

Effects of Intake Temperature and Intake Pressure on Combustion and Exhaust Emissions of HCCI Engine

Fridhi Hadia, Soua Wadhah, Hidouri Ammar, Omri Ahmed

Abstract—In this paper, the effect of the intake temperature (IT) and intake pressure (IP) on ignition timing and pollutants emission of Homogeneous Charge Compression Ignition (HCCI) engine is investigated. Numerical computations are performed using the CHEMKIN computer code. The numerical temperature obtained using different boundary conditions is compared to published data and a good agreement is assigned. Results show that the HCCI combustion engine is significantly improved by increasing the IT. With a value of IT lower than 390 K, combustion cannot occur. However, with an IT greater than 420 K, the cylinder pressure decreases. An optimum crank rotation angle is achieved by using IT of 420 K. So, we can conclude that the variation of the IT and IP influence notably the emission concentration.

Keywords—HCCI engine, CEMKIN, intake temperature, intake pressure.

I. INTRODUCTION

HCCI is the abbreviation of Homogeneous Charge Compression Ignition engine. It was discovered by Onishi et al. [1]. It combines the best features of spark-ignition (SI) gasoline engine and compression-ignition (CI) diesel engine [2], [3], (Fig. 1 [4]). In SI engines, the fuel and air are mixed together before combustion initiation. Then, the charge is compressed and ignited. In the CI engine, air is compressed to a high-pressure and the fuel is injected into the hot compressed air than the auto-ignition will be occurring. So, the HCCI engines work using a homogeneous mixture like the SI, and then it compresses this mixture to auto-ignition like the CI. Recently, the scientists and engineers have focused on HCCI engines. This is due to the significant fuel efficiency gains they achieve compared to SI gasoline engines.

The control of combustion phenomenon and emission characteristics in HCCI engines are the most challenging and demanding issues. Several operating parameters are proposed by researchers to control the combustion phase of HCCI engines such as IT and IP [6]-[10].

A more profound discussion on the effect of intake temperature on ignition timing has been performed. In the following, we are going to underline the research works that have been investigated in the field of IT in IC engine. Soyhan et al. [5] studied the effect of the inlet temperature on the ignition delay for an inlet temperature higher than 700 K.

They concluded that a higher initial pressure and inlet temperature result in a lower ignition delay. Najafabadi et al. [6] determined the influence of the inlet temperature on the combustion chamber pressure for a 2-stroke engine at a speed of 6000 rpm. They concluded that increasing the intake temperature (525-575 K) advances the HCCI combustion timing. Fieweger et al. [7] found that the ignition delay in a shock tube decreases at an increasing inlet temperature.

A review of the intake pressure impact on CI engine has been investigated in following orders. Sayin et al. [8] reported the effect of IP on the performance and emissions of a diesel engine using biodiesel diesel blends. They found that hydrocarbons (HC), and carbon monoxide (CO) emissions decrease and nitrogen oxide (NOx) emissions increase with the increase in IP for the all fuel blends. Karhale et al. [9] investigated the performance of Karanja methyl ester and its blends for running a diesel engine. The engine was tested at two injection pressures 180 and 245 kg/cm² and temperatures of 30, 50 and 70 °C. They concluded that the injection pressure and fuel temperature have significant effects on engine performance parameters. Sukumar et al. [10] studied the effect of injection pressures (200, 220 and 240 bar) on performance, emissions and combustion characteristics of the diesel engine. They also investigated a high linolenic linseed oil methyl ester. They found that, at 240 bar injection pressure, the thermal efficiency improved with increased emissions and the thermal efficiency is comparatively lower than that of diesel. IT and IP play an important role in HCCI combustion. For this reason, it is important to understand the effects of these parameters on HCCI combustion.

In this study, the pressure range spanned 0.9 and 2 atm and the inlet temperature range covered 392, 393, 395, 400, 420, 440 and 460 K.

II. THEORETICAL EQUATION

A. Overview of Simulation Software

The software used to model the isooctane combustion was CHEMKIN [11]. It is a chemical kinetics program firstly developed at Sandia National Laboratory, then it is managed by the company Reaction Design. CHEMKIN has several models representing simple geometries (Fig. 2). These models are used to test the different reaction mechanisms of combustion. In the present work, the ICE (Internal Combustion Engine) model was used (Fig. 3). The software includes an extensive library of gas-phase kinetics, gas transport and thermodynamic data.

Fridhi Hadia is with the Research Unit of Materials, Energy and Renewable Energies MEER, Gafsa Faculty of Science, Zaroug Gafsa 2112, Tunisia (Corresponding author; e-mail:fridhi.hadia@yahoo.com).

Soua Wadhah, Hidouri Ammar and Omri Ahmed are with the Research Unit of Materials, Energy and Renewable Energies MEER, Gafsa Faculty of Science, Zaroug Gafsa 2112, Tunisia.

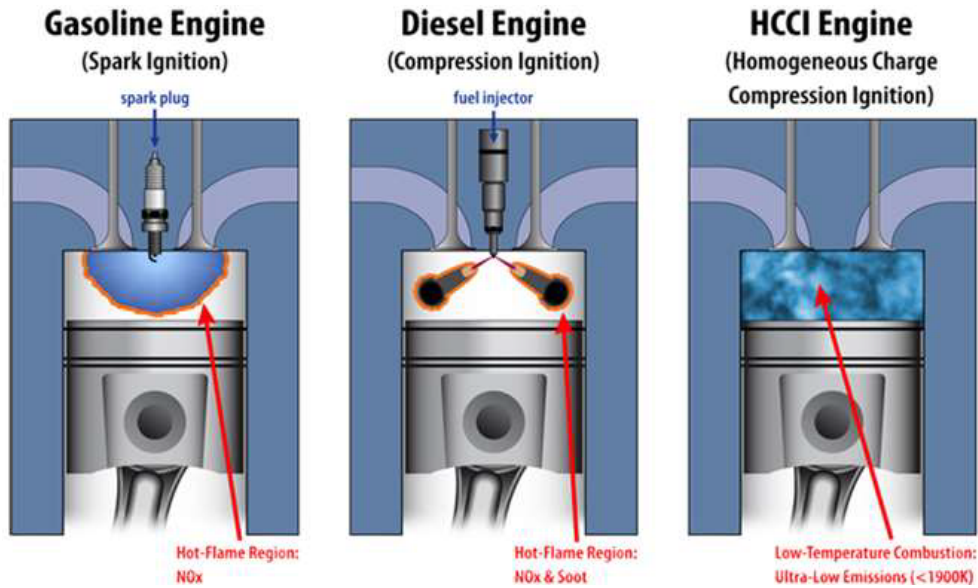


Fig. 1 The differences among SI, CI and HCCI engines [4]

- **The pre-processing stage:** The user should create a chemistry set that specifies applicable data. Indeed, the kinetic mechanism with all elementary reaction and associated parameters like a , b , and EA are loaded into a gas-phase kinetics file.
- **The thermodynamic data file:** Here complicated mechanisms with many different species may require external thermodynamic data not supplied by CHEMKIN.
- **A FORTRAN computer programming language:** It is used to communicate modeling conditions and reactor parameters.
- **The post-processing:** The user is required to select specific data of interest for plotting or exporting.

B. Engine Geometric Parameters

In this study, we put our simulation starting crank angle to -142 degrees in the software. Other simulation parameters, we used in the simulation software, were cylinder cycles end time as 0.043 sec or for 257 degrees crank angle to 115 degrees after TDC.

Fig. 3 illustrates an engine cylinder used in this study. The relative position of piston centre/crank axis at any crank angle is given by [15]:

$$s(\theta) = a \cos \theta + (l^2 - a^2 \sin^2 \theta)^{1/2} \quad (1)$$

When the piston is at TDC (Top Dead Centre) where ($s=l+a$), cylinder volume is named the clearance volume V_c . When the piston is at BDC (Bottom Dead Centre) where ($s=l-a$), cylinder volume is named the maximum displacement volume V_d [12].

$$V_d = \frac{\pi B^2}{4} L \quad (2)$$

The displacement of the piston generates the change of the volume (Fig. 1). Heywood [12] provides equations that describe the volume as a function of time, it is given by:

$$\frac{V(t)}{V_c} = 1 + \frac{C-1}{2} \left[R + 1 - \cos \theta(t) - \sqrt{R^2 - \sin^2 \theta(t)} \right] \quad (3)$$

The compression ratio C is given by:

$$C = \frac{V_d + V_c}{V_c} \quad (4)$$

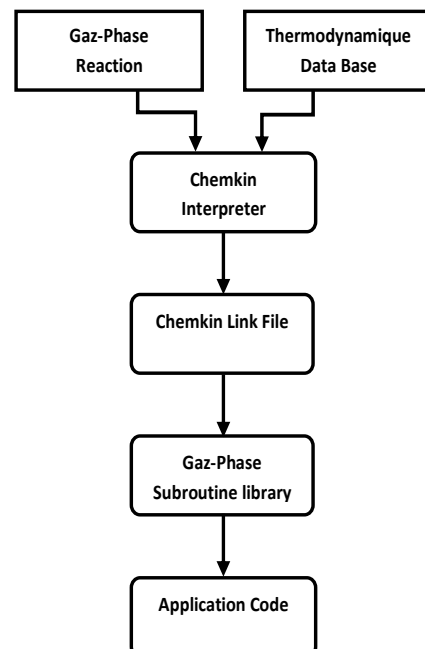


Fig. 2 Chemkin package working flow diagram

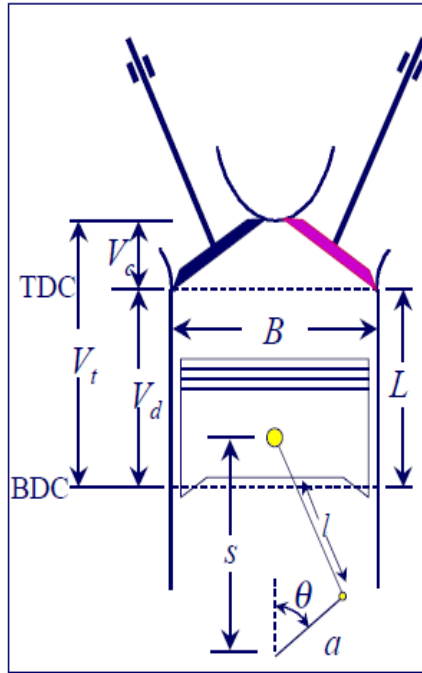


Fig. 3 An engine cylinder: l: The connecting rod length; a: the crank arm radius; B: the bore; s: the distance between crank axis and wrist pin axis; L: stroke length; Vc: The clearance volume; Vd: the displacement volume; Vt: the cylinder volume; TDC: top dead center; BDC: top bottom center

By deriving (1), by the time we obtained [15]:

$$\frac{d\left(\frac{V}{V_c}\right)}{dt} = \Omega \left(\frac{C-1}{2}\right) \sin \theta \left[\frac{1 + \cos \theta}{\sqrt{R^2 - \sin^2 \theta}} \right] \quad (5)$$

The rotation rate of the crank arm Ω is given by:

$$\Omega = \frac{d\theta}{dt} \quad (6)$$

C. Heat-Transfer in the Cylinder

The heat transfer coefficient between the gas and cylinder wall can be obtained in terms of a Nusselt number.

The heat transfer correlation is given by [12]:

$$Nu = a Re^b Pr^c \quad (7)$$

where Nu the Nusselt number, Re the Reynolds number and Pr the Prandlt number:

$$Nu_h = \frac{h B}{\lambda} ; Re = \frac{\rho w D}{\mu} \text{ and } Pr = \frac{c_p \mu}{\lambda} \quad (8)$$

w is the average cylinder gas speed given by Woschni Correlation [13]. The Woschni Correlation of Average Cylindre Gas Velocity is given by:

$$w = \left[\left(C_{11} + C_{12} \frac{V_{swirl}}{S_p} \right) \right] \bar{S}_p + C_2 \frac{V_d T_i}{P_i V_i} (P - P_{motored}) \quad (9)$$

The heat transfer correlation coefficients (denoted by a, b and c) and the Woschni correlation coefficients (denoted by C11, C12 and C2) were taken from [13], as seen in Tables I and II.

TABLE I
HEAT TRANSFER CORRELATION COEFFICIENTS [13]

Coefficient	Values
a	0.035
b	0.071
c	0.0

TABLE II
THE WOSCHNI CORRELATION COEFFICIENTS [13]

Coefficient	Values
C11	2.28
C12	0.308
C2	0.324

The engine geometry and engine operating parameters, used in this study, are specified in Table I. The engine cylinder had a bore of 72 mm, stroke, 120.65 mm and ratio of the connecting road length to crank radius, 3.59. The internal combustion engine initial set speed was 1000 rpm. The IT and pressure were 430 K and 1.065 atm. The fuel was isooctane (C_8H_{18}) with an equivalence ratio of 1. The gas mixture temperature and pressure at IVC are 415 K and 1.065 atm, respectively. Engine specifications are given in Table III.

In this paper, we used a detailed chemical kinetic reaction mechanism consisting of 3606 step elementary reactions among 857 species for the oxidation of isooctane, generated by Curran et al. [14].

TABLE III
THE ENGINE SPECIFICATIONS

Parameters	Setting
Displaced Volume	587.622 cm ³
Bore ×stroke	72 ×12.065 mm
Ratio of the connecting road length to crank radius	3.59
Inlake temperature (K)	390-460
Engine Speed (rpm)	1000
IP (atm)	0.9-2
Fuel	Isooctane

III. RESULTS AND DISCUSSION

A. Validation

The average cylinder temperatures at five different ITs (480, 485, 492.5, 497.5, and 510 K) are shown in Fig. 4; these were compared with numerical results of [15]. In our study, computations were performed using the CHEMKIN computer code. Yanbin [15] used the KIVA-MZ model. In general, Fig. 5 confirms that our results are in good propinquity with the results of Yanbin [15]; but we noted some differences, especially in the start of the combustion. This distinction can

be explained by the different software tools used. Then, this study was extended to evaluate the effect of IT and IP on ignition timing and emission characteristics of HCCI Engine.

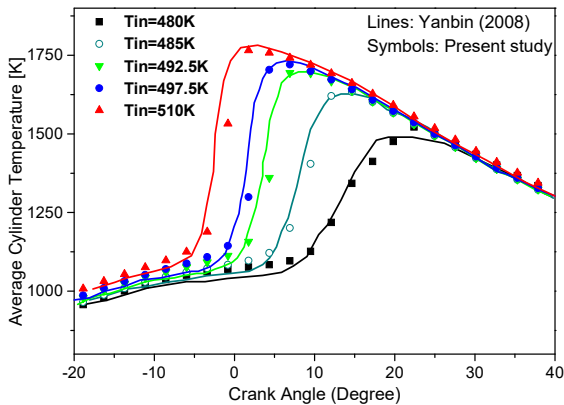
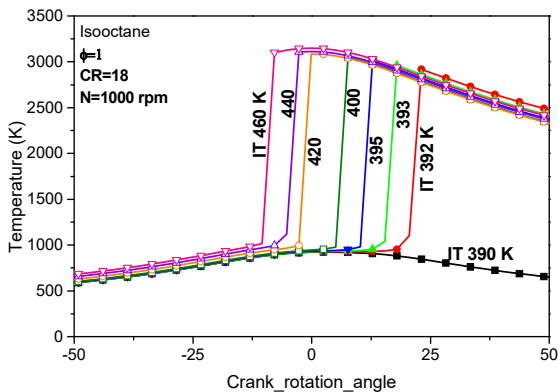
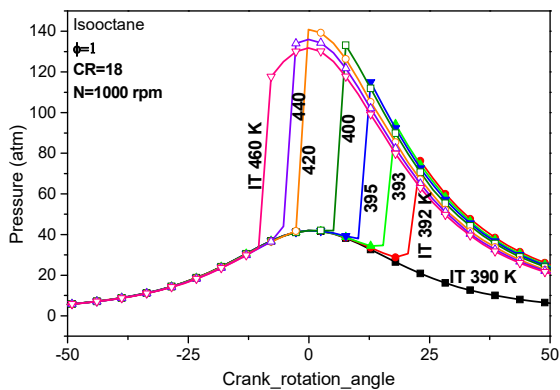


Fig. 4 Comparison between the present study and Yanbin's result [15]



(a)



(b)

Fig. 5 Cylinder temperature (a) and pressure (b) versus crank angle for various IT

B. IT

1. Pressure and Temperature Profiles

This subsection treats the influence of the IT on the HCCI

ignition timing. The operational range for different IT was from 390 to 460 K.

The cylinder temperature and pressure curves were plotted in Figs. 5 (a) and (b). As shown in Fig. 5 (a), the temperature distribution comparison was carried out with three initial temperature cases: good combustion (420 K), marginal combustion (392 K), and misfire (390 K). With lower initial temperature, the ignition was late, and peak pressure and temperature were lower. As can be seen, for the two highest initial temperature cases, the pressure curves decreased. For the temperature cases between 393 and 420 K, the combustion was of high combustion efficiency, and the two lower initial temperature cases are distinctively below, especially the lowest initial temperature case.

Fig. 6 depicts the effect of IT on ignition timing obtained from the data presented in Figs. 5 (a) and (b). As observed, the in-cylinder temperature and pressure increase with the increase of IT. The reason is that the temperature of compression process and gas mixture augments with the increase of IT at the same time. It contributes to the good evaporation and atomization of fuel; thus, the mixture is more homogeneous and the combustion is more complete. Furthermore, high temperatures enhance combustion intensity, reduce combustion duration and advance ignition timing.

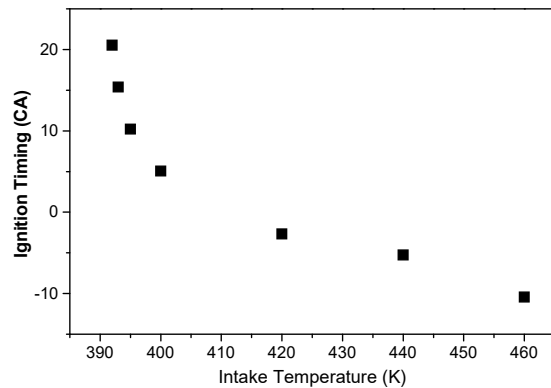


Fig. 6 Effect IT on the ignition timing of HCCI engine

2. Emissions

The effects on emissive species by varying the IT have been illustrated in Fig. 7. The increase of air IT resulted in shorter combustion duration. Therefore, the increase of air IT decreases the CO₂ emissions and increases CO emissions. At lower air IT, the higher oxygen concentration leads to the complete combustion.

C. IP

In this section of search, we will concentrate on the effect of IP on Ignition timing.

1. Pressure and Temperature Profiles

Simulations were carried out at an IT of 430 K and the equivalence of 1 to see the effects of IP on HCCI ignition timing.

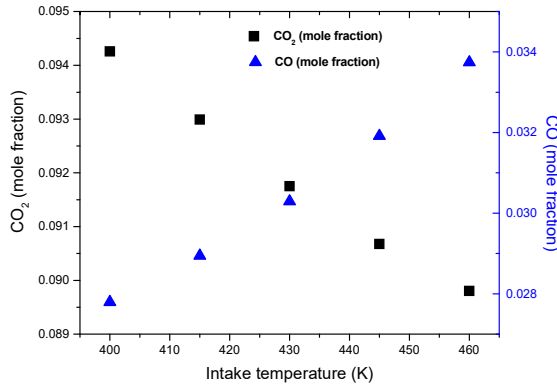
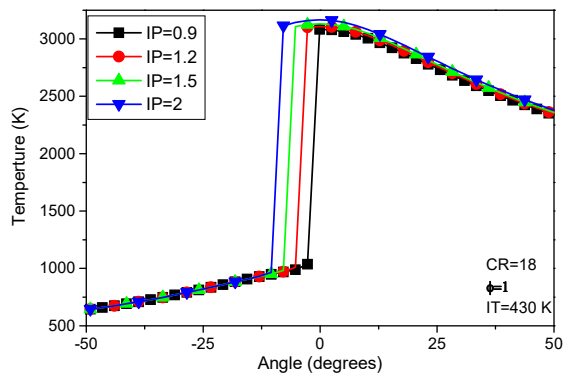
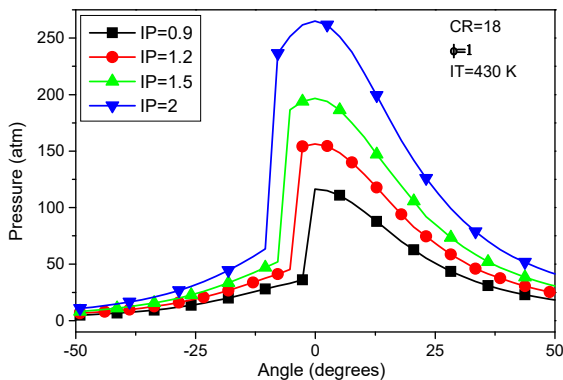


Fig. 7 Emissions versus Crank angle for various IT



(a)



(b)

Fig. 8 Cylinder temperature (a) and pressure (b) versus crank angle for various IP

The IP was varied from 0.9 to 2 atm, while keeping the temperature fixed at 430 K and the compression ratio fixed at 18, as shown in Fig. 8. It can be also seen that the peak cylinder pressure increased significantly with the increase of the IP. Changing the IP has a significant impact on the ignition delay and the high-temperature reaction time, Fig. 10. This is a result of the increased trapped air and fuel concentrations at these higher inlet pressures generating more free radicals that in turn accelerate the chemical reactions.

2. Emissions

Fig. 9 illustrates the effects of varying IP on each emissive species. We noted that with increasing air IP, CO₂ increases but CO decreases.

The increasing of air IP is necessary to increase air density in allowing for better combustion within a limited time to improve fuel economy and exhaust emissions. Therefore, complete combustion leads to the reduction of CO emission and the increasing of CO₂ emissions.

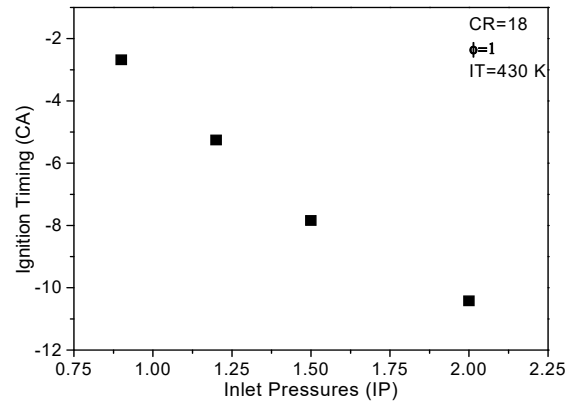


Fig. 9 Effect IP on the ignition timing of HCCI engine

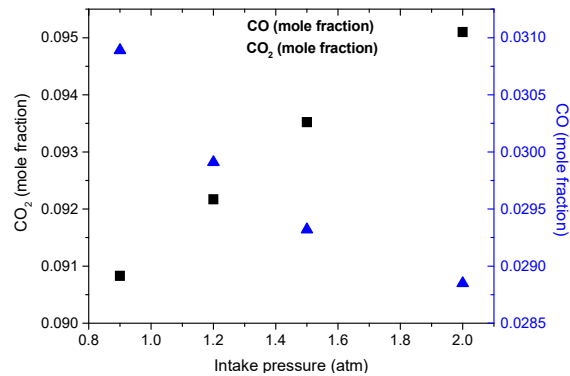


Fig. 10 Emissions versus Crank angle for various IP

IV. CONCLUSIONS

This paper presents the results of an investigation conducted to determine the effects of IT and IP on ignition timing and emission characteristics of HCCI engine fuelled with isooctane. Based on the results of this study, the conclusions can be drawn as:

- (1) Results show that increasing the IT significantly advanced the HCCI combustion ignition. It can be distinguished that a good combustion for IT was equal to 420 K, a marginal combustion for IT was equal to 392 K, and a misfire for IT was equal to 390 K, using isooctane as a fuel.
- (2) Results show that increasing the IT by 60 K, HCCI combustion timing can be advanced up to 15.48° crank angle.

- (3) The IT has great influence on the combustion characteristics of HCCI engine with isooctane, so it can effectively control HCCI combustion by controlling IT.
- (4) For the emission analysis, with the increase of air IT, CO emission was increased however CO₂ was decreased.
- (5) Concerning the IP, by increasing it from 0.9 to 2 atm the ignition timing was advanced by up to 7.74° crank angle and the maximum cylinder pressure and temperature increased.
- (6) In general, increasing the IP, decrease the CO emission and increased CO₂ emission.

Other means of ignition control can also be used for HCCI combustion, such as compression ratio, engine speed and equivalence ratio, will be studied in future works.

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