Theoretical Investigation Over the Effects of EGR, Fuel Reforming and Compression Ratio on the HCCI Engine Operation Fuelled with Ethanol

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THEORETICAL INVESTIGATION OVER THE EFFECTS OF EGR, FUEL REFORMING AND COMPRESSION RATIO ON THE HCCI ENGINE OPERATION FUELLED WITH ETHANOL

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Alireza Rahbari [α], Ahmad Afsari [σ], Bamdad Barari [ρ] & Iman Rostamsowlat [ω]

Abstract - HCCI engines have great potentials in high efficiency with ultra-low NO and particulates emissions and high flexibility in using various fuels. They operate on the basis of auto-ignition and therefore the fuel chemical kinetic is the main ignition controller. Thus initiation of combustion is a determining parameter in the prediction of combustion behavior. The main focus of the present study is to comprehensively investigate the impact of exhaust gas recirculation, fuel reforming and compression ratio on the combustion behavior of HCCI engines using ethanol as fuel. Results show that the increase of EGR delays the ignition timing, slows down the combustion reaction rate and reduces the temperature and pressure in cylinder. Also associating with the effect of exhaust gas fuel reforming, it is shown that hydrogen in the form of reformed gas helps in lowering the intake temperature required for stable HCCI operation. Also, the addition of hydrogen advances the start of combustion in the cylinder. Consequently, it is elucidated that increasing the compression ratio results in advancing the ignition timing, decreasing the burn duration and increasing the temperature and pressure. It is worth noting that the obtained results from the present model have the great compatibility with experimental data.

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I. Introduction

Homogenous charge compression ignition (HCCI) engines are being actively developed worldwide as they can have efficiencies close to that of diesel engines, with low levels of oxides of nitrogen (NOx) and particulate matter (PM) emissions.[1] In addition, HCCI engines have been shown to operate with a range of fuels including ethanol.[2] Research on alternative renewable fuels has become very important worldwide due to concerns about the effects of fossil fuel usage on global warming.[3] Ethanol is a renewable fuel when it is produced from non-petroleum based products.[4] Megaritis *et al.*,[5] investigated the effect of water blending on bioethanol HCCI combustion with forced induction and residual gas trapping. Rahbari and Mozafari,[6] used two step reaction mechanisms in order to model HCCI engines using ethanol as a fuel and also the performance prediction and in-cylinder heat transfer were calculated.

Controlling the combustion phenomenon in HCCI engines is one of the most challenging and demanding issues and the important factors in controlling the combustion phenomenon of HCCI engine are the ignition timing and operating range which is limited to low load because of knocking and misfiring. Several control techniques have been proposed by researchers to control the combustion phase of HCCI engines such as inlet air heating, supercharging, equivalence ratio, EGR, fuel reforming and compression ratio. These techniques depend on engine or fuel types and operating conditions. Associating with these controlling strategies, Martinez-Frias *et al.*,[7] elaborated the ignition timing control system that involved varying the intake temperature of an HCCI engine. Their control system consisted of an exhaust-fresh charge heat exchanger, a supercharger, an EGR addition valve, and an intercooler. Chen and Milovanovic,[8] studied the effect of EGR on the combustion of HCCI engine fuelled with methane. It was found that the thermal energy contained in the EGR is essential for the combustion initiation.

Then, Dec,[9] used Chemkin to explore the bulk generation of CO and HC at low loads and to identify the physical mechanisms by which EGR influenced the energy release rate. Yap *et al.*,[10] investigated the application of bioethanol HCCI operation on a gasoline type engine, in conjunction with residual gas trapping. Then he explored two different approaches to achieve HCCI with bioethanol namely, trapping of internal residual gas and intake temperature heating with a high compression ratio.[11] Kim and Lee,[12] recently investigated partial HCCI combustion as a control mechanism for HCCI combustion. Cooled EGR was introduced for the suppression of advanced auto-ignition of the premixed fuel. The results showed that a simultaneous decrease of NOx and soot was obtained by increasing the premixed ratio. Dubreuil *et al.*,[13]

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studied the effect of the addition of variable initial NO concentration. A detailed kinetic model was performed to rationalize the results. Experiments were also performed using a HCCI engine to characterize the effect of EGR rates with NO addition.

Megaritis et al.,[14] investigated the Effect of inlet valve timing and water blending on bioethanol HCCI combustion with forced induction and residual gas trapping which resulted in a greatly extended engine load range compared to normal aspiration operation.[15] In the recent study, Rahbari,[16] studied the effect of EGR on HCCI combustion characteristics such as ignition timing, burn duration, temperature, pressure and NOx emission containing ethanol as fuel. In addition, Jothi et al.,[17] experimentally investigate the effect of EGR on a 100% Liquefied Petroleum Gas (i.e., LPG) fueled HCCI engine throughout the load spectrum.

Fuel reforming refers to the dissociation of the hydrocarbon fuel into CO and H_2 and the effect of this phenomenon is less studied by the researchers.[18] The addition of hydrogen to the main hydrocarbon fuel is reported to be beneficial in terms of brake power, thermal efficiency and reduction of HC, CO, CO_2 and particulates in.[19] Another important parameter is the use of different engine exhaust gas temperatures and compositions due to different engine conditions requires adjustments of the reactant flows in the reforming process, so that both maximum hydrogen production and process efficiency can be achieved.[20]

Hosseini and Checkel,[21] performed a series of experimental studies using reformer gas blending to control ignition timing with three base fuels: n-Heptane, isooctane, and natural gas. It was found that, reformer gas blending could provide combustion timing control despite the wide range of reformer gas composition. Rahbari et al.,[22] studied the role of fuel reforming on the combustion characteristics of the HCCI engines fuelled with ethanol. Tsolakis et al.,[23] documented the application of exhaust gas fuel reforming of two alternative fuels, biodiesel and bioethanol. The benefits of exhaust gas fuel reforming were demonstrated by adding simulated reformed gas to HCCI engine fuelled by bioethanol.

Another key parameter in controlling the HCCI engines is the role of compression ratio. In relation to the importance of the controlling factors, the comprehensive investigation is recently done by Machrafi and Cavadiasa,[24] over the effects of the inlet temperature, the equivalence ratio and the compression ratio on the pressure, the heat release and the ignition delays. These three parameters appeared to decrease the ignition delays, with the inlet temperature having the least influence and the compression ratio the most.

In this research after following the comprehensive study on the HCCI engines fuelled with ethanol with the emphasis on the role of EGR, fuel reforming and compression ratio on the combustion characteristics. Therefore, this study is followed by a computational model using a mechanism containing ethanol reactions and the governing equations consist of conservation of mass, conservation of energy, conservation of species and heat transfer phenomena are simultaneously solved. This theoretical model declares the effects of these parameters on the in-cylinder pressure and temperature, ignition timing and burn duration. The result obtained from the model has been compared with the experimental data published in the literature and the comparison showed a reasonable compatibility.

## II. CHEMICAL KINETICS OF AUTO-IGNITION

In general, the chemical reaction of fuel and air is initiated by:[25]

$$RH + O_2 \rightarrow R + HO_2 \qquad (1)$$

Auto-ignition of hydrocarbons occurs when chain branching processes become significant. At lower temperatures, reaction (2) produces chain branching precursors which can be decomposed by the temperature above 800K.[26]

$$R + O_2 \rightarrow RO_2 \qquad (2)$$

A highly exothermic cycle is initiated by (2) to produce H_2O and an alkylperoxide. The low temperature reactions are terminated when the reaction (3) becomes competitive as temperature rises.

$$R + O_2 \rightarrow Olefin + HO \qquad (3)$$
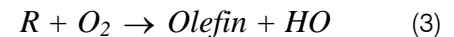
At intermediate temperature 900-1100K, chain branching occurs by:

$$RH + HO_2 \rightarrow H_2O_2 + R \qquad (4)$$

$$H_2O_2 + M \rightarrow OH + OH + M \qquad (5)$$
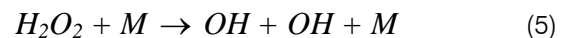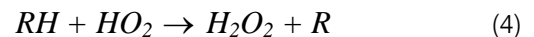
When temperature is reaching up to 1000K, the slightly exothermic reaction (4) is more dominant, thus the temperature and the concentration of H_2O_2 increases steadily. When temperature reaches 1000-1050K, reaction (5) catches up and the concentration of OH increases rapidly. Most OH will react with the fuel molecules, producing water and heat, thus increasing the temperature of the mixture. When the temperature is greater than 1100K, the high temperature chain branching reaction (6) dominates.

$$H + O_2 \rightarrow O + OH \qquad (6)$$
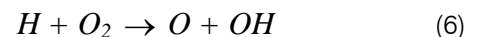
In this stage, OROOH and Olefin decompose to HO and CO. Finally, H_2O is formed by:

$$OH + HO_2 \rightarrow O_2 + H_2O \qquad (7)$$
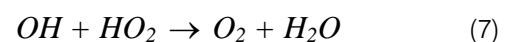
And CO is converted to CO_2 through:

$$CO + OH \rightarrow CO_2 + H \qquad (8)$$
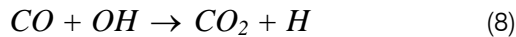
Where (7) and (8) are highly exothermic and release most of the energy.[27] In an HCCI engine, auto-ignition first occurs at local in-homogeneities which overcome their threshold energies to the chain branching reactions faster due to specific heat differences. This is very similar to the knocking phenomenon in SI engines.

It is recognized in all studies that HCCI combustion can't be initiated at atmospheric temperature, i.e. preheating of the engine mixture is necessary in addition to the temperature rise from compression. The energy released will quickly bring the remainder of the mixture to ignite and gives a very rapid heat release rate.

III. No Formation

The thermal NO route is the major constituent to the NO_x emission and can be described by three elementary reactions, called the extended Zeldovich mechanism:

$$O + N_2 \rightarrow NO + N \qquad (9)$$

$$N + O_2 \rightarrow NO + O \qquad (10)$$

$$N + OH \rightarrow NO + H \qquad (11)$$
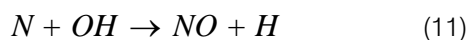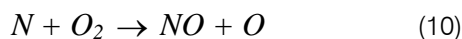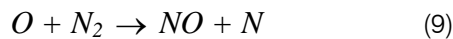
Reaction (9) has very high activation energy. The triple bond of the N_2 molecule is strong, thus the reaction rate is slow unless the temperature is high.

IV. Effect of EGR

EGR is one of the most important HCCI combustion control mechanisms affecting the auto-ignition timing. In this section, the effects of internal EGR are considered as HCCI combustion controller. EGR is defined as:

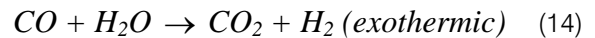$$EGR = \frac{m_e}{m_e + m_f + m_i} \qquad (12)$$

Where m_e is the mass of exhaust gas in one cycle, m_i and m_f are the mass of intake air and fuel respectively in the same cycle. At different engine operating conditions, the amount of EGR was regulated through the EVC timing, which is Exhaust Valve Close timing. The mass of exhaust gas in every cycle was calculated by using the ideal gas law, and the mass of the intake air was measured by the mass flow meter, also the fuel mass was calculated in combined with the intake air mass and Lambda which was get by the oxygen sensor.

V. Fuel Reforming

Fuel reforming can be used to dissociate hydrocarbons and alcohols into CO and H_2.[28] Ethanol, the fuel used in this study, can be reformed into 2 moles of CO and 3 moles of H_2 through the reaction:

$$C_2H_5OH + \tfrac{1}{2} O_2 \rightarrow 2CO + 3H_2O \text{ (endothermic)} \qquad (13)$$
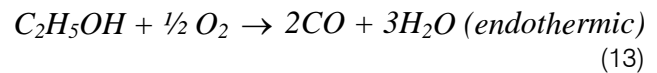
Reaction (13) can be further converted to more H_2 and CO_2 through:

$$CO + H_2O \rightarrow CO_2 + H_2 \text{ (exothermic)} \qquad (14)$$

Hydrogen has the advantages of relatively wide flammability limits and a higher burning velocity than most hydrocarbons, thus it is often studied as an additive to extend the operating limit of engines.[29] Although more hydrogen means a higher heating value of the fuel, further conversion of CO into extra H_2 is found to be unfavorable, because the formation of extra CO_2 will increase the heat capacity and decrease the combustion promotion by hydrogen.

The study by Shudo and Ono,[30] found that CO and H_2 in dimethyl ether (DME) are able to increase IMEP and the indicated efficiency. Also, CO and H_2 have lower cetane number and less low temperature reactions compared to DME, therefore decreasing the ratio of DME to CO and H_2 can be used to retard the ignition timing by reducing the low temperature heat release.[31] When ignition is retarded using H_2, A comparison between the effects of H_2 and CO found that both are equally effective in increasing IMEP, but H_2 is more effective in retarding the ignition timing. In addition, CO is more effective in lowering the unburned HC emissions than H_2 at higher equivalence ratios, because ignition with CO addition is less retarded, so the peak temperature occurs closer to TDC and therefore is higher. In addition, the use of reformed gas to complement the fuel helps to avoid the need for dual fuel storage onboard.

Fuel reforming can be carried out on-board by a catalytic reactor; higher thermal efficiency can be resulted because the endothermic reaction promotes waste heat recovery from the engine exhaust gas. In-cylinder fuel reforming, which makes use of negative valve overlap and direct fuel injection, is a newly researched technique. This technique injects fuel into the in- cylinder residual gas during the negative valve overlap interval. When the fresh intake charge and the trapped exhaust gas are compressed together, the energy supplied by the piston and also the water in the exhaust gas can be used to perform a steam reforming on the intake fuel. This technique has the potential of reducing fuel consumption by optimizing the split ratio between the fuel injected during negative valve overlap and in the take stroke.[32]

In this research, in order to simulate the fuel reforming effect, the reforming product CO/ H_2- ethanol/air mixture with fuel reforming rate is described by:

$$(1-y)C_2H_5OH + \frac{3}{\varphi}\left[\frac{1}{2}(1-y)\right]O_2 + 2yCO + 3yH_2 + 3/\varphi 3.76 N_2 \qquad (15)$$
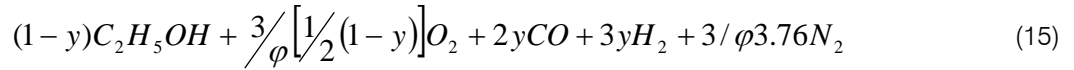
Where y is defined as the fraction of ethanol converted into carbon monoxide and hydrogen (e.i. fuel reforming rate). Thus, the role of different fuel reforming rate is obtained by adding the H_2 and CO concentration at the start of the cycle and changing the quantity of y. It is worth noting that equivalence ratio refers to the ethanol/air mixtures before they are mixed with exhaust gas or converted into reforming products.

VI. Theoretical Equation

In order to define any thermodynamic state within the cylinder, two independent properties and the mixture composition should be known. The general equations governing conservation of mass, species, and energy will be developed for a variable volume reactor, shown in Figure (1). It is assumed that the working fluid behaves as an ideal gas.

a) Conservation of Mass
The rate mass change within any control volume is:

$$m = \sum_{k=1}^{K} m_k = cte \Rightarrow \frac{dm}{dt} = 0 \qquad (16)$$

b) Conservation of Species
Equations tracking the evolution of species within the combustion chamber will be developed on a mass basis corresponding to the definition in Eq. (17).

$$Y_k = \frac{m_k}{m} \qquad (17)$$

It must be said that mixing and stratification are important and can be used for controlling HCCI, but in this research, these effects are ignored for simplicity. As a result the rate of heat release is solely depended on the chemical kinetic reaction rates. To model the combustion process of HCCI engine, it is necessary to describe the heat release via a suitable chemical mechanism and a well-matched, stiff chemical kinetic solver. The model includes the effects of heat transfer, for different engine configurations. The rates of creation/destruction of chemical species are modelled using mass-action kinetics and the specific reaction rate constants strongly depend on the temperature. An elementary reaction that involves 'K' chemical species in 'I' reactions can be represented in the form of:

$$\sum_{k=1}^{k} \upsilon'_{ki} \chi_k \Leftrightarrow \sum_{k=1}^{k} \upsilon''_{ki} \chi_k \qquad (18)$$
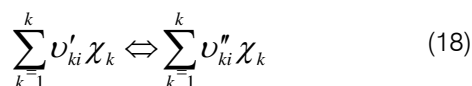
$$\omega_k = \sum_{i=1}^{i} \upsilon_{ki} q_i \qquad (19)$$

$$\upsilon_{ki} = \upsilon''_{ki} - \upsilon'_{ki} \qquad (20)$$

$$q_i = k_{fi} \prod_{k=1}^{K} [X_k]^{\upsilon'_{ki}} - k_{ri} \prod_{k=1}^{K} [X_k]^{\upsilon''_{ki}} \qquad (21)$$

Where υ_{ki}, χ_k, $\dot{\omega}_k$, q_i, k_{fi} and k_{ri} are the stoichiometric coefficients, the chemical symbol for the kth species, the production rate of the kth species, the rate of progress variable for the ith reaction, the forward and reverse rate constants of the ith reaction respectively. The forward and reverse rate constants of the ith reaction rate could be expressed as:

$$k_{fi} = A_i T^{\beta_i} \exp(\frac{-E_i}{R_c T}) \qquad (22)$$

$$k_{ri} = \frac{k_{fi}}{K_{ci}} \qquad (23)$$

$$K_{ci} = K_{pi} \left(\frac{P_{atm}}{RT}\right)^{\sum_{k=1}^{K} \upsilon_{ki}} \qquad (24)$$

$$K_{pi} = \exp\left(\frac{\Delta S_i^\circ}{R} - \frac{\Delta H_i^\circ}{RT}\right) \qquad (25)$$

Where A_i, β_i, E_i and K_{pi} are the pre-exponential factor, temperature exponent, activation energy and the equilibrium constants respectively. The mass conservation equation is employed:

$$\frac{dy_k}{dt} = \frac{\dot{\omega}_k W_k}{\rho} \qquad (26)$$

c) Thermodynamic Property Treatment
The working fluid composition in the cylinder is constantly changing. Once at any time the composition is determined, mixture properties can be calculated. This method simply considers thermodynamic properties of the mixture containing residual gas,

exhaust gas recirculation, unburned gaseous fuel and air. For thermodynamic properties such as constant pressure specific heat, enthalpy and entropy, the NASA curve fits in the following forms are used:

$$\frac{C_{pk}^{\circ}}{R} = \sum_{n=1}^{5} a_{nk} T^{n-1} \qquad (27)$$

$$\frac{h_k^{\circ}}{RT} = a_{1k} + \sum_{n=1}^{5} \frac{a_{n,k}}{n} T_k^{n-1} + \frac{a_{6k}}{T_k} \qquad (28)$$

$$\frac{s_k^{\circ}}{R} = a_{1k} \ln T_k + \sum_{n=1}^{4} \frac{a_{n+1,k}}{n} T_k^n + a_{7k} \qquad (29)$$

d) Conservation of Energy

The generalized energy equation for a thermodynamic control volume may be written as:

$$\dot{T} = \sum_{k=1}^{N} \frac{\omega_k h_k \overline{W}_k}{\rho \overline{C}_p} - \frac{\dot{Q}_{wall}}{\overline{C}_p m} - \frac{1}{\rho \overline{C}_p} \frac{dp}{dt} \qquad (30)$$

The ideal gas law can be written as:

$$PV = nR_u T \qquad (31)$$

These equations can be solved using Dvode subroutine.

e) Heat Transfer

Heat transfer from in-cylinder contents to surrounding surfaces is modelled by Woschni correlation.[33] In this model, in-cylinder gas velocity and convective heat transfer coefficient are as:

$$\omega = [(C_{11} + C_{12} \frac{v_{Swirl}}{\overline{S}_p})]\overline{S}_p + C_2 \frac{V_d T_i}{P_i V_i}(P - P_m) \quad (32)$$

$$h = 3.25 T^{-0.53} B^{-0.2} P^{0.78} \omega^{0.78} \qquad (33)$$

In which B is assumed to be cylinder bore. Thus the heat transfer correlation is obtained as:

$$Q_{wall} = hA(T - T_{wall}) \qquad (34)$$

VII. RESULTS AND DISCUSSION

For the purpose of finding the accuracy of the present model, the predicted values of in-cylinder pressure for ethanol are compared with the experimental obtained results.[34]The specifications of the engine used in the experiments and test conditions are presented in Table 1.

a) Cylinder Pressure

Variations of pressure with crank angle are shown in Figures (2) and (3) for φ=0.287 and φ=0.334 respectively. As seen, the maximum predicted pressures for the present model are about 71.48 bar and 78.123, fairly comparable with corresponding experimental values of 72 bar and 78.7 bar. It must be said that in the present model 386 reactions are considered in order to simulate these engine so that the trend and peak pressures of predicted results are more compatible in comparison with the theoretical predicted model.[35] In order to investigate the effects of important parameters on HCCI engine performance, the engine specification used for this part is shown in Table 2.

b) Cylinder Temperature

Variation of temperature with crank angle is shown in Figure (4) for φ=0.287 and φ=0.334. As observed, the maximum anticipated temperatures for the present model are about 1882 K and 2026 K for φ=0.287 and φ=0.334, respectively. As perceived, the ignition process is started earlier for φ=0.287 in comparison with φ=0.334 and also the expected maximum temperature is higher for φ=0.334 compared with φ=0.287.

c) Mole Fraction

The species mole fractions as a function of CAD are analyzed for different equivalence ratios. Figures (5) and (6) show that at the maximum in-cylinder temperature, the concentrations of ethanol and oxygen sharply decrease but the concentrations of CO and H_2 increase. Then because of equation (33), the CO is converted to CO_2 thus concentration of CO_2 increases.

d) The Effect of EGR Rate on In-Cylinder Pressure and Temperature

Figures (7) and (8) show the variation of pressure and temperature with the EGR rate. The intake charge temperature is maintained at 430K and the equivalence ratio is assumed to be 0.6. Figure (7) shows that the peak pressure decreases with increase in EGR rates, which indicates that pressure is affected by the reduction in the chemical energy content in the engine charge. As seen in Figure (8), the peak temperature starts to decrease rapidly with higher EGR rates where more unreactive molecules are available due to increase in EGR rate. As observed, the increase of EGR delays the ignition timing and slows down the combustion reaction rate.

e) The Effect of Fuel Reforming Rate on In-Cylinder Pressure and Temperature

Figures (9) and (10) show the variation of pressure and temperature with the fuel reforming rate respectively where the inlet temperature is maintained at 421K and the equivalence ratio is considered 0.6 for all cases. As seen in Figure (9), while shooting up the fuel reforming rate increases the in-cylinder pressure when the fuel reforming rate reaches to 27%, there is a considerable decrease in the pressure behavior above this percentage which implies that the variation of pressure is directly proportional to the ignition timing. This matter elucidates that when ignition occurs after top dead center, raising the fuel reforming rate culminates to

increase in peak pressure on contrary to the case in which ignition happens before the top dead center.

As seen in Figure (10), increasing the fuel reforming rate to H_2 ,CO causes to have a remarkable raise in the temperature trend. It must be said that the engine performance is acceptable only for the case in which fuel reforming rate is under 27% due to the fact that the knock phenomena occurs in the range above this value.

f) The Effect of Compression ratio on In-Cylinder Pressure and Temperature

Figures (11) and (12) illustrate the variation of pressure and temperature for different values of compression ratio. As observed, by increasing the compression ratio, the initiation of ignition is retarded and burn duration is decreased and therefore the in-cylinder heat release and chemical reaction rates are enhanced and as a result the maximum pressure and temperature goes up. It is worth noting that the combustion process does not occur for the compression ratio equal and lower than 14 and also at compression ratios higher than 17, the auto-ignition phenomenon occurs.

VIII. CONCLUSION

In this research, a single-zone thermodynamic model is used to predict and analyze the combustion of HCCI engines fuelled with ethanol. For this purpose, a mechanism containing the important reactions of ethanol is employed. Control of combustion phenomenon, ignition timing and burn duration are the main issues of these engines and the exhaust gas recirculation (EGR), fuel reforming and compression ratio are the attempts in order to achieve the acceptable operation of these engines. The aim of this study is to investigate the performance of HCCI engines using these controlling methods.

From this analysis, the predicted pressures have been compared with both experimental and theoretical results and they show a good agreement with the published data. In addition, the in-cylinder mole fraction at different equivalence ratios shows that the concentrations of ethanol and oxygen sharply decrease but the concentrations of CO and H_2 increase and because the CO is converted to CO_2 thus the concentration of CO_2 increases. The effect of EGR rate on the engine specification indicate that the ignition timing and burn duration increase and the in-cylinder pressure and temperature decrease as a result of increasing the EGR rate.

As elucidated in this research, ignition timing advances and burn duration shortens when the fuel reforming rate enhances and above 27% of reforming rate, higher amount of heat release may lead to knocking in the cylinder. Furthermore, it is shown that higher amount of fuel reforming rate results in increasing

the peak temperature. The peak pressure is dominated by the location of ignition timing which implies that the increase in the fuel reforming rate under 27% causes the enhancement in the pressure profile because the initiation of ignition is after TDC, but above this percentage, the pressure profile declines because the initiation of ignition is before TDC. Consequently, increasing the compression ratio results in advancing the ignition timing, decreasing the burn duration and increasing the in-cylinder pressure and temperature.

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Table 1 : Engine specifications

Displaced Volume	1600 cm^3
Bore	120.65 mm
Stroke	140 mm
Connecting Road Length	260 mm
Inlet Temperature (K)	383
Comp. Ratio	18
Engine Speed	1000 RPM
φ	A=0.278 , B=0.334
Exhaust Valve Opening	39° BBDC
Exhaust Valve Closing	10° BTDC
Inlet Valve Opening	5° ATDC
Inlet Valve Closing	13° ABDC

Table 2 : Engine specifications

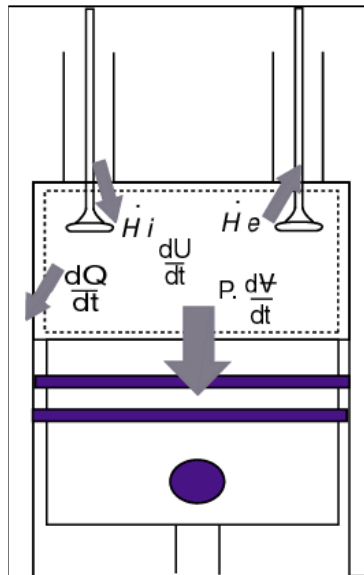| Displaced Volume	587.622 cm^3
Bore	72 mm
Ratio of the Connecting Road length to Crank Radius	3.59
Inlet Temperature (K)	415 to 460 K
Engine Speed	1000 RPM
φ	0.2 to 0.6
EGR Rate (%)	10 to 50
Reforming Rate (%)	0 to 100
Compression Ratio	14 to 19

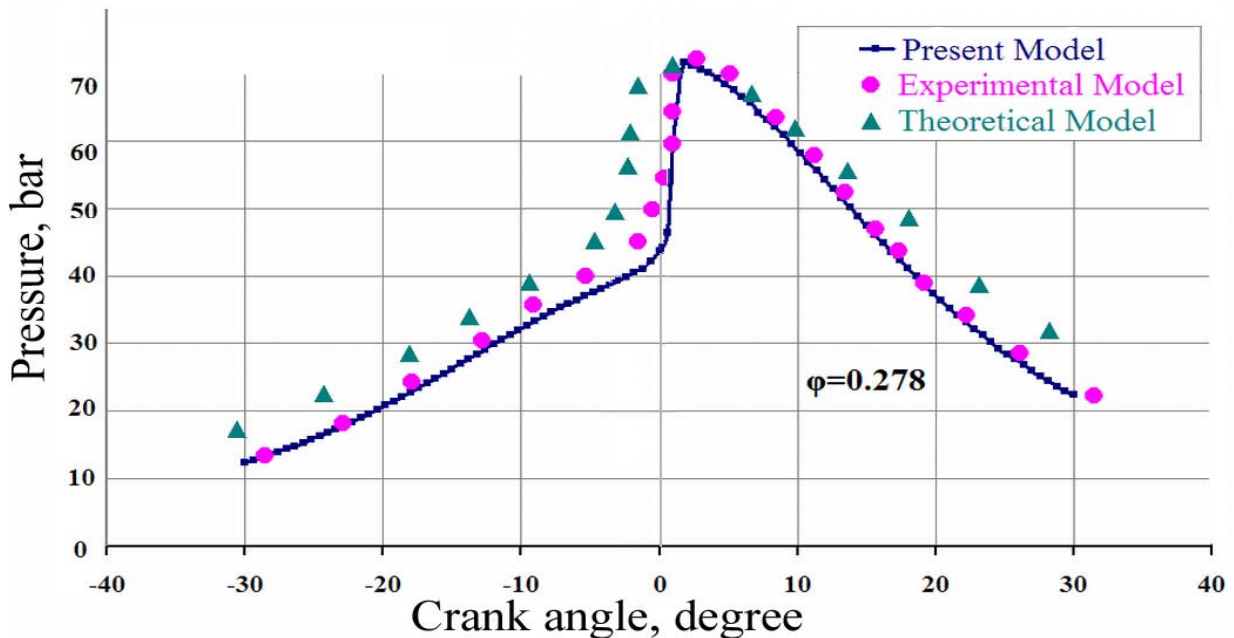*Figure 1 :* Schematic view of control volume interaction with surrounding

Figure 2 : Variation of predicted, experimental and theoretical values of pressure with crank angle for $\varphi = 0.278$
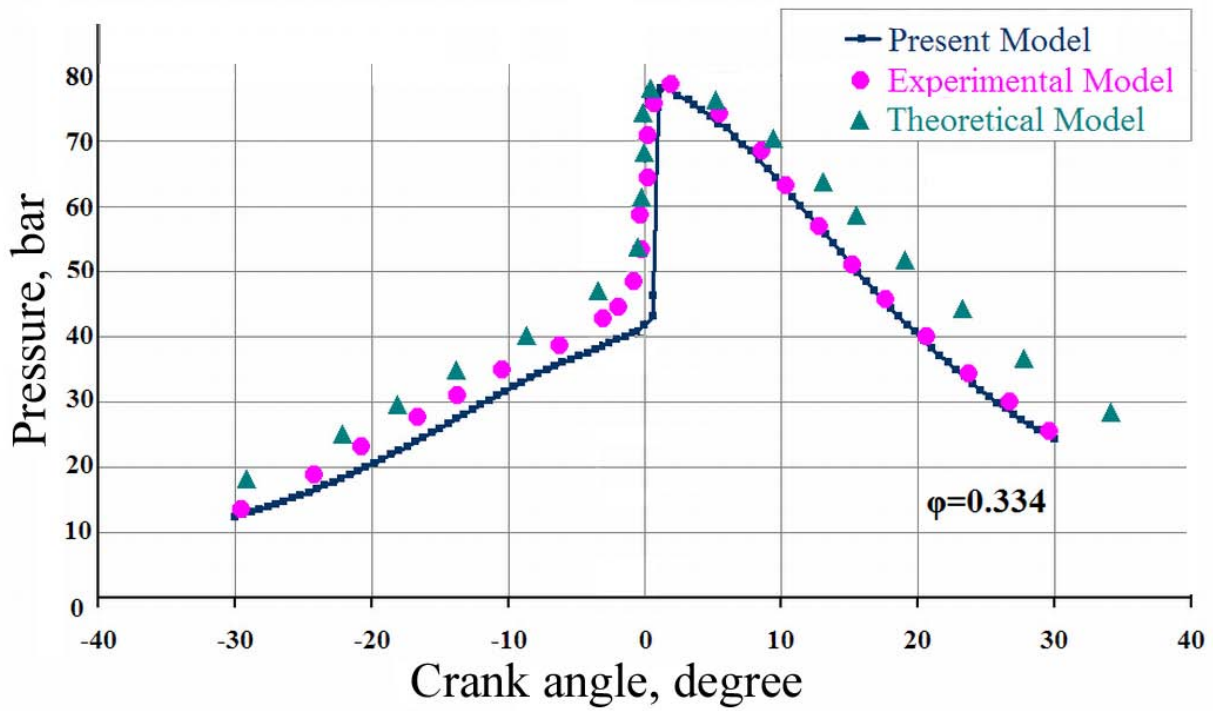
Figure 3 : Variation of predicted, experimental and theoretical values of pressure with crank angle for $\varphi = 0.334$
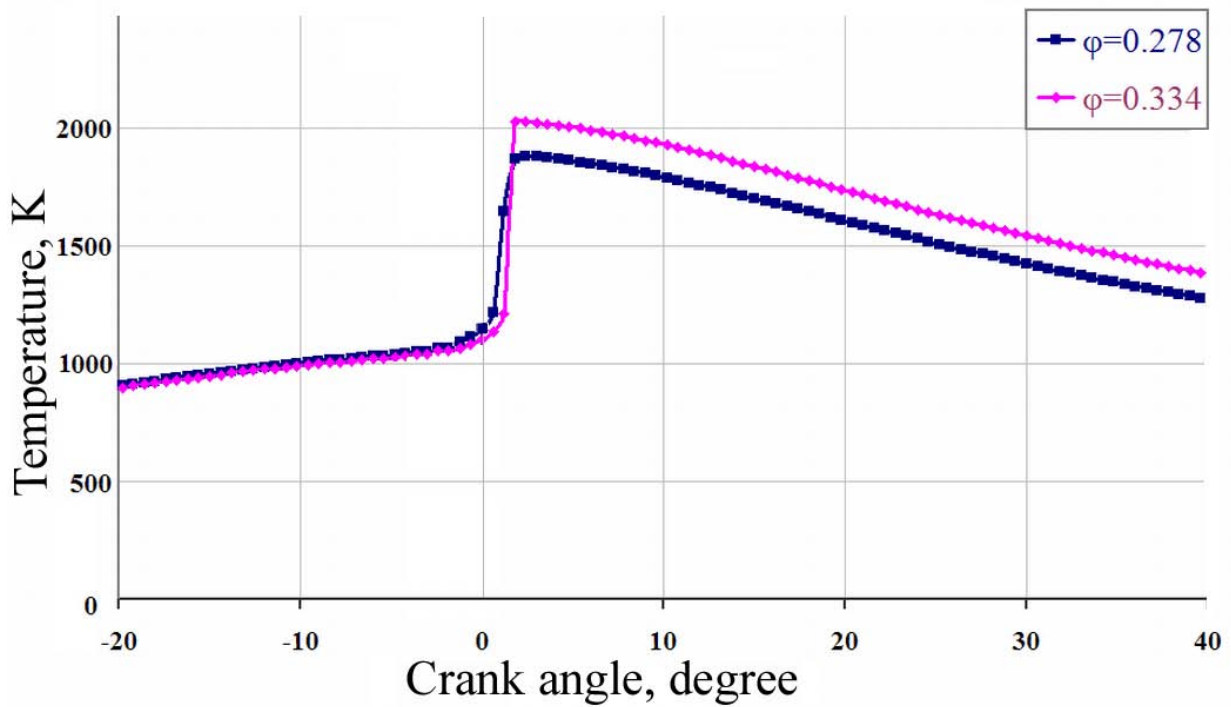
Figure 4 : Variation of temperature with crank angle for $\varphi = 0.278$ and $\varphi = 0.334$
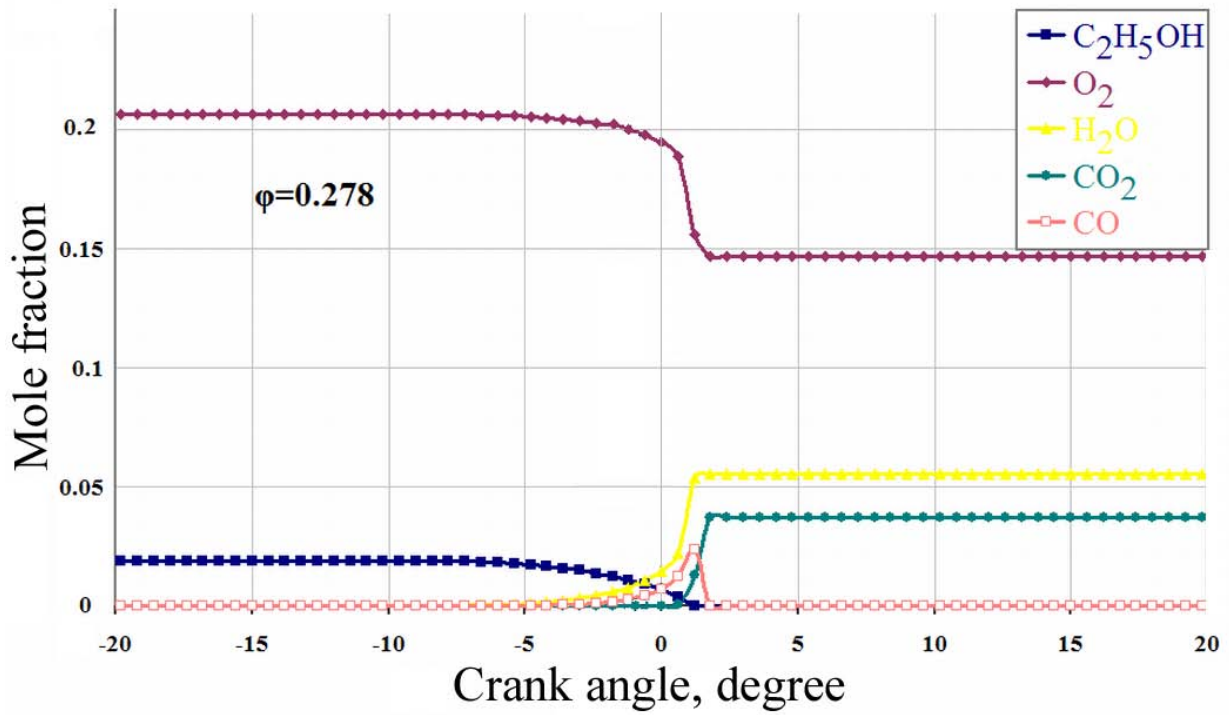
Figure 5 : Variation of in-cylinder mole fraction with crank angle for $\varphi = 0.278$
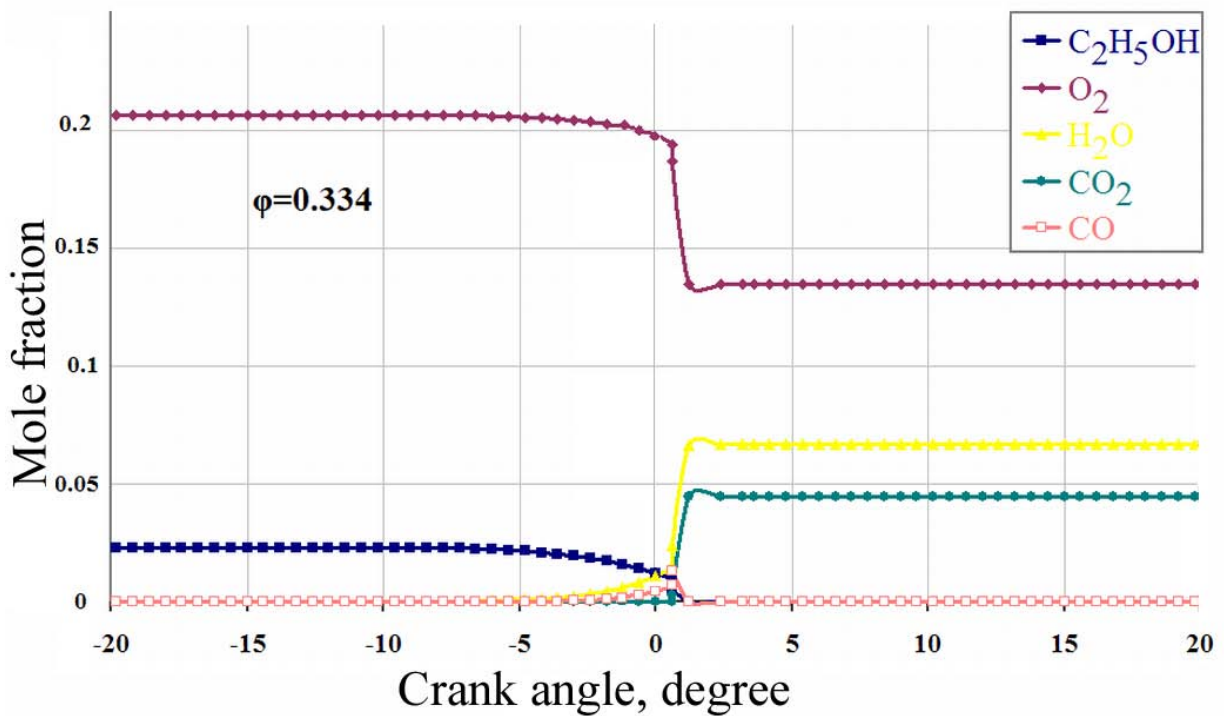
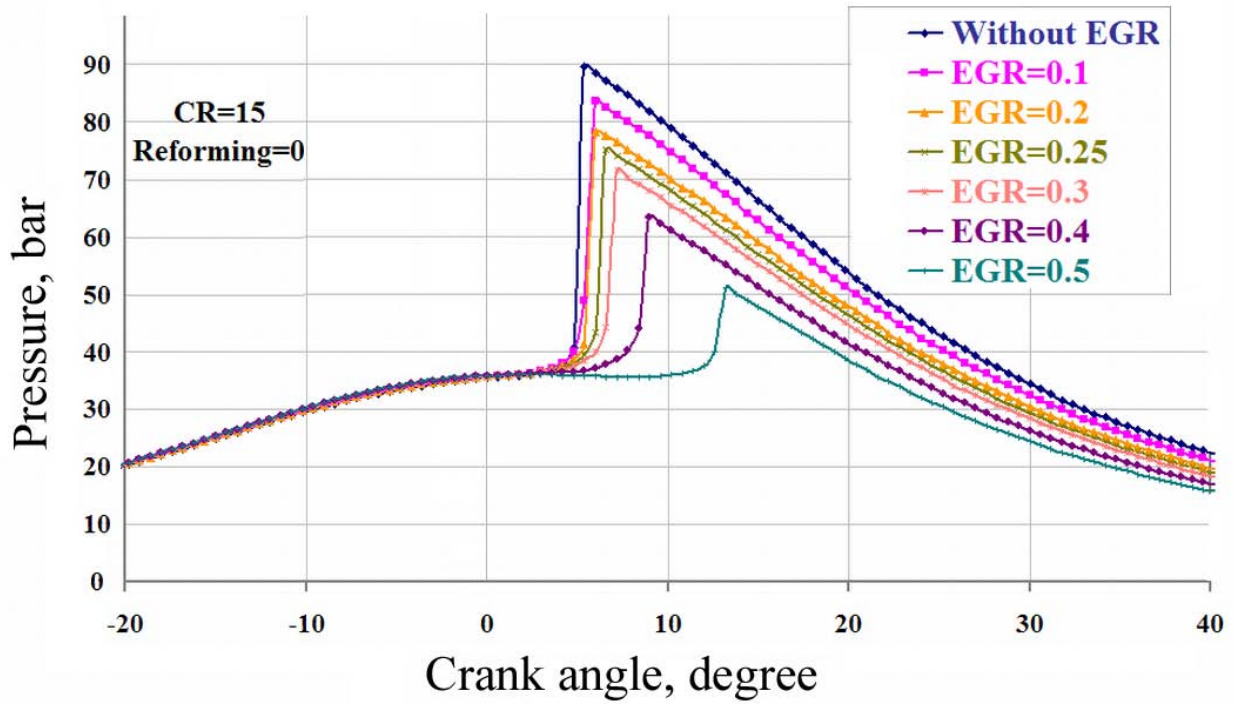Figure 6 : Variation of in-cylinder mole fraction with crank angle for $\varphi = 0.334$

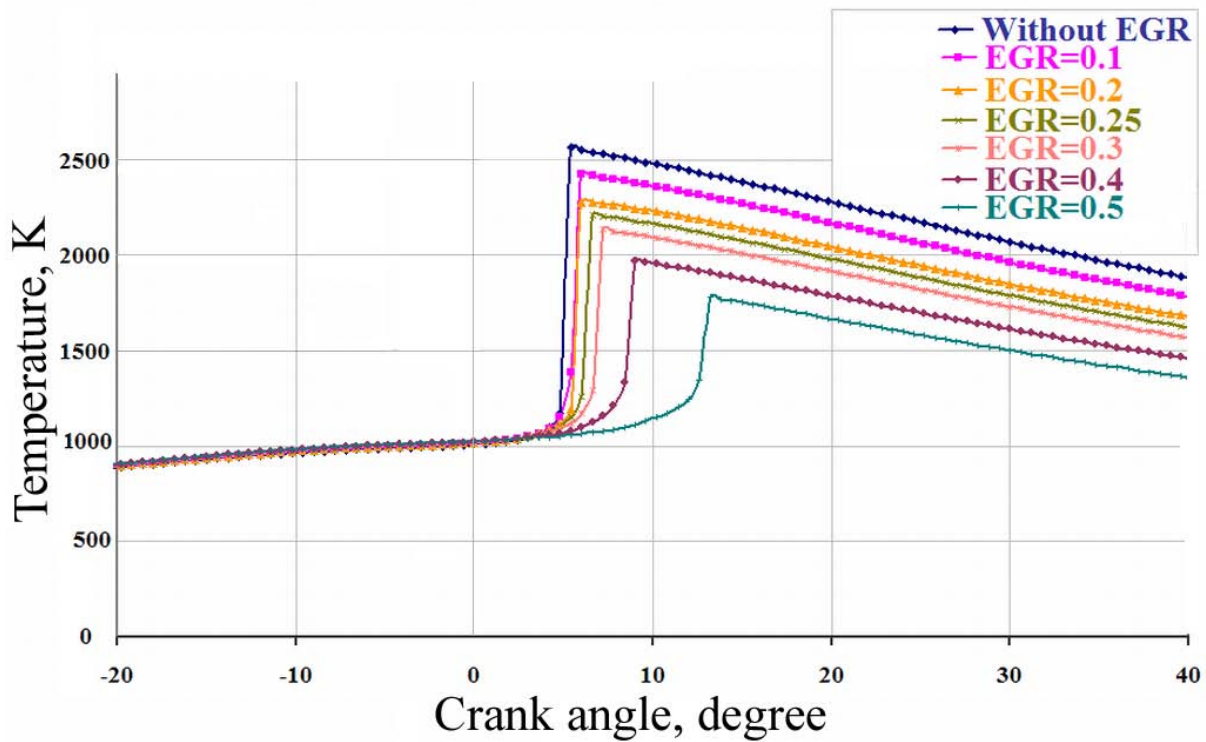Figure 7 : Variation of pressure with crank angle for different EGR rate

Figure 8 : Variation of temperature with crank angle for different EGR rate
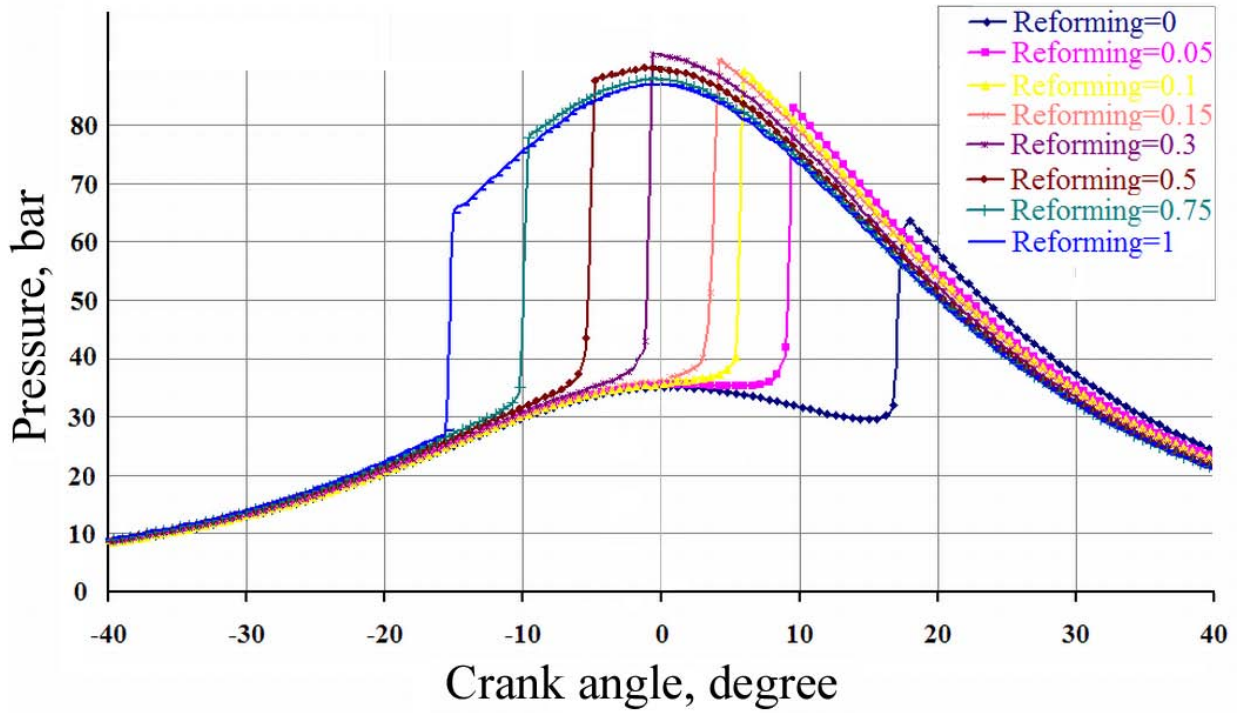
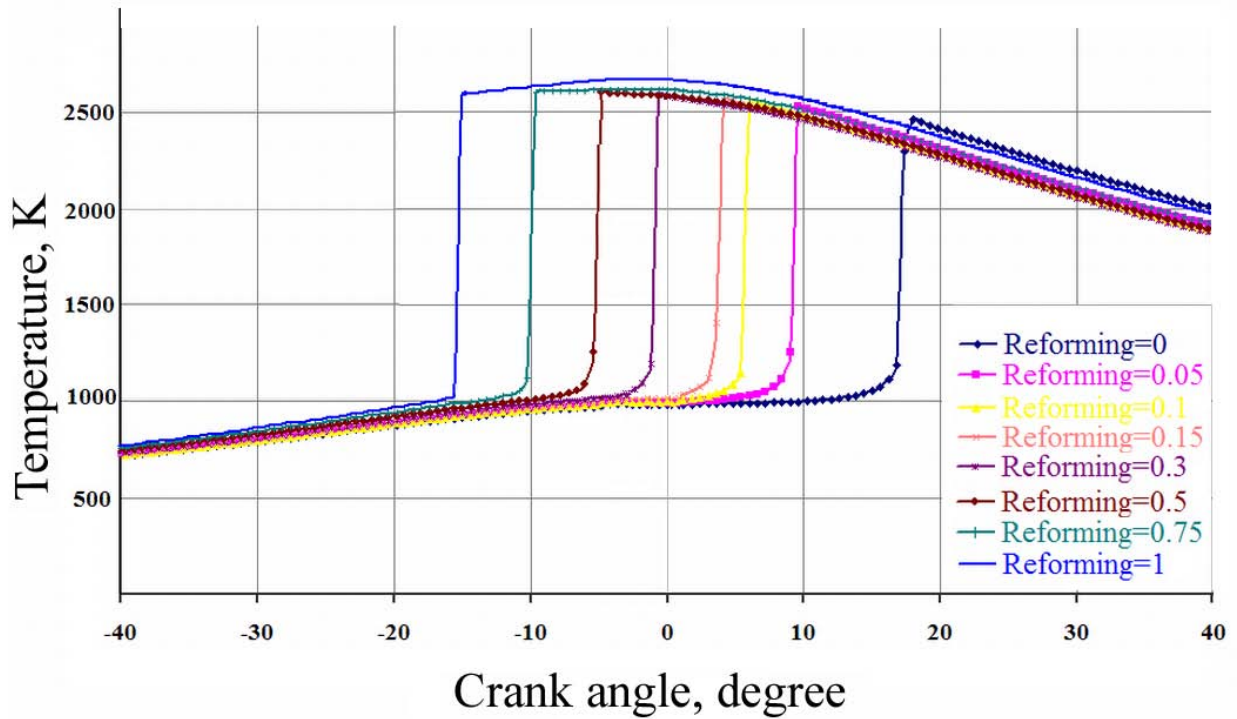*Figure 9 :* Variation of pressure with crank angle for different fuel reforming rate

Figure 10 : Variation of temperature with crank angle for different fuel reforming rate
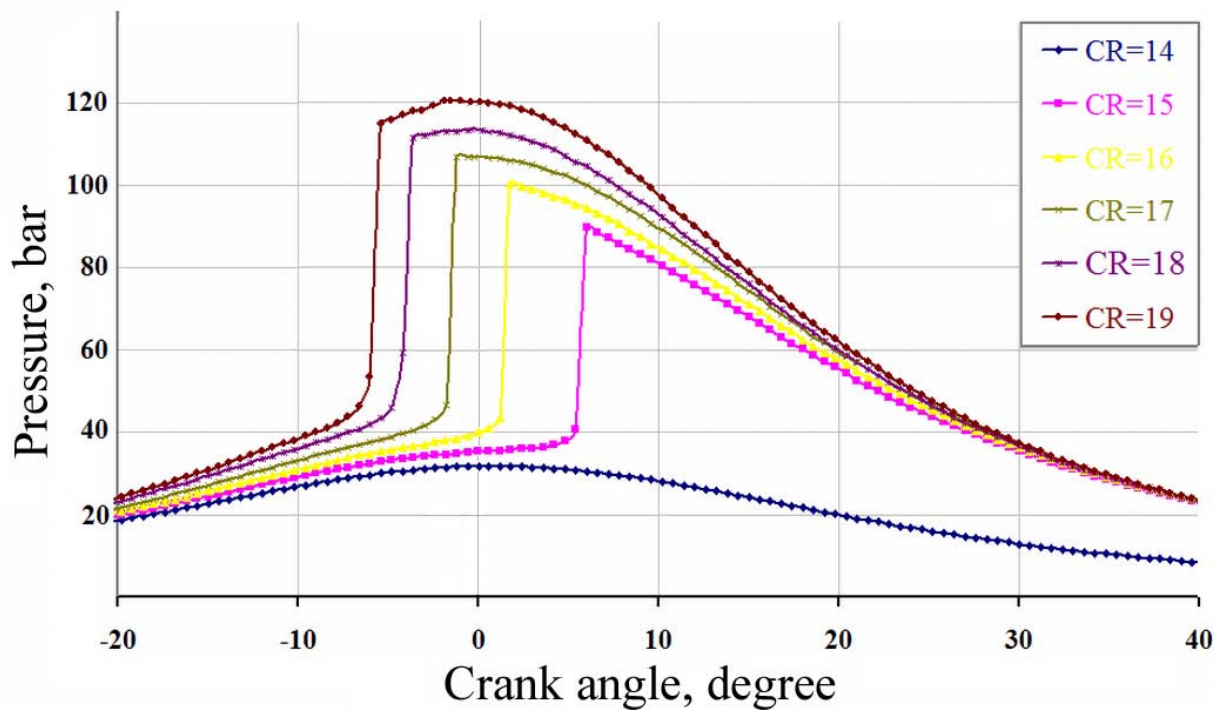
Figure 11 : Variation of pressure with crank angle for different compression ratio
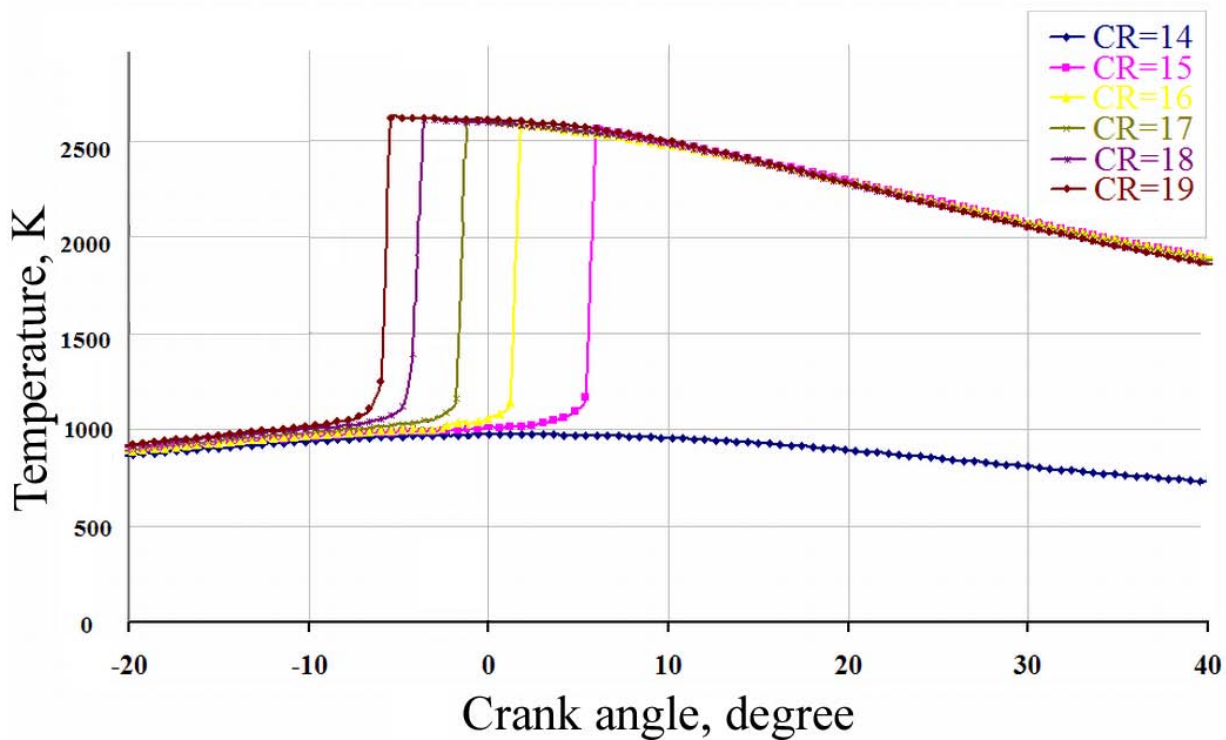

Figure 12 : Variation of temperature with crank angle for different compression ratio