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THERMODYNAMIC MODELING FOR ANALYSING THE PERFORMANCE AND EMISSIONS OF SPARK-IGNITION ENGINES

Dr. Mahmoud Attallah Mashkour[1], *Ahmed Yaseen Ahmed[2]

1) Assistant Prof., Mechanical Engineering Department، University of Technology, Baghdad, Iraq.
2) Assistant Lecturer, Ministry of Education, Al Nahda Vocational School, Baghdad, Iraq.

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Abstract: A thermodynamic modeling for the simulation of a spark ignition engine running on gasoline fuel and other alternate hydrocarbon fuel is presented. This paper aims to develop a simple, fast and accurate engine simulation model by using **MATLAB (GUI)** program. The model is based on the classical two-zone approach, wherein parameters like the first law of thermodynamics, equations for energy, mass conservation, equation of state and mass fraction burned، heat transfer from the cylinder. Curve-fit coefficients are employed to simulate air and fuel data along with frozen composition and practical chemical equilibrium routines. The mathematical model has the ability to predict the cumulative heat release, cylinder pressure, cylinder gas temperature, heat transfer from the gases to cylinder wall and work done for hydrocarbon fuels using a Zero-dimensional combustion model. In addition, the program has the ability to predict engine performance and exhaust emissions at any condition of engine. The validity of the mathematical model has been tested against experimental data obtained from four-stroke S.I. (***Mercedes Benz 200E***) engine. A good agreement was obtained between the results of the present model and the experimental results. Remarkable similarity has been found with the literature published. In comparing model predictions with experimental data, it is found that all brake specific fuel consumption predictions are accurate to within $\pm$ **3%**, while all brake thermal efficiency predictions are accurate to within $\pm$ **4%**.

Keywords: *Thermodynamic modeling, MATLAB (GUI), Zero-dimensional, SI engine*

النمذجة الثرموديناميكية لتحليل معاملات الأداء والانبعثات في محركات الأشتعال بالشرر

الخلاصة: في الدراسة الحالية تم بناء نموذج رياضي لمحاكاة عملية الاحتراق في محركات الأحتراق بالشرر رباعي الأشواط ولوقود مفرد ووقود ثنائي .تهدف هذه الدراسة الى بناء وتطوير نموذج محاكاة بسيط وسريع ودقيق وباستخدام برمجة الحاسوب الآلي وبلغة الماتلاب وباستخدام (واجهة المستخدم الرسومية). تم بناء البرنامج باستخدام القانون الأول في الثرموداينمك وقانون حفظ الطاقة وقانون حفظ الكتلة . يقوم البرنامج بحساب الضغط ودرجة الحرارة للغازات داخل الأسطوانة وانتقال الحرارة من الغازات الى جدران الأسطوانة والشغل المسلط على مكبس الأسطوانة للوقود المفرد والوقود الثنائي بأستخدام النموذج اللابعدي (**Zero – Dimensional Model**). النموذج الرياضي المعد يمكنه كذلك حساب معاملات الأداء للمحرك وتحليل الأنبعاثات الناتجة من المحرك للوقود المفرد والوقود الثنائي عند أي ظرف .لغرض التحقق من صحة نتائج البرنامج تم أستحصال نتائج معامل الأداء لمحرك مختبري أشتعال بالشرر (**Mercedes Benz 200E**)رباعي الأسطوانة عند نسبة أنضغاط (**9**) وسرعة ثابتة (**1500 rpm**) لوقود مفرد (بنزين) ووقود ثنائي (بنزين + أيثانول) بنسب خلط مختلفة ثم قورنت هذه النتائج مع النتائج المتحصلة من النموذج الرياضي وبينت المقارنة وجود توافق جيد جدا.إضافة الى ذلك قورنت هذه النتائج مع النتائج في البحوث المنشورة عالميا وكانت متوافقة بدرجة كبيرة وهذا يثبت ان النموذج الثرموديناميكي قد تم بناءه بشكل صحيح . وعند مقارنة النتائج المتحصلة من النموذج

*Corresponding Author: ahmed.yaseen3333a@gmail.com

الرياضي مع النتائج العملية ، تبين ان نسبة الخطأ في معدل استهلاك الوقود المكبحي بنسبة $\pm 3\%$ وفي الكفائة الحرارية المكبحية بنسبة $\pm 4\%$.

1. Introduction

Four-stroke spark ignition (SI) engine was developed by Nikolaou's Otto in 1876.That engine produced power output 3 hp[1]. The engine developing has been done constantly over 100 years. Even now, some spark ignition engines can produce power output more than 1000 hp[2]. Nevertheless, developments of spark ignition engine along 100 years has been directing very slow due to lots of parameters, such as physical geometries (bore, stroke, compression ratio, crank radius,) valve timing, advanced ignition, combustion characteristic etc. Several studies on effects of each parameter were done by experiment. However, this approach spent lots of expenses and time such as building test engine setting up laboratory etc. The simulation method that allows spark ignition engine designer to variate and test many different parameters without building real parts or even real engines leads to low cost and minus time consumption. Many mathematical models have been developed to assist understand, correlate, and investigate the process of engine cycles. Computer simulations of internal combustion engine cycles are desirable because of the help they provide in design studies, in predicting trends, in serving as diagnostic tools, in giving more data than are normally obtainable from experiments, and in helping one to understand the complex processes that occur in the combustion chamber. In the present work, a Zero-dimensional combustion model is employed to simulate a 4-stroke cycle of a spark ignition engine fueled with various types of fuels, i.e. gasoline, methane, ethanol, and their mixture. The most important assumptions were that [1, 2, 3]:

1- The contents of the cylinder are fully mixed and spatially homogeneous in terms of composition and properties during intake, compression, expansion, and exhaust processes.

2- For the combustion process, two zones (each is spatially homogeneous) are used. The two zones are the burned and the unburned zones. The two zones are always separated by an infinitesimally thin flame.

3- The cylinder pressure is assumed to be the same for the burned and unburned zones.

4- The heat transfer between the two zones is neglected.

5- The working medium was considered, in general, to be a mixture of **10** species $\mathbf{CO_2}$, $\mathbf{H_2O}$, $\mathbf{CO}$, $\mathbf{O_2}$, $\mathbf{N_2}$, $\mathbf{H_2}$, $\mathbf{OH}$, $\mathbf{NO}$, $\mathbf{H}$ and $\mathbf{O}$.

6- All **10** species were considered as ideal gases.

2. Literature Review

Sitthichok Sitthiracha [4] 2006 developed a mathematical model of spark ignition engine, which combines both physical formulae such as burning duration, empirical formulae and engine geometries. The engine performance, torque and power, could be calculated by integrating the pressure inside cylinder within one engine cycle. The model was verified by data obtained from (Mercedes-Benz 250SE) engine. It can detention torque and power characteristics very well. The overall errors were between (-6% to 4%). In addition, this model was used for simulating in order to predict the burning duration of the different fuels. Arun Singh Negi et.al [5] 2016 studied the engine performance characteristics with ethanol-gasoline blend, a cylinder-by- cylinder model designed in MATLAB/Simulink. The mathematical models build in the form of blocks differential and empirical formulas of engine parameters, which were describing the engine behavior with respect to crank angle. The parameters mathematical model calculated were brake power, brake specific fuel consumption, fuel consumed, brake thermal efficiency, burn duration and exhaust gas temperature. Finally, in this study it is found that ethanol blend with gasoline increases brake power and brake thermal efficiency by lowering the exhaust gas temperature.

3. Thermodynamic Modeling of the Spark Ignition Engine

For the present study, a Zero-dimensional combustion model is employed. In this model the combustion chamber was divided into burned and unburned zones and separated by a flame front as shown in (Figure1) [1]. The first law of thermodynamics, equation of state, mass conservation ،mass fraction burned and heat transfer from the cylinder were applied to the burned and unburned zones. The pressure was assumed to be uniform throughout the cylinder charge .A system of first-order ordinary differential equations was obtained for the pressure, temperature, volume, work and heat transfer with respect to crank angle.

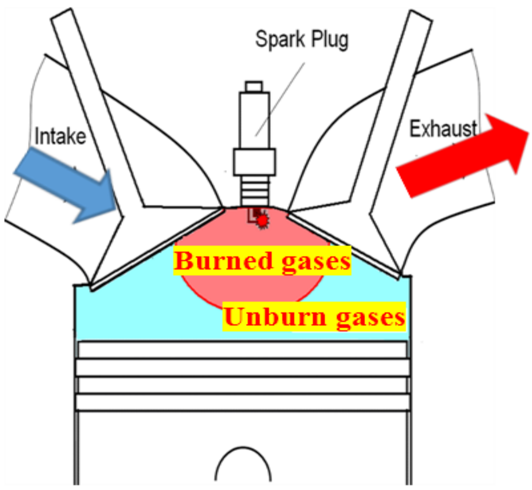


Figure1. Two-zone thermodynamic model of combustion

3.1 Mass and Energy Balance

The total mass is assumed to be constant, since valve leakage and blow-by are neglected [2]:

$$m = m_u + m_b \qquad (1)$$

The volume of the two zones is equal to the total cylinder volume, which is a function of the cylinder geometry and crank angle [1].

$$V = V_u + V_b \qquad (2)$$

$$V = \frac{V_d}{r-1} + \frac{V_d}{2}\left[\epsilon + 1 - cos\theta - (\epsilon^2 - sin^2\theta)^{1/2}\right] \qquad (3)$$

$$\epsilon = \frac{2L}{S}$$

The energy equations were written for each zone as follows [2]:

$$\frac{d(m_u u_u)}{d\theta} = -p\frac{dV_u}{d\theta} + \sum\nolimits_i \frac{dQ_{ui}}{d\theta} - h_u\frac{dm_u}{d\theta} \qquad (4)$$

$$\frac{d(m_b u_b)}{d\theta} = -p\frac{dV_b}{d\theta} + \sum\nolimits_i \frac{dQ_{bi}}{d\theta} - h_b\frac{dm_b}{d\theta} \qquad (5)$$

Where: $\sum$ is the summation of the heat transfer rates through the different engine's parts surfaces in contact with the cylinder gases. In each zone, assuming ideal gases and the same pressure, the equation of state gives [1]:

$$p_u V_u = m_u R_u T_u \qquad (6)$$

$$p_b V_b = m_b R_b T_b \qquad (7)$$

3.2 Fuel Burning Rate Model

Many experiments show that the burning rate depends mostly on the combustion chamber shape and the position of the spark plug. The mass fraction burned $\boldsymbol{x_b}$, is represented by the following finite heat release equation [2]:

$$x_b = 1 - exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^n\right] \qquad (8)$$

Where: θ = crank angle, θ_s = start of heat release, θ_d= duration of heat release
a = Weibe efficiency factor n = Weibe form factor

The parameters $\boldsymbol{a}$ and $\boldsymbol{n}$ are adjustable parameters used to fit experimental data. Values of $a = 5$ and $a = 3$ have been reported to fit well with experimental data [2].

At the beginning of the combustion, the burn fraction is zero and at the end of the combustion, it is almost one. The heat release the basis on the crank angle becomes:

$$\frac{\partial Q}{d\theta} = Q_{in}\frac{\partial x_b}{\partial\theta} \qquad (9)$$

With considering the Wiebe function the derivative of burn fraction is [2]:

$$\frac{\partial x_b}{\partial\theta} = \frac{(1 - x_b)\, n\, a}{\theta_d}\left(\frac{\theta - \theta_s}{\theta_d}\right)^{n-1} \qquad (10)$$

3.3 Air and Combustion Products Data

To calculate the thermodynamic properties for air and combustion products **Gordon and McBride** [3], proposed by the following expressions that are curve-fitted to the tabulated **JANAF** thermo-chemical tables are obtained as:

$$\frac{C_P}{R} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 \qquad (11)$$

$$\frac{h}{R\,T} = a_1 + \frac{a_2}{2}T + \frac{a_3}{3}T^2 + \frac{a_4}{4}T^3 + \frac{a_5}{5}T^4 + \frac{a_6}{T} \qquad (12)$$

$$\frac{s^{\circ}}{R} = a_1 LnT + a_2T + \frac{a_3}{2}T^2 + \frac{a_4}{3}T^3 + \frac{a_5}{4}T^4 + a_7 \qquad (13)$$

Where: $\boldsymbol{c_p}$ is the specific heat at constant pressure $\boldsymbol{h}$ is the specific enthalpy، $\mathbf{s}$ is the specific entropy. The coefficient constants $\boldsymbol{a_1}$ to $\boldsymbol{a_7}$ are calculated at different temperature [3].

3.4 Equivalence Ratio

When the modeling treatment with a single fuel, the equivalence ratio $(\boldsymbol{\varphi})$ is given by [1]:

$$\varphi = \frac{\left(\frac{F}{Air}\right)_{Act}}{\left(\frac{F}{Air}\right)_{Sto}} \qquad (14)$$

Where: subscript $(\boldsymbol{Act})$ denotes the actual and $(\boldsymbol{Sto})$ denotes to stoichiometric fuel/air ratios.

For blending of hydrocarbons or (alcohol) with gasoline fuel, the equivalence ratio changes to [6]:

$$\varphi = \frac{\left(\frac{[F]}{[Air] - [\text{Alcohol}]/\left(\frac{[\text{Alcohol}]}{Air}\right)_{Sto}}\right)}{\left(\frac{F}{A}\right)_{Sto}} \qquad (15)$$

3.5 Practical Chemical Equilibrium

In the atmospheric air composition assumption (**21%v Oxygen and 79%v Nitrogen**), the species including $\boldsymbol{O}$, $\boldsymbol{H}$, $\boldsymbol{OH}$ and $\boldsymbol{NO}$ are important due to dissociation. The combustion reaction develops [2]:

$$C_\alpha H_\beta O_y N_\delta + \frac{a_s}{\varphi}(0.21O_2 + 0.79N_2) \longrightarrow$$
$$y_1CO_2 + y_2H_2O + y_3N_2 + y_4O_2 + y_5CO + y_6H_2 + y_7H + y_8O + y_9OH + y_{10}N \qquad (16)$$

Where: $\boldsymbol{y_1}$ to $\boldsymbol{y_{10}}$ represent the products mole fractions and $\boldsymbol{a_s}$ the stoichiometric molar air-fuel ratio.

This model includes six gas phase equilibrium reactions which contain dissociation of hydrogen, oxygen, water, carbon dioxide and equilibrium $\boldsymbol{OH}$ and $\boldsymbol{NO}$ formation [2]:

$$\frac{1}{2}H_2 \rightleftharpoons H \qquad (17)$$

$$\frac{1}{2}O_2 \rightleftharpoons O \qquad (18)$$

$$\frac{1}{2}H_2 + \frac{1}{2}O_2 \rightleftharpoons OH \qquad (19)$$

$$\frac{1}{2}O_2 + \frac{1}{2}N_2 \rightleftharpoons NO \qquad (20)$$

$$H_2 + \frac{1}{2}O_2 \rightleftharpoons H_2O \qquad (21)$$

$$CO + \frac{1}{2}O_2 \rightleftharpoons CO_2 \qquad (22)$$

The calculations were based on the equilibrium assumption, except for $\boldsymbol{NO_x}$ formation, where the extended Zeldovich mechanism was used as [2]:

$$N + NO \rightleftharpoons N_2 + O \qquad K_1 = 3.1 * 10^{10} * exp\left(\frac{-160}{T}\right) \qquad (23)$$

$$N + O_2 \rightleftharpoons NO + O \qquad K_2 = 6.4 * 10^{6} * T * exp\left(\frac{-3125}{T}\right) \qquad (24)$$

$$N + OH \rightleftharpoons NO + H \qquad K_3 = 4.2 * 10^{10} \qquad (25)$$

Where: $\boldsymbol{K_1}$, $\boldsymbol{K_2}$ and $\boldsymbol{K_3}$ are the forward rate constants, taken from the model of Benson et al. (1975) [7].

3.6 Heat Transfer Model

Heat transfer into the modeling is expressed in terms of heat loss from the burned and unburned gas respectively as [2]:

$$\frac{dQ}{d\theta} = \frac{-\dot{Q}_1}{\omega} = \frac{-\dot{Q}_b - \dot{Q}_u}{\omega} \qquad (26)$$

To express the heat loss in terms of temperature requires the introduction of a heat transfer coefficient $\boldsymbol{h_c}$ [2]:

$$\dot{Q}_b = h_c A_b (T_b - T_w) \qquad (27)$$

$$\dot{Q}_u = h_c A_u (T_u - T_w) \qquad (28)$$

Where: $\boldsymbol{h_c}$ is convection heat transfer coefficient, and $\boldsymbol{A_b}$ and $\boldsymbol{A_u}$ are the areas of burned and unburned gases in contact with the cylinder walls as temperature $\boldsymbol{T_w}$.We have assumed, for convenience, that $\boldsymbol{h_u = h_b = h_c = constant}$. For the areas $\boldsymbol{A_b}$ and $\boldsymbol{A_u}$ let us suppose that the cylinder area $\boldsymbol{A_c}$ can be divided as follows [2]:

$$A_c = \frac{\pi b^2}{2} + \frac{4V}{b} \qquad (29)$$

$$A_b = A_c {x_b}^{1/2} \qquad (30)$$

$$A_u = A_c\left(1 - {x_b}^{1/2}\right) \qquad (31)$$

This simulation model has the convenience to adapt the heat transfer correlation proposed by Woschni [1].

$$\dot{Q} = h_c\, A_c\, (T_g - T_w) \qquad (32)$$

$$h_c = 3.26\, b^{-0.2} p^{0.8}\, T^{-0.55} w^{0.8} \qquad (33)$$

$$w = \left[C_1 U_p + \frac{C_2 (V_d\, T_r)}{p_r V_r}(p - p_m)\right] \qquad (34)$$

Where:-

$$\overline{U_p} = 2SN \qquad (35)$$

Where $\boldsymbol{C_1}$ and $\boldsymbol{C_2}$ are constants that varied depending on the combustion period, $\overline{\boldsymbol{U_p}}$ is the mean piston velocity, $\boldsymbol{T_r}$ isthe reference temperature, $\boldsymbol{p_r}$ is the reference pressure, $\boldsymbol{V_r}$ is the reference volume, $\boldsymbol{p_m}$ is the motored cylinder pressure, $\boldsymbol{S}$ is piston stroke and $\boldsymbol{N}$ is engine speed. The constants $\boldsymbol{C_1}$ and $\boldsymbol{C_2}$ are defined as [1]:

$\mathbf{C_1}$ =2.28 and **$\mathbf{C_2}$=0** $(-180 \le \theta \le \theta_s)$ during compression.

$\mathbf{C_1}$ =2.28 and **$\mathbf{C_2}$=3.24*$\mathbf{10^{-3}}$** (during combustion and expansion)

Watson and Janota suggested the motored cylinder pressure for modeling as a polytrophic process [1]:

$$p_m = p_r \left(\frac{V_r}{V}\right)^{\lambda} \qquad (36)$$

Where: $\mathbf{V}$ is the instantaneous cylinder volume and $\boldsymbol{\lambda}$ is the polytrophic constant.

3.7 Adiabatic Flame Temperature

The adiabatic flame temperature is the maximum temperature that the combustion of products will reach in the limiting case of no heat loss to the surroundings through the combustion development. The adiabatic flame temperature ranges its maximum value when complete combustion occurs with the theoretical value of air. Reminding the description of enthalpy, this can be stated as [1]:

$$H_{react}(T_i\,,p) = H_{prod}(T_{ad}\,,p) \qquad (37)$$

3.8 Principle Governing Equations

The mathematical modeling is used to predict the temperature ,pressure and work done from the first law of thermodynamics, the open system can be described as following equations which be used only for compression stroke, combustion stroke and expansion stroke [2 ,7]:

$$\frac{dp}{d\theta} = \left[\left(\frac{V_b}{m_b} - \frac{V_u}{m_u}\right)\frac{dm_b}{d\theta} + \frac{m_u R_u}{p}\frac{dT_u}{d\theta} + \frac{m_b R_b}{p}\frac{dT_b}{d\theta} - \frac{dV}{d\theta}\right] * \frac{p}{V} \qquad (38)$$

$$\frac{dT_u}{d\theta} = \frac{V_u}{m_u c_{pu}}\frac{dp}{d\theta} - \frac{1}{m_u c_{pu}}\frac{dQ_u}{d\theta} \qquad (39)$$

$$\frac{dT_b}{d\theta} = \frac{p}{m_b R_b}\frac{dV}{d\theta} - \left(\frac{R_b T_b}{p} - \frac{R_u T_u}{p}\right)\frac{dm_b}{d\theta} - \frac{R_u}{c_{pu}}\frac{V_u}{p}\frac{dp}{d\theta} + \frac{1}{p}\frac{R_u}{c_{pu}}\frac{dQ_u}{d\theta} + \frac{V}{p}\frac{dp}{d\theta} \qquad (40)$$

$$\frac{dW}{d\theta} = p\frac{dV}{d\theta} \qquad (41)$$

When the temperature is known the values of $\boldsymbol{R_u}\,,\boldsymbol{R_b}\,,\boldsymbol{c_{vu}}\,,\boldsymbol{c_{vb}},\boldsymbol{c_{pu}}$ and $\boldsymbol{c_{pb}}$ are calculated from the thermodynamic properties of burned and unburned mixtures [2, 8].

4. Master Program

The simulation of spark ignition engine was programmed using **MATLAB (GUI)** language (Figure 2). The computer program contains of a main part and number of MATLAB routines .The program predicts cylinder pressure, temperature of burned and unburned zones, heat release rate, accumulated heat release, heat transfer by radiation and convection, engine performance and concentration of pollutants emitted. The flow chart of the main program is shown in (Figure 3).

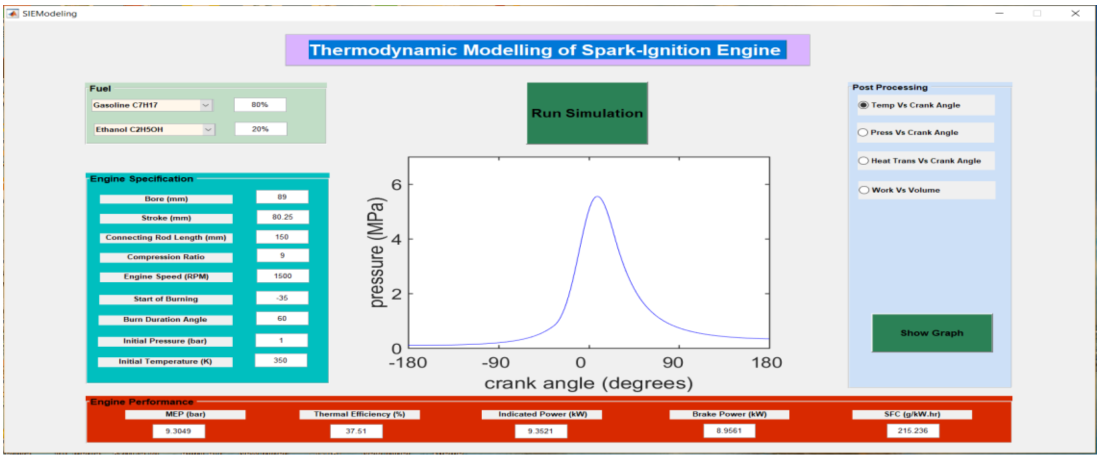


Figure 2. MATLAB interface of the SI engine simulation program.

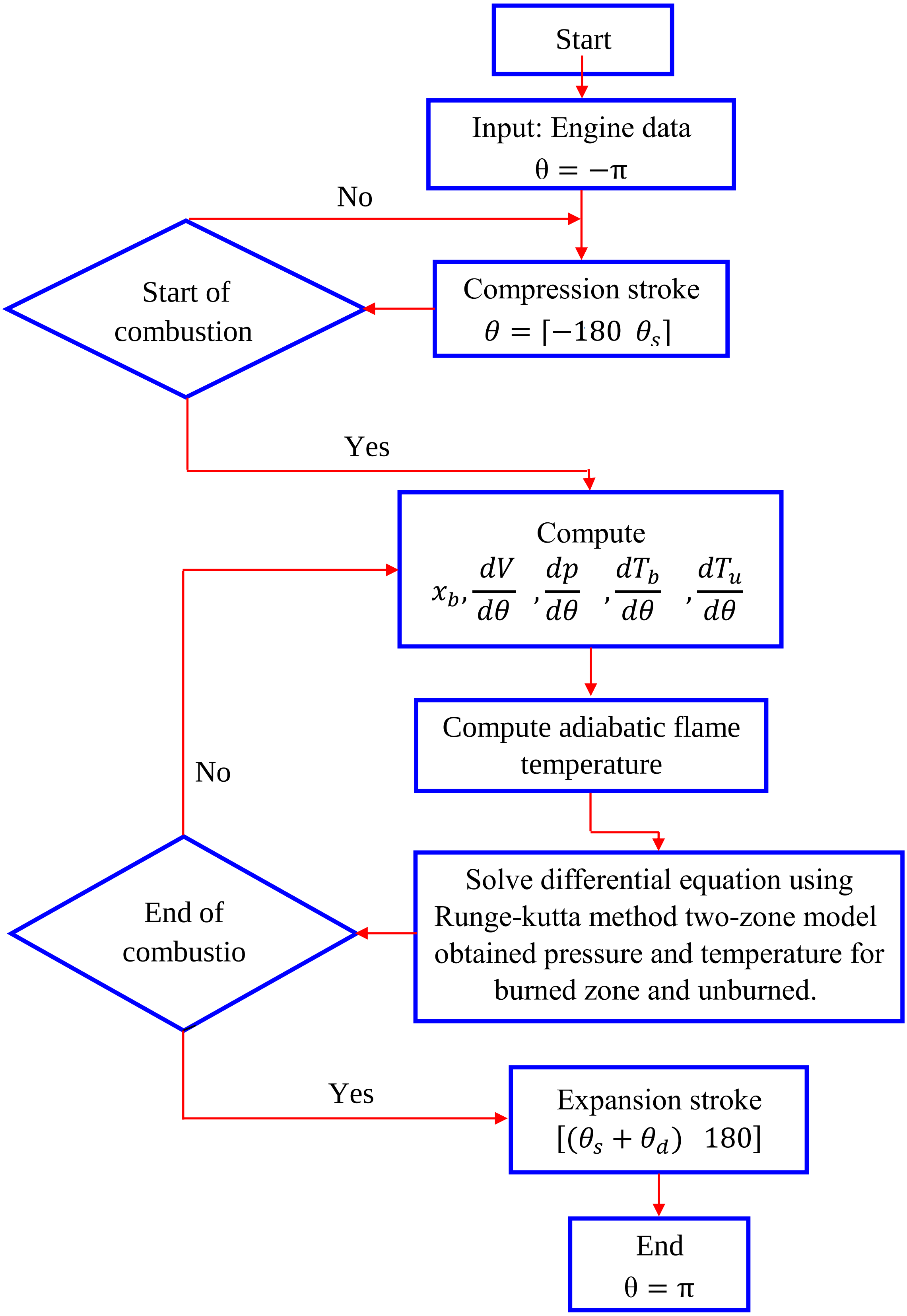


Figure 3. Solution procedure of the combustion of two zone (SI) engine model

5. Experimental Setup Description

For comparison and validation modeling, experimental data are used. The experimental setup consists of a Mercedes Benz model 200E four-cylinder, four stroke research engine has been used in this research (Figure 4). The technical details of spark ignition engine are given in table1.The engine torque has been measured using a hydraulic dynamometer. The fuel consumption of the spark ignition engine has been measured using a glass tube.The exhaust gas analyzer type (model 550 Korean) was used to analyze the emission of exhaust for spark ignition engine. The samples were prepared gasoline only and mixing gasoline with certain ratios of ethanol with various blended rates 5%, 10%, 15% and 20% by volume. The properties of gasoline and ethanol are given in table 2 and table 3in **(**Appendix).

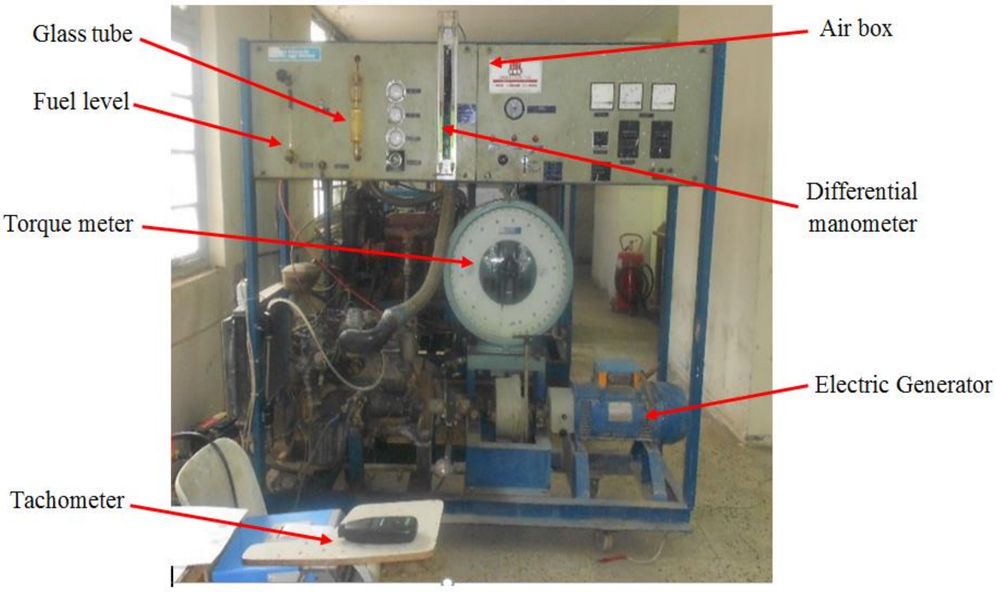


Figure 4 .The experimental setup of (S.I. engine)

Table1. Specification of test engine

Engine type:	Mercedes-Benz 200E
Number of strokes:	4
Number of cylinders:	4
Bore x Stroke:	89mm x 80.25 mm
Compression ratio:	9:1
Max. Power @ rpm:	80 kW (107.5 hp) @ 5500 rpm
Max. Torque @ rpm:	165 N.m (118 lb) @ 3000 rpm
Displacement:	1997cc 2L
Fuel System:	Carburettor
Cooling:	Water

6. Engine Performance

The relationships used to calculate engine performance are [8]:

6.1. Brake power

$$bp = \frac{2\pi * N * \tau b}{60 * 1000} \quad (42)$$

6.2. Fuel mass flow rate:

$$\dot{m}_f = \frac{V_F}{time} \times \rho_F \quad (43)$$

6.3. Brake specific fuel consumption:

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad (44)$$

6.4. Brake thermal efficiency:

$$\eta_{bth.} = \frac{bp}{\dot{m}_f * L.C.V} \quad (45)$$

7. MATLAB (GUI) Model Validation

After setting up the MATLAB script, the corresponding outputs are analyzed and compared with results from experimental work to determine their validity. This research begins by defining the model inputs that are used to simulate outputs for the **Mercedes-Benz 200E** engine. In addition to, the results are prepared which obtain from experimental work represented by engine performance for comparing with the results of the mathematical model. The experimental results are used for a comparison model such as brake efficiency $(\boldsymbol{\eta_b})$ and brake specific fuel consumption $(\boldsymbol{bsfc})$ at compression ratio 9 and constant engine speed 1500 rpm by using gasoline fuel and gasoline-ethanol blends that shown in figure (5) to figure (10) are measurable.

In comparing model predictions with experimental data, it is found that all brake specific fuel consumption predictions are accurate to within ± 3%, while all brake thermal efficiency predictions are accurate to within ± 4%. Differences between theoretical and experimental results, which obtain by mathematical model due to the error in measurement accuracy and real conditions of the engine, such as friction and engine life. The model is verified against published results of previously (SI) engine models and then used to analyze the performance and emission of spark ignition engine with gasoline and ethanol-gasoline mixture. After the model is validated, the mode is able to predict and simulate any various parameters in the combustion of spark ignition engine at any condition with range wide from engine speed and torque.

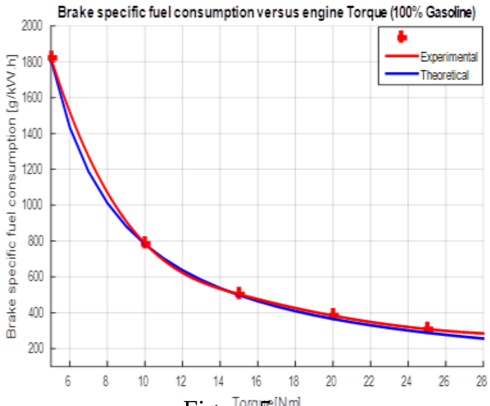


Figure 5

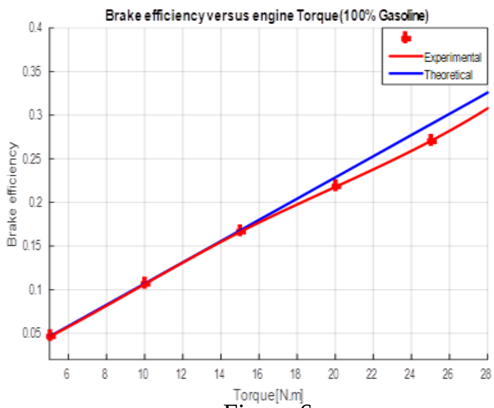


Figure 6

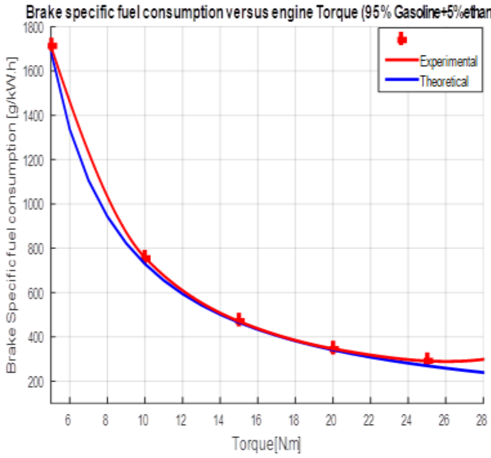


Figure 7

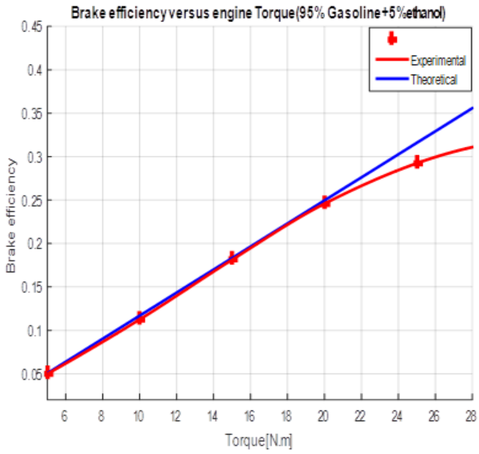


Figure 8

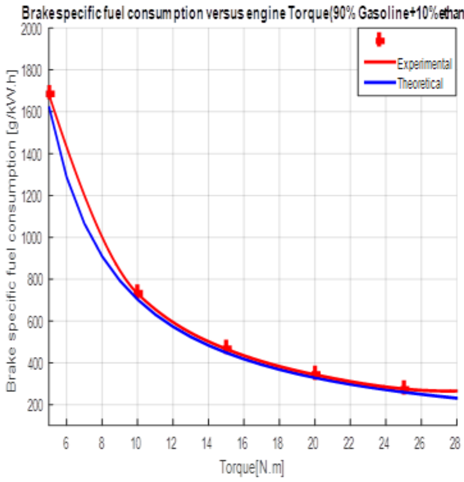


Figure 9

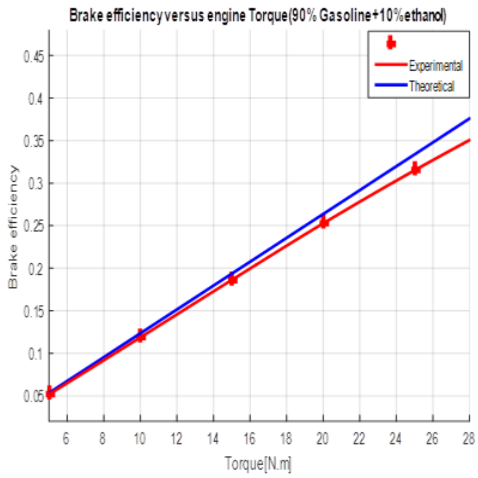


Figure 10

8. Results and Discussion

Figures (11),(12),(13) and (14) show cylinder pressure, cylinder temperature, work done and heat transfer as function of crank angle which is predicted by **MATLAB (GUI)** program inside combustion chamber for gasoline fuel at constant speed 1500 rpm compression ratio 9, spark timing is 35° before (TDC) and equivalence ratio ($\varphi = 0.8$) of spark ignition engine two-zone model. In two-zone model, the cylinder volume is divided into burned and unburned zones by an infinite small thin flame-front with a spherical shape and separation between the zones flam front. The results obtained by model are logical when comparing with published results[8]. The simulation is based on relation for energy and mass conservation, equation of state and mass fraction burned.

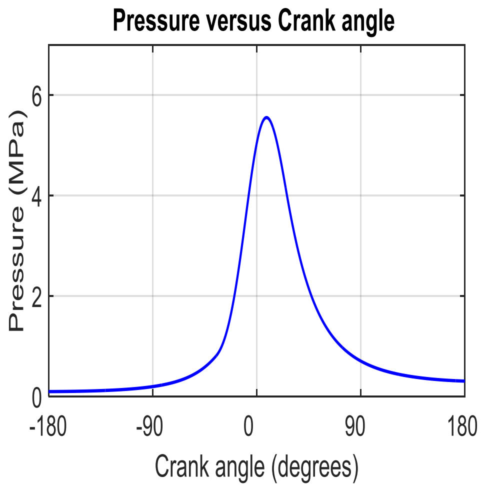


Figure 11

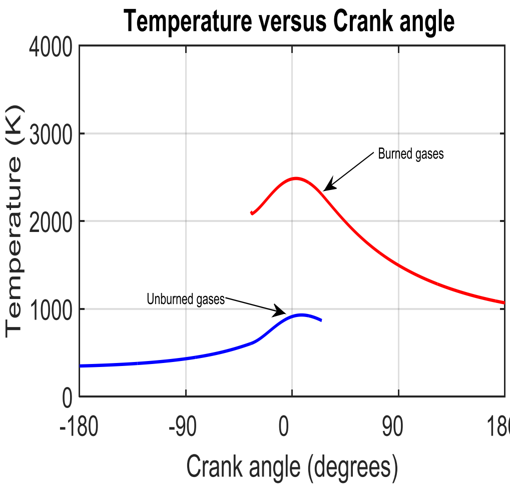
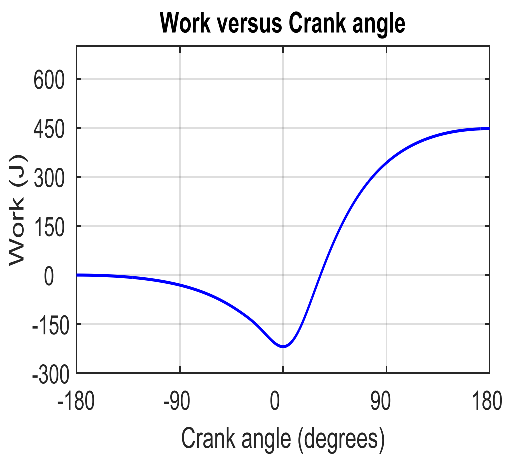


Figure 12



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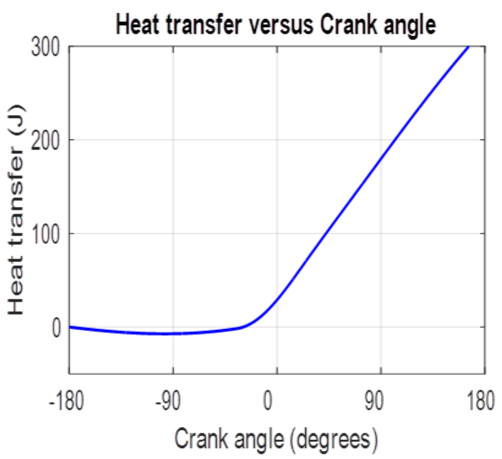
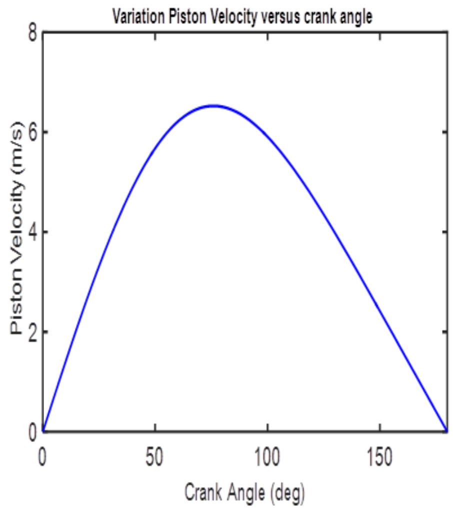


Figure 14

Figures(15),(16),(17) and (18) show the mean piston velocity, cylinder volume variation, heat transfer coefficient and mass fraction burned respect to crank angle which is predicted by **MATLAB (GUI)** program inside combustion chamber for gasoline fuel at same condition. It is found that the mass fraction burned is zero before the spark advance. After reaching the spark timing, the profile suddenly increased, before plateauing at one. In changing the spark timing and burn duration, the plot expanded and contracted, as expected and logical. Based on the provided information, it is decided that the mass fraction burned is working correctly. The cylinder volume is found less when piston moves from bottom dead center toward top dead center as expected. Also the mean piston velocity can be predicted at this condition, it reached to maximum velocity ($6.5\ m/s$) at crank angle ($76°$) .finally heat transfer coefficient increased at combustion stroke due to heat release and increased dramatically.



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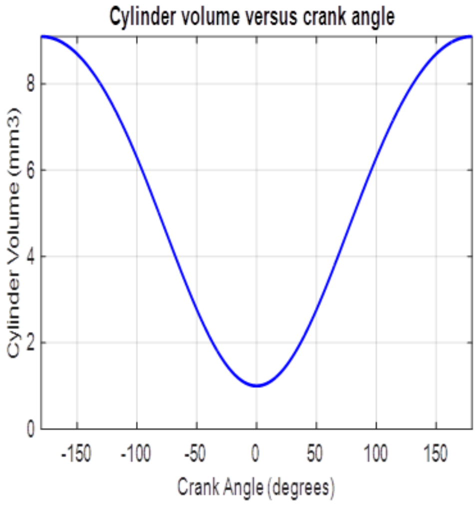

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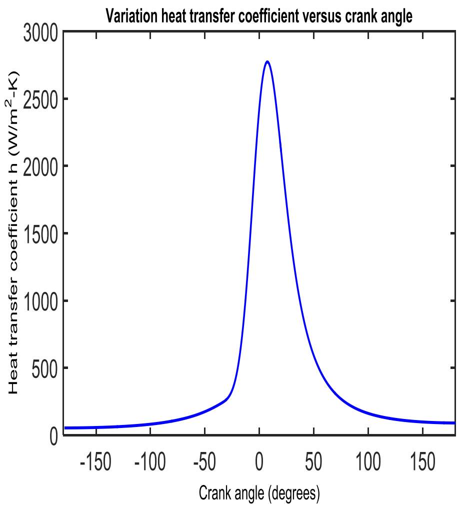

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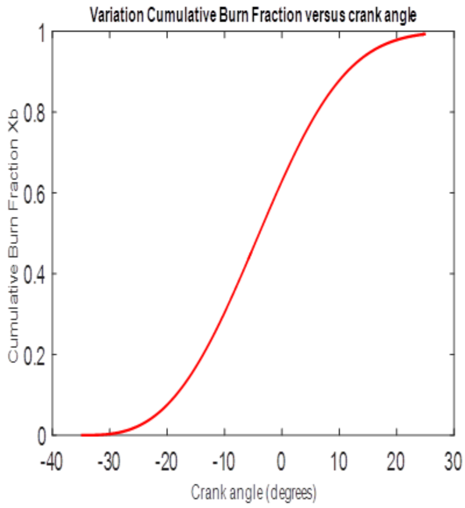
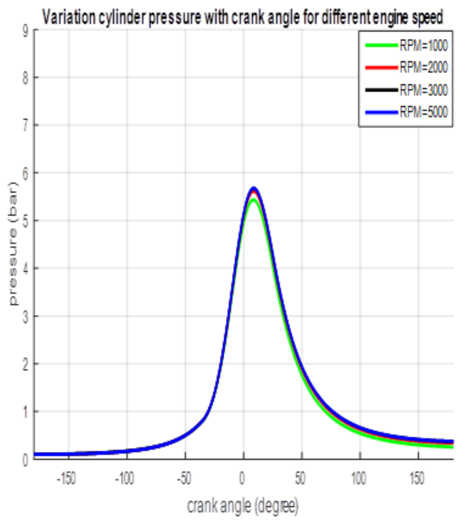


Figure 18

Figures (19),(20),(21) and (22) show the effect engine speed on cylinder Pressure, cylinder temperature, work done, and heat transfer which predicted by **MATLAB (GUI)** program inside combustion chamber for gasoline fuel at same condition. The cylinder Pressure, cylinder temperature and work done are increased when increasing engine speed up to (3000 rpm) due to shortened the heat transfer period from burn gases to cylinder wall and fastest burning mixture.

In addition to, acceleration of the engine speed would accelerate the turbulence inside the clearance volume and flame speed would result with shortening the combustion duration as a time [9]. Heat transfer is decreased due to shortened burn duration.



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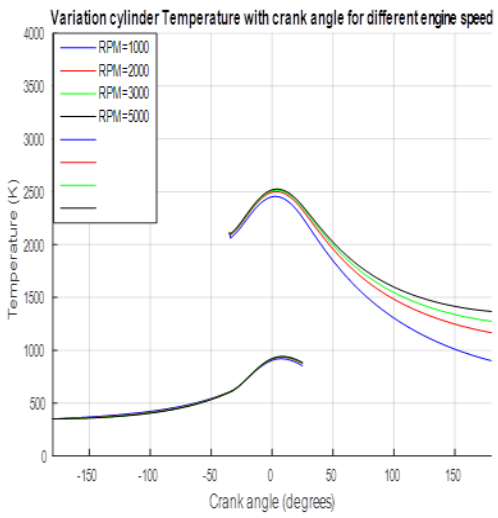

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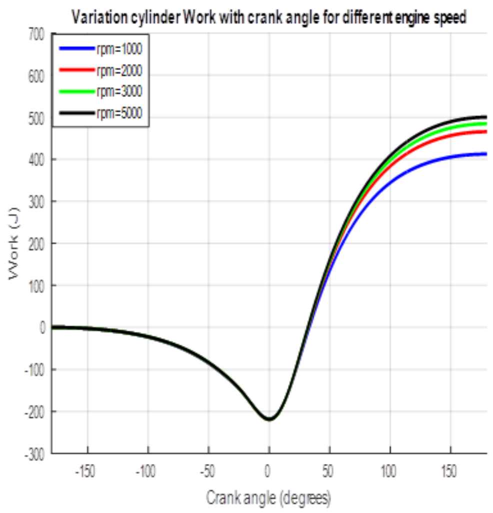


Figure 21

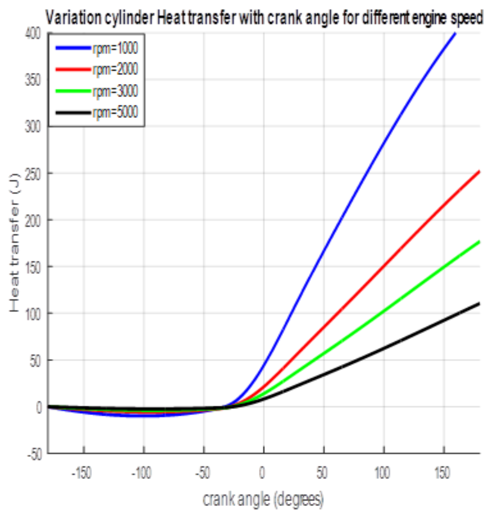
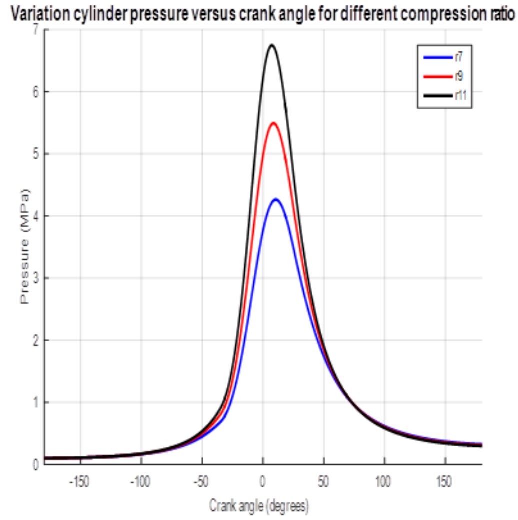


Figure 22

Figures (23), (24), (25), and (26) show the effect of variation of compression ratio on cylinder pressure, cylinder temperature, work done and heat transfer which is predicted by **MATLAB (GUI)** program inside combustion chamber for gasoline fuel at same condition. It is found that increasing engine compression ratio led to increase cylinder pressure , cylinder temperature and work done due to the piston compressed a large volume of mixture (fuel + air) to very small space that led to burn faster and produced high-pressure gases leading to raise the cylinder temperature . Work done increases because increasing gases pressure on piston. However, heat transfer decreased with increasing compression ratio due to shorten the combustion period. In addition, the results obtained by model are logical when compared with published results.



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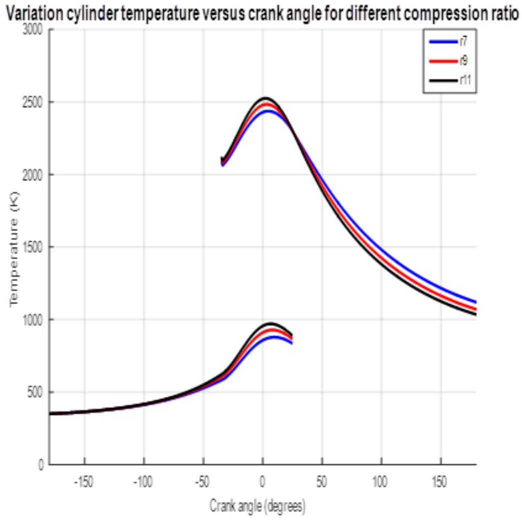

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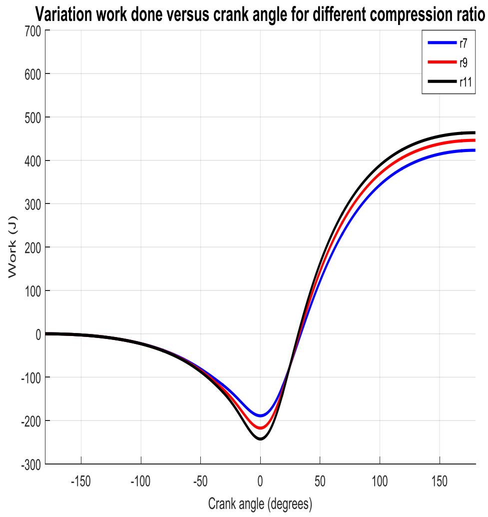


Figure 25

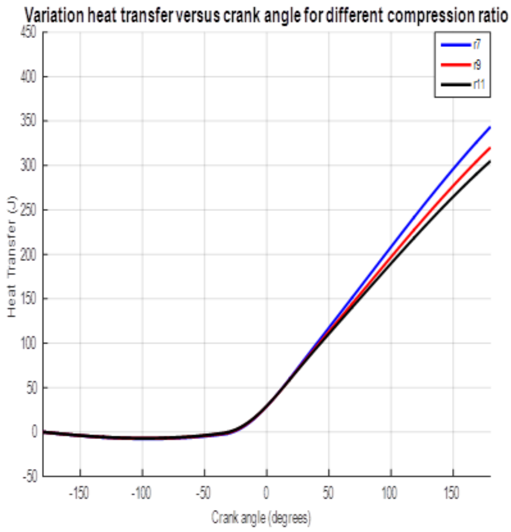


Figure 26

Figures (27), (28), (29) and (30), show the effect of variation of equivalence ratio on cylinder pressure, cylinder temperature, work done, and heat transfer as function of crank angle which predicted by **MATLAB (GUI)** program inside the combustion chamber for gasoline fuel at same condition. It is found that cylinder gas temperature and pressure values for combustion at ($\boldsymbol{\varphi = 1}$) mixtures are higher than($\boldsymbol{\varphi = 0.8}$, $\boldsymbol{\varphi = 1.2}$) due to lower combustion durations of stoichiometric mixture comparing with leaner and rich mixtures that led to increase flame speed. In addition to increasing cylinder temperature in stoichiometric mixture due to provide enough fuel to use up all of the oxygen in the cylinder that led to increase the amount of heat emitted inside the engine cylinder therefore increasing cylinder pressure, work done, and heat transfer.

Figures (31) and(32), show cylinder pressure and cylinder temperature as function of crank angle which predicted by **MATLAB (GUI)** program inside the combustion chamber for gasoline and ethanol fuels. Despite the low heating value of ethanol fuel compared to gasoline, we note a convergence in pressures and temperatures due to chemical composition of ethanol which characterized by found oxygen which helps oxidize hydrogen and carbon to improve and fasten combustion.

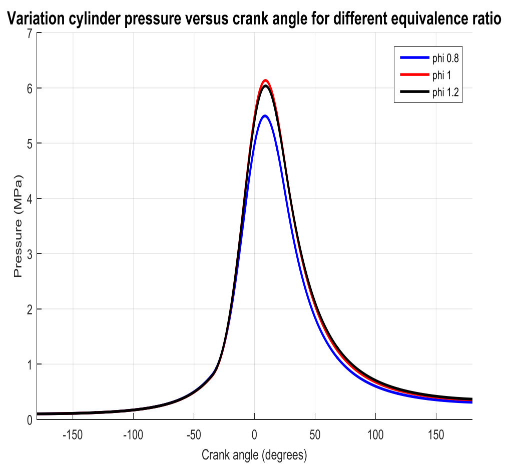
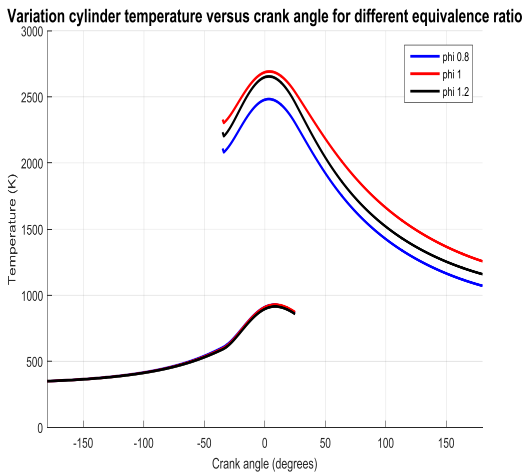


Figure 27



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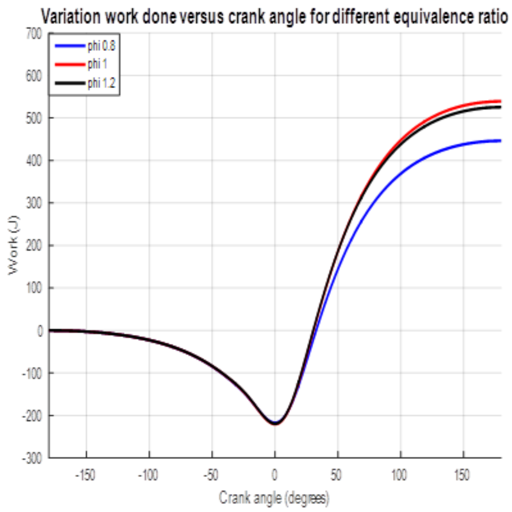

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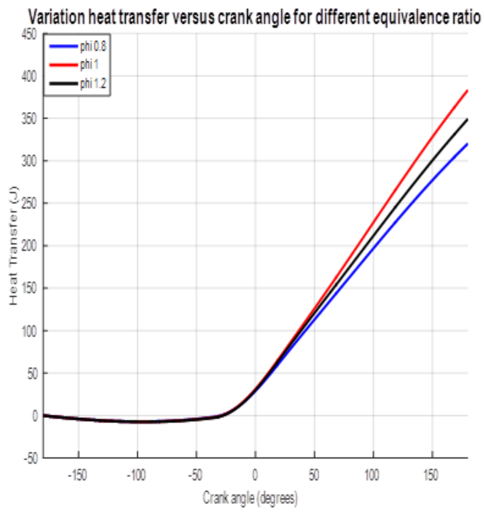

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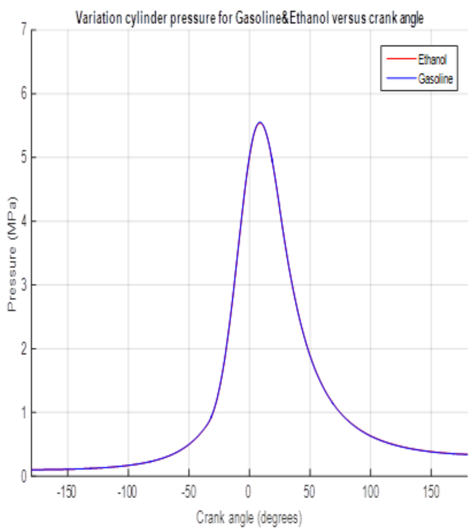

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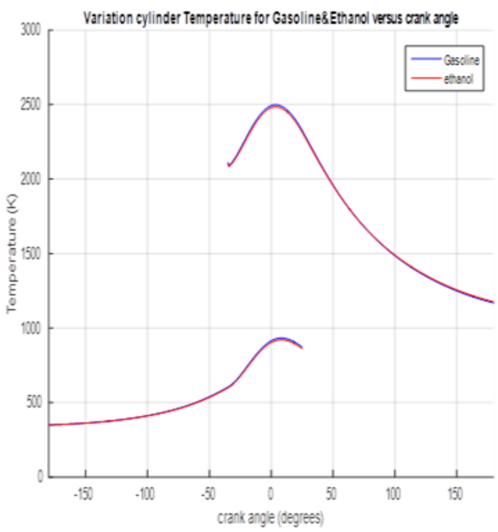
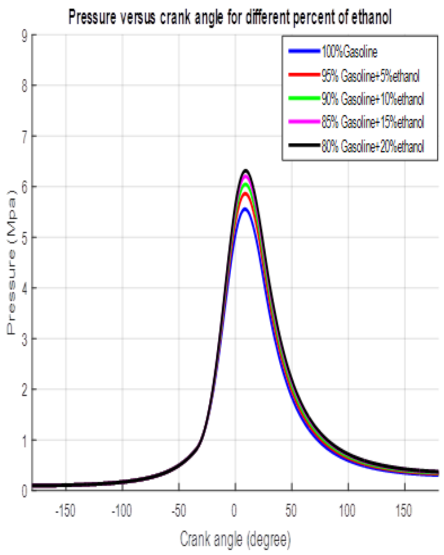


Figure 32

Figures (33), (34), (35) and (36) show the effect of addition ethanol on cylinder pressure, cylinder temperature, work done, and heat transfer respectively as function of crank angle for various blend of ethanol- gasoline fuel, which are predicted by **MATLAB (GUI)** program inside combustion chamber at same condition. The cylinder pressure, cylinder temperature and heat transfer increased return to chemical composition of ethanol which characterized by found oxygen which helps oxidize hydrogen and carbon to improve and fasten combustion. In additional to, the amount of inlet air and volumetric efficiency increased when increase of the percentage of ethanol in blended fuel due to higher heat of vaporization of ethanol led to cool in the end of induction process. The work done increased due to increase gases pressure on piston.



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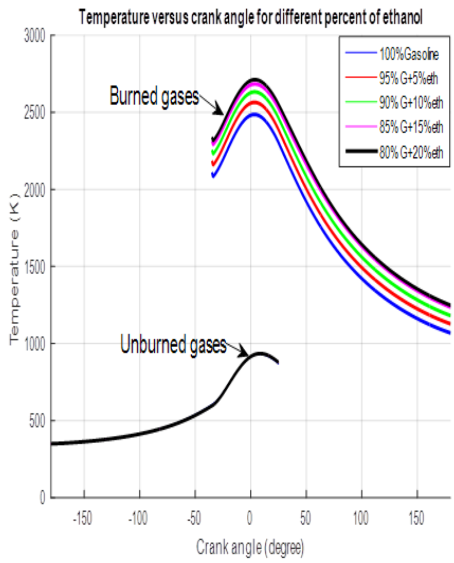

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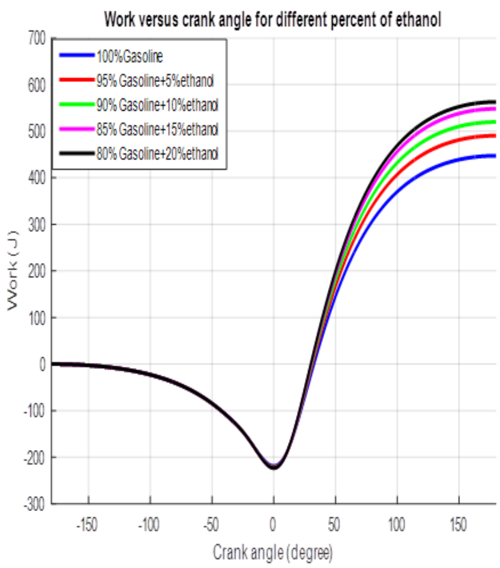

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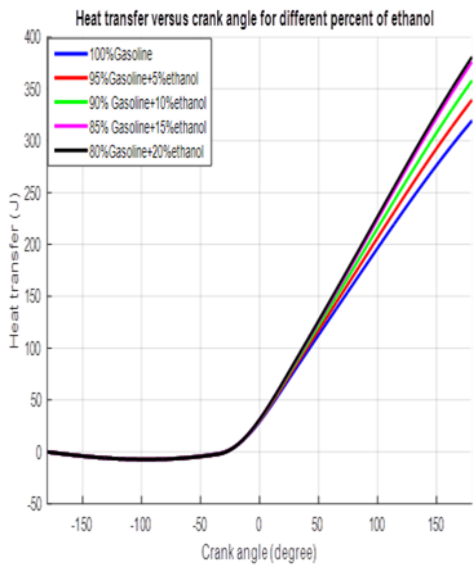
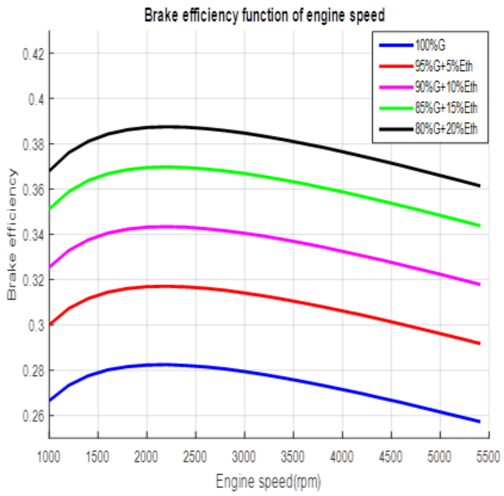


Figure 36

Figures (37), (38), (39) and (40) show the effect addition ethanol on engine performance as function of engine speed for various blend of ethanol- gasoline fuel, which was predicted by MATLAB (GUI) program inside combustion chamber at same engine condition [12 ,13]. The simulation is based on relation of engine performance with respect to engine speed .The thermal efficiency, brake efficiency increased up to (80% gasoline+20% ethanol) due to increase of the indicated mean effective pressure, cylinder pressure and volumetric efficiency. The brake specific fuel consumption increased due to higher heat of vaporization for ethanol or lower heating value for ethanol.



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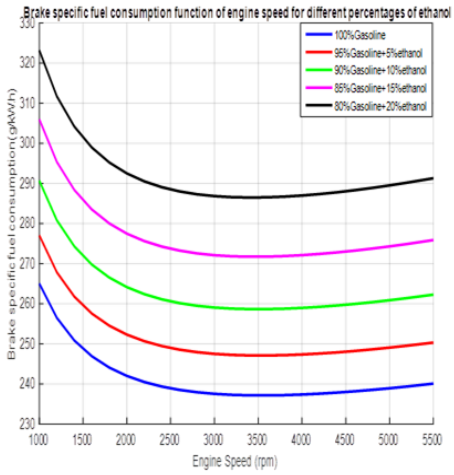

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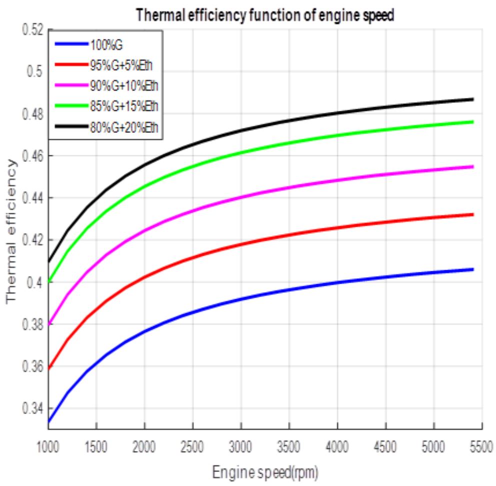


Figure 39

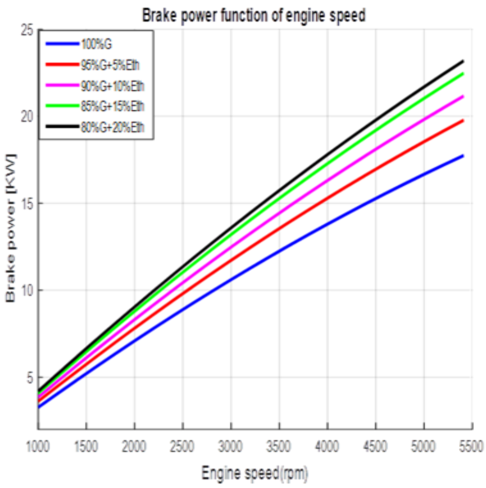
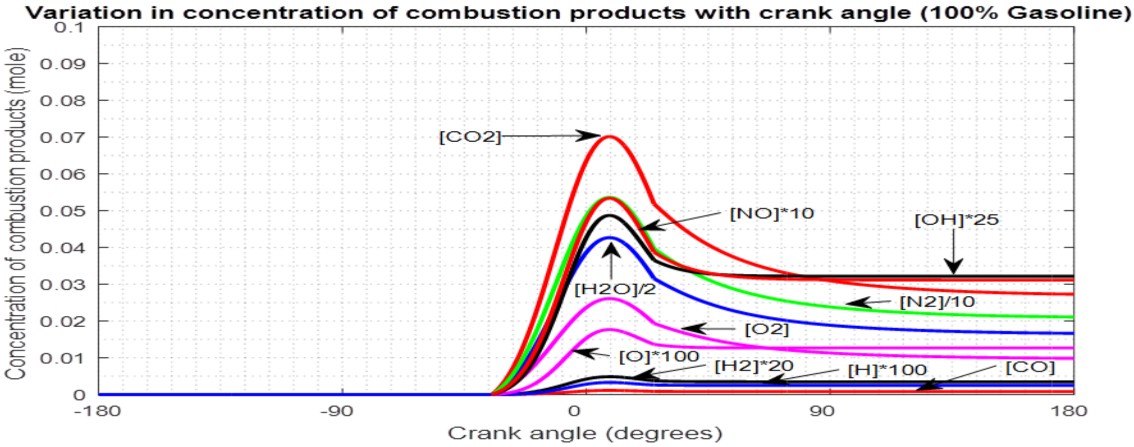


Figure 40

Figures (41), (42), (43) , (44) and (45) show the effect of addition of ethanol on variation of combustion products as function of crank angle which predicted by **MATLAB (GUI)** program inside combustion chamber at same engine condition for various blend percentage of ethanol-gasoline fuels. The simulation is based on the relation of the combustion reaction at added ethanol and equilibrium combustion products with respect to crank angle at low temperature and high temperature, which led to dissociation to occur. It was noted that adding ethanol to gasoline improved the combustion of spark ignition engine. The $\mathbf{CO_2}$ emission increases at added percent of ethanol (5%, 10%, 15%, 20%) because the improvement of combustion. The **NO** emission decreased dramatically due to higher heat of vaporization of ethanol that reduced the peak temperature inside the engine cylinder. The **CO** emission increased at this condition due to increase fuel-air ratio led to reduce $\mathbf{O_2}$ in combustion chamber. In addition to, the emissions $\mathbf{H_2O}$, $\mathbf{N_2}$, $\mathbf{H}$, $\mathbf{H_2}$, $\mathbf{O}$ and $\mathbf{OH}$ the increase and decrease have no effect on engine performance and human health. Finally, the emissions $\mathbf{H}$, $\mathbf{H_2}$, $\mathbf{O}$ and $\mathbf{OH}$ are changed when adding ethanol for different percentages dependent on cylinder gas temperature.



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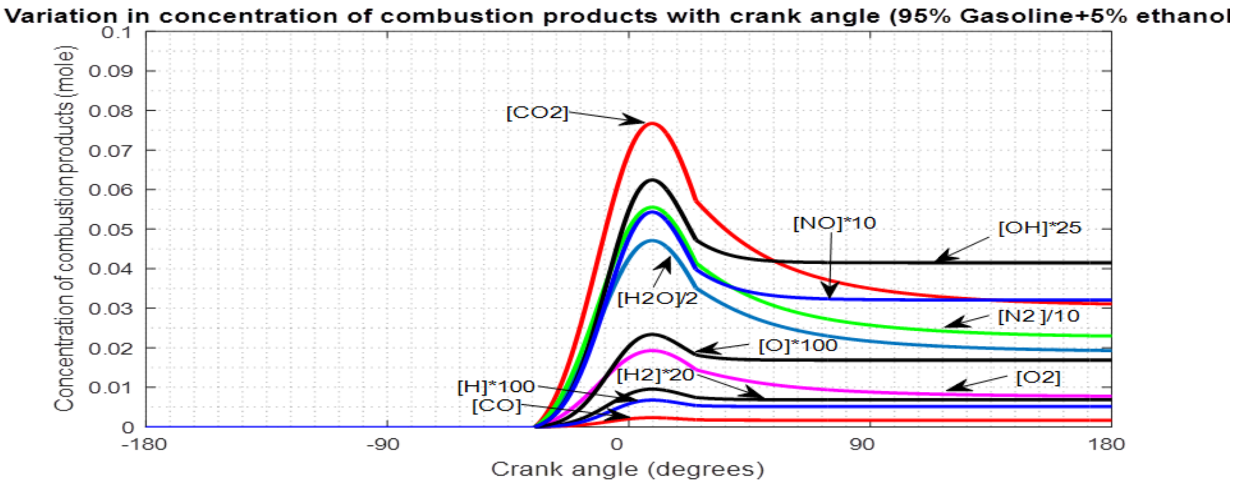

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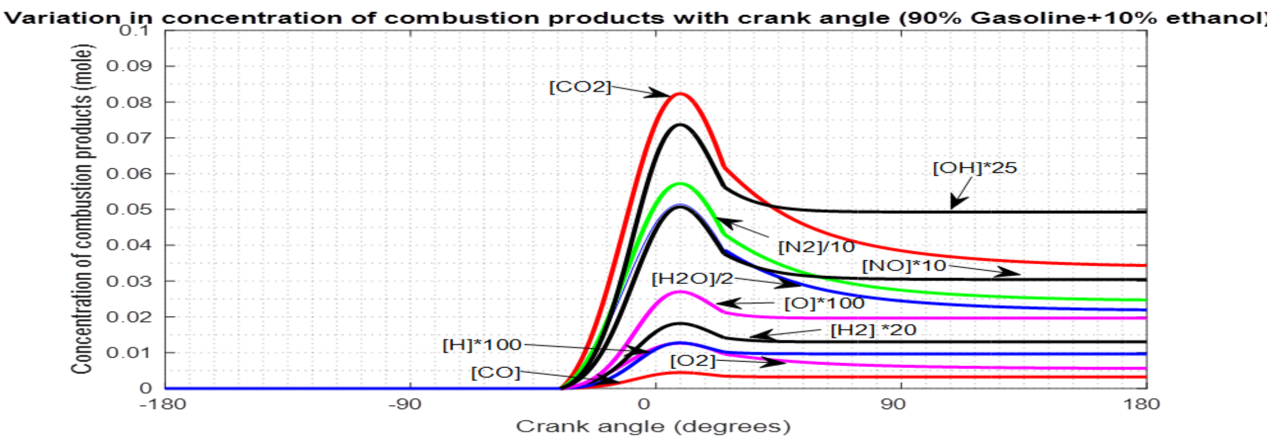

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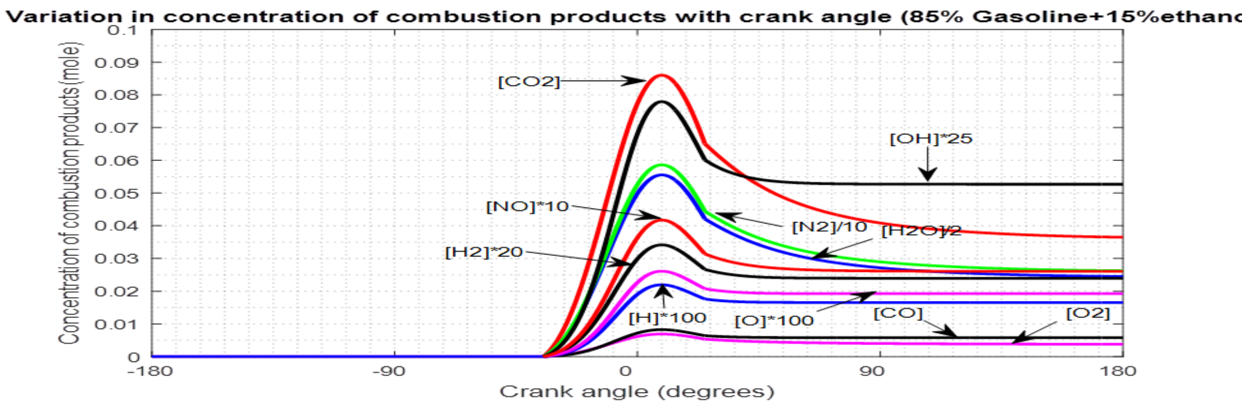

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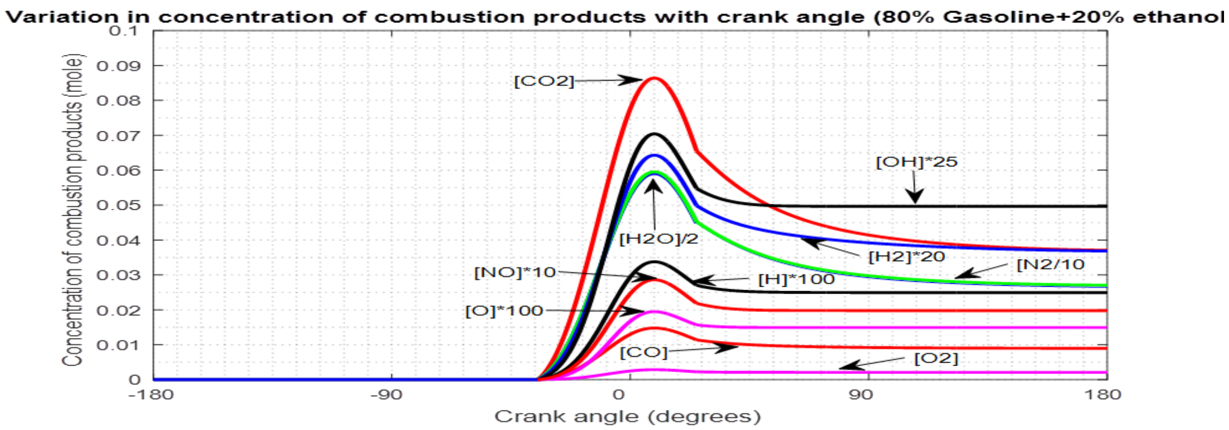

Figure 45

Figure (46) shows equilibrium and rate-limited $\mathbf{NO_x}$ concentrations as function of crank angle for various mass burn which predicted by **MATLAB (GUI)** program inside combustion chamber at same engine condition. There are large differences between the nitric oxide concentrations in the first mass to burn $(x = 0)$and last mass to burn$(x = 0)$. Furthermore, it can be seen that when the temperatures drop to about $\mathbf{2000\ K}$, the decomposition rate becomes very slow, and for practical purposes, it may be said that the nitric oxides freeze at a concentration greater than the equilibrium values.

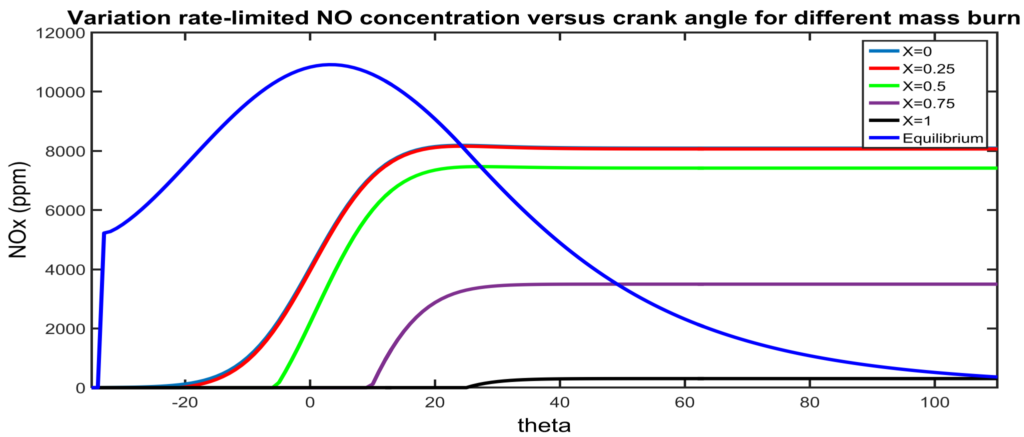

Figure 46

9. Conclusions

A mathematical model achieves its goal by being a simple, fast and accurate engine simulation model. The mathematical model can be used to help in the design of a spark ignition engine for alternative fuels as well as to study various problems such as pollutant emissions, engine performance, pre-ignition, knocking and misdistribution of the fuel-air mixture. Also, many other parameters can be studied using this mathematical model such as the effect of combustion duration for each fuel on the performance and emission of the engine, the best amount of fuel supplement, and high suitable compression ratio for each fuel. The model can predict and analyze the engine thermodynamic characteristic; engine exhausts emission and performance parameters. The results of the model had a good correspondence with the experimental data. Due to its simplicity and computational efficiency, the model can also be used as a preliminary test on a wide range of alternate hydrocarbon fuels. From the mathematical model can also be conclusion ethanol gasoline blend increases the cylinder pressure and temperature up to a blending ratio of 20%. From the mathematical model can also be conclusion ethanol gasoline blends, the engine performance, the power output is increased and the brake specific fuel consumption increased up to a blending ratio (80% gasoline+20% ethanol). Also increasing in compression ratio leads to an increase in cylinder pressure, work and cylinder temperature. Generally, the addition of the ethanol shows an increase in the $\mathbf{CO}$, $\mathbf{CO_2}$ emissions and a decrease in the $\mathbf{NO}$ emissions according to the engine condition.

10. Nomenclature

S	Cylinder stroke (m)
Q	Heat input (J)
b	bore (m)
p	Pressure (bar)
V	Volume (m^3)
U	Internal energy (kJ)
$\dot{m}_1$	Mass flow rate (kg/s)
h_1	Specific enthalpy (kJ/kg)
ω	Engine speed (rad/sec)
m	Mass (kg)
V_d	Displacement volume (m^3)
r	Compression ratio
c_p	Specific heat at constant pressure (kJ/kg K)
c_v	Specific heat at constant volume (kJ/kg K)
R	Universal gas constant. (kJ/kg K)
φ	Equivalence ratio
L	Connecting rod length (m)
m_{IVC}	Mass entering of the mixture when inlet valve closing (kg)
c_{pu}	Unburned specific heat at constant pressure (kJ/kg K)
c_{pb}	Burned specific heat at constant pressure (kJ/kg K)
C_b	The blow by coefficient (sec^{-1})
T_b	Burned gas Temperature (K)
T_u	Unburned gas Temperature (K)
v_b	Burned specific volume m^3/kg
v_u	Unburned specific volume m^3/kg
τb	Torque of engine (N.m).
N	Rotational Speed (rpm).
V_F	Volume of Fuel Consumption (m^3).
ρ_F	Volume of Fuel Consumption (kg/m^3).
h_u	Specific enthalpy of unburned gases (kJ/kg).
h_b	Specific enthalpy of burned gases (kJ/kg).
θ_{IVC}	Angle of the inlet valve closing (degrees).
θ_{IVO}	Angle of the inlet valve opening (degrees)

11. References

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12. Appendix

Table 2. Gasoline C_7H_{17} Proprieties

Density@ 20	721.10 kg/m^3
Octane NO	88.8
Heating value (LHV)	43000 kJ/kg

Table 3. Gasoline C_2H_5OH Proprieties

Density@ 20	789 kg/m^3 kg/m^3
Octane NO	102
Heating value (LHV)	28000 kJ/kg