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Experimental Evaluation of Regression Prediction Analysis After Testing Engine Performance Characteristics

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ABSTRACT

Using ethanol in gasoline is considered one of the most significant goals in the 2030 agenda, which has been set a 15-year plan in order to achieve it since 2015. Appropriately, this project was planned for predicting the value of the most important engine parameters such as the equivalence air-fuel ratio (φ), fuel consumption ($\dot{m}f$), and brake thermal efficiency $n_{b.\ th}$, and brake-specific fuel consumption (BSFC) by regression models. According to the protocol of this project, first, the determined percentages of ethanol were added (0, 20, 40, 60, and 80%) to gasoline at different engine speeds (850, 1000, 2000, 3000, and 4000 rpm and the New European Driving Cycle test). After testing, calculating, mathematical programming, and fitting the regression models for the two SI-engine (TU5 and EF7) with different properties of engine design,12 regression equations have been determined for each of the ‘ (positive linear model), ($\dot{m}f$) (negative linear model), $n_{b.th}$ (negative second-order polynomial model), and BSFC (positive second-order polynomial model), respectively. Clearly, these 48 regression equations with different line slopes will be able to predict the exact value of the ‘, ($\dot{m}f$), $n_{b.th}$, and BSFC for each concentration of ethanol at different engine speeds in order to help automotive industries for trend predicting them in other similar engines.

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INTRODUCTION

The fuel conversion efficiency of alcohol that has higher heat of vaporization is better in comparison to gasoline (Balabin, 2007). Temperature of air is decreased by alcohol, as it enters the engine; moreover, it increases the brake thermal efficiency as the engine power output goes up (Uslu & Celik, 2020). In the meantime, alcohol fuels vaporize more easily during compression stroke because of their high heat of vaporization. This is attributed to absorption of heat from the cylinder by the fuel during vaporization process, as well as the more easily compression of fuel mixture, resulting in an improvement in thermal efficiency of alcohol-gasoline blend in comparison to pure gasoline (Najafi et al., 2009; Chansauria & Mandloi, 2018).

However, the higher heat of vaporization of alcohol negatively effects on its efficiency, particularly on when the engine is to start in cold climates (Armas et al., 2012). An intelligent cooling system and control model for improved engine thermal management is studied by Arya K. Haghighat (Haghighat et al., 2017).A more comprehensive and detailed knowledge of how the released energy from the fuel is distributed (between brake power output, coolant energy, exhaust energy, and unmeasured heat losses) will be vital for a better understanding of the overall thermal behavior of engines, considering the fact that

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some heat losses are unavoidable (Elfasakhany, 2020; Schifter, 2011).

The efficiency and performance of fuels can be improved by fuel additives, making them really important in this industry (Zaharin et al., 2018). One of the most crucial additives to enhance fuel performance is such oxygenating as ethanol (Manzetti & Andersen, 2015). It was reported that nearly all oxygenated blends provide a better anti-knock performance in comparison to hydrocarbon fuels having the same octane range (Luo et al., 2008); this is especially true during low-speed acceleration. There are some comparisons at a fixed compression ratio between oxygenated blends with those hydrocarbon fuels that were presented by Ricardo; an improvement in efficiently of about 5% was reported in the fuels containing ethanol (Al-Baghdadi, 2008; Jeuland et al., 2004).

Effect of the used ethanol-gasoline blends is analyzed in practice, since 15 and 20% ethanol are considered in the fuel, that spark ignition performance of Ricardo and Peugeot 405 engines is studied. Noticing the results, the effects of gasoline-ethanol blends (containing 15% and 20% ethanol) on the performance of spark ignition engines was also studied (similar to Ricardo and Peugeot 504 GR engines). An average drop in power was reported for the Ricardo engines (test range of 8:1 to 10:1 compression ratio), when compared to gasoline (2.5% on blend with 15% ethanol, and 7.5% on blend with 20% ethanol). However, it is to be noted that an increase in the specific fuel consumption was observed in the ethanol-gasoline blend; the increase was about 0.5% and 4% for blends containing 15% and 20% ethanol, respectively.

In Peugeot engine, the tests demonstrated a power decrease of about 1% and 2.5% on blends with 15% and 20% ethanol, respectively. Moreover, the blend with 15% ethanol demonstrated an increase of about 5% in the specific fuel consumption, while that of blend 20% was 1% (Al-Baghdadi, 2008; Jeuland et al., 2004). Many researchers (Al-Baghdadi, 2008; Jeuland et al., 2004) studied the performance of spark ignition engines used in engine-dynamometers. The cited articles evaluated the effects of ethanol-gasoline blends on the performance, as well as controlling the ratio of fuel-air, under a variety of conditions including steady- state.

The main findings by these researchers is that the effect of blends with nearly low ethanol content (less than 20%, v/v ethanol) on the overall power and torque of the engines can be ignored. The effects of addition of ethanol alcohol to gasoline in two engines and under different engine speeds were experimentally evaluated to ascertain and measure combustion performance of the blends. Then, we measured variables in engines for drawing regression models. Finally, we evaluated the same variables in the same engines of vehicles running under the New European Driving Cycle test on a chassis dynamometer.

MATERIAL AND METHODS

Defining experimental levels

According to the protocol of this project, the determined percentages of ethanol were added (0, 20, 40, 60 and 80%) to gasoline at different engine speeds (850, 1000, 2000, 3000 and 4000 rpm along with the NEDC test). After that, some engine parameters such as the φ, $\dot{m}f$, $n_{b.th}$, and BSFC were evaluated for two four-cylinder SI-engines, which are used in all around the world (TU5 and EF7).

Engine specification and experimental facilities

In order to test and evaluate the mechanical performance of the engine, a mechanism similar to Super Flow 902 (SF-902) was used. (Hydraulically based, with a range of 0-1627 N.m and also an accuracy of 5 N.m, USA). In order to read engine sensors, the Vgate OBD scan was utilized, and a LabVIEW software interface as well (Table 1). At the first step, calibration was done on the dynamometer. Then, the control panel was connected and fitted with a PC hardware, followed by adjustment in all PCI data card, data acquisition system, engine speed (RPM), torque (N.m), pressure (N.m $^{-2}$), mechanical power (kW), and finally the temperature calibration (C). At the third state, the fuel tank similar to the model (SUM-290122) was checked and connected with a 6 kg electronic scale. (Phoenix Lexus) so that consumption rate of fuel can be monitored.

Table 1. Technical characteristics of the instrumentation used in this work.

	Trademark	Super Flow SF902
Dynamometer	Range	0-1627 Nm
	Power capacity	1119 kW
	Continuous	9000 RPM
	Intermittent	11000 RPM

In order to determine the ratio of air to fuel, a gas analyzer similar to the model (Galio Smart 2000X) has been used. These factors were measured under NEDC test conditions, φ, ṁf, nb.th, and BSFC and placed on a chassis dynamometer for static monitoring. A more comprehensive description of NEDC test can be found in this article (Ahmed et al., 2018). Finally, the data was stored in the Microsoft excel in tabular and graphical form.

Fuel and Fuel properties

Unleaded gasoline was obtained from Tehran Oil Refining Company. Moreover, ethanol (99% of purity) was used in gasoline blends. The properties of different fuel types are detailed in Table 2. In order to get five different test blends, ethanol was blended unleaded gasoline; ethanol content ranged from 0% to 80%, with 20% incremental intervals.

Procedures

Some engine parameters such as φ, $\dot{m}f$, $n_{b.th}$, and BSFC were calculated with the following equations. First, according to Eq. (1), the $\dot{m}f$(kg/h)represents fuel consumption calculated by using the Qf (cm^3) which represents volume flow of fuel, the p_b (g/cm^3) representing density of fuel blend, and t (s) that represents time required to consume 100 (cm^3) of fuel. Second, by considering Eq. (2), the p_b is calculated by using the p_i (g/cm^3) which represents density of given component in fuel blend and the V_i (vol. %) that represents volume fraction of given component in fuel blend.

$$\dot{m}f = \frac{3.6\ \mathrm{Q_f\, p_b}}{\mathrm{t}} \tag{1}$$

$$\mathrm{p_b = \sum p_i v_i} \tag{2}$$

The equivalence air-fuel ratio φ) is calculated by using Eqs. (3) and (4). Eq. (3), the ($AF\,R_{st:b}$) and ($AF\,R_{act}$) represent the stoichiometric air-fuel ratio of fuel blend and actual air-fuel ratio of fuel blend, respectively. For calculating ($AF\,R_{st:b}$) in Eq. (4), ($AF\,R_{st:i}$) and V_i are the molar stoichiometric air-fuel ratio of fuel blend and volume fraction of given component in fuel blend (vol. %).

$$\varphi = \frac{\left(AF\,R\right)_{st.b}}{\left(AF\,R\right)_{act}} \tag{3}$$

$$\left(AFR\right)_{st.b} = \sum\left(AFR\right)_{st.i} V_i \tag{4}$$

Regarding Eq. (5), the BSFC is brake specific fuel consumption (kg/(kWh)), *inf* (see Eq. (1)) and *Bp* (see Eq. (6)) are fuel consumption (kg/h) and brake power (kW), respectively. According to Eq. (6), the brake

Table 2. Specification of spark-ignition engines subjected to the testing.

Engine parameters	Unit	First engine	Second engine
Model of car	-	Dena	Peugeot 206 SD
Kind of engine	-	EF7	TU5
Fuel type	-	gasoline	gasoline
Number of cylinders	-	4	4
Number of valves / cylinder	-	4	4
Fuel system	-	MPFI	MPFI
Transmission	-	5-speed manual	5-speed manual
Date of manufacture	year	2020	2020
Registration date	year	2015	2003
Odometer reading	km	120	70
Cylinder displacement	CC	1761	1587
Bore and stroke	mm	78.6 × 85	78.5 × 82
Top speed	Km/h	189	190
Compression ratio	-	11.0 : 1	10.5 : 1
Nozzle orifice diameter	mm	0.225	0.217
Tail pressure	bar	30	31
Swirl ratio	-	2.8	2.9
Connecting rod length	mm	134.5	139.0
Maximum power	-	113 PS/6000 rpm	105 PS/5800 rpm
Maximum torque	-	153 Nm at 3300 rpm	142 Nm at 4000 rpm
Cooling system	-	WP and OP	WP and OP
Engine oil capacity	L	5.5	3.75
Cooling capacity	kW	51	42
Weight of car	kg	1285	1054
Fuel consumption (urban)	L	8.9	8.6
Fuel consumption (extra urban)	L	6.2	5.4
Fuel consumption (combined)	L	7.74	6.6
Engine cylinder head material	-	AL SI9 CU3	AL SI7 CU1
Cylinder block material	-	Gray cast iron (gjlb1)	Gray cast iron (gjl250)
Thermal coefficient (head)	k-1	24.12×10^{-6}	23.38×10^{-6}
Thermal coefficient (block)	°C	12.50×10^{-6}	12.50×10^{-6}

power is calculated by measuring the engine speed (N) and the engine torque (T).

$$BCFC = \frac{\dot{m}f}{B_P} \tag{5}$$

$$B_P = \frac{NT}{9549.29'} \tag{6}$$

Equation (7) show that the brake thermal efficiency ($n_{b.th}$) is considered as the ratio of the brake power (Eq. (6)) to the heat input for each blend (Eq. (8)). The $(LHV)_b$ is lower heating value of fuel blend (kJ/kg). The p_i, V_i and $(LHV)_i$ are density of given component in fuel blend (g/cm^3), the

Table 3. The overall design of measured experiments.

Input layer for TU5			Input layer for EF7		
Test no.	Speed (rpm)	Ethanol in gasoline (%)	Test no.	Speed (rpm)	Ethanol in gasoline (%
1	850	0	31	850	0
2	850	20	32	850	20
3	850	40	33	850	40
4	850	60	34	850	60
5	850	80	35	850	80
6	1000	0	36	1000	0
7	1000	20	37	1000	20
8	1000	40	38	1000	40
9	1000	60	39	1000	60
10	1000	80	40	1000	80
11	2000	0	41	2000	0
12	2000	20	42	2000	20
13	2000	40	43	2000	40
14	2000	60	44	2000	60
15	2000	80	45	2000	80
16	3000	0	46	3000	0
17	3000	20	47	3000	20
18	3000	40	48	3000	40
19	3000	60	49	3000	60
20	3000	80	50	3000	80
21	4000	0	51	4000	0
22	4000	20	52	4000	20
23	4000	40	53	4000	40
24	4000	60	54	4000	60
25	4000	80	55	4000	80
26	NEDC test	0	56	NEDC test	0
27	NEDC test	20	57	NEDC test	20
28	NEDC test	40	58	NEDC test	40
29	NEDC test	60	59	NEDC test	60
30	NEDC test	80	60	NEDC test	80

φ, $\dot{m}f$, $_{nb.th}$, and BSFC for TU5
$\Downarrow$
φ is shown in Fig. 2
$\dot{m}f$ is shown in Fig. 3
$_{nb.th}$ is shown in Fig. 4
BSFC is shown in Fig. 5

φ, $\dot{m}f$, $_{nb.th}$, and BSFC for EF7
$\Downarrow$
φ is shown in Fig. 2
$\dot{m}f$ is shown in Fig. 3
$_{nb.th}$ is shown in Fig. 4
BSFC is shown in Fig. 5

volume fraction of given component in fuel blend (%), and lower heating value of given component in fuel blend (kJ/kg), respectively.

$$n_{b.th} = \frac{3600\ B_p}{\dot{m}_f \cdot (LHV)_b} \tag{7}$$

$$(LHV)_b = \Sigma\left(\frac{p_{i\,v_i}}{p_b}\right)(LHV)_i \tag{8}$$

Data analysis

The interaction effects (Table 3) of engine kinds, percentage of ethanol, and engine speeds related to the regression were determined via analysis of variance in the SPSS software (version 20).In addition, the SigmaPlot software (version 12) was used for drawing the regression models. With regard to the data of φ and $\dot{m}f$, simple linear regression ($y= b_0+ b_1x$)was designed to ascertain possible significant correlations between thevariables. In addition, according to the type of some data (BSFC and $n_{b.th}$) the second-order polynomial regression ($y= b_0+ b_1x_1+ b_2 x_1^2$) was used. Maximum or minimum line slope in simple linear ($y= b_0+ b_1x$) and second-order polynomial ($y= b_0+ b_1x_1+ b_2 x_1^2$) regression models demonstrate that the value of engine characteristics increase or decrease under increase or decrease of percentages of ethane in gasoline at different engine speeds. For each and every model, the model Equation and probability values were determined, in addition to the coefficient of determination (R2). All statistical analyses were carried out at 99% level of confidence.

RESULTS AND DISCUSSION

Experimental results are carried out by laboratory set as shown in Fig. 1. Various test are performed and discussion are illustrated in next parts.

Equivalence Air-Fuel Ratio (φ)

The implications of these findings are discussed in negative linear regression relationship between increase- ing ethanol content and *φ* at all engine speeds in EF7 (Fig. 2a) and TU5 engines (Fig. 2b). We found a significant relationship between changing ethanol content and *φ* under all engine speeds conditions. For example, there is a significant relationship ($p < 0.01$) between increasing ethanol and

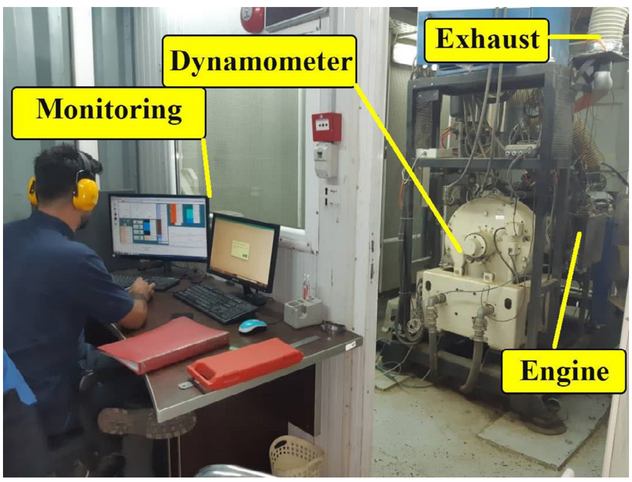
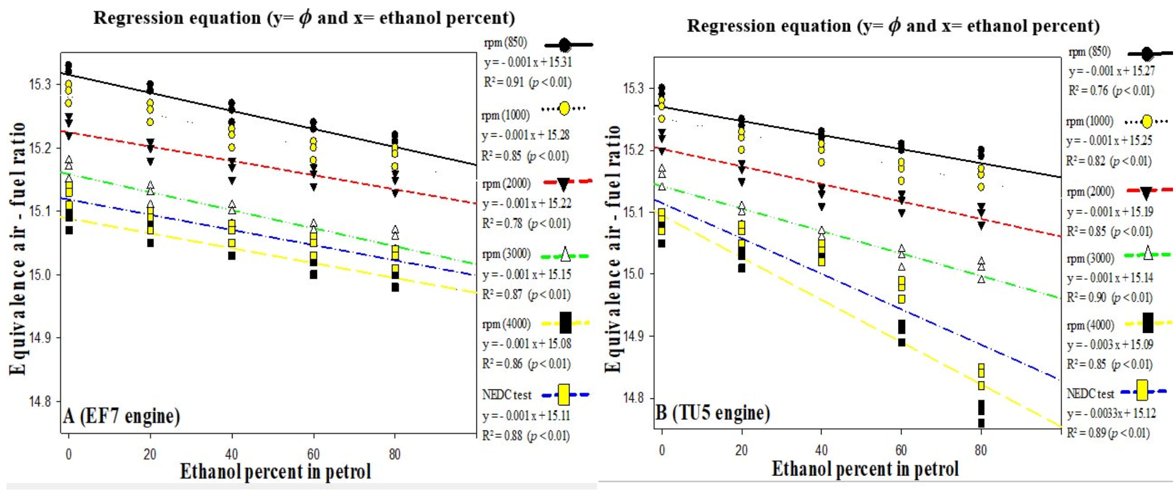


Fig. 1. Used experimental setup



(a) (b)

Fig. 2. The negative linear regression model for predicting the equivalence air-fuel ratio trait (φ) under the interaction between add ethanol (from 0 to 80%) to gasoline and engine speeds (850, 1000, 2000, 3000 and 4000 rpm along with the NEDC test) (a)-in the EF7 and (b)- in the TU5 engines.

Table 4. Analysis of variance for the interaction effects of engine kinds (EK), percentage of ethanol (EP) and engine speeds (ES) related to the regression on air-fuel ratio (φ), fuel consumption ($\dot{m}f$), brake thermal efficiency ($n_{b.th}$) and brake specific fuel consumption (BSFC).

Source of variation	d.f	Mean squares			
		$\dot{m}f$	φ	$n_{b.th}$	BSFC
EK.EP.ES	24	0.125**	0.002**	0.044*	1.031**
Standard error	138	0.001	0.001	0.001	0.001
C.V. (%)	-	12.723	29.879	14.564	29.864
** Significant at the 0.01 probability level					

decreasing φ under the highest engine speed (4000 rpm) in both engines (Table 4). In addition, R2 and line-slope are 0.85

and 15.09 for TU5 engine, respectively (Fig. 2b). Moreover, R2 and line-slope are 0.86 and 15.08 for EF7 engine, respectively (Fig. 2a). It leads to good results, even if the improvement is negligible. The results demonstrate two things. First, increasing ethanol decreases *j* in both engines. Second, in the TU5 engine *j* decreases faster than EF7 engine under the highest engine speed (4000 rpm). It is important to correctly interpret the results. It indicates that to achieve the same goal, minimum*j*, in TU5 engine [*y* (*j* for TU5engine) = -0.003 (80) + 15.09= 14.85, Fig. 2b] decreased by 0.15 compared with EF7 engine [*y* (*j* for EF7engine) = -0.001 (80) + 15.08= 15.00, Fig. 2a] under the highest engine speed (4000 rpm).

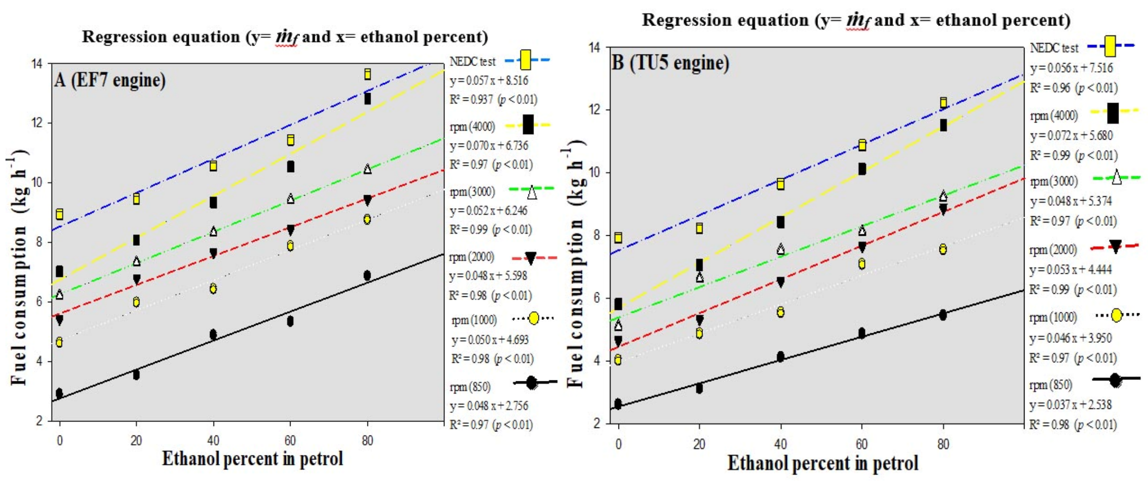
The effect of the ethanol-gasoline blends on the *j* is shown in Fig. 2a. From these results, it is clear that the *j* decreases as the percentage of ethanol in gasoline increases from 0 to 80 percent ethanol in gasoline under all engine speeds in both engines Fig. 2a. This was attributed to two main reasons. The first reason was the decrease that occurred in the in the AFRst.b (based on Equation (3) for the fuel blends), with regard to the fact that the $AFR_{st.b}$ of ethanol fuel can usually be considered lower than that of the gasoline (based on the data mentioned in Table 2). The second reason was the increase of AFRact in blends and the consequent oxygen content in ethanol (Table 2). The oxygen mass fraction ranged 0% for pure gasoline to

35% in blends with 80% of ethanol. The findings are directly in line with previous findings (Jeuland et al., 2004; Canakci et al., 2013; Yucesu & Topgul, 2006). According to equations (3) and (4), in summary, these results show that for increase engine speed from 850 to 4000 rpm, the behavior is reversed because the AFR_{act} decreases (Fig. 2). Results clearly demonstrated that φ decreased as engine speed increased from 850 to 4000 rpm (Fig. 2). This was attributed to an increase in the AFR_{act} as a result of an increase of air introduced into the cylinder, ultimately resulting in a decrease in φ. Moreover, the above mentioned increase in air into cylinders was accompanied by an increase in the pressure drop from atmospheric pressure; ultimately, further decrease in the cylinder pressure occurred. The results about ‘ are comparable to the results reported by the other researchers (Armas et al., 2012; Jeuland et al., 2004; Canakci et al., 2013). From the results, it is clear that regarding Fig. 2 on average the ‘ for all engine speeds was higher in the EF7 engine compared to the TU5 engine. The design of the TU5 engine would clarify this. The compression ratio of TU5 is quite high (10.9: 1) in comparison to EF7 engine (9.1: 1). Therefore, the early stages of combustion were dominated by the properties that are typical of high compression work (that is, charge density and in-cylinder turbulence). In addition, ratios lower than stoichiometric are considered” rich”. Ratios higher than stoichiometric are considered” lean”. The TU5 engine is designed with features to allow lean-burn.

Fuel consumption ($\dot{m}f$)

Another promising finding was that there is positive linear regression relationship between increasing ethanol content and $\dot{m}f$ at different engine speeds in EF7 (Fig. 3a) and TU5 engines (Fig. 3b). We found a significant relationship between ethanol and $\dot{m}f$ under all engine speeds conditions. For instance, there is a significant relationship ($p < 0:01$) between increasing ethanol and increasing $\dot{m}f$ under the highest engine speed (4000 rpm) in both engines. R2 and line-slope are 0.97 and 6.736 for EF7 engine, respectively (Fig. 3a).

While R2 and line-slope are 0.99 and 5.680 for TU5 engine, respectively (Fig. 3b). The results demonstrate two things. First, increasing ethanol increases $\dot{m}f$ in both engines. Second, in the EF7 engine $\dot{m}f$ increases faster than TU5 engine under the highest engine speed (4000 rpm). It is important to correctly interpret the results. This suggests that to achieve the same goal, maximum $\dot{m}f$ in EF7 engine [y ($\dot{m}f$ for EF7 engine) = 0:070(80) + 6:736 = 12:336 $\dot{m}f$ kg/h, Fig. 3a] increased by 0.89 $\dot{m}f$kg/h compared with TU5 engine [y ($\dot{m}f$ for TU5 engine) = 0:072(80) + 5:680 = 11:440 $\dot{m}f$ kg/h, Fig. 3b] under the highest engine



(a) (b)

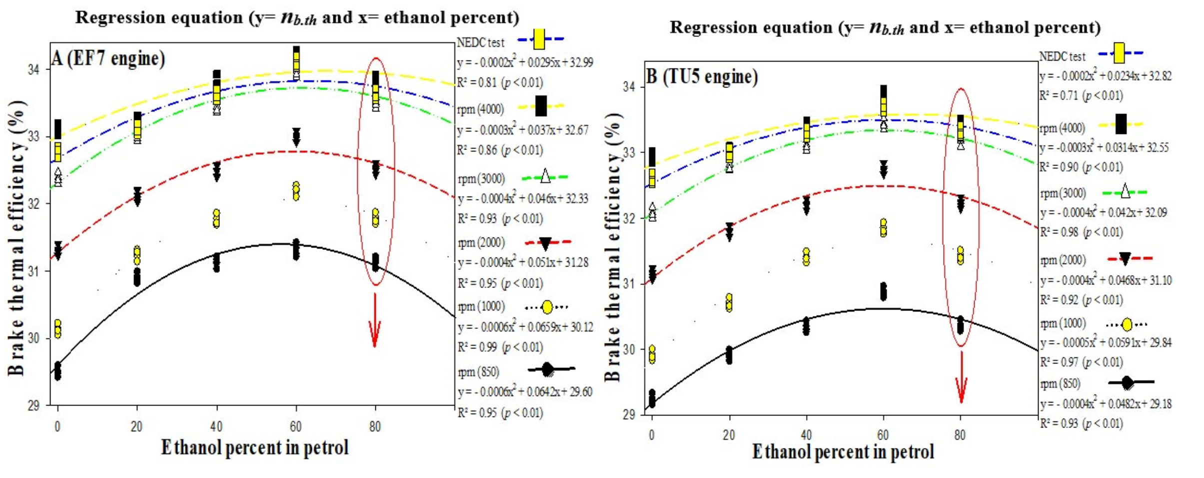
Fig. 3. The positive linear regression model for predicting the fuel consumption trait ($\dot{m}f$) under the interaction between add ethanol (from 0 to 80%) to gasoline (0, 20, 40, 60 and 80 %) and engine speeds (850, 1000, 2000, 3000 and 4000 rpm along with the NEDC test) in the (a)- EF7 and (b)- TU5 engines.

speed (4000 rpm). We describe the results of $\dot{m}f$, which show the $\dot{m}f$ increases as the percentage of ethanol in gasoline increases for all engine speeds in EF7 and TU5 engines (Fig. 3).

Equations 1 and 7 describe the $n_{b.th}$ per unit mass of the ethanol fuel. The mass was significantly lower than that of pure gasoline (Table 2), consequently increasing the amount of fuel that was introduced into cylinders for a given energy input. This is consistent with what has been found in previous researches (Al-Baghdadi, 2008; Celik, 2008; Elfasakhany, 2017). Moreover, the consequent decrease in the volumetric energy content of the blend as a result of an increase in the alcohol portion, it would be expected that the flow rate of fuel will increase with an increase in ethanol concentration. In fact, the energy content of ethanol is lower than that of gasoline, both on a mass basis and on a volume basis (Li et al 2019; Daniela, 2016; Costagliola et al., 2016). Based on the above mentioned results, and with the pre-assumption that the efficiencies of the fuels were compatible, it was concluded that higher ethanol flow-rate was needed to produce the same power output similar to that of pure gasoline.

These findings are also effective when considering lower heating value per unit mass of the evaluated fuels. The main reason can be found in the fact that ethanol fuel produces lower heating value per unit mass, which is considerably lower than that of unleaded gasoline (Table 2). Based on the above mentioned discussion and facts, it was concluded that higher volume of ethanol fuel is needed for a given energy input, as compared to pure gasoline. The above mentioned results clarified that as the engine speed goes up (from about 850 rpm to near 4000 rpm), $\dot{m}f$ increases by four times in both engine types (Fig. 3). The increase in $\dot{m}f$ is attributed to an increase in air velocity and a decrease in the pressure at the injector venture, as the engine speed goes up. The outcome of the above mentioned fluctuations are a drop in pressure between the injector venture, as well as an increase in atmospheric pressure inside the float chamber, resulting in more $\dot{m}f$. Overall, other results were broadly in line with results of this study (Al-Baghdadi, 2008; Daniela, 2016; Koc et al., 2009).

They have demonstrated that when 40 percent of ethanol was used instead of gasoline. The $\dot{m}f$ for ethanol operation was increased by about 90% in the entire engine speeds range (from 1000 to 4000 rpm) investigated. The amount of fuel consumed depends on the engine, the type of fuel used, and the efficiency with which the output of the engine is transmitted to the wheels. Our results demonstrated that regarding Fig. 3 on average the $\dot{m}f$ for all engine speeds conditions were higher in the EF7 engine compared to the TU5 engine. First, this behavior is attributed to the $n_{b.th}$ that in EF7 engine (Fig. 4a) was higher than TU5 engine (Fig. 4b). Second, this behavior is attributed to



(a) (b)

Fig. 4. The negative second-order polynomial regression model for predicting the brake thermal efficiency character ($n_{b.th}$) under the interaction between add ethanol (from 0 to 80%) to gasoline and engine speeds (850, 1000, 2000, 3000 and 4000 rpm along with the NEDC test) (a)- in the EF7 and (b) in the TU5 engines.

the cylinder displacement that in EF7 engine (1761 CC) was more than TU5 engine (1587 CC). Third, this behavior is attributed to the nozzle orifice diameter that in EF7 engine (0.225 mm) was more than TU5 engine (0.217 mm. Fourth, this behavior is attributed to the compression ratio that in EF7 engine (9.1: 1) was lower than TU5 engine (10.9: 1).

A decrease in $\dot{m}f$ usually happens due to the fact that internal combustion engines are considered as heat engines; therefore, an increase in compression ratios will provide similar combustion temperature with lower fuel consumption.It would be desirable if the compression ratio is high, as this provides an engine with a higher mechanical energy to extract from the same mass of air-fuel mixture, resulting in the efficiency to increase significantly.

Compression ratios of ethanol are reported to be significantly higher than that of gasoline. In this regard, the compression ratio of racing engines with ethanol fuel is frequently measured by different researchers to be 14:1 to 16:1 (Costa & Sodre, 2011).

Brake Thermal Efficiency ($n_{b.th}$)

The $n_{b.th}$ results illustrate that there is negative second-order polynomial regression relation between in- creasing ethanol ratio and $n_{b.th}$ at different engine speeds (Fig. 4). Regression analysis was performed on$n_{b.th}$ as variable and ethanol-gasoline blends (0, 20, 40 and 80 percent of ethanol in gasoline) as treatmentat each level of engine speeds (850 rpm, 1000 rpm, 2000 rpm, 3000 rpm, 4000 rpm and NEDC) in EF7(Fig. 4a) and TU5 engines (Fig. 4b). We describe the results of negative second-order polynomial regression, which show that the optimal percentage of ethanol in gasoline for all engines speeds are estimated to be 60% ethanol in gasoline to achieve maximum $n_{b.th}$ in both engines (Fig. 4).

As seen in the figure,$n_{b.th}$ at each level of engine speeds was negatively related to the ethanol-gasoline blends in EF7 engine (850 rpm: R2= 0.94, 1000 rpm: R2= 0.98, 2000 rpm: R2= 0.93, 3000 rpm: R2= 0.92, 4000 rpm: R2= 0.78 and NEDC: R2= 0.84) (Fig. 4a) and TU5 engine (850 rpm: R2= 0.92, 1000 rpm: R2= 0.96, 2000 rpm: R2= 0.91, 3000 rpm: R2= 0.96, 4000 rpm: R2= 0.69 and NEDC: R2= 0.87)(Fig. 4a).

These results go beyond previous reports, showing that we can estimate the amount of $n_{b.th}$ by placing the percentage of ethanol in the formula for all engine speeds. This yields increasingly good results on data. For example, it indicates that to achieve the same goal, maximum $n_{b.th}$ in the EF7 engine [y ($n_{b.th}$ for EF7 engine) = -0.0003 (602) + ((0.037(60)) + 32.67= 33.81%, Fig. 4a] increased by 0.41% compared with TU5 engine [y ($n_{b.th}$ for TU5 engine) = -0.0003 (602) + ((0.0314(60)) + 32.55= 33.35%, Fig. 4b] under the highest engine speed (4000 rpm). The $n_{b.th}$ is normally reported as the ratio between the output power and the fuel energy content. This, in turn, can be calculated by two main factors, first the fuel mass flow rate, and second, the low heating value. It is to be noted that the $n_{b.th}$ can be improved by an increase in the proportion of energy which is ultimately transferred into useful brake power (Eq. 7). The $n_{b.th}$ of the engine can be improved to a great extent by increasing ethanol content, in comparison to pure gasoline. An increase in the combustion efficiency can be mentioned as the main reason for this phenomenon. In connection to this issue, Fig. 4 presents the effect of using ethanol-gasoline blends on $n_{b.th}$. As shown in the figure, the $n_{b.th}$ increases as the percentage of ethanol in gasoline increases.

The maximum $n_{b.th}$ recorded with 60 percent of ethanol to gasoline for all engine speeds in EF7 and TU5 engines (Fig. 4). As seen in Fig. 4, the higher $n_{b.th}$ for adding 60 percent of ethanol to gasoline in EF7 engine that found in the present study attributed to charge cooling in the intake system and the combustion characteristics of ethanol. Addition of 60% ethanol to a gasoline-ethanol blend demonstrated a higher cooling effect in the compression stroke, when compared with the other fuel blends studied here. This was attributed to two main factors, including the higher enthalpy of vaporization in ethanol, and the increase in the amount of fuel that was injected into cylinders. This suggests that first, ethanol oxygen availability will improve combustion and reduce its burning duration (Schifter et al., 2011; Manzetti & Andersen, 2015; Ahmed et al., 2018).

Higher oxygen flow caused a better combustion of fuel. This, in turn, results in an increase in the $n_{b.th}$

(Najafi et al., 2009; Al-Baghdadi, 2008; Jeuland et al., 2004; Costa & Sodre, 2011). Second, however, ethanol higher enthalpy of vaporization (Table 2) and lower fuel lower heating value (Table 2) will decrease gas temperature during compression resulting in slower combustion duration (Najafi et al., 2009; Al-Baghdadi, 2008; Balki et al., 2014). The combined influence of the two factors will affect burning duration inside the cylinder and increases the $n_{b.th}$ (Najafi et al., 2009; Costa & Sodre, 2011). It was demonstrated that during the compression stroke, the vaporization of fuel still went on (Zaharin et al., 2018; Costa & Sodre, 2011). Consequently, the temperature of the working charge tended to decrease which in turn, it reduced the compression work and increased the quantity of vapor for each single working charge (that is, the compression work was increased (Uslu & Celik, 2020; Chansauria & Mandloi, 2018; Al-Baghdadi, 2008; Kareddula & Puli, 2018). Similar to cases when gasoline is used, the effect of cooling will not be sufficient to put the effect of additional vapor in perspective when the latent heat of fuels is not high (Uslu & Celik, 2020; Chansauria & Mandloi, 2018; Al-Baghdadi, 2008; Kareddula & Puli, 2018). The cooling will be more effective if the latent heat of fuel blends increases by using higher percentages of ethanol (i.e., reduces the compression work) (Jeuland et al., 2004; Celik, 2008; Li et al., 2019; Balki et al., 2014).

On the other hand, higher percentages of ethanol in the fuel blend will cause the pressure and temperature to decrease at the initial stage of combustion (that is, the delay period prolongs along with an increase in the crank angle at which the maximum pressure happens) (Jeuland et al., 2004; Celik, 2008; Costa & Sodre, 2011). The combined influence of above factors will affect the burning duration inside the cylinder and increases the $n_{b.th}$ and brake power. In line with previous studies, researchers during their investigation using 20-40% ethanol in gasoline under partial load found the brake thermal efficiency to improve by 4-12% (Uslu & Celik, 2020; Chansauria & R. Mandloi, 2018; Kareddula & Puli, 2018).

From these results, it is clear that a further increase in the percentage of ethanol in gasoline beyond 60 percent of ethanol in gasoline result in decreasing $n_{b.th}$ for all engine speeds in both engines (Fig. 4). Because increasing percentage of ethane in gasoline decreases the ‘ (that is, the heat transfer to the cylinder wall decreases as a result of incomplete combustion) (Fig. 2), consequently, $n_{b.th}$ is decreased (Al-Baghdadi, 2008; Celik, 2008; Costa & Sodre, 2011). Researchers' results show that energy balance an improvement in $n_{b.th}$ proportional to the increase in ethanol ratio (Celik, 2008; Balki et al., 2014). Previous studies have also attributed this to an improvement in combustion efficiency and reduction in coolant and exhaust losses (Chansauria & Mandloi, 2018; Celik, 2008; Costa & Sodre, 2011; Hasan et al., 2018).

Another promising finding was that for all engine speeds conditions, an increase can be achieved in the ratio of brake output to the total heat that was released by burning of fuel as ethanol content is increased in the blend (the $n_{b.th}$ improves) (Fig. 4). The ‘ can clearly explain the effects of engine speed on $n_{b.th}$. As the engine speed increases from 850 to 4000 rpm, $n_{b.th}$ increases (Fig. 4), whereas ‘ decreases (Fig. 2). This behavior validates the fact that at points where ‘ is minimum (i.e., leaner mixture), the $n_{b.th}$ is maximum (Figs. 2 and 4). Besides that, $n_{b.th}$ increases with the increase in vehicle speeds because of the decreases in $\dot{m}f$ (Fig. 2). The higher the oxygen rate, the better of the combustion and thus increases the $n_{b.th}$. Overall, these findings are in accordance with findings reported by other researchers (Jeuland et al.,2004).

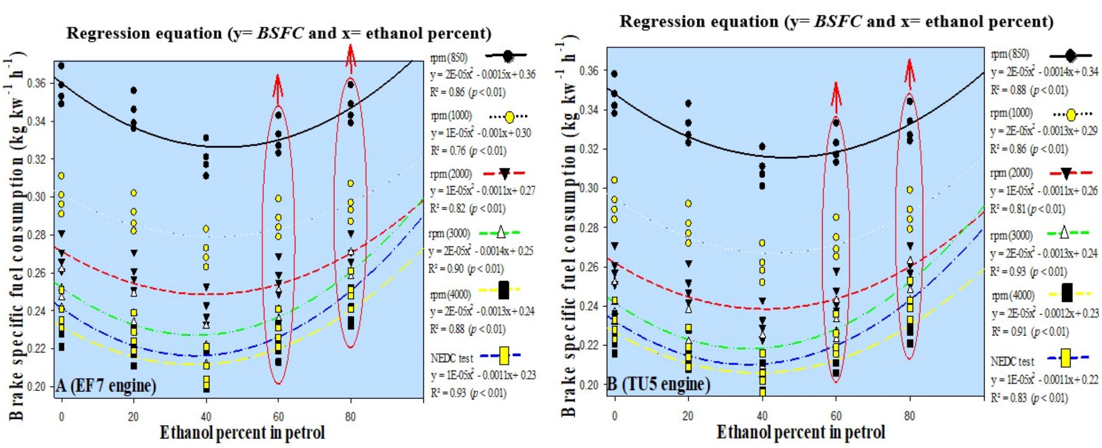
The present findings confirm that regarding Fig. 4 on average the $n_{b.th}$ for all engine speeds were higher in EF7 engine compared to TU5 engine. The higher $n_{b.th}$ for adding all the percentages of ethanol to gasoline in the EF7 engine compared to the TU5 engine that was found in the present study is attributed to charge cooling in the intake system in EF7 engine are compared to the TU5 engine (thermal coefficient for head cylinder and cooling capacity are 23:38 × 10 - 6k^{-1} and 42 kW, respectively).

Brake Specific Fuel Consumption (BSFC)

A further novel finding is that the BSFC illustrates that there is positive second-order polynomial regression relation between increasing ethanol ratio and BSFC at different engine speeds (Fig. 5). Regression analysis was performed on BSFC as variable and ethanol-gasoline blends (0, 20, 40 and 80 percent of ethanol in gasoline) as treatment at each level of engine speeds (850 rpm, 1000 rpm, 2000 rpm, 3000 rpm, 4000 rpm and NEDC) in EF7 (Fig. 5a) and TU5 engines (Fig. 5b). This suggests that the optimal percentage of ethanol in gasoline for all engine speeds are estimated to be 40% ethanol in gasoline to achieve minimum BSFC in both engines (EF7 and TU5) (Fig. 5). Also, it showed that BSFC at each level of engine speeds was positively related to the ethanol-gasoline blends in EF7 engine (850 rpm: R2= 0.65, 1000 rpm: R2= 0.48, 2000 rpm: R2= 0.53, 3000 rpm: R2= 0.69, 4000 rpm: R2= 0.63 and NEDC: R2= 0.68) (Fig. 5a) and TU5 engine (850 rpm: R2= 0.64, 1000 rpm: R2= 0.60, 2000 rpm: R2= 0.55, 3000 rpm: R2= 0.70, 4000 rpm: R2= 0.55 and NEDC: R2= 0.68) (Fig. 5b). These results go beyond previous reports, showing that we can estimate the amount of BSFC by placing the percentage of ethanol in the formula for all engine speeds.

These result indicate that to achieve the same goal, maximum BSFC, in EF7 engine [y (BSFC for EF7 engine) = 0:00002(402)((0:0013(40)) + 0:24 = 0:223 kg=(kwh), Fig. 5a] increased by 0.005 kg=(kwh) compared with TU5 engine [y (BSFC for TU5 engine) = 0:00002(402)((0:0012(40)) + 0:23 = 0:214 kg/(kwh), Fig. 5b] under the highest engine speed (4000 rpm). As far as we know, no previous research has investigated regression analysis for BSFC in the world. The BSFC is basically defined in terms of the mass of fuel that is required in each and every hour to produce 1 kW of B_p. In this definition, the value of BSFC can be determined by the amount of B_p that is produced, and mf rate as well. As Eq. 6 reveals, the BSFC defines the amount of fuel flow rate that is required to produce the engine per unit power. As the low heating value of ethanol (26.95 MJ/kg) is lower than that of gasoline (44.52 MJ/kg) (Table 2), it can be deduced that the amount of ethanol mass should be higher so that the same per unit power is produced, when compared to that of gasoline. Nevertheless, that explains why the BSFC of pure gasoline is higher than that of gasoline-ethanol blends in both engines, as shown by Fig. 5. Adding the 20 and 40 percent of ethanol to gasoline decreased the BSFC for all engine speeds in both engines (Fig. 5). In fact, the BSFC decreases as the percentage of ethanol increases up to 40 percent in both engines (Fig. 4). Research by Eyidogan et al. (2010) shows that ethanol contains a higher oxygen rate than ethanol (Table2); it is to be noted that better combustion efficiency can be achieved by higher oxygen content, resulting in a reduction in the BSFC. Others have shown similar results (Uslu & Celik, 2020; Balki et al., 2014; Eyidogan et al., 2010; Prasad et al., 2020).

Moreover, mf has an increasing effect on both $n_{b.th}$ and vehicle speed. The higher the oxygen rate, the better combustion and thus the $n_{b.th}$ increases. However, adding too much ethanol to gasoline (60% and 80%) increased BSFC for all engine speeds in both engines (Fig. 5). This can be



(a) (b)

Fig. 5. Positive second-order polynomial regression model for predicting the brake specific fuel consumption trait (BSFC) under the interaction between add ethanol (from 0 to 80%) to gasoline and engine speeds (850, 1000, 2000, 3000 and 4000 rpm along with the NEDC test) (a)-in the EF7 and (b)- in the TU5 engines.

considered as a normal outcome of the engine $n_{b.th}$ (Fig. 5). Because the $n_{b.th}$ is reduced by adding too much ethanol. It is a general knowledge that BSFC can be affected by the heating value of fuel. Addition of too much ethanol content to the gasoline-ethanol blend causes BSFC to increase when no modification is done. The amount of increase (increment) will be dependent on ethanol content in the blend. Ethanol has a heating value of about 35% which is considerably lower than that of gasoline. Therefore, higher volume of blends is needed so that the same power is produced, considering that the opening conditions remains the same. This is due to the fact that the heating value of the blend would be lower than that of pure gasoline. Consequently, BSFC increases as a result of the addition of 60% and 80% of ethanol to gasoline. In addition, this analysis found evidence for, as the engine speed increases from 850 to 4000 rpm, the BSFC decreases (Fig. 5). This is due to the increase in $n_{b.th}$ (Fig. 5) and decreases in ‘ (Fig. 2) under increasing engine speed in both engines. In this connection, an increase in throttle opening causes higher volume of fuel to be burnt; this means more energy input for the engine.

This leads to an increased torque output along with the increase in the throttle opening. Therefore, the BSFC should be decreased with the increase of engine speed. From the results, it is clear that regarding Fig. 5 on average the BSFC for all engine speeds were higher in EF7 engine compared to TU5 engine. This is due to the $n_{b.th}$ (Fig. 4) for EF7 engine was more than TU5 engine, while the average of ‘ (Fig. 2) for TU5 engine was less than EF7 engine. The higher the oxygen rate (from ethanol and increases the j), the better combustion and thus the $n_{b.th}$ increases in EF7 engine compared to the TU5 engine. In addition, it is by now generally accepted that the reduction in BSFC values at a higher compression ratio is lower than those of the lower compression ratio (Chansauria & Mandloi, 2018; Costa & Sodre, 2011). It is important to highlight the fact that a high compression ratio allows for improved fuel conversion efficiency, as the engine $n_{b.th}$ is increased (Fig. 4) and therefore, BSFC is reduced (Fig. 5). It is interesting to note that the compression ratio for EF7 engine is 9.1 and it produced the highest BSFC for all engine speeds range (Fig. 5a), while the compression ratio for TU5 engine is 10.9 and it produced the lowest BSFC for all engine speeds range (Fig. 5b).

CONCLUSION

Clearly, these 48 regression equations with different line slopes will be able to predict the exact value of the j, $\dot{m}f$, $n_{b.th}$, and BSFC for each concentration of ethanol at different engine speeds in order to help automotive industries for trend predicting them in other similar engines. In addition, the j decreases as the percentage of ethanol in gasoline increases from 0 to 80 percent ethanol in gasoline under all engine speeds.

Alteration in the oxygen content of the final fuel blend was highly influential in the above mentioned outcome. Oxygen mass fraction in the fuel increases from approximately 0% for gasoline to 35% for adding 80 percent of ethanol to gasoline. Moreover, as the engine speed increases from 850 to 4000 rpm, ‘ decreases, because the AFR_{act} decreases.

Average j for all engine speeds was higher in the EF7 engine compared to the TU5. The design of the TU5 engine would clarify this. The $\dot{m}f$ increases as the percentage of ethanol in gasoline increases for all engine speeds. The $n_{b.th}$ per unit mass of ethanol was significantly lower than that of pure gasoline (Table 2), consequently increasing the amount of fuel that was introduced into cylinders for a given energy input.

The $\dot{m}f$ increases about 4 times as the engine speed increases from 850 to 4000 rpm in both engines. The increase in $\dot{m}f$ is attributed to an increase in air velocity and a decrease in the pressure at the injector venture, as the engine speed goes up. The outcome of the above mentioned fluctuations are a drop in pressure between the injector venture, as well as an increase in atmospheric pressure inside the float chamber, resulting in more $\dot{m}f$.

The $\dot{m}f$ for all engine speeds conditions were higher in theEF7 engine compared to the TU5 engine. This behavior is attributed to the $n_{b.th}$ that in EF7 engine was higher than TU5 engine. On the other hand, this behavior is attributed to the cylinder displacement that in EF7 engine (1761CC) was more than TU5 engine (1587 CC). Another reason is the nozzle orifice diameter that in EF7 engine (0.225 mm) was more than TU5 engine (0.217 mm). Another reason is the compression ratio that in EF7 engine (9.1: 1) was lower than TU5 engine (10.9: 1).

The $n_{b,th}$ increases as the percent of ethanol in gasoline increases. The higher $n_{b.th}$ for adding 60 percent of ethanol to gasoline in EF7 engine found in the present study attributed to charge cooling in the intake system and the combustion characteristics of ethanol. Due to ethanol's higher enthalpy of vaporization and the increase for fuel injected, adding 60 percent of ethanol to gasoline show a higher cooling effect in the compression stroke than the other fuel blends. a further increase in the percentage of ethanol in gasoline beyond 60 percent of ethanol in gasoline results in decreasing $n_{b.th}$ for all engine speeds in both engines. The ‘ can be decreased by an increase in the percentage of ethanol that is blended with gasoline; this consequently results in decreasing of heat transfer rate to cylinder walls because of incomplete combustion.

The end result would be a decrease in $n_{b.th}$. In addition to this, along with an increase in the ethanol content, the ratio of brake work output to total heat that is released as a result of the combustion of fuel goes up; this means that the $n_{b.th}$ improves. The *j* can also explain the effect of engine speed on nb.th. Increasing the engine speed from 850 to about 4000 rpm results in an increase in nb.th though it causes *j* to decrease. This is considered a corroborating evidence of the fact that when ‘ is in its minimum level (that is, leaner mixture), the $n_{b.th}$ will be at its maximum amount.

Also, the higher $n_{b.th}$ for adding all the percentages of ethanol to gasoline in the EF7 engine compared to the TU5 that was found in the present study is attributed to charge cooling in the intake system in EF7 engine (thermal coefficient for head cylinder and cooling capacity are 24.12×10^{-6} k^{-1} and 51 kW, respectively) compared to the TU5 engine (thermal coefficient for head cylinder and cooling capacity are 23.38×10^{-6} k^{-1} and 42 kW, respectively).

As a result, BSFC increases when adding 60% and 80% ethanol to gasoline. In addition, as the engine speed increases from 850 to 4000 rpm, the BSFC decreases. This is due to the increase in $n_{b.th}$ and decreases in φ under increasing engine speed in both engines. Basically, more fuel is normally provided by a higher throttle opening; consequently, more energy input is provided. Ultimately, the increased opening in the throttle valve results in an increased torque output. Therefore, the BSFC should be decreased with the increase of engine speed. In addition, the BSFC for all engine speeds were higher in EF7 engine compared to TU5 engine. This is due to the nb.th for EF7 engine was more than TU5 engine while the average *j* for TU5 engine was less than EF7 engine.

The higher the oxygen rate (from ethanol and increase the *j*), the better of the combustion and thus increases the $n_{b.th}$ in EF7 engine compared to the TU5 engine.

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CONFLICT OF INTEREST

The authors declare that there is not any conflict of interests regarding the publication of this manuscript. In addition, the ethical issues, including plagiarism, informed consent, misconduct, data fabrication and/ or falsification, double publication and/or submission, and redundancy has been completely observed by the authors.

LIFE SCIENCE REPORTING

No life science threat was practiced in this research.

NOMENCLATURE

AFR_{act}	Ratio of actual air-fuel in fuel blend
$AFR_{st:b}$	Ratio of stoichiometric air-fuel in fuel blend
$AFR_{st:i}$	ratio of molar stoichiometric air-fuel in fuel blend
Bp	Power of Brake
BSFC	Brake specific fuel consumption (kg/kWh)
MPFI	Multi point fuel injection
N	Engine speed (rpm)
NEDC	New European driving cycle
OP	Oil pump
p_i	Given component density in fuel blend (g/cm^3)
p_b	Fuel blend density (g/cm^3)
t	Required time for 100 cm^3 of fuel consume (s)
T	Engine torque (N.m)
Vi	Fraction volume of given component in the fuel blend (vol.%)
WP	Water pump
$n_{b.th}$	Brake thermal efficiency (%)
$\dot{m}f$	Fuel consumption (kg/h)
φ	Equivalence air-fuel ratio
LHV_b	Lower heating value of fuel blend (kJ/kg)
LHV_i	Lower heating value of given component in fuel blend (kJ/kg)
Qf	Volume flow of fuel
RPM	Revolutions per minute

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