

# THEORETICAL INVESTIGATION OF THE EFFECTS OF ETHANOL-DIESEL FUEL BLENDS ON DIESEL ENGINE PERFORMANCE CHARACTERISTICS AND COMBUSTION

**Z. Şahin**

Karadeniz Technical University  
[zsahin@ktu.edu.tr](mailto:zsahin@ktu.edu.tr)

**O. Durgun**

Karadeniz Technical University  
[odurgun@ktu.edu.tr](mailto:odurgun@ktu.edu.tr)

## ABSTRACT

*In the presented study, the effects of the using of ethanol-diesel fuel blends on the engine performance characteristics such as brake specific fuel consumption (BSFC), brake effective power, brake effective efficiency and exhaust emissions have been investigated theoretically in different direct-injection (DI) diesel engines by using a multi-zone thermodynamic model. The results indicate that as ethanol percentage in the mixture increases, BSFC reduces and brake effective efficiency improves significantly and brake effective power increases slightly. On the other hand, equivalence ratio decreases and ignition delay increases for ethanol blends and combustion duration indicates generally a decreasing tendency. The concentrations of nitric oxide (NO), carbon monoxide (CO) and hydrogen (H<sub>2</sub>) increase at low ethanol ratios because of increment of cylinder temperatures. But at high ethanol ratios they decrease because of decrement of cylinder temperatures. In the presented study, cost analysis has also been performed by using semi empirical relation given by Durgun. It was determined that ethanol blends are not economic for these engines because cost of ethanol is higher than diesel fuel in Turkey as well as in many of the other countries and decrease ratio of brake specific fuel consumption is low.*

**Keywords:** Ethanol blends, combustion model, cost analysis, exhaust emissions

## 1. Introduction

Ethanol is a promising renewable energy source which can be locally produced and used to extend petroleum fuel sources. The use of anhydrous ethanol as a gasoline blending component is readily accepted in many markets around the world. Over the past few decades, researchers have also investigated different methods of using ethanol in compression ignition engines to extend diesel fuel supplies and gain the benefits of reduced smoke and particulate matter emissions. These techniques can be generally divided into the following three categories [1-3]:

(a) Ethanol fumigation to the intake air charge, by using carburetion or manifold injection techniques. It seems there are limiting values of the ethanol amount that can be used in this manner, due to engine knocking tendency at high loads, and prevention of flame quenching and misfire at low loads. (b) Dual injection system that is not considered very practical, as requiring an extra high-pressure injection system for the ethanol and, thus, related major design modifications of the cylinder head. (c) Blends (emulsions) of ethanol and diesel fuel by using an emulsifier to mix different fuels in order to prevent separation; these require no technical modifications on the engine side [2]. *The most attractive and the simplest one of these techniques which we are concerned with in this study is using ethanol blends for diesel engines. Many relevant experimental studies on ethanol-diesel fuel blends have been performed in the literature [1-7]. However theoretical studies for ethanol blends are very scarce [2]. For this reason; in the presented study, it has been proposed to investigate the effects of the using of ethanol blends on the engine performance characteristics such BSFC, brake*

*effective power, brake effective efficiency and exhaust emissions theoretically in different direct-injection (DI) diesel engines by using a multi-zone thermodynamic model.*

## 2. Description of the Model for NDF

In the theoretical model used here, multi-zone thermodynamic based model developed by Shahed [8] and then Ottikkutti [9] has been used and developed with new assumptions to determine complete engine cycle and engine characteristics. Detailed information about this model has been given in the authors' previous studies [10-12]. Here, a brief description of the model has been presented. In this model, the spray injected into combustion chamber is divided into several zones. The boundaries of these zones are determined from lines of constant equivalence ratios. Applying the first law of thermodynamics, ideal gas equation and other basic relations to the cylinder charge (these zones), a system of ordinary differential equations for cylinder pressure and zones volumes have been obtained. By solving this ordinary differential equations simultaneously during engine cycle by using Runge-Kutta 4 method, cylinder pressure and zones volumes can be calculated. Also, zones temperatures can be computed from the ideal gas equation by using cylinder pressure and zones volumes.

Here, Dent's [13] correlation for spray penetration and Reitz's correlation given by Heywood [14] and Wakuri's [15] correlation for spray angle are used. The effect of swirl on spray penetration and cone angle is incorporated by using

Hiroyasu's approach given by Heywood [14]. For determination of the instantaneous total mass and instantaneous mass rates of spray zones, it is required to know the spatial distribution of diesel fuel in the zones. In this model, concentration distribution of the fuel along the spray axis is assumed to be hyperbolic, while across of the spray it is taken as a normal distribution curve by benefit from information given in literature. Here thermodynamic properties and their partial derivatives are computed for unburned mixture and for burned equilibrium products by using Olikara's method [16]. The instantaneous total heat transfer from the cylinder content to the combustion chamber walls is calculated using Annand's [17] correlation.

Along the compression and expansion processes, differential equations for  $\dot{p}$  and  $\dot{T}$  given by Heywood are solved to determine cylinder pressure and temperature variation. On the other hand, intake and exhaust processes are computed by semi-empirical method given by Durgun [18]. In the presented model, residual gasses temperature has been chosen approximately at the beginning of cycle calculations. Then, after completing the cycle calculations, chosen and calculated exhaust temperatures have been compared. If the difference ratio between these values is higher than 2 %, final value has been taken as  $T_r$  and the cycle calculations have been repeated again. This calculation procedure has been applied iteratively until the difference ratio between these values becomes smaller than 2 %. Thus, complete cycle control has been performed. In some theoretical cycle models, correction factor of indicator diagram  $\phi_i$  has been used to take into account injection advance, ignition delay and valve timing effects. The numerical values of the correction factor  $\phi_i$  were given as 0.92-0.95. But in some presented model, injection advance and ignition delay have been considered and the pressure-volume variation during the combustion process has been determined at a rounded character. Here, only exhaust valve opening advance must be taken into account. For this reason  $\phi_i$  was chosen somewhat higher than the usual values. Here  $\phi_i$  has been selected as 0.98 [18].

After determining the complete diesel engine cycle, engine performance parameters such as brake effective power, brake effective efficiency, BSFC etc. have been calculated by using relationships given by Heywood [14] and Durgun [18]. The presented model can be used for both naturally aspirated and turbocharged direct injection diesel engines. In the turbocharged version, a simple calculation procedure given by Durgun [18] has been added to the computer code. Detailed information about this model and a flow chart are given in authors' previous studies [10-12].

## 2.1 Modifications in the Developed Model for Ethanol Blends

Adaptations done for blends in presented model which originally developed for neat diesel fuel (NDF).

For using of ethanol blends, ethanol and diesel fuel mixed in the fuel tank and injected by using injector. The properties of any mixture such as lower heating value, stoichiometric air requirement, gas constant and cetane number (CN) etc have been calculated by using following the equations.

$$Q_{LHV,mix} = \frac{x_d \rho_d H_{LHV,d} + x_{eth} \rho_{eth} H_{LHV,eth}}{x_d \rho_d + x_{eth} \rho_{eth}} \quad (1)$$

$$h_{min,mix} = \frac{x_d \rho_d h_{min,d} + x_{eth} \rho_{eth} h_{min,eth}}{x_d \rho_d + x_{eth} \rho_{eth}} \quad (2)$$

$$R_{mix} = \frac{x_d \rho_d R_d + x_{eth} \rho_{eth} R_{eth}}{x_d \rho_d + x_{eth} \rho_{eth}} \quad (3)$$

$$CN_{mix} = \frac{x_d CN_d + x_{eth} CN_{eth}}{100} \quad (4)$$

where  $x_d$  and  $x_{eth}$  are the volumetric percentages of diesel fuel and ethanol respectively.  $\rho_d$  and  $\rho_{eth}$  are densities of diesel fuel and ethanol respectively. In the presented study  $h_{min}$  is taken equal to  $1/F_{st}$ .

As any empirical equation to calculate for calculation spray penetration and spray angle for ethanol mixtures have not been found in the literature yet, similar empirical equations developed for NDF has been used to calculate spray penetration and spray angle in this study.

## 2.2 Accuracy Control of the Presented Model for NDF and Ethanol Blends

In the presented study; accuracy controls of the developed model have been performed for NDF and ethanol blends. For the cases of NDF detailed comparisons with experimental and other relevant theoretical results have also been done in the authors' previous studies [12]. A satisfactory conformity can be observed between these results. Here some examples of these comparisons results for NDF and ethanol blends have been given. In Table 1-2 brake effective power and brake effective efficiency obtained from the presented model have been compared with experimental results for NDF and Bilgin et al.'s [5] experimental results for ethanol blends respectively. Fig. 1 shows the variations of cylinder pressure values obtained from the presented model and Rakopoulos et al.'s [2] experimental results for 15 % ethanol blends. Obtained cylinder pressure values from the developed model are lower than experiment values during combustion process. But cylinder pressure values are higher than Rakopoulos's values during expansion process. It can be said that this difference

might have arisen from using the same spray penetration equations given for NDF in the presented study.

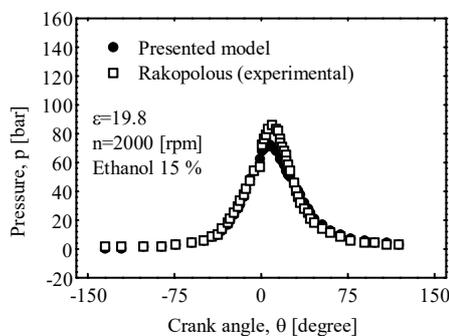
Because different equations for spray penetration for ethanol blends must be used or some correction factors have to be added to usual spray penetration equations. But any spray penetration equations for ethanol blends has not been found in the literature because theoretical ethanol studies are very scarce. Nevertheless, brake effective efficiency obtained from the presented model is about at the levels of 0.3517 and brake effective efficiency given by Rakopoulos is 0.3288 for 15 % ethanol. Thus, the difference in brake effective efficiencies between the presented model prediction and Rakopoulos’s experimental data is 6.96 %. Also, for same ethanol ratios exhaust temperature values obtained from the presented model and given by Rakopoulos are 608 K and 660 K respectively. Hence the difference between these temperatures is 7.88 %. Thus satisfactory conformity can be observed in Fig. 1 and Table 1-2 for NDF and ethanol blends.

**Table 1.** Comparison of the obtained results from the presented model with experimental results.

n=1300 [rpm]			
$\epsilon$	$\Delta N_c/N_c$ [%]	$\Delta b_c/b_c$ [%]	$\Delta \eta_c/\eta_c$ [%]
19	10.671	-7.485	7.087
20	10.605	-8.459	8.595
23	8.632	-7.879	7.782
n=1450 [rpm]			
$\epsilon$	$\Delta N_c/N_c$ [%]	$\Delta b_c/b_c$ [%]	$\Delta \eta_c/\eta_c$ [%]
19	12.889	-12.5	13.278
20	10.556	-11.144	11.245
23	10.911	-10.557	11.290

**Table 2.** Comparison of the obtained results from the presented model with Bilgin et al. [5] experimental results for ethanol blends.

$\epsilon=21$ , ethanol [%]	$\Delta N_c/N_c$ [%]	$\Delta \eta_c/\eta_c$ [%]
0	11.691	11.661
2	11.190	14.235
4	9.545	12.892
6	11.767	13.194



**Fig. 1.** Comparison of pressure values obtained from the presented model and Rakopoulos [2] experimental results as functions of crank angle.

### 3. Results and Discussion

In this section, comparisons and variations of the main engine characteristics and exhaust emissions for various ethanol blends have been shown in Figs. 2-4 and Table 3-4. Here, two turbocharged DI diesel engine given by Ottikkutti [9] and Li [20] have been used. Various numerical calculations have been performed for three engine speeds. Selected speeds of Ottikkutti’s and Li’s engines are (1500, 1700, 2100 and 1600, 1900, 2100) [rpm] respectively. Nominal engine speeds of Ottikkutti’s and Li’s engines are 1700 and 1900 [rpm] respectively. In the presented study, higher speed than nominal speed and lower speed than nominal speeds are mentioned as high and low engine speed respectively.

Fig. 2 (a-c) present the effects of ethanol blends on equivalence ratio, ignition delay and combustion duration. Equivalence ratio decreases with increasing ethanol percentage in mixture because molecular structure of ethanol contains of oxygen. This property of ethanol improves the combustion process. Ignition delay increases as the percentage of ethanol in the mixture increases because of its lower cetane number. Combustion duration increases slightly at low blend ratios (such as until 2-4 %) at high engine speeds. At low engine speeds and nominal speeds it decreases with increasing ethanol ratio in the mixture. This can be attributed to containing ethanol molecular structure oxygen and decreasing equivalence ratio with increasing ethanol percentages. The effects of ethanol at high ratios are dominant than low ratios. Thus decrement ratios of combustion duration at high ratios become longer [1].

As shown in Fig.3 (a), there is no significant difference between the cylinder pressures for ethanol blends and NDF. Fig. 3 (b) indicates the variations of temperature for different ethanol blends at 1900 rpm. It can be seen from this Fig. 3 (b) temperature of the mixture increases until approximately 6% ethanol blends. After this ratio, temperature starts to decrease. Because latent heat of vaporization of ethanol is about 1.5 times greater than that of diesel fuel which decreases the temperature in the cylinder [21].

As shown in Fig. 4 (a), brake effective power increases slightly until approximately (6-8) % ethanol-diesel fuel blends. But after these ratios, brake effective power starts to decrease. The obtained increase ratios of brake effective power at nominal engine speeds are approximately at the levels of 1.8 and 1.2 for Ottikkutti’s engine and Li’s engine respectively. The reasons of obtained improvement of brake effective power can be explained as follows: Because ethanol molecular

structure contains of oxygen and equivalence ratio decreases with increasing ethanol ratios, burning of the charge improves. Also, most of diesel fuel burns closer to TDC due to the increment of ignition delay and thus more energy of diesel fuel are released than NDF.

It can be seen from Fig. 4 (b) that brake effective efficiency for NDF is 0.3768 whereas it becomes 0.3817, 0.3845, 0.3865, 0.3875 and 0.3872 for 2, 4, 6, 8 and 10 %

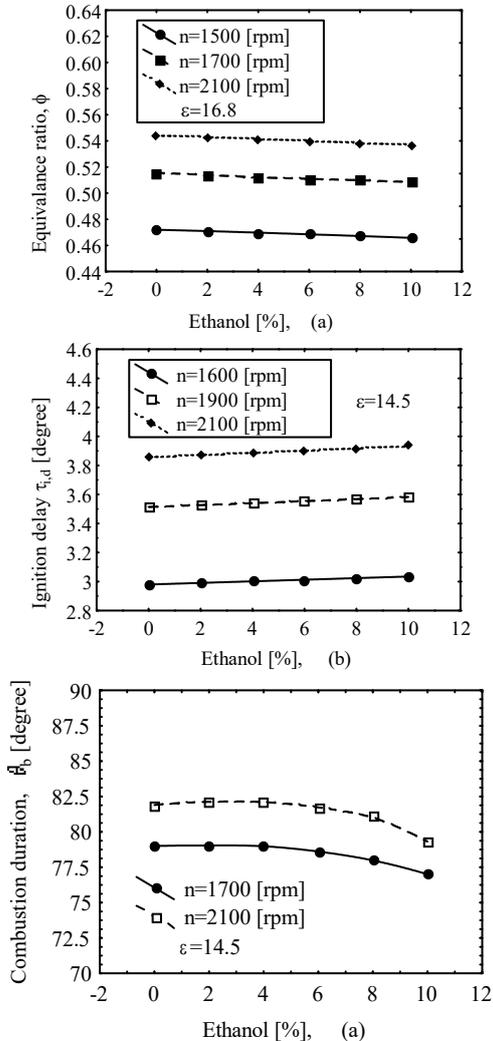


Fig. 2. (a-b) Variations of ignition delay, equivalence ratios and combustion durations as functions of ethanol ratio % respectively.

ethanol-diesel blends at high engine speed for Li's engine, respectively. As the percentages of ethanol in the mixtures increased, an increment tendency can be observed in the brake effective efficiency compared to NDF at low ethanol blends. These effects occur because of the cooling effect of ethanol as well as more efficient combustion compared to NDF. Since

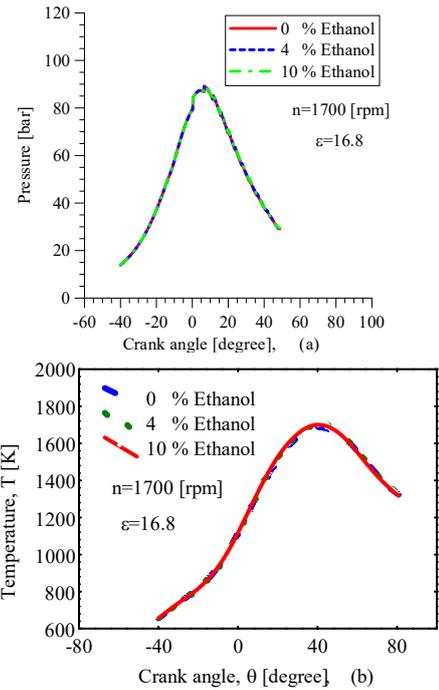


Fig. 3. (a, b) Variations of pressure and temperature as functions of crank angle, respectively.

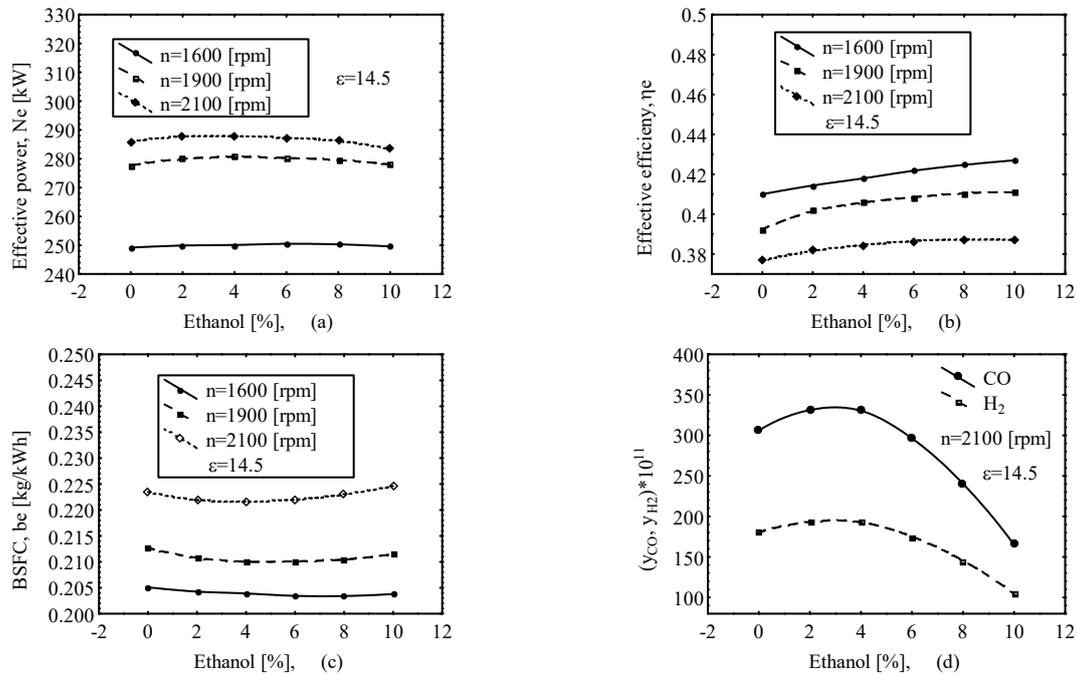
the exhaust gases temperature is lower in the case of ethanol-diesel fuel blend, less heat loss occurs through exhaust channels, and higher brake effective efficiency can be attained. Also, as combustion duration is shortened with increasing ethanol percentage in the mixture, combustion efficiency might be improved. Similar results have been obtained by other researchers [1-3]. These can also explain the reasons of being BSFC lower when ethanol blended fuels is used. After 6 % ethanol ratio, brake effective power begins to decrease and increase ratio of brake effective efficiency starts to decrease. The formation of NO is highly dependent on in-cylinder temperatures, the oxygen concentration, and residence time for the reaction to take place. The variation ratios in the NO concentration at different ethanol blends are shown in Table 4. NO concentration generally increases because combustion temperature increase slightly at low ethanol ratios and after (6-8) % ethanol ratios it starts to decrease because of lower temperature. CO ratio results are presented in Fig. 3 and Table 4 for different ethanol blends. The results show that when ethanol ratio in the mixture increases, CO ratio in the exhaust increases until approximately (2-4) % ethanol ratios. This trend is due to the fact that ethanol has less carbon than diesel fuel and ethanol gives lower temperature. After these ratios it starts to decrease because of lower flame temperature [3].

**Table 3.** Variation ratios of brake effective power, BSFC, brake effective efficiency and cost.

Ethanol [%]	n=1900 [rpm], ε=14.5			
	$\Delta N_e/N_e$ [%]	$\Delta b_e/b_e$ [%]	$\Delta \eta_e/\eta_e$ [%]	$\Delta C/C_1$ [%]
2	0.859	-0.940	2.446	1.878
4	1.092	-1.269	3.415	4.359
6	0.894	-1.175	4.052	7.290
8	0.704	-1.081	4.612	10.235
10	0.118	0.564	4.740	14.971

**Table 4.** Variation ratios of NO concentration and CO rates at 1900 [rpm].

Eth. [%]	n=1600 [rpm], ε=14.5	
	$\Delta NO/NO$ [%]	$\Delta CO/CO$ [%]
2	3.268	3.965
4	10.271	27.865
6	3.695	0.161
8	-0.786	-21.876
10	-4.755	-25.376



**Fig. 4.** (a-d) Variations of brake effective power, brake effective efficiency, BSFC, CO and H<sub>2</sub> ratios as functions of ethanol percentage % respectively.

#### 4. CONCLUSIONS

The results and recommendations achieved from the presented study can be summarized as follows:

1. The introduced theoretical model can satisfactorily predict the diesel engine cycles for neat diesel fuel and ethanol blends. Low computational cost is required for the presented model and this model could assist scientists and engineers in their efforts towards the development of more efficient and cleaner engines using alternative fuels.
2. As ethanol percentage in the mixture increases, BSFC reduces and brake effective efficiency improves significantly and brake effective power increases slightly. The obtained maximum increase ratios in brake effective efficiency and brake effective power are at the levels of 5.1 % and 2.7 %, respectively for Ottikkutti’s engine. For Li’s engine, the maximum increase ratios in brake effective efficiency and brake effective power have been calculated as 4.7 and 1.1 % respectively. The obtained maximum decrease ratios in BSFC are 2.8 and 1.3 for Ottikkutti’s and Li’s engine respectively. **Experiences gained in the developments and applications of the presented model show that the most favorable percentage of ethanol is between 4 % and 6 % for engine performance characteristics.**

3. Equivalence ratio decreases and ignition delay increases for ethanol blends and combustion duration indicates generally a decreasing tendency for two engines. NO concentration and CO and H<sub>2</sub> ratios increase at low ethanol ratios because of increment of cylinder temperatures. But at high ethanol ratios they decrease because of decreasing of cylinder temperatures. It was determined that ethanol blends are not economic for these engines because cost of ethanol is higher than diesel fuel in Turkey as well as in many of the other countries and decrease ratio of BSFC is low.

## 5. REFERENCES

- [1] He B-Q, Wang J-X, Yan , X-G, Tian X and Chen H. Study on Combustion and Emission Characteristics of Diesel Engines Using Ethanol Blended Diesel Fuels, SAE Tech. Paper Series 2003-01-0762.
- [2] Rakopoulos CD, Antonopoulos KA, Rakopoulos DC, Hountalas DT. Multi-zone modeling of combustion and emissions formation in DI diesel engine operating on ethanol–diesel fuel blends. *Energy Conversion and Management* 2008; 49: 625–643.
- [3] Abu-Qudais M, Haddad O, Qudaisat M. The effect of alcohol fumigation on diesel engine performance and emissions. *Energy Conv. and Mang.* 2000; 41: 389-399.
- [4] Li D, Zhen H, Xingcai L, Wu-gao Z. Physico-chemical properties of ethanol-diesel blend fuel and its effect on performance and emissions of diesel engines. *Renewable Energy* 2005; 30: 967-976.
- [5] Bilgin A, Durgun O, Şahin Z. The effect of diesel-ethanol blends on CI engine performance. *Energy Sources* 2002; 24: 431-440.
- [6] Can O, Çelikten I, Usta N. Effects of ethanol addition on performance and emissions of turbocharged indirect injection diesel engine running at different injection pressure. *Conversion and Management*, 2004; 45: 2429-2440.
- [7] J. Czerwinski. Performance of HD-DI-diesel engine with addition of ethanol and rapeseed oil. SAE Tech. Paper 940545.
- [8] Shahed SM, Chiu WS, Lyn WT. A mathematical model of diesel combustion. *Proc. I. Mech. E.* 1975; C94/75: 119-128.
- [9] Ottikkutti P. Multizone modeling of a fumigated diesel engine. Ph.D. Thesis, Iowa State Univ. of Science and Techn., Ames, Iowa, 1989
- [10] Şahin Z, Durgun O. Multi-zone combustion modeling for the prediction of CI engine cycles and engine performance parameters. *Applied Thermal Engineering*, (Accepted, ATE-2007-447R3).
- [11] Şahin Z, Durgun O. High speed direct injection (DI) light-fuel (gasoline) fumigated vehicle diesel engine, *Fuel* 2007; 86:388-399.
- [12] Şahin Z, Durgun O. Theoretical investigation of light-fuel fumigation on diesel engine performance and emissions, *Energy Conversion and Management*, 2007; 48: 1952-1964.
- [13] Dent JC. A basis for comparison of various experimental methods for studying spray penetration. SAE 1984; 710571: 1881-1884.
- [14] Heywood JB. *Internal combustion engines fundamentals*. McGraw Hill Book Company, New York, 1989.
- [15] Wakuri Y, Fujii M, Amitani T, Tsuneya R. Studies on the penetration of fuel spray in a CI engine, *JSME.3* 1960; 9: 123-130.
- [16] Olikara C, Borman GL. A computer program for calculating properties of equilibrium combustion products with some applications to I.C. engines. SAE 1975; 750468: 1-21.
- [17] Annand WJD. Heat transfer in the cylinders of reciprocating internal combustion eng. *Proc. Instn. Mech. Engrs.* 1963; 177: 973-990.
- [18] Durgun O. A practical method for calculation engine cycles, Union of Chambers of Turkish Engineers and Architects, Chamber of Mech. Eng. 1991; 383: 18-29: (in Turkish).
- [19] Durgun O, Ayvaz Y. The use of diesel fuel-gasoline blends in CI engines. *First Trabzon Int. Energy and Env. Syst.*, Trabzon, 1998; 2: 905-912.
- [20] Li Q. Development of a quasi-dimensional diesel engine simulation for energy and availability analysis. Ph.D. Thesis, University of Illinois, Urbana-Champaign, 1992.
- [21] Sayin C, Uslu K, Canakcı M. Influence of injection timing on the exhaust emissions of a dual-fuel CI engine. *Renewable Energy*, 2007.