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Effects of Operating Conditions on Performance of a Spark Ignition Engine Fueled with Ethanol-Gasoline Blend

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A B S T R A C T

Nowadays, two main deals of researchers in different fields of industries are emissions and fuel consumption. The political turmoil of crude oil besides stricter environmental laws in the world tends researchers to find novel ways for fuel consumption and emissions reduction. Using Ethanol-Gasoline blend as fuel in spark ignition engines is considered as a promising idea to achieve this goal for internal combustion engines industries. Providing a model to investigate the performance of Ethanol-Gasoline fueled engine in different operating conditionsis still needed to reduce experimental test costs. In this study, a thermodynamic model of ethanol-gasoline fueled spark ignition engine is provided and the effects of operating conditions on engine performance are investigated in detail after validating simulation results via experimental data. Results show the provided model generates reliable data of engine performance in the full range of fuel composition, from pure ethanol to pure gasoline. In addition, studied engine produces maximum power besides best fuel consumption when it is run at 3000 rates per minute. Also, the best performance is achieved with E-45 composition while NOx emission raise 60 percent in comparison to pure gasoline. So, it can be introduced as design point for studied engine.

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INTRODUCTION

Ethanol has been introduced and met an enormous production in recent decades to decrease fossil fuel usage dependence and likewise extension in use of renewable energy. The policies performed by governments are one of the most important causes of this orientation [1]. Ethanol is used in wide range of applications such as a fuel for direct electricity production [2], fuel cells [3] and also working fluid of refrigeration systems [4]. In addition, ethanol is employed in wide range as fuel additive with gasoline in spark ignition (SI) engines in transportation usage. Although the proportion of ethanol from total used fuel in overall is considered fixed, it may not be suitable strategy to achieve optimum efficiency in this approach. Ethanol has higher octane number in compare with gasoline and offers better antiknock features. So, it is possible to increase compression ratio and engine efficiency when it used as fuel [5, 6]. Ethanol has higher evaporation rate than gasoline; thus lower charge temperature, higher mixture density and higher volumetric efficiency are achievable by injecting it in inlet manifold. In contrast, its heating value is less than gasoline, so ethanol fueled engine has more fuel consumption than gasoline fueled ones [7].

There are extensive researchers experimentally investigating the effect of ethanol-gasoline mixture as fuel in engines. Generally, these researches can be categorized in two groups; investigating engine performance and emissions. Li et al. [8] have experimentally investigated the combination of alcoholic fuels; Isopropanol– Butanol – Ethanol (IBE) and gasoline; they stated that in comparison with pure gasoline, the blend of 30 percent IBE would decrease CO, UHC and NOx by 4, 20.3 and 18.6 percent, respectively. Phuangwongtrakul et al. [9] have experimentally studied the performance of a SI engine to find optimal ethanol-gasoline blend. They have noted that while E-40 and E-50 provide maximum thermal efficiency, E-20 to E-40 bring maximum produced torque. Besides experimental studies, numerical simulation methods are well known to engine improvement in different approaches [10-14]. Many researchers try to simulate ethanol-gasoline fueled SI engines performance due to high cost of experimental tests and impossibility of doing tests in some cases. These studies are mainly categorized in three groups; engine performance [15, 16], provided fuel combustive characteristics [17, 18] and engine downsizing [19]. But extending previous models to achieve reliable fast

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response model which is able to consider ethanol both; as fuel and additive is still needed. Najafi et al. [20] have proposed a model using Support Vector Machine (SVM) and Adaptive Neuro-Fuzzy Inference System (ANFIS) due to their experimental data. They asserted that provided model is able to predict ethanol-gasoline fueled SI engine performance and emissions. Lodice et al. [21] have also reported that HC and CO is decreased noticeably using E-20 in cold start. Thangavel et al. [22] have also expressed that E-30 and E-50 are suitable blends to achieve optimal efficiency and no-knock operation, respectively. In addition, produced torque increase with low emissions is achievable by adding Butanol to the charge. Akanso et al. [23] have also represented that addition of hydrogen to Ethanol and gasoline brings both; enginethermal efficiency increase and emissionreduction.

In this paper, a thermodynamic model of ethanol-gasoline fueled SI engine is provided to predict engine performance and emissions. The model is sufficient for all range of fuel composition from pure gasoline to pure ethanol. Provided model is calibrated and its results is validated via experimental data, then the model is used to investigate the effects of engine inlet parameters such as engine speed, equivalence ratio and the proportion of ethanol in fuel, on performance and emissions and the design point of each study is introduced due to engine best performance.

Model description

Engine performance can be evaluated by separately investigating mass and energy flows of each component [24]. In this section the most important correlations used in simulator model is divided into two subsections namely, component and combustion simulations.

Component simulation

One dimension as simulation needs to solve mass conservation, energy and Navier–Stokes equations, simultaneously. Existent mass of each component can be calculated as follows [25]:

$$\dot{m}_{sub} = \sum \dot{m}_e - \sum \dot{m}_i \quad (1)$$

Here, $\dot{m}_i$ and$\dot{m}_e$are inlet and outlet mass flow rates, respectively; which are characterized as follows:

$$\dot{m} = \rho U A \quad (2)$$

Where, ρ indicates density, A represents cross sectional area perpendicular to flow and U indicates the flow velocity. Energy equation is defined as follows [26]:

$$\frac{dE}{dt} = \frac{dW}{dt} + \frac{dQ}{dt} \quad (3)$$

Here, E is energy, W is work and Q is heat transfer that it will be expanded as,

$$\frac{d(me)}{dt} = P\frac{dV}{dt} + \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e - h_g A(T_{gas} - T_{wall}) \quad (4)$$

Where, $e$and$h$indicate specific internal energy and enthalpy, $h_g$is convective heat transfer coefficient and $T_{gas}$and T_{wall} indicate in-cylinder and cylinder wall temperature. Convection heat transfer coefficient is described as follows:

$$h_g = \frac{1}{2}\,\rho\, C_f U_{eff} C_p\, P\, r^{-\frac{2}{3}} \quad (5)$$

C_f, U_{eff}, C_p and P_r define friction coefficient, effective velocity out of boundary layer, specific heat coefficient and Prandtl number respectively. Friction coefficient depends to Reynolds number which is described as follows:

$$Re = \frac{\rho\, U_c L_c}{\eta} \quad (6)$$

Here, $L_c$andU_c are length and speed of flow andη is dynamic viscosity. Considering pipes roughness, friction coefficient is calculated by Nikuradse equation [27]:

$$C_{f\,(rough)} = \frac{0.25}{\left(2\log_{10}\left(\frac{1D}{2h}\right) + 1.74\right)^{0.25}} \quad (7)$$

Where, D is diameter of pipe and h is height of roughness. Momentum conservation equation in 1D is described as follows:

$$\frac{\dot{m}}{dt} = \frac{dpA + \sum_i \dot{m}_i u + \sum_e \dot{m}_e u}{dx} - \frac{4\, C_f \frac{\rho u^2}{2}\frac{dxA}{D} - C_p\,(\frac{1}{2}\rho u^2)A}{dx} \quad (8)$$

Here, C_p is pressure loss coefficient which is defined as follows:

$$C_p = \frac{P_1 - P_2}{\frac{1}{2}\,\rho\, {V_1}^2} \quad (9)$$

Subscripts 1 and 2 show inlet and outlet conditions.

Combustion simulation

Combustion process can be simulated considering first law of thermodynamics, ideal gas equation of state and engine geometrical correlations. Assuming charge and combustion products as ideal gas, energy term of correlation (3) can be rewritten as follows:

$$\frac{dE}{dt} = m\, C_v \frac{dT}{dt} \quad (10)$$

Here, C_v is volumetric special heat capacity of the fluid. Work is defined by follows:

$$W = \int P\, dV \quad (11)$$

Where, V is the volume of combustion chamber defined by engine geometrical correlation [28].

$$V = V_c + \frac{\pi B^2}{4}\left(l - a - a\cos(\theta) - \sqrt{l^2 - a^2 \sin^2(\theta)}\right) \quad (12)$$

Here, V_c, B, l and a are clearance volume, bore, connection rod length and crank radio respectively. Also, θ refers to the crank angle. The heat rate is defined as sum of convection heat transfer to cylinder wall and heat release due to combustion. Convection heat transfer coefficient is described by Woschni correlation [29]

$$\frac{dQ_{HT}}{dt} = h_c A \frac{dT}{dt} \tag{13}$$

$$h_c = 130\, P^{0.8} U^{0.8} B^{-0.2} T^{-0.55} \tag{14}$$

Where, U is gas local velocity and considered as a function of mean piston velocity. Heat release rate also describe by Wiebe function which is modified for ethanol-gasoline blend combustion,

$$\frac{dQ_{HR}}{dt} = x_b\, m\, LHV \tag{15}$$

$$x_b = 1 - \exp\left(-Ea\left(\frac{\theta - \theta_{ig}}{\Delta\theta}\right)^{m+1}\right) \tag{16}$$

Here, x_b is the fraction of burnt fuel, Ea is activation energy, θ_{ig} is spark time angle and LHV is the fuel low heating value.
Finally to close the system of equations, in cylinder pressure, temperature and volume connect to each other according to ideal gas law, the equation stated follows:

$$PV = mR\,T \tag{17}$$

Validation

One dimension engine components model coupling via thermodynamic combustion model of ethanol-gasoline fueled SI engine is provided in simulation environment of GT-POWER commercial software shown in Figure 1. In order to validate the model, results are compared via experimental data reported in literature [30]. Also, engine characteristics are presented in Table 1. In Figure 2, the results of simulated in-cylinder pressure were compared with experimental data in 3 different fuel compositions. According to Figure 2 it can be claimed that the provided model have enough accuracy, less than 10 percent maximum local error, to be used as engine simulator and evaluate its performance.

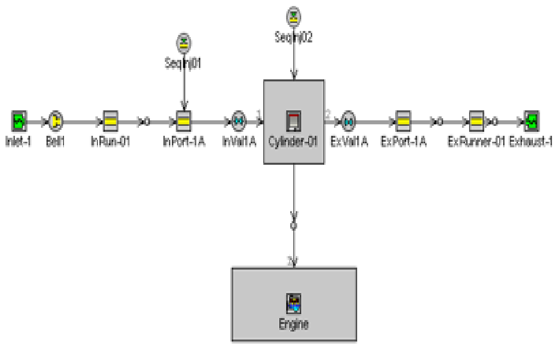
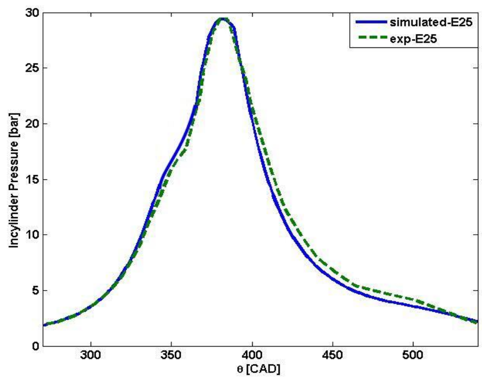


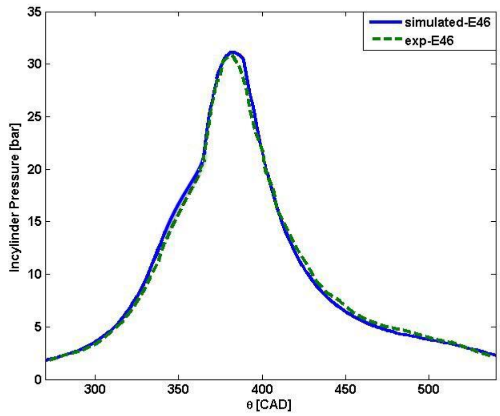
Figure 1. Schematic of engine

TABLE 1: Engine specifications [28]

Characteristics	Value (Unit)
Engine type	Single cylinder, air cooled, four-stroke
Fuel	E-0, E-25, E-46, E-58, E-69, E-76, E-85, E-100
Bore	74(mm)
Stroke	58(mm)
Connection rode length	102(mm)
Compression ratio	9.8
N	4000(rpm)
IVO	22.20 (CAD bTDC)
IVC	53.80 (CAD aBDC)
EVO	54.60 (CAD bBDC)
EVC	19.30 (CAD aTDC)



(a)



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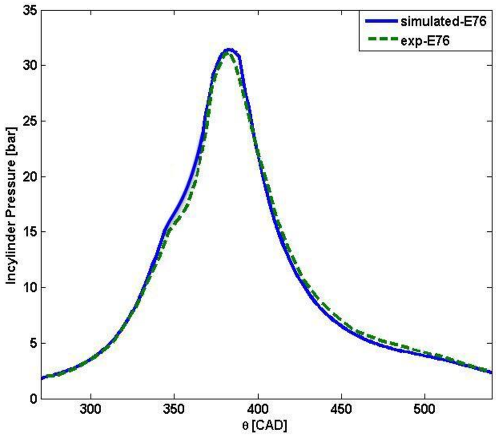

(c)

Figure 2. Comparison simulated in-cylinder pressure via experimental data [30], a) E10, b) E20, c)E30

RESULTS AND DISCUSSION

In this section, the main results of current study are presented and the effects of engine inlet variables such as inlet air pressure and temperature, engine speed, equivalence ratio and ethanol proportion on engine performance is investigated in details.

Fuel composition

To investigate the effect of fuel composition, it is changed from E-0, pure gasoline, to E-100, pure ethanol. The results of engine power and torque changes via ethanol percentage enhancement are shown in Figure 3. Both power and torque sharply increased up to E-45, then almost fixed to E-75 and then fall after that. The same trend is reported for Indicated Mean Effective Pressure (IMEP) while Indicated Specific Fuel Consumption (ISFC) reduction converts to the enhancement after E-45 shown in Figure 4. Power performance of studied engine is almost the same between E-45 and E-75, but engine has the least fuel consumption when it is run via E-45. Consequently, E-45 can be a design point of fuel composition for achieving high power efficiency from the studied engine. However, NOx emission is sharply increased to 800 ppm up to E-45 and then reduced and the same trend is observed for NOx per power curve shown in Figure 5. Due to the least produced NOx emission per power for E-25 shown in Figure 5, this point can be also introduced as a design point having cleaner engine.

Engine Speed

To investigate the effects of engine speed on its performance, the range of 950 to 7000 rpm is selected considering engine operating limitations. The other inlet parameters of engine were fixed and inlet fuel is selected as E-25. Increasing engine speed decreases the time of heat transfer; therefore, mean in-cylinder temperature and pressure would rise. Figure 6 reports a 90 K increase in cylinder peak temperature and 33 percent enhancement in peak pressure simultaneously due to increasing engine speed. These can raise IMEP and output power and consequently ISFC decreases.

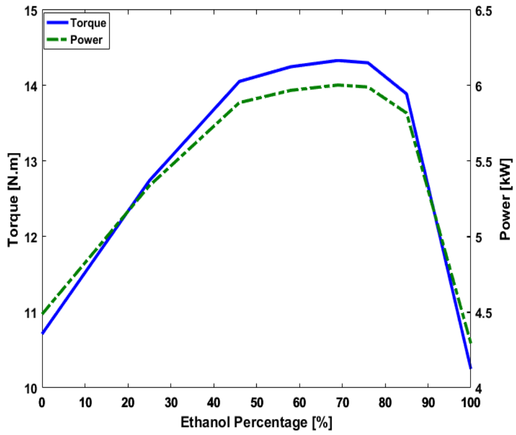

Figure 3. Torque and power via fuel composition

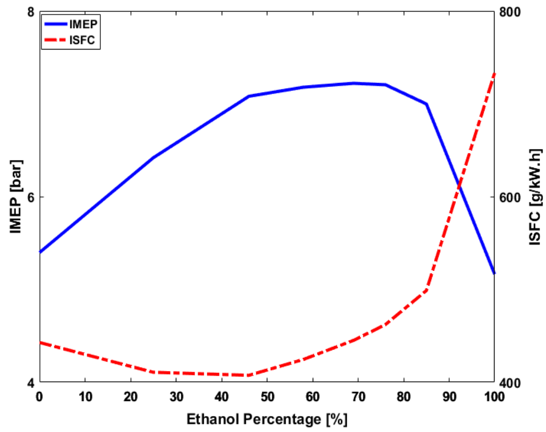

Figure 4. IMEP and ISFC via fuel composition

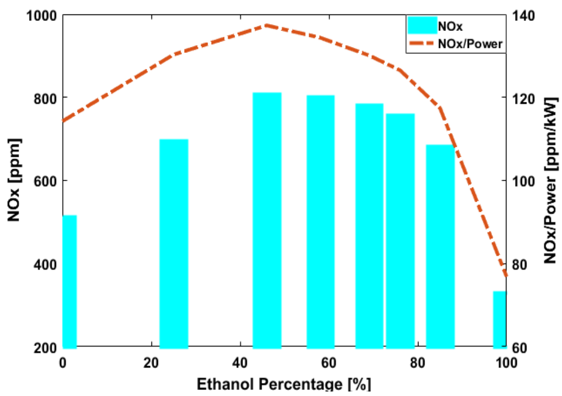

Figure 5. NOx emission and NOx/Power via fuel composition

Combustion efficiency is reduced by engine speed enhancement more than 3000rpm. So, engine thermal efficiency decreases and ISFC increases after this point. Therefore, the speed of 3000 rpm can be introduced as a design point for studied engine with E-25 as the fuel, shown in Figure 7.

Charge pressure and temperature

The effects of using supercharger can be modeled via charge pressure raise. According data shown in Figure 8, the trends of both IMEP and ISFC are linear, increase and decrease, respectively. In addition, mean and peak temperature of cylinder would increase due to isentropic correlation.

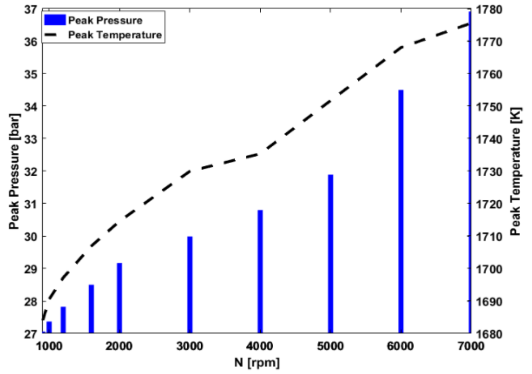

Figure 6. Peak pressure and temperature via engine speed

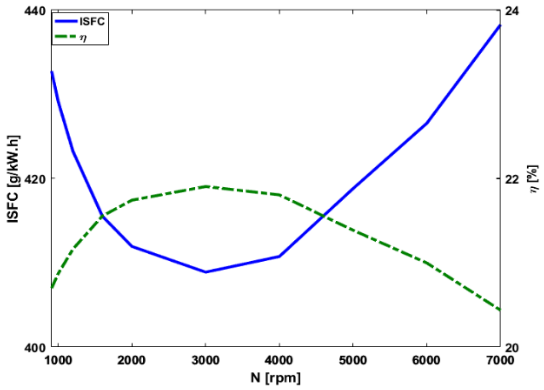

Figure 7. Thermal efficiency and ISFC via engine speed

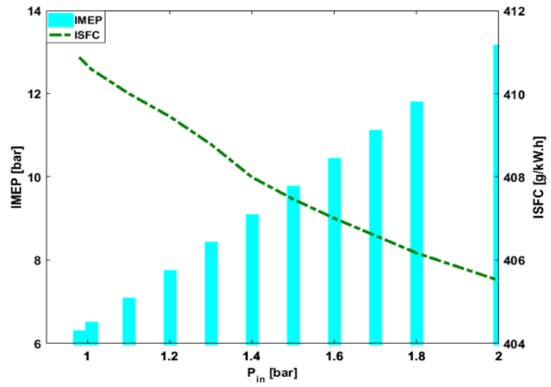

Figure 8. IMEP and ISFC via inlet pressure

Therefore, NOx emission increases by 270 ppm via charge pressure enhancement from 1 to 2 bar as shown in Figure 9.

Engine volumetric efficiency reduces by charge density reduction caused by charge temperature increasing. Engine power and torque changes are reported in Figure 10 which had 2 and 2.5 percent reduction, respectively; due to the charge 20 ℃ enhancement. Also, ISFC increases while IMEP decreases via charge temperature raise shown in Figure 11.

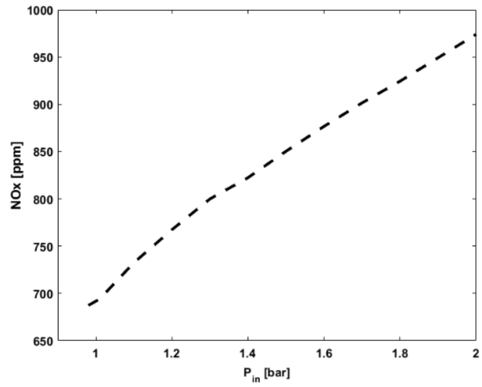

Figure 9. NOx emission via inlet pressure

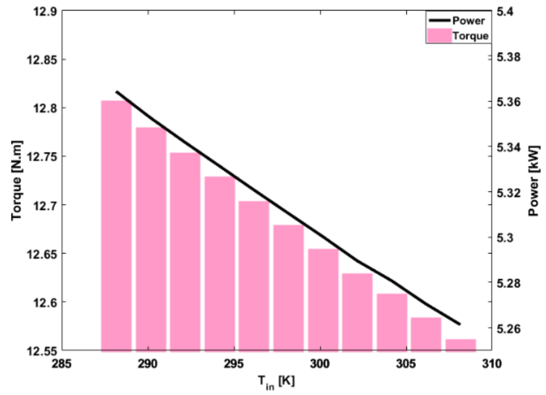

Figure 10. Torque and power via inlet temperature

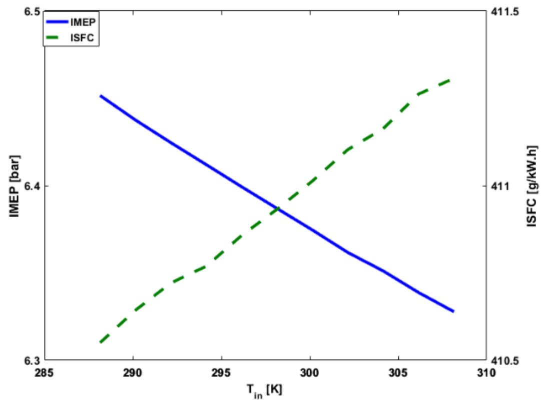

Figure 11. IMEP and ISFC via inlet temperature

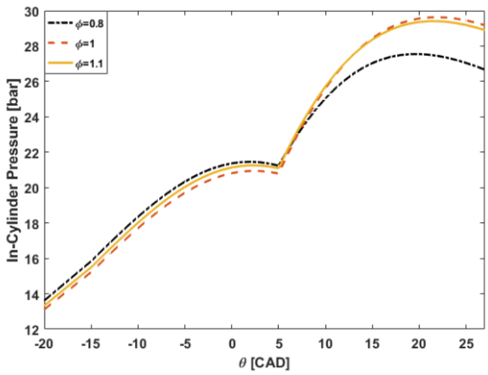

Figure 12. In-Cylinder pressure via equivalence ratio

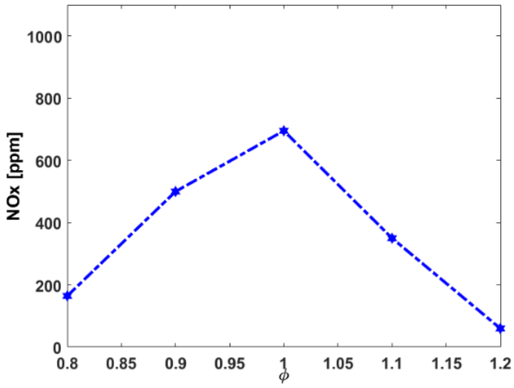

Figure 13. NOx emission via equivalence ratio

Equivalence ratio

Equivalence ratio enhancement, theoretically up to 1, can improve engine performance. Considering combustion efficiency, engine performance improvement continue up to almost 1.1. This fact is verified in Figure 12 showing in-cylinder pressure. Better combustion cause of richer charge releases more heat and causes more peak temperature. Therefore, NOx emission is increased up to stoichiometric equivalence ratio while after one, in-cylinder gas density enhancement beside combustion efficiency reduction bring less peak temperature and NOx emission; data are shown in Figure 13.

CONCLUSIONS

One dimension engine components model coupling via thermodynamic combustion model of ethanol-gasoline fueled SI engine is provided in simulation environment of GT-POWER commercial software. The effects of operating conditions on engine performance are investigated in detail and main results are summarized below:

- Provided model has acceptable accuracy, less than 10 percent maximum local error, and is able to consider full range of fuel composition; from E-0 to E-100.
- E-45 and E-25 can be the points of design to achieve best power performance and clean engine, respectively.
- 3000 rpm can be the point of design to achieve best performance for studied engine when engine is run by E-25.
- Stoichiometric charge brings more combustion performance besides maximum NOx emission production.

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چکیده

امروزه دو عامل اصلی پژوهشگران در زمینه های مختلف صنایع عبارتند از: انتشار و مصرف سوخت. آشفتگی سیاسی نفت خام علاوه بر قوانین سختگیرانه محیط زیست در جهان، محققین را برای یافتن راه های جدید برای مصرف سوخت و کاهش انتشار گازهای گلخانه ای، به کار می گیرد. استفاده از اتانول-بنزین به عنوان سوخت در موتورهای احتراق جرقه ای به عنوان یک ایده امیدوار برای دستیابی به این هدف برای صنایع موتور احتراق داخلی در نظر گرفته شده است. ارائه مدل برای بررسی عملکرد موتور سوخت اتانول-بنزین در شرایط عملیاتی مختلف، هنوز برای کاهش هزینه های آزمایشی آزمایش نیاز است. در این مطالعه، یک مدل ترمودینامیکی از موتور احتراق سوخت اتانول-بنزینی ارائه شده است و اثرات شرایط عملیاتی بر عملکرد موتور، پس از تایید نتایج شبیه سازی شده از طریق داده های تجربی، مورد بررسی قرار می گیرد. نتایج نشان می دهد که مدل ارائه شده، اطلاعات قابل اعتماد از عملکرد موتور را در طیف کاملی از ترکیب سوخت، از اتانول خالص به بنزین خالص تولید می کند. علاوه بر این، موتور مورد مطالعه حداکثر قدرت را علاوه بر بهترین مصرف سوخت تولید می کند در حالی که با ۳۰۰۰ نرخ در دقیقه اجرا می شود. همچنین بهترین عملکرد با ترکیب E-45 به دست می آید، در حالی که انتشار NOx در مقایسه با بنزین خالص ۶۰ درصد افزایش می یابد. بنابراین، می توان آن را به عنوان نقطه طراحی برای موتور مورد مطالعه معرفی کرد.