numerical analysis, design of experiment (DOE) method was employed. The results showed that by increasing the ethanol fraction in gasoline/ethanol blend, the brake specific fuel consumption (BSFC) improved between 2.3% and 4.5%. However, the level of NOx increased between 22% to 48%. With port injection of water up to 8%, there was up to 1% increase in engine power whereas NOx and BSFC were reduced by 8% and 1%, respectively. The impacts of simultaneous changing of the start of combustion (SOC) and water injection rate on engine power and NOx production was also investigated. It was found that the NOx concentration is very sensitive to SOC variation.

Keywords: E10 biofuel; Ethanol; NOx; water port injection; start of combustion

1. Introduction

Vehicle emissions are one of the main concerns around the world from both environmental and health perspectives. When the concentration of emissions, such as carbon monoxide (CO) and oxides of nitrogen (NOx) in the air exceeds a threshold limit, inhalation of these emissions by humans can result in deleterious health effects [1–3]. Recently, researchers have reported a link between air pollution and increased death rates from the Corona virus [4,5]. Due to such detrimental effects, authorities and researchers around the world are developing measures that can mitigate the adverse effects of using fossil fuels for internal combustion (IC) engines.

One of the solutions is to use alternative renewable fuels instead of fossil fuels. In IC engines, biofuels extracted from different renewable sources can be added into the fossil fuels at various percentages to provide a fuel mixture, the thermo-physical properties of which are better than pure gasoline. For example, ethanol has been used as an additive for gasoline in various countries for a number of years and is known as E5 [6,7]. One advantage of biofuels such as E5 is that they can be produced from renewable energy sources such as food waste through thermal processes [8–12]. The ethanol share in E5 fuel is about 5% which has increased to 10% in E10 and it has been recently employed in some countries based on their emission regulations (i.e., WLTP regulation) and fuel standards [6]. However, there are a few reservations that held out the increase in the ethanol level within the fuel blend (i.e., from E5 to E10) in some countries such as the UK despite the potential environmental benefits of biofuels, especially on CO_2 reduction [13].

Changing the ethanol percentage in the fuel will affect the thermo-physical properties of the fuel blend with positive impacts on combustion and some emissions [14,15]. The effects of adding ethanol to gasoline fuel on engine performance and combustion characteristics have been investigated in the literature [2,16–19]. Adding ethanol to fuel can increase the mixed fuel octane number. Increasing the octane number of the fuel can reduce the engine knock intensity [20,21]. This means a better combustion efficiency, increasing the power and decreasing different exhaust emissions for the engine, except for NOx [22]. It has been reported that injection of biofuels into the engines leads to higher NOx emission production. This can be due to the addition of the fuel oxygen content which results in higher flame temperature during combustion [23,24].

The hydrous ethanol was also introduced as an alternative to ethanol for reducing engine NOx emissions [1,3,22]. The hydrous ethanol contains water, the injection of which into the combustion chamber results in a reduction in flame temperature, leading to lower NOx formation. However, as reported by Costa [22], this may bring some drawbacks at lower engine speeds such as a reduction in engine thermal efficiency. Apart from fuel solutions, other technical modifications are recommended in the literature to reduce the extra emissions produced by biofuel [25]. Zhuang et al. [16] studied the effects of ignition timing in a gasoline/ethanol dual-fuel engine using microscopic combustion in a numerical analysis. In this study, the start of combustion was changed in each configuration and its impacts on the combustion process that directly affect the emission characteristics were investigated. According to their result, retarding the start of combustion can improve the combustion efficiency. Huang et al. [17] conducted an experimental study in which the impacts of ethanol injection timing for an engine with an ethanol direct injection system was investigated. In this study, the ethanol was injected directly into the combustion chamber in a gasoline engine with a port fuel injection (PFI) system. Based on the results of their investigation, delaying the ethanol injection timing led to overcooling the combustion chamber which resulted in a reduction in NO emissions and an increase in CO and hydrocarbons (HC).

In this paper, the effect of the port injection of water into a spark ignition gasoline engine equipped with multi-point fuel injection system and fueled by gasoline, E5 and E10 was investigated with respect to performance and emissions. The first part of the work involved the experimental analysis of a gasoline fueled engine to obtain its main functional outputs. After that, the engine was modeled in AVL BOOST software and the results of the model were validated against experimental data. The Latin Hypercube sampling method was employed to define the number of experiments and values of each design point in each experiment, and the performance of the engine and its emissions were evaluated in various rates of water injection. This study also investigated the impacts of variation of the start of combustion angle on engine output parameters when different rates of water are injected.

2. Methodology

2.1. Experimental Study

The KIA Cerato engine was used for this study and the performance data was collected experimentally to validate the AVL model. The technical specification of the KIA Cerato engine is provided in Table 1. The engine was tested on an engine test bed and its main output parameters were measured at various speeds under full load condition (Figure 1). During the experimental tests, the engine functional outputs such as BSFC, NOx emission rate, BMEP, torque and power at various speeds had been recorded.

Table 1. Technical specifications of KIA Cerato engine.

Parameter	Unit	Value
Bore	mm	86
Stroke	mm	86
Connecting rod length	mm	143.5
Number of Cylinders	-	4
Displacement volume	cc	2000
Maximum RPM	RPM	7000
Rated RPM	RPM	6000
Compression Ratio		10.5

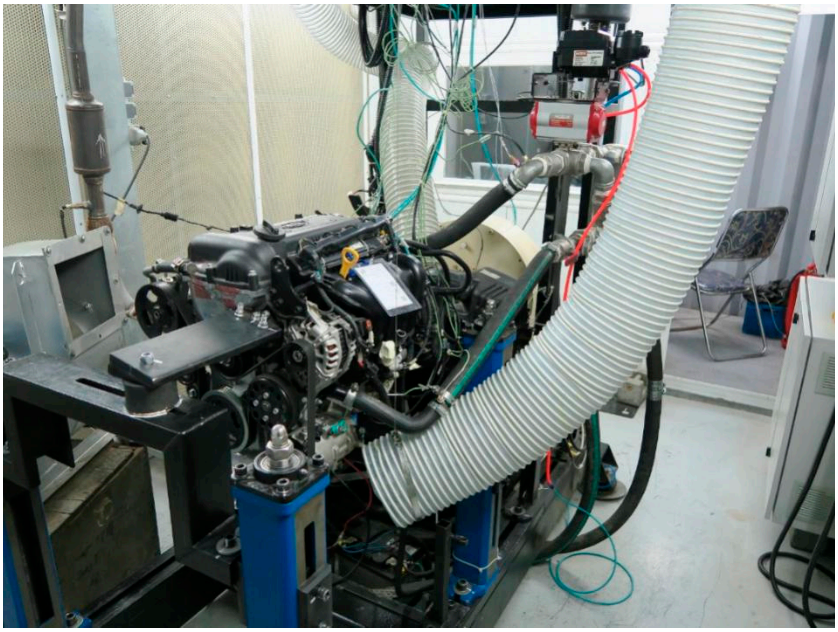


Figure 1. Experimental testing of KIA Cerato engine on engine test bed.

2.2. Engine Mathematical Model

After completion of the experimental phase, the engine was mathematically modelled using the AVL BOOST software. AVL BOOST provides a 1-D thermodynamic precise model for internal combustion engines and it is widely used in academia and the automotive industry. The AVL model of the KIA Cerato engine is shown in Figure 2. The engine contains 4 gasoline/ethanol injectors (I1, I2, I3, I4) and one water injector (I5). The engine cylinders are named C1, C2, C3 and C4, and the catalyst converter and air cleaner are indicated as CAT1 and CL1, respectively. PL3 and PL4 are the engine muffler and the measure points (MP), which are used for measuring air thermophysical properties in various sections.

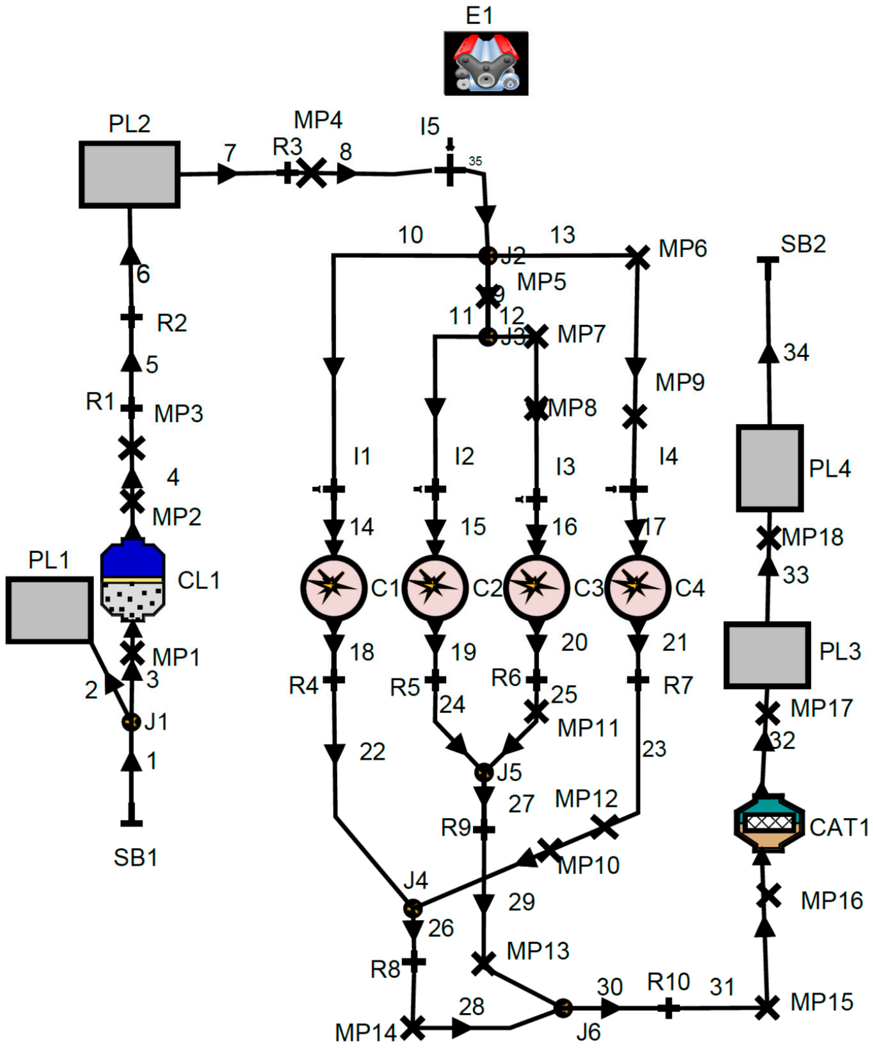


Figure 2. Block diagram of engine model in AVL BOOST software.

Due to the variation of the lower heating values with different fuels in dual-fuel (and blended fuels) engines, the equivalent brake specific fuel consumption (BSFC) should be calculated and used to compare the fuel consumptions, as shown in Equation (1):

$$BSFC_{eqv} = \frac{\dot{m}_{gasoline} + \dot{m}_e \frac{LHV_e}{LHV_{gasoline}}}{\dot{W}_{engine}} \tag{1}$$

where, $\dot{m}_{gasoline}$, $\dot{m}_e$ and $\dot{W}_{engine}$ are the gasoline mass flow rate, ethanol mass flow rate and engine power output parameters, respectively.

The ethanol was added to the fuel at 5% and 10% volume fractions and the fuels used in this study are shown in Table 2. The ethanol volume fraction in gasoline/ethanol mixture injected by I1, I2, I3 and I4 injectors are also indicted in this table.

Table 2. The specification of fuels used in this study.

Fuel Type	Ethanol Volume Fraction(%)
Gasoline	0
E5 (5% ethanol, 95% gasoline)	5
E10 (10% ethanol, 90% gasoline)	10

Water was then injected into the engine in various ratios when needed. The injection of water was continuous. The water injection ratio (WIR) is defined by the Equation (2):

$$\mathrm{WIR} = \dot{m}_{water} / \dot{m}_{fuel} \quad (2)$$

where the WIR and $\dot{m}_{water}$ are the water injection ratio and water mass flow rate, consecutively.

2.2.1. Combustion and Heat Transfer Models

The Wiebe two-zone combustion model has been chosen for modelling the combustion procedure in the engine AVL model. In this combustion model, the combustion chamber is divided into two zones (burnt and unburnt zones) in which the mixture temperature for each zone is calculated separately [26,27]. This model is widely used for the modelling of dual-fuel engines with species transport in AVL BOOST [26,27]. In the Wiebe two-zone model, the fuel mass burned fraction (x) during combustion is expressed by Equation (3):

$$x = 1 - \exp[-a\left(\frac{\propto - SOC}{BDUR}\right)^{m+1}] \quad (3)$$

SOC, *BDUR*, $\propto$, m and a are the start of the combustion, burn duration, crankshaft angle, Wiebe shape and Wiebe parameter, respectively. Applying first law of thermodynamics to each zone provides:

$$\frac{dm_b u_b}{d\propto} = -P_c\frac{dV_b}{d\propto} + \frac{dQ_f}{d\propto} - \sum\frac{dQ_{wb}}{d\propto} + h_u\frac{dm_b}{d\propto} - h_{BB,b}\frac{dm_{BB,b}}{d\propto} \quad (4)$$

$$\frac{dm_u u_u}{d\propto} = -P_c\frac{dV_u}{d\propto} - \sum\frac{dQ_{wu}}{d\propto} - h_u\frac{dm_B}{d\propto} - h_{BB,u}\frac{dm_{BB,u}}{d\propto} \quad (5)$$

where dm_u, $P_c\frac{dV_b}{d\propto}$, $\frac{dQ_f}{d\propto}$, $\frac{dQ_w}{d\propto}$, $h_u\frac{dm_b}{d\propto}$ and $h_{BB,b}\frac{dm_{BB,b}}{d\propto}$ are variation of in-cylinder internal energy, piston work, fuel input energy, wall heat losses, enthalpy flow from unburnt to burnt zone and blow by enthalpy, respectively. Furthermore, for modelling heat transfer between walls and mixture, the Woschni 1978 model was employed [28,29].

2.2.2. Emission Model

For calculation of the NOx formation rate in AVL BOOST, the Pattas and Hafner equation [26] combined with Zeldovich mechanism [26] were used:

$$r_{NO} = C_{PPM}C_{KM}(2{,}0)(1 - a_{NO}^2)[\frac{r_1}{1+a_{NO}AK_2} + \frac{r_4}{1+AK_4}] \quad (6)$$

$$a_{NO} = \frac{c_{NO.act}}{c_{NO.equ}}\frac{1}{C_{KM}} \quad (7)$$

$$AK_2 = \frac{r_1}{r_2+r_3} \quad (8)$$

$$AK_4 = \frac{r_4}{r_5+r_6} \quad (9)$$

where C_{PPM}, C_{KM}, C_i and r_{NO} are the post processing multiplier, kinetic multiplier, molar concentration and reaction rate of NOx, respectively.

2.2.3. Knock Model

The knock model is used for calculating the minimum octane number required for working free of knock. It is employed for the assessment of engine knock intensity variation when various fuels are injected [30]. As indicated in Equation (10), K_c is the parameter that identifies the knock onset. In this equation, τ is the ignition delay and t is the elapsed time from start of compression; when its value reaches 1 the knocking is initiated [31].

$$K_c = \int_0^t \frac{1}{\tau_{iD}}dt \quad (10)$$

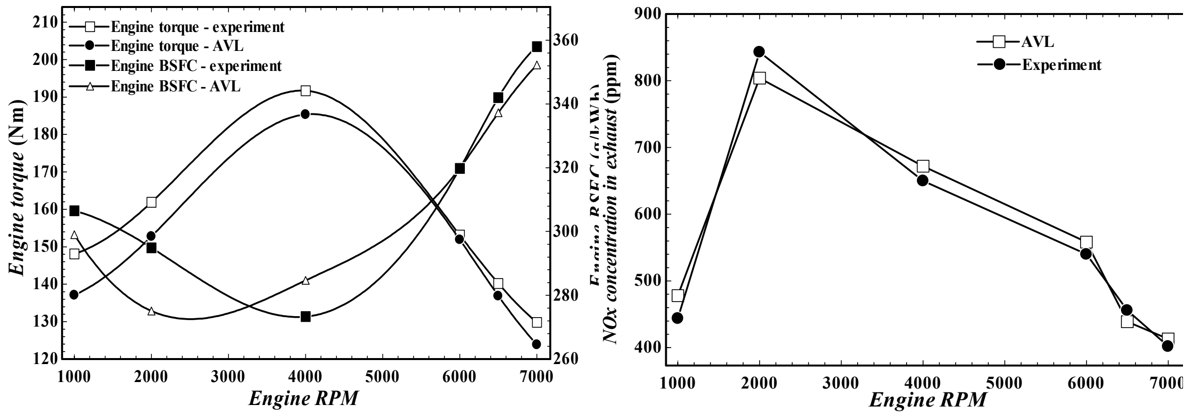
Furthermore, in knock modelling, the ignition delay has a strong relation with fuel octane number and in-cylinder gas thermophysical condition [30]:

$$\tau_{iD} = A.(\frac{ON}{100})^{a}.p^{-n}e^{B/T} \tag{11}$$

where τ_{iD}, *ON*, *p* and *T* are ignition delay, minimum octane number, pressure and temperature of in-cylinder gas, consecutively. Moreover, *A*, *a*, *n* and *B* are constant parameters used in the model which are equal to 17.68, 3.402, 1.7 and 3800 in this study, respectively [30].

2.3. Validation

The validation of the computer model was performed by comparing the results of running the gasoline fueled engine at various RPMs in the engine test room with the AVL model outputs. The engine BSFC, torque and NOx production were used for validation purposes. Figure 3a shows the engine torque and BSFC from the experiments as well as the AVL model. Furthermore, the NOx concentrations from the engine exhaust for both the AVL and experiment are compared in Figure 3b. By comparing the experimental and simulation values, it can be found that the maximum error of the AVL model is below 8%, which proves that the developed model can represent the engine performance with a high degree of accuracy.



(**a**) (**b**)

Figure 3. (**a**) Engine Torque and BSFC and (**b**) NOx concentration in the exhaust in the AVL model and experimental tests obtained from running the engine by gasoline at full load and various RPMs.

2.4. Design of Experiments Method and Analysis

The parameters studied in this research (design variables) were the start of combustion (SOC) angle, water injection ratio and ethanol volume fraction as presented in Table 3. As it can be seen in this table, the maximum value of the water injection ratio is 8%, because the R-square value of different responses in regression analysis is higher than 99%, when the water injection ratio is below 8%. The Latin Hypercube sampling method was used to define the sampling space. As shown in Figure 4, the Latin Hypercube recommended 200 points for the design variables for this study. These points were used in the regression analysis to obtain the engine output parameters and emissions as the function of design variables at the engine rated RPM (6000 RPM) at which the maximum power will be reached. The flow chart of the research method proposed in this paper is also shown in Figure 5.

Table 3. Maximum and minimum values of each design parameter.

Design Parameter	Unit	Minimum Value	Maximum Value
Ethanol volume fraction	-	0	0.1
Water injection ratio	-	0	0.08
Start of combustion	Degree	−20	5

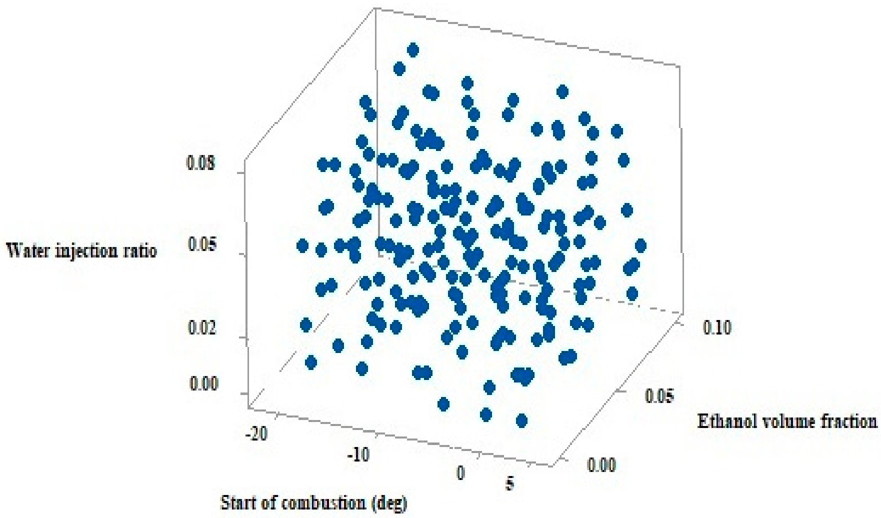


Figure 4. The sampling space and the design points in Latin Hypercube DOE method.

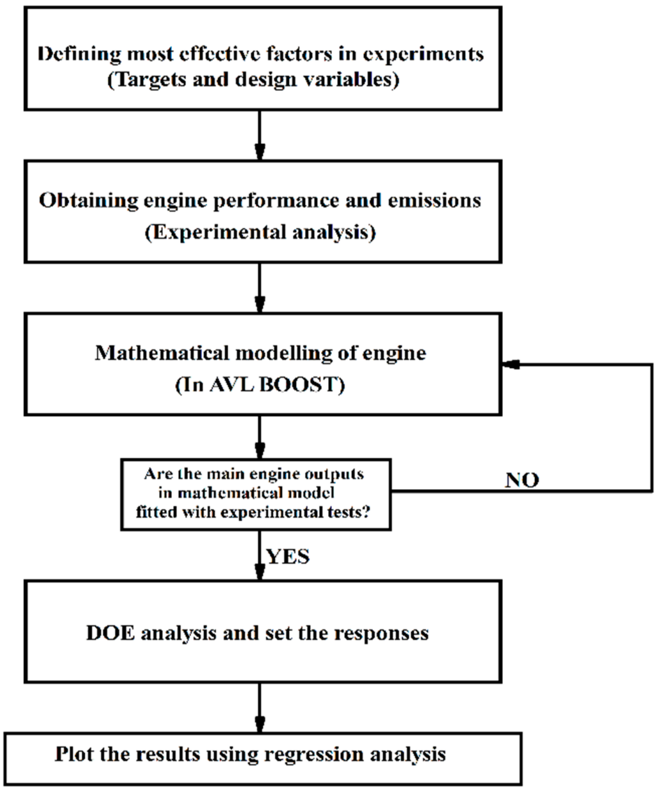


Figure 5. The research method flow chart proposed in this paper.

3. Results and Discussion

As discussed, the mathematical model of the KIA Cerato engine in AVL BOOST was developed for this study. The water port injector was added to inject water into the engine at different ratios.

The range of the design variables (SOC, water injection ratio and ethanol volume fraction) were defined in the AVL design explorer and engine output parameters were calculated at engine rated speed (6000 RPM). The general equation format which was developed for each engine output parameter (response) by regression analysis is shown below:

$$Response = cte + (a \times SOC) + (b \times mf_e) + (c \times WIR) + (d \times SOC \times mf_e) + (e \times SOC \times WIR) + (f \times mf_e \times WIR) + (g \times SOC^2) + \left(h \times {mf_e}^2\right) + (i \times WIR^2) \quad (12)$$

where SOC, mf_e and WIR are start of combustion, ethanol volume fraction and water injection ratio design parameters, respectively. Equation (12) consists of some constant and multipliers which are calculated and provided in Table 4 for each response, with the goodness of fitness analysis for the equations (see Table 5). In Table 4, *cte* denotes the constant parameter for each response equation in the regression analysis.

3.1. Engine Performance Parameters

The effects of water port injection into the engine for pure gasoline, E5 and E10 at rated RPM and a constant SOC of −5 degree are presented in this section. Figure 6 shows the engine power output for different fuels (gasoline, E5 and E10) and different water injection ratios. As can be seen, when no water is injected into the engine, there was a slight increase in power when using E10 compared to E5 and pure gasoline. This increase in power by injection of ethanol is due to an increase in flame velocity, which is achieved by ethanol injection and resulted in an improvement of the engine performance. Furthermore, with the injection of water, there was an increase in engine power for all three fuels with the maximum engine power output of about 96.75 kW for E10 when the water injection was approximately 8%. Therefore, the combination of E10 fuel and 8% water injection can deliver around a 1.3% increase in engine power output when compared with the gasoline fueled engine.

Engine BMEP at varying water injection ratios and fuel types is shown in Figure 7. As can be seen, as in the case of power, a small increase in BMEP was observed by shifting to E10 from gasoline or E5. The engine BMEP for each fuel increased slightly with water injection; this increase was approximately 0.02 bar for E5 and 8% water injection compared to pure gasoline without any water injection, and about 0.05 bar when E10 was used.

Figure 8 presents the impacts of water port injection on engine brake thermal efficiency at various injection ratios for various fuels. As shown, there is not a significant difference between E5 and E10 in terms of the brake thermal efficiency. Compared with gasoline, brake thermal efficiency increased by about 1% for E10. Furthermore, water port injection increased the engine brake thermal efficiency for all the fuels. It was also found that the combination of ethanol with water injection does not have a negative effect on brake thermal efficiency.

Table 4. The Equation (6) constant parameters and multipliers.

Responses	cte	a	b	c	D	e	f	g	h	i
Power	89.840	−1.053	3.807	10.930	−0.056	−0.053	3.817	−0.035	2.957	−12.480
BMEP	8.992	−0.105	0.381	1.094	−0.006	−0.005	0.382	−0.004	0.296	−1.249
Brake power efficiency	25.120	−0.294	8.725	2.744	−0.067	−0.008	1.797	−0.010	6.177	−4.030
Peak cylinder temperature	2511	−5.705	230	−256.400	3.682	0.904	36.830	0.211	−170.600	87.120
NOx concentration	3.787×10^{-4}	-2.651×10^{-5}	1.688×10^{-3}	-7.633×10^{-4}	-1.019×10^{-4}	-1.545×10^{-5}	−0.002	2.357×10^{-7}	0.004	4.331×10^{-4}

Table 5. The fitness goodness for each response equation.

Responses	*p*-Value	R^2
Power	<0.0001	0.99
BMEP	<0.0001	0.99
Brake power efficiency	<0.0001	0.99
Peak cylinder temperature	<0.0001	0.99
CO concentration	<0.0001	0.99
HC concentration	<0.0001	0.99
NOx concentration	<0.0001	0.99

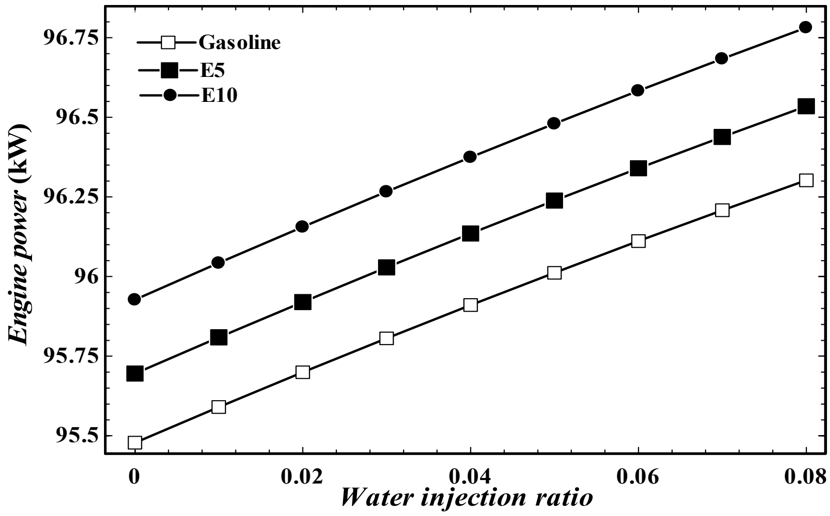


Figure 6. Engine power output in various water injection ratios for each fuel.

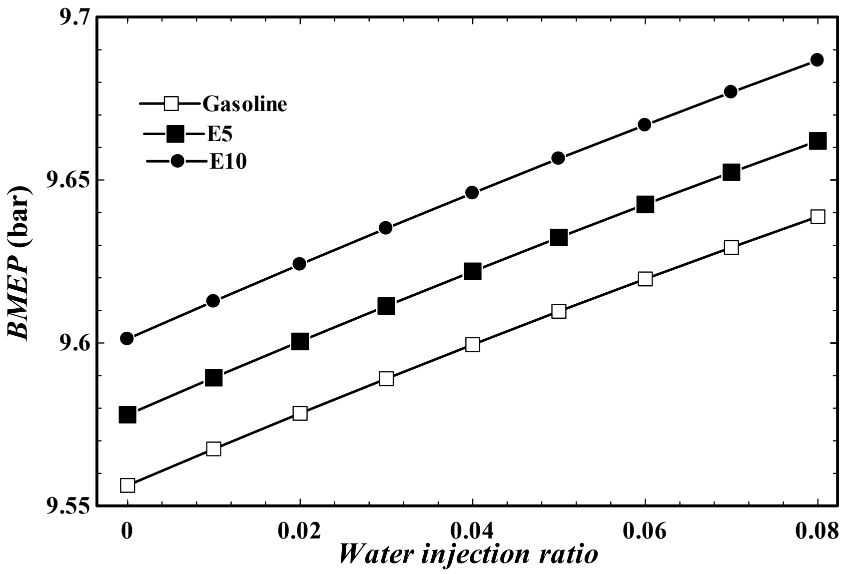


Figure 7. Engine BMEP in various water injection ratios for each fuel.

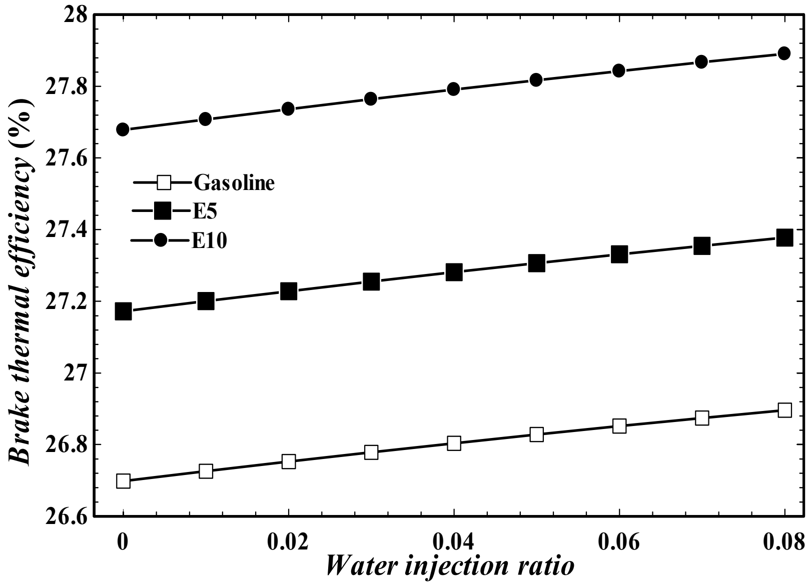


Figure 8. Engine brake power efficiency in various water injection ratios for each fuel.

The engine equivalent BSFC for different water injection rates and fuels is presented in Figure 9. It can be seen that by using E10 instead of E5, BSFC was decreased by almost 2% (to 307 g/kWh). Comparing E5 and E10 fuels with gasoline, the engine BSFC decreased by nearly 2.3% and 4.5%, respectively. The reason is due to an increase in combustion efficiency as a result of ethanol injection into the combustion chamber. It was also found that the injection of water had a positive impact on BSFC and the engine equivalent BSFC decreased by up to 1% with water injection for each fueling mode.

Knock onset is one of the most limiting factors in PFI spark ignition engines. It limits the range of fuels, compression ratio and ignition advances that can be used. It is also dependent on the resulting thermal level of the combustion chamber. All these factors change significantly with the use of different degrees of ethanol and water injection. The mathematical model used for simulating the Knock onset was presented before based on ignition delay, which depends on various parameters including octane number, pressure and temperature of in-cylinder gas. From this analysis, the minimum fuel octane number required for preventing engine knock is obtained in this study and the results are presented in Figure 10. As shown, with injection of ethanol, there is a minimal increment to the minimum octane number required to avoid the knocking, so the new fuel should satisfy this minimum requirement. However, by water injection up to 8% the minimum octane number was decreased by nearly 1.5. Therefore, it can be concluded that port injection of water is helping in reducing the knock intensity, which has been caused by the injection of E5 and E10 fuels.

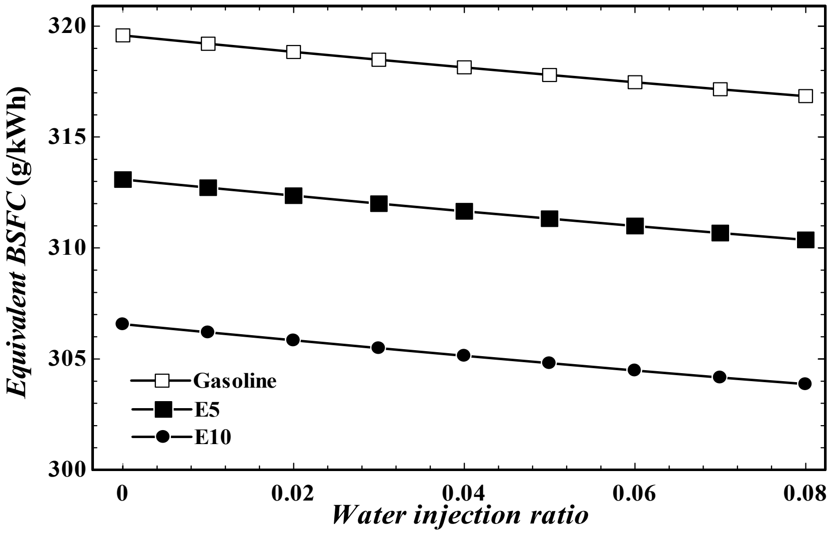


Figure 9. Engine equivalent BSFC in various water injection ratios for each fuel.

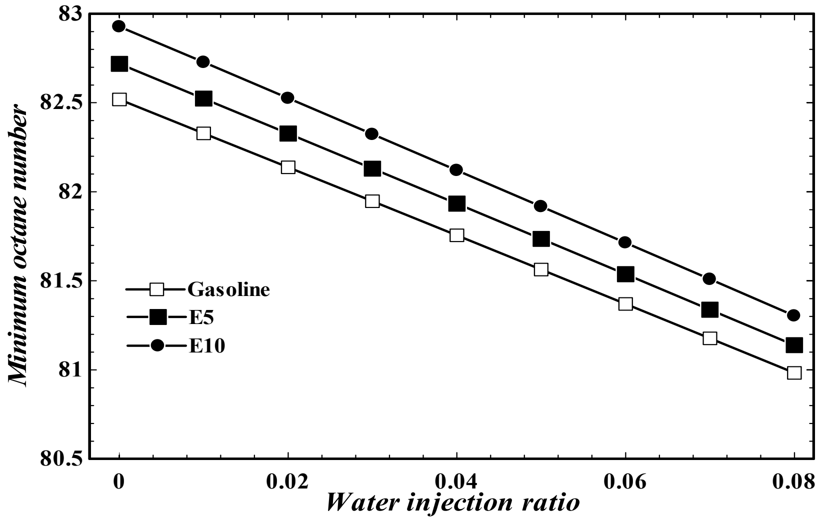


Figure 10. Minimum octane number in various water injection ratios for each fuel.

3.2. NOx Emissions and Peak Cylinder Temperature

Figures 11 and 12 show the effect of using gasoline, E5 and E10 fuels on peak cylinder temperature and NOx formation at various water injection ratios. Since the cylinder temperature is a key parameter on NOx production, it is presented and discussed together with NOx in this section. As shown, switching to E5 and E10 from pure gasoline resulted in an increase in peak cylinder temperature, thus increasing the formation of NOx by 22% and 48%, respectively. However, the peak cylinder temperature showed a decreasing trend with water injection for all fuels. Therefore, water injection can be used for reducing NOx production when using biofuels. Comparing E5 and E10, NOx concentration increased by about 22% by fuel shift, however, the injection of water showed a positive effect in reducing NOx, with a reduction in NOx concentration of up to 8%.

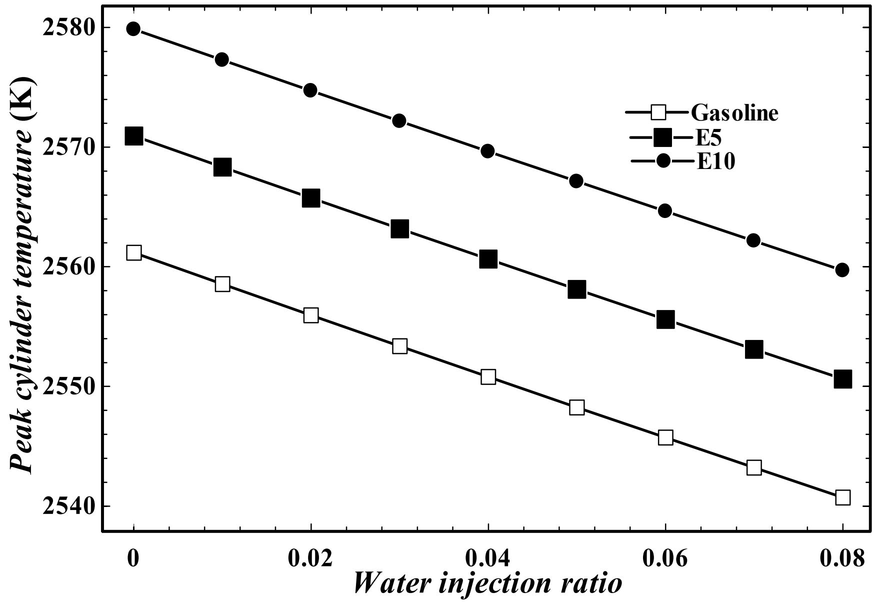


Figure 11. Peak cylinder temperature in various water injection ratios for each fuel.

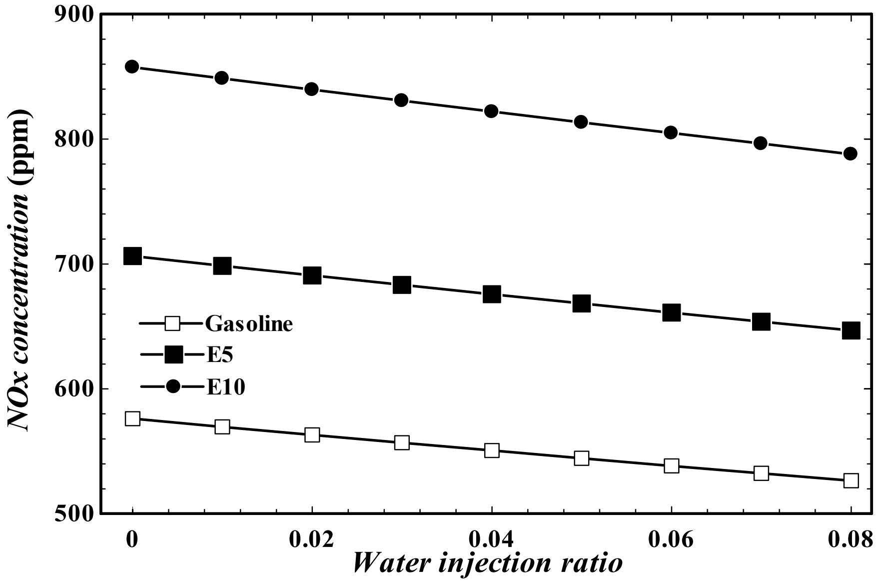
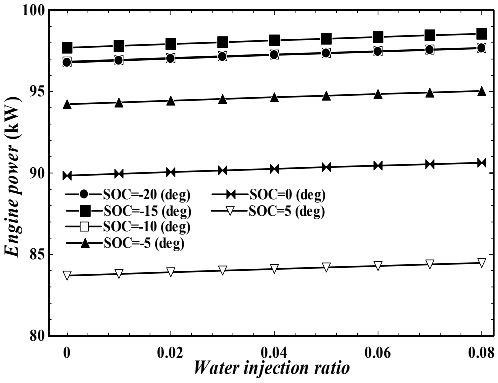


Figure 12. NOx concentration in various water injection ratios for each fuel.

3.3. The Effects of Start of Combustion Parameter

In this section, the effect of water injection with various ratios on engine main parameters such as engine power and NOx production for E5, E10 and gasoline fuel at varying ignition timing (SOC) are presented. The start of combustion parameter is one of the important parameters in engine research. It shows the crank angle position when the combustion starts. The effects of start of combustion on engine power output at various water injection ratios are shown in Figure 13a–c. As can be seen, the engine ignition timing has a significant effect on engine power for all the fuels. The variation of the engine power by SOC was much more significant compared to water injection. Therefore, with water injection up to 8%, there was no significant effect on engine power output, as compared with SOC. For instance, increasing the water injection rate by up to 8% led to an increase in power of nearly 1%, while advancing the start of combustion from 5 to −20 degrees resulted in an increase in engine power by 17%. The maximum power production occurred when SOC was equal to −15 degrees.



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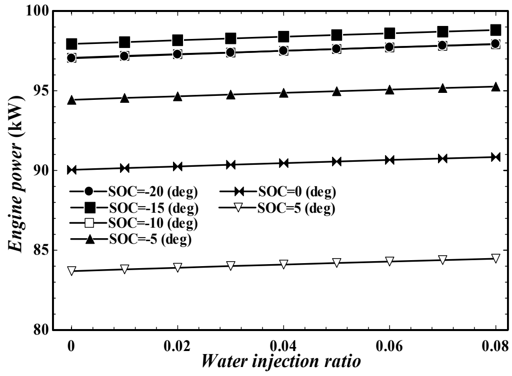

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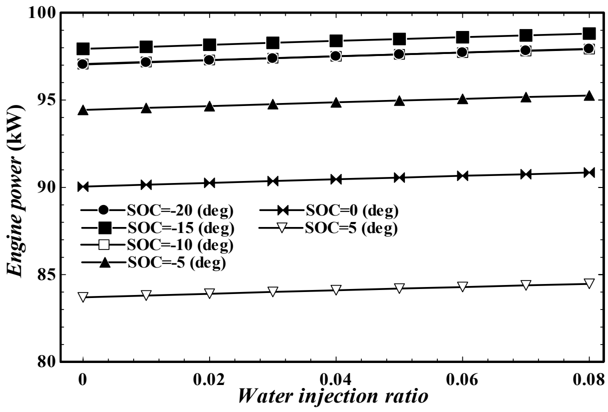
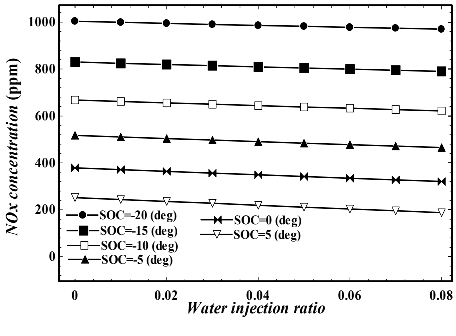


(**c**)

Figure 13. Engine power in various water injection ratios and start of combustion angles for gasoline (**a**), E5 (**b**) and E10 (**c**).

Figure 14a–c demonstrate the variation of NOx concentration in engine exhaust gas with different water injection ratios and start of combustion for gasoline, E5 and E10 fuels, respectively. Despite the positive effect of SOC on power, it resulted in a negative effect on NO_x. This is due to the extra energy released in the combustion chamber and the simultaneous increase in flame temperature by changing the SOC. As can be seen from this figure and was discussed before, the injection of water can reduce the NOx concentration. Thus, water injection and tuning SOC should be combined to achieve the best performance of the engine for power delivery as well as NOx emission.



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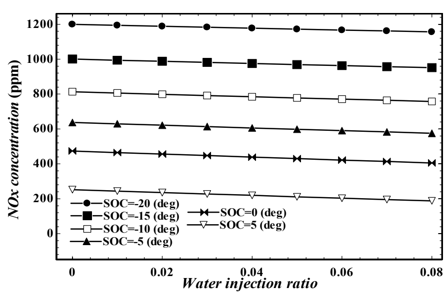
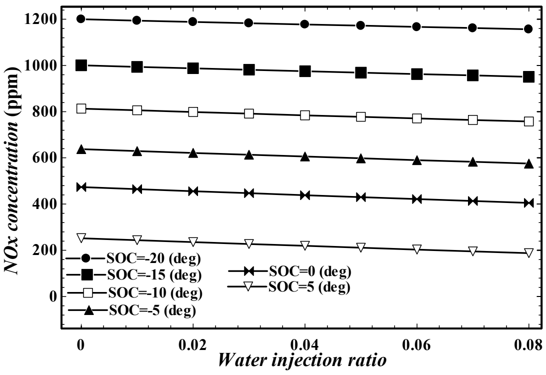


(**b**)



(**c**)

Figure 14. NOx concentration in various water injection ratios and start of combustion angles for gasoline (**a**), E5 (**b**) and E10 (**c**).

4. Conclusions

The impact of port water injection on performance and emissions in a spark ignition engine fueled by gasoline and biofuel were investigated in this paper. Experimental tests on a KIA Cerato engine were performed to obtain the main engine functional parameters at various RPMs. Then, the proposed engine was modeled numerically in AVL BOOST software, and the results of modelling were compared with the experimental test results for validation purposes. The Latin Hypercube sampling method was used to apply the design of experiment (DOE) method for defining the minimum number of design points required in the AVL software. Regression analysis was employed following the DOE method to define the equations for each engine output parameter based on the design variables. The main conclusions drawn from this paper are presented as follows:

- Increasing the ethanol fraction from E5 to E10 resulted in a small increase in the power output of the engine.
- Adding ethanol also showed a small positive impact on BMEP.
- Comparing E5 and E10 fuels with gasoline, the engine BSFC decreased by nearly 2.3% and 4.5%, respectively. So, a better fuel economy and reduction in greenhouse gas emissions are expected as a result of an increase in the ethanol content.
- With the combination of E10 fuel and 8% water injection, an increase of approximately 1.3% in engine power output can be obtained, when compared with a gasoline fueled engine.
- The break thermal efficiency was not changed significantly between E5 and E10. Compared with gasoline, the brake thermal efficiency increased by about 1% for E10. Water injection did not make any significant difference in improving the thermal efficiency for all fuels.
- By using E10 instead of E5, BSFC decreased by approximately 2% to 307 g/kWh. Comparing E5 and E10 fuels with gasoline, the engine BSFC decreased by nearly 2.3% and 4.5%, consecutively.
- Changing the fuel from gasoline to E5 and E10 resulted in an increase in peak cylinder temperature, which led to an increase in NOx formation of up to 22% and 48%, for E5 and E10, respectively.
- Water injection can reduce the NOx formation rate by up to nearly 8%.
- The engine power generation rate has changed slightly as a result of the injection of water at various ratios for different values of SOC.

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Nomenclature

DOE	Design of experiment
I	Injector
C	Cylinder
CAT	Catalyst converter
CL	Air cleaner
MP	Measure point
PL	Plenium
BSFC	Brake specific fuel consumption [g/kWh]
WIR	Water injection ratio
mf	Mass fraction
SOC	Start of combustion
BDUR	Burn duration
$\propto$	Crank shaft angle
m	Wiebe shape
a	Wiebe parameter
η	Overall efficiency
E	Engine
SB	System boundary
CO	Carbon monoxide
NOx	Nitrogen oxide
COVID-19	Corona virus
BMEP	Brake mean effective pressure [bar]
WLTP	Worldwide harmonised light vehicle test procedure
Subscripts	
eqv	Equivalent
u	Unburnt
b	Burnt
w	Wall heat loss
f	Fuel

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