



Development of a Mathematical Model that Calculates Combustion, Emission and Engine Performance Values of Internal Combustion Engines

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ARTICLE INFO

Received: Jul., 22. 2022
Revised: Sep., 22. 2022
Accepted: Oct., 10. 2022

Keywords:

Internal combustion engines
Engine performance
Exhaust emission
Graphical user interface (GUI)

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ISSN: 2536-5010 / e-ISSN: 2536-5134

DOI: <https://doi.org/10.36222/ejt.1147020>

ABSTRACT

In this study, a simulation program was developed that can calculate combustion, emission and engine performance values depended on variable parameters for internal combustion engines by mathematical modelling. This simulation program was created with the Java programming language. Graphical user interface (GUI) was used in this simulation program. With the simulation program created, it can calculate many values such as in-cylinder pressure, temperature, gas amounts generated as a result of combustion, power, torque, specific fuel consumption, both in spark ignition engines and compression ignition engines. This simulation program presents the results both numerically and graphically. The calculations are obtained at every crankshaft angle at 0.25 ° intervals. With this simulation program, analyses were made in different excess air coefficients (1-1.1-1.2) and the effects of in-cylinder pressure, temperature, mass ratios of gas components and engine performance values were compared. According to the simulation results, as the excess air coefficient increases, the temperature and pressure values decrease. CO_2 and NO mass ratios decrease as the excess air coefficient increases in case the excess air coefficient is greater than 1. H_2O mass ratio increased as the excess air coefficient increased.

1. INTRODUCTION

Nowadays, many problems arise with the increase of automobile use. The most important of these problems are the increase in environmental pollution caused by exhaust emissions and the decrease in fuel reserves. Due to these problems, many improvement studies are carried out to reduce fuel consumption and exhaust emission in internal combustion engines and alternative fuel is sought [1].

At the design process of internal combustion engines, it is not possible to experimentally predict the parameters that affect the engine's exhaust emission, fuel consumption, engine efficiency and performance. Calculating these factors affecting internal combustion engine design is very difficult and time consuming due to complex equations [2].

With the development of software engineering and computer technology, simulation programs started to be used. Thanks to the simulation programs, the analysis results replace the experimental analysis by giving results much closer to the reality [3].

The most important goal in simulation programs is to obtain the closest values to the truth. For this reason, there are many studies in the literature on simulation programs created using different mathematical and thermodynamic modeling

for internal combustion engines. For example, a software program was created with the thermodynamic model of a four-stroke direct injection diesel engine developed by Balci. The created program is written in the GW Basic language. Engine performance values were examined by making calculations with different compression and air-fuel ratios [4]. A software program was created with a thermodynamic model for four-stroke, single-cylinder diesel engines developed by Polat. The software program was created using Matlab programming language. Calculations were made in the intervals of one-degree crank angle. Calculations were made depending on different engine parameters, n-dodecane ($C_{12}H_{26}$) was used as fuel and the results were interpreted with graphics [5]. In the model developed by Esin, a software program was created with a graphical user interface (GUI) by using the combustion model for emission estimation. In the combustion model of the program created, the thermodynamic properties of the combustion products, their changes depending on pressure and temperature were calculated with the equilibrium coefficient method [6]. Kutlar et al. created a single-zone thermodynamic computational model to analyze the 13B multi-port rotary engine on the basis of the Mazda RX-8. With this model, many parameters such as combustion chamber pressure, mass and engine torque were investigated at

different speeds and full load conditions [7]. Experimental study was carried out by Cihan et al. using Wankel engine at 2 bar and 3 bar and different engine speeds. In addition, the results obtained by analyzing in the AVL Boost program under the same conditions were compared with the test results in terms of specific fuel consumption and exhaust emissions. [8].

In this study, a simulation program was developed that can calculate combustion, performance and emission values depending on variable parameters for both spark ignition engines and compression ignition engines. The simulation program was created using the Java programming language. Graphical user interface (GUI) is used in this simulation program. While creating this simulation program, firstly thermodynamic model of internal combustion engines was created. This thermodynamic modeling was created according to the real cycle. Calculations are made at each crankshaft angle at 0.25 degree intervals. Friction losses and heat transferred to the cooling water were included in the calculations. In the calculations, the exhaust gases remaining from the previous cycle in the cylinder due to the engine geometry was taken into account. Specific heat values were calculated for each crank angle (CA) depending on the temperature. In addition, in this simulation program, analysis can be made for the desired fuel by entering the atomic numbers of the fuel. Then, with this simulation program, analyzes were made in different excess air coefficients (1-1.1-1.2) and the effect of excess air coefficient on in-cylinder pressure, temperature, mass ratios of gas components and engine performance values were examined.

2. METHOD

2.1. Thermodynamic and mathematical modeling

The working principle of internal combustion engines is based on thermodynamic laws and principles. Cycle analyzes are performed in internal combustion engines based on thermodynamic principles with certain assumptions and known data [9].

2.1.1. Engine geometry and calculations

For calculations to be made in the simulation program, it is necessary to know the geometric properties of the engine. With the geometric properties of the engine, cylinder volume, combustion chamber volume, cylinder surface area, piston speed values are calculated. These values change depending on the CA.

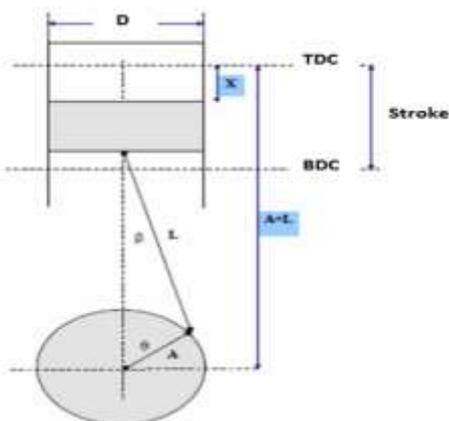


Figure 1. Engine geometry

In calculating the cylinder's volume and area, it is necessary to know the distance between the top of the piston and the top dead center of the piston. Eq. (1) is used to calculate the position of the piston depending on the CA.

$$x(\theta) = A + L - \sqrt{L^2 - A^2 \cdot \sin^2(\theta)} - A \cdot \cos(\theta). \quad (1)$$

Where $x(\theta)$ is the position of the piston relative to the (CA), θ is the crankshaft angle, A is the crank radius, L is the connecting rod length.

The combustion chamber volume is calculated by using the Eq. (2).

$$V_c = \frac{\pi \cdot D^2}{4(r-1)} \cdot S \quad (2)$$

Where V_c is combustion chamber volume, D is cylinder diameter, S is stroke length, r is compression ratio.

The instantaneous cylinder volume depending on the CA is calculated using Eq. (3).

$$V(\theta) = V_c + \frac{\pi \cdot D^2}{4} \cdot x(\theta). \quad (3)$$

The instantaneous cylinder surface area depending on the CA is calculated by using Eq. (4).

$$A(\theta) = \frac{\pi \cdot D^2}{2} + \pi \cdot D \cdot x(\theta). \quad (4)$$

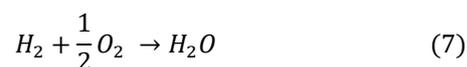
The instantaneous piston speed depending on the CA is calculated using Eq. (5).

$$V(\theta) = \frac{\pi}{2} \cdot \sin(\theta) + \left(1 + \frac{\cos(\theta)}{\sqrt{R^2 - \sin^2(\theta)}} \right) \cdot \frac{2 \cdot N \cdot S}{60}. \quad (5)$$

Where $V(\theta)$ is instantaneous piston speed, N is engine speed, S is stroke length.

2.1.2. Combustion and 1st law of thermodynamic

Combustion is the reaction of fuel and oxygen under suitable conditions. Combustion is a chemical process and is also called oxidation. Heat energy is released as a result of burning reaction. This energy is generated as a result of breaking the bond between the molecules in the fuel during the reaction. The fuel must be at the ignition temperature for the combustion reaction to occur. Each fuel has its own ignition temperature due to its chemical properties. In addition, there must be sufficient oxygen in the environment for the burning reaction [10]. Hydrocarbons (CaHb) are generally used as fuel in internal combustion engines [11]. The combustion of hydrocarbons occurs when the carbon and hydrogen atoms react with oxygen. General combustion equations of carbon and hydrogen with oxygen are given in Eq. (6) and Eq. (7).



The first law of thermodynamics is the energy conservation law. According to this law, it does not disappear when there is energy, it does not exist when it is absent. But energy can be transformed into another form. In internal combustion engines, the chemical energy of the fuel is converted into mechanical energy. For this reason, the first law of thermodynamics is valid in internal combustion engines [10].

According to the first law of thermodynamics;

$$\Delta E = \Delta U = U_2 - U_1 \quad (8)$$

$$\Delta U = \Delta Q + \Delta W \quad (9)$$

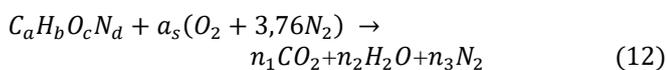
$$\Delta Q = Q_{in} - Q_{out} \quad (10)$$

$$\Delta W = \int_2^1 p \, dV \quad (11)$$

Where ΔU is the internal energy change of the system, ΔQ is the net heat in the system, ΔW is the work in the system.

2.1.3. Stoichiometric air-fuel ratio

The stoichiometric air-fuel ratio is the ratio of the amount of air required for the complete combustion of the fuel to the fuel amount. If the fuel reacts with air less than the stoichiometric ratio, not all of the fuel is burned. The stoichiometric air fuel ratio is calculated by the equilibrium coefficients of the exact combustion equation of the fuel. The exact combustion equation of hydrocarbon fuels is established as in Eq. (12) [12].



In the equation, the coefficient of oxygen and nitrogen (a_s) are calculated using Eq. (13)

$$a_s = a + \frac{b}{4} - \frac{c}{2} \quad (13)$$

Where n_1 , n_2 , n_3 values calculated using Eq. (14), Eq. (15) and Eq. (16).

$$n_1 = a \quad (14)$$

$$n_2 = \frac{b}{2} \quad (15)$$

$$n_3 = \frac{d}{2} + 3,76 a_s \quad (16)$$

The stoichiometric air-fuel ratio (A_s) is calculated using Eq. (17).

$$A_s = \frac{28,84(4,76 a_s)}{(12,01 a + 1,008 b + 16,00 c + 14,01 d)} \quad (17)$$

2.1.4. Determination of air fuel amount air-fuel ratio

When determining the amount of air fuel, the air excess coefficient must be known. Air excess coefficient was calculated using Eq. (18).

$$\lambda = \frac{\left(\frac{m_{air}}{m_{fuel}}\right)_{real}}{\left(\frac{m_{air}}{m_{fuel}}\right)_{stoic}} \quad (18)$$

Where λ is the air excess coefficient, m_{fuel} , fuel mass of the mixture, m_{air} is the air mass of the mixture.

Air-fuel mixture is called a poor and rich mixture in terms of fuel amount. A poor mixture is a condition in which the fuel enters into combustion reactions with more air than the amount of air required to burn. A rich mixture is a condition in which the fuel enters combustion reactions with less air than the amount of air required to burn. Whether the mixture is a poor or rich mixture is determined by the equivalence ratio. Equivalence ratio is indicated by the symbol (Φ). Equivalence ratio and air excess coefficient are inversely proportional [10].

$$\Phi = \frac{1}{\lambda} \quad (19)$$

- Stoichiometric mixture: $\Phi = 1$ or $\lambda = 1$
- Poor mixture: $\Phi < 1$ or $\lambda > 1$
- Rich mixture: $\Phi > 1$ or $\lambda < 1$

During the suction process, it is assumed that air is taken into the cylinder as much as the stroke volume, depending on the volumetric efficiency. The fresh air taken into the cylinder was assumed to be 21% O_2 and 79% N_2 . The amount of air taken into the cylinder;

$$m_{N_2} = \frac{P_{air} \cdot V(\theta)}{R_{N_2} \cdot T_{air}} \cdot \%N_2 \quad (20)$$

$$m_{O_2} = \frac{P_{air} \cdot V(\theta)}{R_{O_2} \cdot T_{air}} \cdot \%O_2 \quad (21)$$

$$m_{air} = m_{N_2} + m_{O_2} \quad (22)$$

The amount of fuel taken into the cylinder;

$$m_{yakut} = \frac{m_{air}}{\lambda \cdot A_s} \quad (23)$$

Where m_{N_2} is mass of N_2 in air, m_{O_2} is mass of O_2 in air, P_{air} is air pressure, T_{air} is air temperature, $V(\theta)$ is volume, R_{N_2}, R_{O_2} is gas constant, $\%N_2, \%O_2$ is percentage of components in air, λ is air excess coefficient, A_s is stoichiometric ratio.

2.1.5. Exhaust gases remaining in the cylinder

Due to the engine geometry, exhaust gases remain in the cylinder as much as the combustion chamber volume from the previous cycle. These exhaust gases remaining in the cylinder are mixed with fresh air or air fuel absorbed during the intake stroke. Since the temperature of the exhaust gases is high, it increases the temperature of the mixture. In order to start the cycle in the created mathematical model, it is necessary to know the molar amount, pressure and temperature of the filler before compression. The temperature of the exhaust gases in the cylinder is calculated using Eq. (24) and Eq. (25).

Temperature at the moment the exhaust valve is opened;

$$T_{ex} = T_{exo} \cdot \left(\frac{P_{atm}}{P_{exo}} \right)^{\frac{(n-1)}{n}} \quad (24)$$

The temperature when the exhaust valve is closed is;

$$T_{exm} = T_{ex} \cdot \left(\frac{P_{atm}}{P_{ex}} \right)^{\frac{(n-1)}{n}} \quad (25)$$

Where T_{ex} is the temperature at the time of opening of the exhaust valve, T_{exo} is the opening temperature of the exhaust valve, T_{exm} is the temperature at the moment of closing of the exhaust valve, P_{atm} is atmospheric pressure, P_{exo} is exhaust back pressure, n , polytropic index.

The total amount of exhaust gas remaining in the combustion chamber is calculated with Eq. (26).

$$m_{eg} = \frac{P_{ex} \cdot V(\theta)}{R \cdot T_{ex}} \quad (26)$$

Where P_{ex} is the exhaust counter pressure, $V(\theta)$ is volume, R is gas constant, T_{ex} is the exhaust gas temperature.

2.1.6. Lower heating value of fuel

Heating value is the amount of energy released as a result of combustion fuel. It is divided into two as higher heating value and lower heating value. The higher heating value is the highest heat value obtained by combustion a fuel. The lower heating value is the lowest heat value obtained by combustion the fuel under the worst conditions [13,14]. In the calculations, the lower heating value is used.

The lower heating value of hydrocarbons is calculated using the Mendeleyev equation in Eq. (27).

$$H_u = [3,4013C + 125,6H - 10,89(O - S) - 2,512(9H + W)]. 1000 \quad (27)$$

Where, C , H , O , S , W values indicate the weight ratios of the components in the fuel. Since there is no sulfur and water in the fuels used in internal combustion engines, S , W values are accepted as zero.

2.1.7. Specific heats

Specific heat is the amount of heat required to increase the unit mass of the substance by one degree. Specific heat is divided into two as specific heat at constant volume (c_v) and specific heat at constant pressure (c_p) [15].

While calculating the specific heat, Janaf tables were used. Janaf tables are tables in which the thermodynamic properties of substances and components are found [16].

For the specific heat of gases at constant pressure, the following equations are used.

$$c_{p_{N_2}} = (39,060 - 512,790 \cdot \theta^{-1,5} + 1072,78 \cdot \theta^{-2} - 820,40 \cdot \theta^{-3}) \quad (28)$$

$$c_{p_{O_2}} = (37,432 + 0,02010 \cdot \theta^{1,5} - 178,57 \cdot \theta^{-1,5} + 236,88 \cdot \theta^{-3}) \quad (29)$$

$$c_{p_{H_2}} = (56,505 - 702,74 \cdot \theta^{-0,75} + 1165,0 \cdot \theta^{-1} - 560,70 \cdot \theta^{-1,5}) \quad (30)$$

$$c_{p_{CO}} = (69,145 - 0,70463 \cdot \theta^{0,75} - 200,770 \cdot \theta^{-0,5} + 176,76 \cdot \theta^{-0,75}) \quad (31)$$

$$c_{p_{NO}} = (59,283 - 1,7096 \cdot \theta^{0,5} - 70,613 \cdot \theta^{-0,5} + 74,889 \cdot \theta^{-1,5}) \quad (32)$$

$$c_{p_{H_2O}} = (43,05 - 183,540 \cdot \theta^{0,25} + 82,751 \cdot \theta^{0,5} - 3,6989 \cdot \theta) \quad (33)$$

$$c_{p_{CO_2}} = (-3,7357 + 30,529 \cdot \theta^{0,5} - 4,1034 \cdot \theta + 0,024198 \cdot \theta^2) \quad (34)$$

Here;

$$\theta = \frac{T}{100} \quad (35)$$

For constant volume specific heat (c_v) values, Eq. (36) was used.

$$R = c_p - c_v \quad (36)$$

2.1.8. Percentage of fuel burned

To calculate the percentage of fuel burned based on crankshaft angle, the Wiebe function was used. The Wiebe function is an equation created by considering ignition delay, sudden and controlled combustion processes. Wiebe function is given in Eq. (37).

$$X_y = 1 - \exp \left[-a \left(\frac{\theta_x - \theta_0}{\Delta\theta} \right)^{m+1} \right] \quad (37)$$

Where X_y is the percentage of instantaneous burning fuel, θ_x , instantaneous CA, θ_0 is the CA at which combustion begins.

According to Heywood, $a = 6.908$, $m = 2$ [10].

2.1.9. Heat transfer

Approximately 10-35% of the heat generated as a result of combustion in internal combustion engines passes to the cooling system. Since heat transfer is temperature dependent, it is calculated for each CA. While calculating the amount of heat transferred to the cooling system, Eq (38) created by Woschni was used [17].

$$Q_w = \frac{A(\theta) \cdot (T_l - T_w)}{\left[\left(\frac{1}{h_g} \right) + \left(\frac{X}{k_s} \right) + \left(\frac{1}{h_c} \right) \right]} \quad (38)$$

Where Q_w is heat transfer, $A(\theta)$ is instantaneous cylinder surface area depending on CA, T_l is cylinder temperature, T_w is cylinder surface temperature, h_g is heat transfer coefficient of gases, X is thickness of cylinder wall, k_s is heat transfer coefficient of cylinder wall and it is taken as 50 W/m².K, h_c is the heat transfer coefficient of the cooling water and it is taken as 30 W/m².K.

Heat transfer coefficient (h_g) was calculated using Eq. (39) [18].

$$h_g = 0,820 \cdot D^{-0,2} \cdot P^{0,8} \cdot W^{0,8} \cdot T^{-0,53} \quad (39)$$

Where D is cylinder diameter, P is cylinder pressure, T is cylinder temperature, W is the average velocity of gases.

$$W = \left[C_1 \cdot C_m + C_2 \cdot \frac{v_d \cdot T_1}{P_1 + v_1} \cdot (P - P_0) \right] \quad (40)$$

Where C_m is piston speed.

C_1, C_2 values;

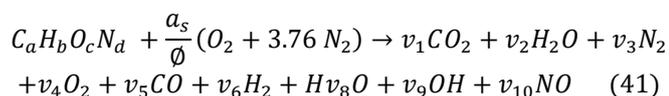
For exhaust time	$C_1=6.18$	$C_2=0$
For compress time	$C_1=2.28$	$C_2=0$
For the burning time	$C_1=2.28$	$C_2=3.24 \cdot 10^{-3}$ [18].

2.1.10. Modeling of combustion

In order to calculate the parameters inside the cylinder, the amount of instantaneous gases must be known. The air fuel mixture entering the cylinder turns into different combustion products as a result of the combustion reaction, the type and quantity of combustion products vary depending on the combustion temperature. The calculations are modeled according to the instantaneous temperature and mass quantities of the components depending on the CA. The combustion products formed as a result of the combustion of high temperature hydrocarbons with air are $CO_2, H_2O, N_2, O_2, CO, H_2, O, OH, NO, H$.

The combustion model was created by the equilibrium coefficient method. The equilibrium coefficient (K) describes the behavior of the reaction and can be formed bidirectional. Equilibrium coefficient (K) is obtained from the ideal gas equation and its partial pressures [19].

The equation of the high temperature combustion reaction and the coefficients of the components are shown in Eq. (41). The equation is constructed for one mole of fuel [20].



Here;

$$x_i = \frac{v_i}{N} \quad (42)$$

$$a = (x_1 + x_5) \cdot N \quad (43)$$

$$b = (2x_2 + 2x_6 + x_7 + x_9) \cdot N \quad (44)$$

$$c + \frac{2a_5}{\phi} = (2x_1 + x_2 + 2x_4 + x_5 + x_8 + x_9 + x_{10}) \cdot N \quad (45)$$

$$d + \frac{2a_5(3,773)}{\phi} = (2x_3 + x_{10}) \cdot N \quad (46)$$

Equilibrium coefficients of combustion products of partial pressure type;

$$\frac{1}{2} H_2 \leftrightarrow H \quad K_1 = \frac{x_7 P^{1/2}}{x_8^{1/2}} \quad (47)$$

$$\frac{1}{2} O_2 \leftrightarrow O \quad K_2 = \frac{x_9 P^{1/2}}{x_4^{1/2}} \quad (48)$$

$$\frac{1}{2} H_2 + \frac{1}{2} O_2 \leftrightarrow OH \quad K_3 = \frac{x_9}{x_4^{1/2} x_6^{1/2}} \quad (49)$$

$$\frac{1}{2} O_2 + \frac{1}{2} N_2 \leftrightarrow NO \quad K_4 = \frac{x_{10}}{x_3^{1/2} x_4^{1/2}} \quad (50)$$

$$H_2 + \frac{1}{2} O_2 \leftrightarrow H_2O \quad K_5 = \frac{x_2}{x_4^{1/2} x_6^{1/2}} \quad (51)$$

$$CO + \frac{1}{2} O_2 \leftrightarrow CO_2 \quad K_6 = \frac{x_1}{x_4^{1/2} x_5^{1/2}} \quad (52)$$

Where K_1, K_2, K_3, K_4, K_5 and K_6 values are calculated using Eq. (53). This equation was obtained from Janaf tables in the range of 600K-4000K by curve fitting method [19].

$$\log K_i = A_i \ln \left(\frac{T}{1000} \right) + \frac{B_i}{T} + C_i + D_i T + E_i T^2 \quad (53)$$

$A_i, B_i, C_i, D_i,$ and E_i values in the equation are given in Table I.

TABLE I
CURVE COEFFICIENTS FOR EQUILIBRIUM CONSTANTS [17].

i	A_i	B_i	C_i	D_i	E_i
1	0.432168	$-0.112464 \cdot 10^5$	$0.267269 \cdot 10^1$	$-0.745744 \cdot 10^{-4}$	$0.242484 \cdot 10^{-8}$
2	0.310805	$-0.129540 \cdot 10^5$	$0.321779 \cdot 10^1$	$-0.738336 \cdot 10^{-4}$	$0.344645 \cdot 10^{-8}$
3	-0.141784	$-0.213308 \cdot 10^4$	0.85346 ¹	$0.355015 \cdot 10^{-4}$	$-0.310227 \cdot 10^{-8}$
4	$0.150879 \cdot 10^{-1}$	$-0.470959 \cdot 10^4$	0.646096	$0.272805 \cdot 10^{-5}$	$-0.154444 \cdot 10^{-8}$
5	-0.752364	$0.124210 \cdot 10^5$	$-0.260286 \cdot 10^1$	$0.259556 \cdot 10^{-3}$	$-0.162687 \cdot 10^{-7}$
6	$-0.415302 \cdot 10^{-2}$	$0.148627 \cdot 10^5$	$-0.475746 \cdot 10^1$	$0.124699 \cdot 10^{-3}$	$-0.900227 \cdot 10^{-8}$

2.2. Engine performance calculations

Engine performance values are examined in two ways, indicative and effective. Indicative parameters are engine parameters calculated with the values obtained in the cylinder. The effective parameters are the values measured from the flywheel or crankshaft [21].

Power is the work done per unit time. Indicated power is derived from the in-cylinder pressure value. Indicated power is the thrust that occurs on the piston with the effect of the pressure created as a result of patching inside the cylinder. The effective power is the power obtained by subtracting the losses such as friction in the engine from the indicated power. Effective power is the real power of the engine. Indicated power is obtained using Eq. (54) [22].

$$N_i = \frac{P_i \cdot V \cdot n \cdot i}{60 \cdot T} \quad (54)$$

Where; N_i is indicated power, P_{me} is indicated pressure (mean pressure), V is stroke volume, n is crankshaft revolution, i is number of cylinders, T is the number of strokes in a cycle (it is 2 for 4-stroke engines).

Effective power is obtained using Eq. (55) [21].

$$N_e = \frac{P_e \cdot V \cdot n \cdot i}{60 \cdot T} \quad (55)$$

$$P_e = P_i \cdot \eta_m \quad (56)$$

Where N_e is effective power, P_e is effective pressure, η_m is mechanical efficiency.

Moment is the force of rotation. The pressure force generated as a result of combustion in the cylinder moves the piston. The connecting rod transfers this linear motion to the crankshaft and creates a moment effect on the crankshaft. The moment value is obtained by using Eq. (57) [22].

$$M_e = 9549 \cdot \frac{N_e}{n} \quad (57)$$

Where M_e is moment, N_e is effective power, n is engine speed.

Specific fuel consumption is the amount of fuel spent in unit time for unit power. Specific fuel consumption is obtained using Eq. (58) [21].

$$B_e = \frac{m_f}{N_e} \quad (58)$$

Where; B_e is specific fuel consumption, m_f is amount of fuel consumed per unit time, N_e is effective power.

2.2. Conversion of mathematical model to software language

Mathematical modeling that created with thermodynamic formulas was turned into a software program with Java programming language.

Java was generated in 1995 by the Sun Microsystems company. Java is an object-oriented programming language. Java is a programming language that can work on all operating systems without rearrangement [23].

NetBeans platform was used while creating the simulation program. NetBeans is the Java development platform developed by Oracle. NetBeans is a code development environment that offers opportunities such as writing, compiling, and debugging [23].

For this simulation program can be easily used by the user, graphical user interface (GUI) was created. While creating the GUI, Java's Swing library was used. The Swing library is a library that is used to develop components such as panels, buttons, tables and menus required for the GUI [23].

3. RESULTS AND DISCUSSION

The generated software contains approximately 3520 lines of code. Codes were created to calculate values at each crankshaft angle at 0.25° intervals. The calculations are obtained by creating arrays in a repeating loop. The calculations of the in-cylinder parameters are set to be made

with the CA values between the closing of the intake valve and the opening of the exhaust valve. Engine performance parameters have been obtained in the range of 900-5500 (rpm).

The created GUI is the screen where the parameters required for the calculations of internal combustion engines are entered. This screen consists of 3 sections. These sections are the section where the menus are grouped with the engine parameters, the section where the engine parameters are entered and the message section. The main screen encountered by the user when the program is opened is shown in Fig. 2.

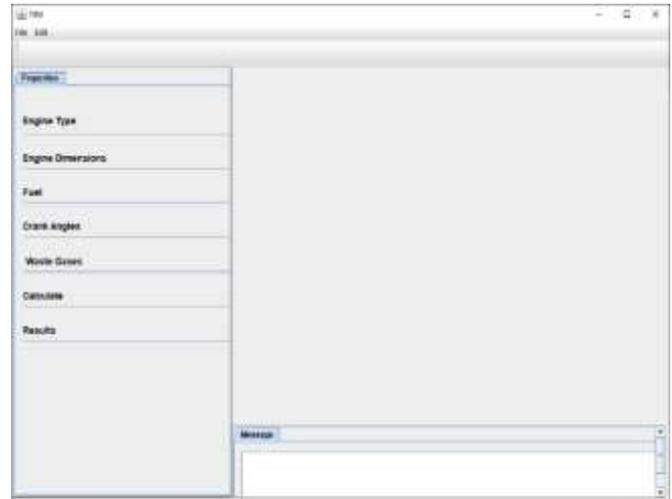


Figure 2. Main screen of created interface.

Properties menu of the interface created; It consists of 7 menu buttons in order to enter engine parameters, to make calculations and to see the results.

The engine type button is the section where the spark ignition engine or compression ignition engine type is entered. The screen where the engine type is entered is shown in Fig. 3.

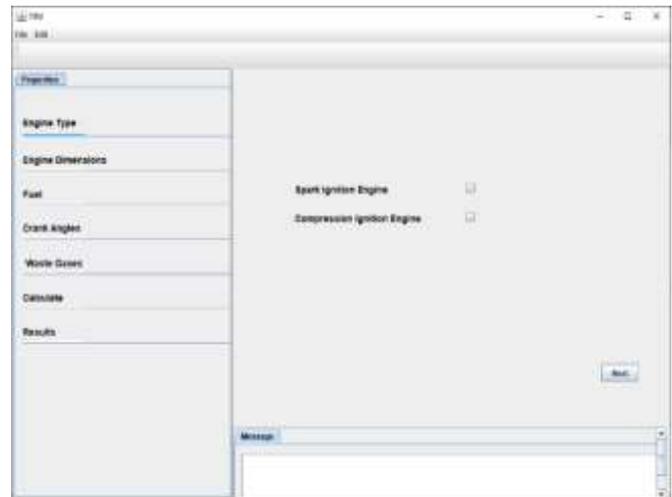


Figure 3. The screen for entering the engine type.

It is the section where the engine dimensions button, cylinder number, cylinder diameter, stroke length, connecting rod length, compression ratio and engine speed are entered. The screen where motor dimensions are entered is shown in Fig. 4.

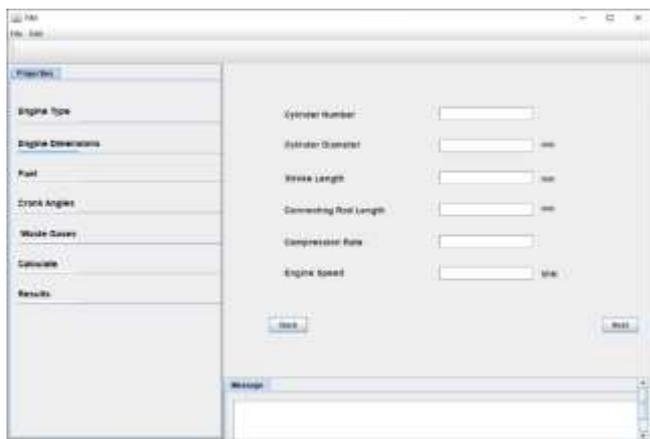


Figure 4. The screen for entering motor dimensions.

The fuel button is the section where the atomic numbers of the fuel, the excess air coefficient, the pressure of the sucked air, the temperature of the sucked air and the temperature of the cooling water are entered. This screen is shown in Fig. 5.

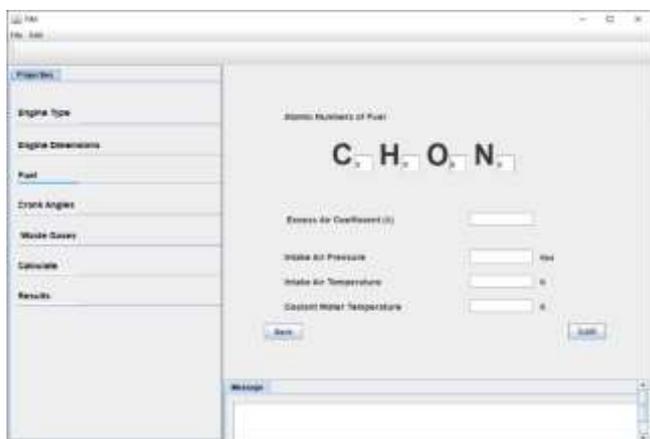


Figure 5. The screen where the properties of fuel, air and cooling water are entered.

The crank angles button is the section in which the closing of the intake valve, the opening of the exhaust valve, the ignition starts and end for spark ignition engines, the start and end of the spraying for compression ignition engines are entered as the CA value. This screen is shown in Fig. 6.

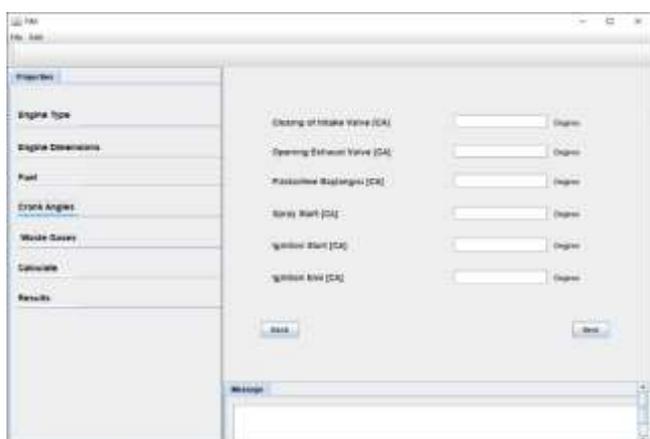


Figure 6. Screen for entering crank angles.

The waste gases button is the section where the temperature of the exhaust gases from the previous cycle due to the engine geometry and the mass ratios of the components are entered. In

the calculations, CO₂, H₂O, O₂, N₂ components in the exhaust gases remaining in the cylinder were taken into account. Since the mass ratios of other components in the exhaust gases are low, they are not taken into account in the calculations. The screen where the properties of exhaust gases are entered is shown in Fig. 7.

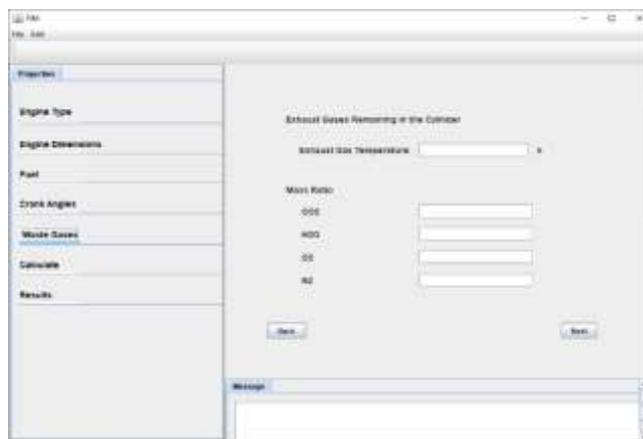


Figure 7. The screen where the properties of fuel, air and cooling water are entered.

There is a button in the calculation menu. When this button is clicked, it provides the calculation by transferring all the entered data to the software. After the calculations are completed, the simulation program creates an Excel file containing the results to analyze the results numerically. After the calculations are completed, the screen with the graphics created is shown in Fig. 8.

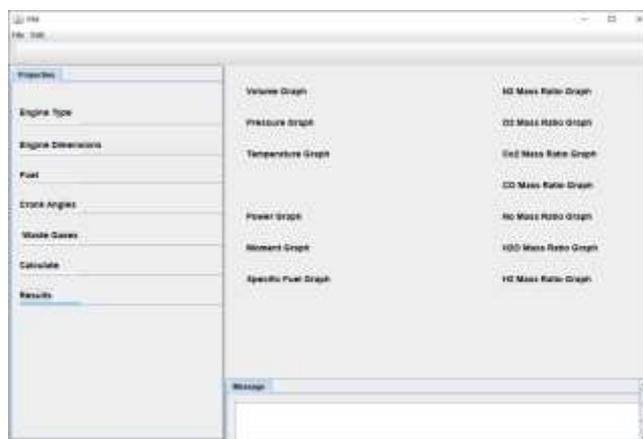


Figure 8. Graphics screen.

With this simulation program, analyzes were made with different excess air coefficients for gasoline engines, and in-cylinder pressure, temperature, mass ratios of gases and engine performance values were compared. Different excess air coefficients (1-1.1-1.2) values were used for the analyzes. The characteristics of the internal combustion engine used in the analysis and the values entered are given in Table 2.

TABLE II
MOTOR CHARACTERISTICS AND ENTERED VALUES

Parameters	Feature
Engine type	Spark ignited
Number of cylinders	4
Cylinder diameter	73 mm

Stroke length	80 mm
Connecting rod length	149mm
Compression ratio	10.8
Engine Speed	3000 (rpm)
Fuel	Gasoline (C_8H_{18})
Excess air coefficient (EAC)	1-1.1-1.2
Pressure of intake air	100 kPa
Temperature of the intake air	300 K
Ignition time	351-368
Coolant water temperature	340 K
Exhaust gas temperature	850K
The ratio of N_2 in waste gas	0.7
The ratio of O_2 in waste gas	0
The ratio of CO_2 in waste gas	0.17
The ratio of H_2O in waste gas	0.13

The graph of the change in the cylinder temperature depending on the CA created from the values obtained as a result of the calculations in the simulation program is shown in Figure 9, and the change in the cylinder pressure depending on the CA is shown in Figure 10.

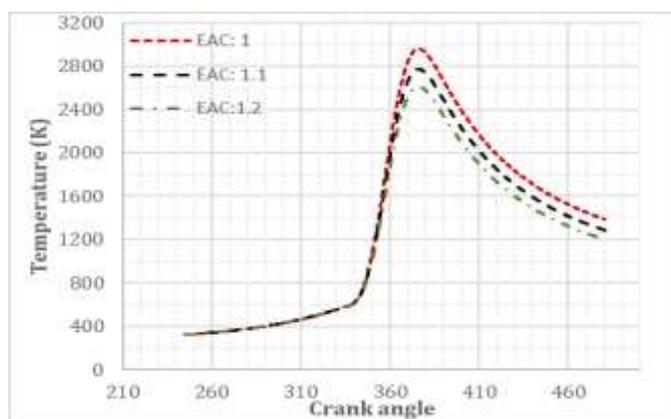


Figure 9. The change of cylinder inside temperature depending on crank angle in different EAC.

When the temperature graph inside the cylinder depending on the CA is examined, the maximum temperature values for 1-1.1-1.2 EAC values respectively; 2960 K at 375 CA, 2772 K at 375 CA, 2612 K at 375 CA. According to these values, as EAC increases, temperature values decrease. The reason for this is that as the EAC increases, the air fuel mixture gets poorer, thus the amount of fuel decreases. Therefore, depending on the amount of fuel, the amount of heat generated as a result of combustion decreases

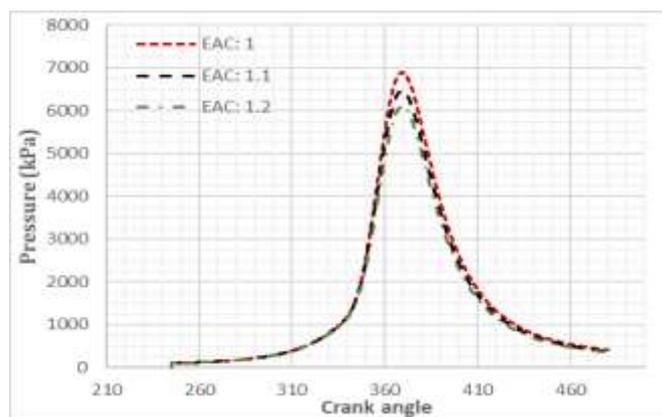


Figure 10. The change of in-cylinder pressure depending on the crank angle in different EAC.

When the in-cylinder pressure graph depending on the CA is examined, the maximum pressure values for 1-1.1-1.2 EAC values respectively; 6892 kPa at 369 CA, 6892 kPa at 369 CA, 6099 kPa at 369 CA. When these values are examined, as the EAC increases, the pressure values decrease as well as the temperature values.

The graphs of the mass ratios of gas components in the cylinder depending on the CA are shown in Figure 11 for CO_2 , Figure 12 for H_2O , Figure 13 for NO .

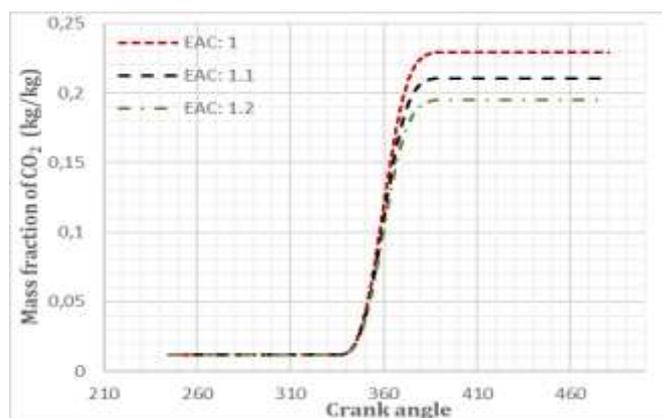


Figure 11. The change of CO_2 mass fraction in different EAC depending on the crank angle.

For CO_2 ; When the mass ratios in the cylinder are examined depending on the CA, as the EAC increases, the in-cylinder CO_2 mass ratio as a result of combustion decreases.

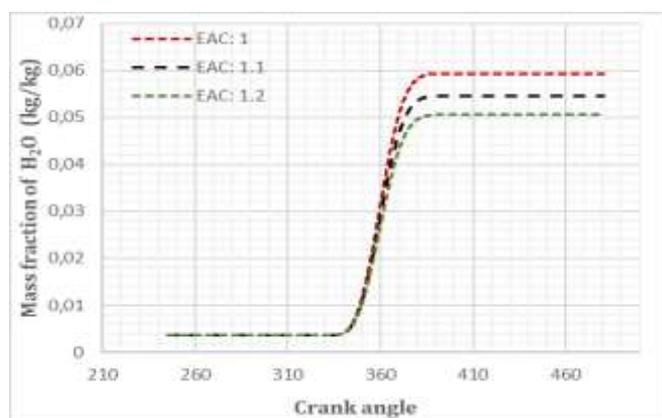


Figure 12. The change of H_2O mass ratio in different EAC depending on crank angle

For H_2O , When the mass ratios in the cylinder depending on the CA are examined, the H_2O mass ratio decreased as the EAC value increased.

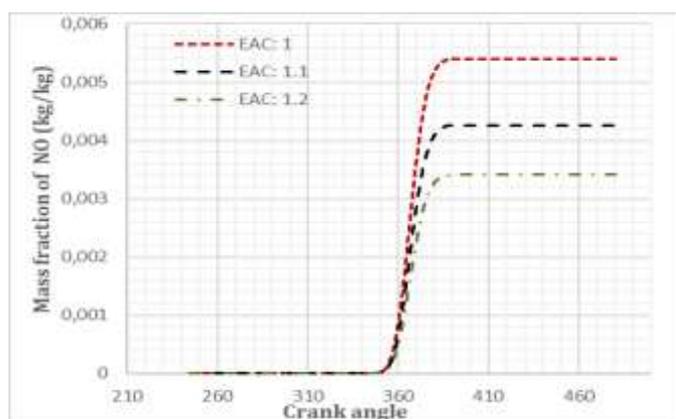


Figure 13. The change of NO mass ratio in different EAC depending on the crank angle.

For NO, when the mass ratios in the cylinder depending on CA are examined, as the EAC value increases, the mass ratio of NO also decreases.

The change of effective power (kW) in different EAC depending on engine speed is shown in Figure 14.

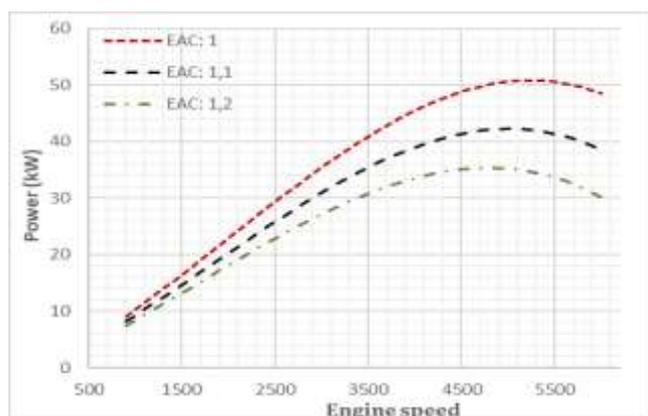


Figure 14. Change of effective power (kW) in different EAC depending on engine speed.

The change of moment in different EAC depending on engine speed is shown in Figure 15.

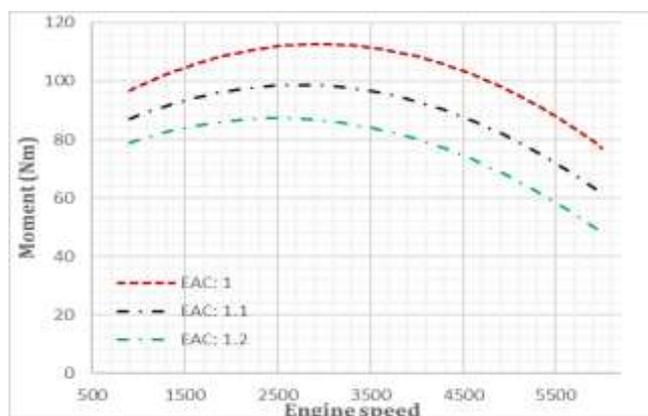


Figure 15. Change of moment in different EAC depending on engine speed.

The change of specific fuel consumption in different EAC depending on engine speed is shown in Figure 16.

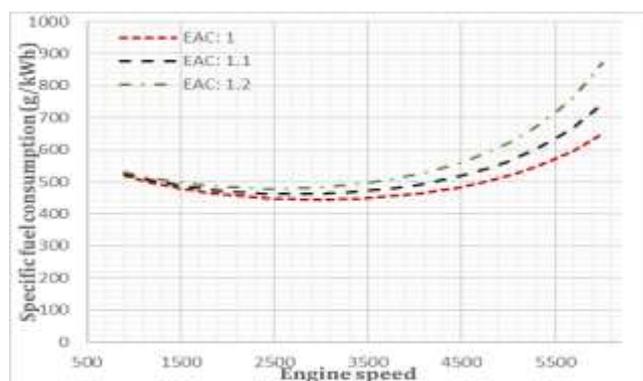


Figure 16. Change of specific fuel consumption in different EAC depending on engine speed.

When effective power and moment graphs are examined, power; 50.82 kW at 5200 rpm at 1 EAC, maximum moment; It was obtained as 112.65 Nm at 2900 rpm at 1 EAC. The most important factor when calculating engine performance values is the in-cylinder pressure value. As the EAC value increases, the in-cylinder pressure value decreases. Therefore, as the EAC value increases, the power and moment values decrease.

When the specific fuel consumption graph is examined, it is seen that the specific fuel consumption increases as the EAC value increases.

The results obtained by using Honda L13A4 i-DSI spark ignition engines in the simulation program created are compared with the data obtained from the experimental study in Table III [24].

The comparison was made with the values of the engine at full load and the mass ratios of the gases obtained from the simulation program were converted into volume ratios for comparison.

TABLE III
COMPARISON OF THE OBTAINED DATA.

Parameters	Experimental	Simulation Results
Power (kW)	70	63.7
Torque (Nm)	127,3	133.2
Specific fuel consumption (kg/kWh) 3000 (rpm)	310	366
CO ₂ (volume%) 3000(rpm)	10,65	12,3
CO (volume %) 3000(rpm)	5,5	5,72
NO (ppm) 3000(rpm)	2220	2153

The most important goal in simulation programs is to obtain values closest to reality. When the results obtained for the Honda L13A4 i-DSI spark ignition engine in the developed software program are compared with the data obtained from the experimental study of the engine, it is seen that they are compatible. The resulting differences can be caused by entered or selected motor parameters, omissions and calculation deviations.

4. CONCLUSION

In this study, a simulation program was developed that can calculate combustion, emission and engine performance values depended on variable parameters for internal combustion engines by mathematical modeling. This simulation program was created with the Java programming language.

Since liquid hydrocarbons are generally used as fuel in internal combustion engines, the simulation program has been created to make calculations based on the atomic numbers of the desired fuel. Thermodynamic calculations were created to be performed for each angle within the 0.25 degree range of the crankshaft. At each crank angle, the masses of the fuel and gases in the cylinder and the specific heat of these gases depending on the temperature, the heat released and the amount of heat transferred to the cooling water were calculated instantly.

With this simulation program, analyzes were carried out in different excess air coefficients, and in-cylinder pressure, in-cylinder temperature, mass ratios of gas components in the cylinder, power, moment and specific fuel consumption values were compared.

When the analysis results for the excess air coefficient (EAC) are examined, as the EAC increases, the pressure and temperature values in the cylinder decreased. Due to the decrease in pressure, the power and moment values decreased as the EAC increased.

When the CO_2 , H_2O , NO values in the cylinder after combustion were examined, it was seen that the mass ratios of CO_2 , H_2O , NO decreased as EAC increased.. H_2O mass ratio decreased as the EAC value increased. On the other hand, specific fuel consumption increased as EAC increased.

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BIOGRAPHIES

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