

Performance of Homogenous Charge Compression Ignition (HCCI) Engine with Premixed Methane/Air Supported by DME for Electrical Power Generation Application

Mojtaba Keshavarz*

Performance Analysis Engineer,
Iran Heavy Diesel Mfg. Co.
m.keshavarz@desa-co.com

Seyed Ali Jazayeri

Faculty of Mechanical Engineering,
K. N. Toosi University of Technology
jazayeri@kntu.ac.ir

Corresponding Author*
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Abstract

Homogenous Charge Compression Ignition (HCCI) is a mode of combustion in IC engines in which premixed fuel and air are ignited spontaneously. There is a belief that HCCI engines have a great potential to improve fuel consumption and reduce NOx emissions.

In this study, a single zone, zero dimensional, thermo-kinetic model has been developed and a computer program with MATLAB software is used to predict engine performance characteristics. This model has been used to predict the principal parameters of controlling auto-ignition to acceptable level and this work leads to achieving the working region with two limitations for knock and misfire. The cycle is simulated with premixed blend of methane and DME with air. Also, NOx emissions are compared in a diesel engine working as a conventional diesel and at HCCI mode.

Keywords: HCCI Engine, EGR, IVC, NOx, Emission, Lubricity

Introduction

HCCI engines have a potential to reduce about 15 to 20 percent in fuel consumption and they can have a significant reduction in NO_x emissions compared with conventional SI engines [1]. Despite the potential advantages of this mode of combustion, there are some problems with the HCCI combustion. The most important challenge of these engines is having a limited operating region between knock and misfire and also is significant amount of CO and UHC emissions.

To overcome the challenges facing the HCCI engines, some concepts should be developed to control combustion phasing and heat release rate.

All these concepts are limited by choosing the base fuel. For the gasoline fuel, the higher octane number needs more preheating of mixture and it causes lower volumetric efficiency and lower power density. The lower octane number also leads the engine to knock; therefore, the working region becomes smaller.

For the diesel fuel, the situation is vice versa. Moreover, low volatility of diesel fuel makes it difficult to form a premixed homogeneous charge [2].

Some concepts are developed by many researchers to overcome these problems. But natural gas is a significant fuel in some applications, such as power generation because of its lower cost or availability of gas pipeline network. The homogenous mixture formation is the most important feature of gas fuel and makes it the favorite fuel for the HCCI engines.

Nevertheless, natural gas has a narrow operating region and achieving appropriate combustion phasing is difficult because the auto-ignition in a short time near TDC (to have the best efficiency and low emissions) causes pressure to exceed design limits [3]. This issue is more critical in power generation application in which high power density and the boost pressure about 2.5 to 3 bar is needed because this pressure makes the operating region narrower than that for naturally aspirated engines.

To control combustion timing and to decrease the initial temperature of mixture, we can blend another fuel to change the kinetic of methane/air mixture [4].

DME is already used as a blending agent to improve the ignition delay of diesel fuels and there is an interest to replace the diesel fuels with DME in a diesel engine. The cost of transportation, storage, and conditioning of DME is almost twice as expensive as that of the diesel fuel. But

there is no need for NO_x absorber or particulate trap for DME fuel and the common rail injection system for DME is cheaper than that for diesel fuel. Therefore, the overall cost is nearly the same [5]. However, because of high volatility and low lubricity of DME, this fuel is usually used as a blend.

It was shown that blending DME and natural gas expands the working region of HCCI [6]; also, it improves the manifold temperature needed to achieve auto-ignition.

In this study, the effect of adding DME to the homogenous mixture of the methane and air is investigated via a zero dimensional thermo-kinetic model. Using this model, the effects of significant parameters such as the initial temperature, the initial pressure and the residual gas level on auto-ignition and combustion are investigated. Then, an appropriate region is selected for electrical power generation application.

The most important feature of stationary engines is working on constant speed that is often held between 1000 r/min to 1500 r/min in which the HCCI combustion is easily achieved.

The International Organization for Standardization (ISO) has organized the operating cycles for electrical power generation applications. In this study, two standard operating cycles including continuous power (COP) and prime power (PRP) are considered and the operation of a stationary diesel engine in both cases of the conventional diesel and the HCCI with CH₄/DME fuel has been determined.

Modeling of hcci combustion

The modeling of HCCI engines has already been investigated in different ways:

- zero dimensional thermo-kinetic modeling [7]
- quasi dimensional thermo-kinetic modeling [8]
- sequential CFD based multi-zone modeling [9]
- experiment based multi-zone modeling [10]
- coupled kinetic- CFD modeling [11 and 12]

A zero dimensional model is selected to study the engine performance in an electrical power generation application because of the Simplicity of zero dimensional models and their ability to predict the performance of HCCI engine [13].

This model has been developed for the closed cycle (from intake valve closing to exhaust valve opening) with full chemical kinetic considerations both in compression

and expansion strokes.

The following assumptions are considered for this single zone model:

- all the **chemical species are considered as ideal gas**
- the **mixture is considered to be homogenous** and there is no **spatial gradient throughout the combustion chamber**
- the homogenous mixture of fuel and air is formed just before IVC

The **thermodynamic system** of the combustion chamber for which the governing equation has been developed is shown schematically in Figure 1.

For this system, the conservation law of energy in differential form is:

$$\delta Q - \delta W = dU \quad (1)$$

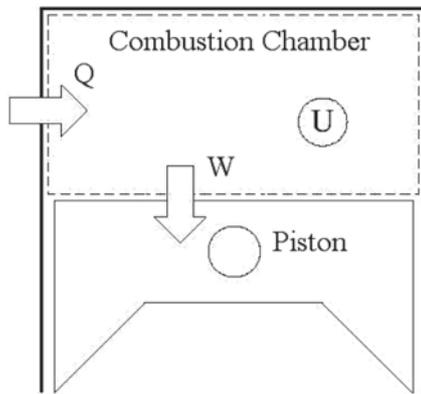


Fig 1. Thermodynamic consideration of combustion chamber

The internal energy of a system consisting of the mixture of ideal gases is:

$$U = \sum_{i=1}^{Ns} m_i u_i \quad (2)$$

With differentiating the above equation, we have:

$$dU = \sum_{i=1}^{Ns} (m_i du_i + dm_i u_i) \quad (3)$$

The specific internal energy of each species with the ideal gas assumption is:

$$du_i = c_{v,i} dT \quad (4)$$

Considering the **work, heat transfer** and mass fraction of each species, **as well as the time differential equation** form of 1st law of thermodynamic is:

$$\frac{dq_{ht}}{dt} - p \frac{dV}{dt} = m \bar{C}_v \frac{dT}{dt} + m \sum_{i=1}^{Ns} u_i \frac{dy_i}{dt} \quad (5)$$

The equation (5) can be rewritten to present the variation of the mixture temperature:

$$\frac{dT}{dt} = \frac{\frac{1}{V} \left(\frac{dq_{ht}}{dt} - p \frac{dV}{dt} \right) - \rho \sum_{i=1}^{Ns} u_i \frac{dy_i}{dt}}{\rho C_v} \quad (6)$$

From the conservation law of chemical species:

$$\frac{dy_i}{dt} = \frac{M_i \dot{\omega}_i}{\rho} \quad (7)$$

And from the kinematics of crank and slider mechanism:

$$\left\{ \begin{aligned} V &= V_c \left[1 + \frac{r_c - 1}{2} \right. \\ &\left. \left(R + 1 - \cos \theta - \sqrt{R^2 - \sin^2 \theta} \right) \right] \end{aligned} \right. \quad (8)$$

Chemical kinetics model

The production rate of each species is:

$$\dot{\omega}_i = \sum_{k=1}^{Nr} (v''_{i,k} - v'_{i,k}) (RP)_k \quad (9)$$

The rate of the progress variable or the k th reaction is given by the difference of the forward and reverse rates are:

$$(RP)_k = K_{f,k} \prod_{i=1}^{Ns} [x]_i^{v'_{i,k}} + K_{r,k} \prod_{i=1}^{Ns} [x]_i^{v''_{i,k}} \quad (10)$$

for elementary reactions and

$$\left\{ \begin{aligned} (RP)_k &= \left(\sum_{i=1}^{Ns} a_{i,k} [x]_i \right) \\ &\left(K_{f,k} \prod_{i=1}^{Ns} [x]_i^{v'_{i,k}} - K_{r,k} \prod_{i=1}^{Ns} [x]_i^{v''_{i,k}} \right) \end{aligned} \right. \quad (11)$$

for third-body reactions.

The rate constants for the reactions are generally assumed to have the following Arrhenius temperature dependence:

$$K_k = A_k T^{\beta_k} \exp\left(-\frac{E_k}{R_u T}\right) \quad (12)$$

All the above equations related to the chemical kinetics and rate expressions (E&S.9 to 12) are derived from chemkin4.0 Theory manual [13].

The Arrhenius constants for both forward and reverse reaction and thermodynamic properties of each species are derived from a file which is downloaded from reaction mechanism developer's website.

In this study, the GRI mechanism [14] is used to predict methane oxidization and NO_x emissions in addition, and LLNL mechanism for DME oxidization [15] is used to predict DME oxidization process.

In this paper, the Cantera [16] which is an open-source, object-oriented software package for problems involving chemically-reacting flows and has an interface to work with MATLAB is used. This interface and free cost availability of Cantera makes it a suitable alternative option for CHEMKIN.

The input files which are downloaded from the related website are converted into the xml format that can be read by Cantera.

Heat transfer model

Among the existing heat transfer models suggested for IC engines, the Woschni model is used because of its simplicity.

$$\begin{cases} h = 129.8 B^{-0.2} P^{0.8} T^{-0.55} [2.28 S_p + f(P)]^{0.8} \\ f(P) = 3.34 \times 10^{-4} \frac{V_d T_r}{P_r V_r} (P - P_{mot}) \end{cases} \quad (13)$$

The motored pressure is derived from the following differential equation:

$$\frac{dP_{mot}}{dt} = \frac{(\gamma_{mot} - 1)}{V} \frac{dq_{ht,mot}}{dt} - \gamma_{mot} \frac{P_{mot}}{V} \frac{dV}{dt} \quad (14)$$

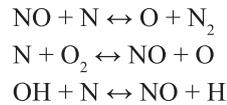
Finally, the cylinder pressure derived from the ideal gas law:

$$P = \rho R_g T \quad (15)$$

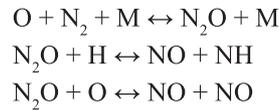
Nox emission model

In order to model NO_x formation for the HCCI combustion, both the extended Zeldovich mechanism and the N₂O mechanism have been used.

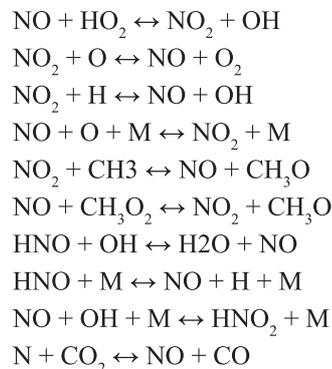
Extended Zeldovich Mechanism is:



N₂O Mechanism is:



In GRI mechanism, some additional reactions are proposed that present the role of NO in changing heat release behavior. These equations are [3]:



Model validity

Figures 2 and 3 show the validity of this model compared with the experimental results by Assanis and Fiveland for CH₄ fuel. Similar to all zero dimensional models, the pick cylinder pressure is higher than the experimental results. Also, maximum pressure occurs earlier. This is the main disadvantage of zero dimensional models that is originated from considered uniform temperature and uniform composition of charge throughout the cylinder.

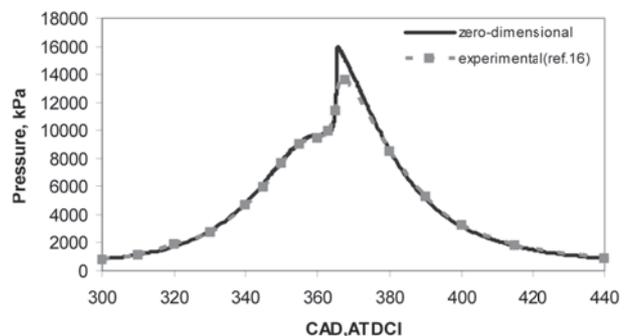


Fig 2. Model validation with the experimental results [17]

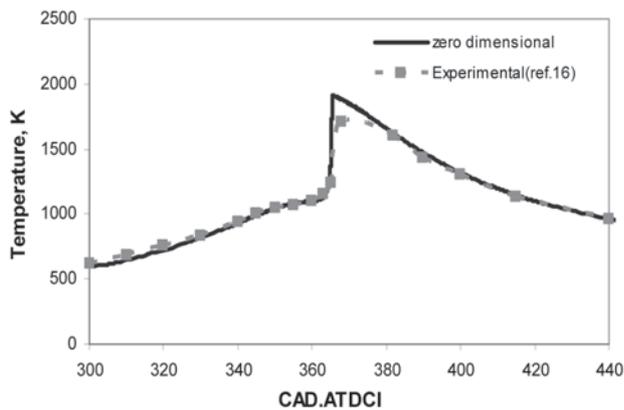


Fig 3. Model validation with the experimental results [17]

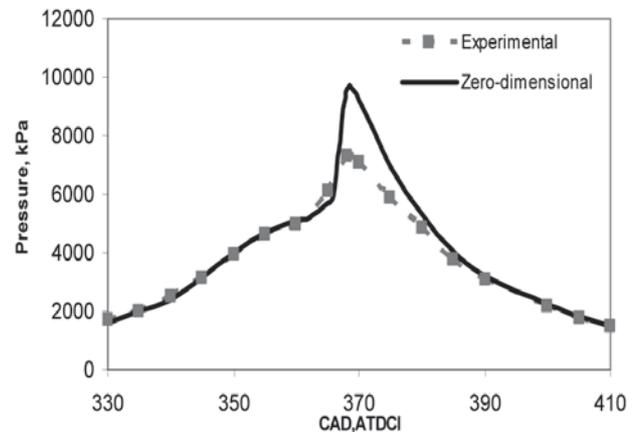


Fig 4. Model comparison with Experimental results [5]

For the mixture of DME and CH₄, the results are compared with the experimental data [6]. The simulation program has been run with a blend of natural gas (92% methane, 5% ethane, 1.5% n-butane, 0.4% CO₂ in Volume percent.) and DME. The mixture is composed of 65.1% natural gas and 34.9% DME.

Tables 1 and 2 show the specifications of the engines used in the above references.

Table 1. Specification of the engine used in Reference 16

Bore (mm)	Stroke (mm)	Speed (rpm)	CR
170	190	1500	17
IVC (ATDCI)	Equivalence ratio	Intake pressure (bar)	T ₀ (°C)
200	0.3	2	118

The Comparison of zero dimensional model and experimental results of ref 5 are shown in Figure 4. There is the same disadvantage for zero dimensional model as the previous, but the differences are more. This increase of differences between model and experimental results is because of the two-stage-pressure rise related to LTR and HTR. Anyway, the model can be used for performance analysis in order to find out combustion characteristics and evaluate the feasibility of HCCI combustion with DME and CH₄ mixture.

Table 2. Specification of the engine used in reference 5

Bore (mm)	Stroke (mm)	Speed (r/min)	CR
92	96	960	17.7
IVC (ATDCI)	Equivalence ratio	Intake pressure (bar)	T ₀ (°C)
220	0.36	1	28

Simulation results

The developed computer code for simulating HCCI combustion has been run with the engine specifications that are presented in table 3. This engine is currently under development in Iran heavy diesel engine Mfg (DESA) in order to improve the efficiency and exhaust emission of this engine.

It should be noted that in order to decrease the initial temperature needed for auto-ignition of natural gas, the compression ratio is assumed to be 14.5:1. In this work, the effect of the temperature at IVC, Pressure at IVC, internal EGR and equivalence ratio on auto-ignition of HCCI has been investigated. Also, it was assumed that the fuel consists 10% volumetric fraction of DME.

The effect of the initial temperature on the cylinder pressure and heat release, for the case in which the initial pressure is equal to 100 kPa is shown in figure 5 and figure 6. These figures show that the start of ignition will advance with increasing temperature because the higher temperature at IVC leads to higher compression temperature and it means that the auto-ignition temperature will achieve in advanced time. The bulk HR shape is mainly the same because of the same level of mixture dilution. But the last shape of heat release is a little different because the second stage of heat release period related to CO oxidation has hardly occurred.

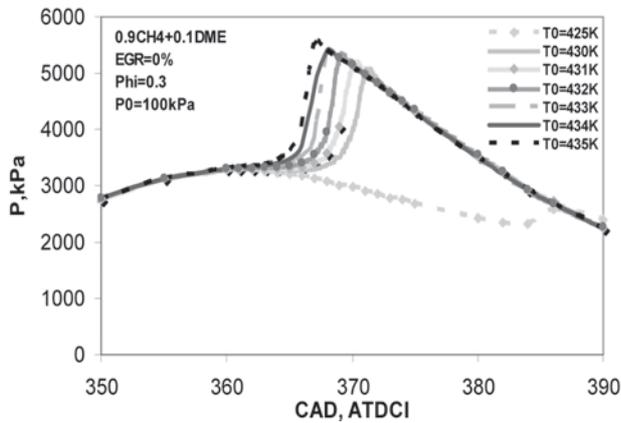
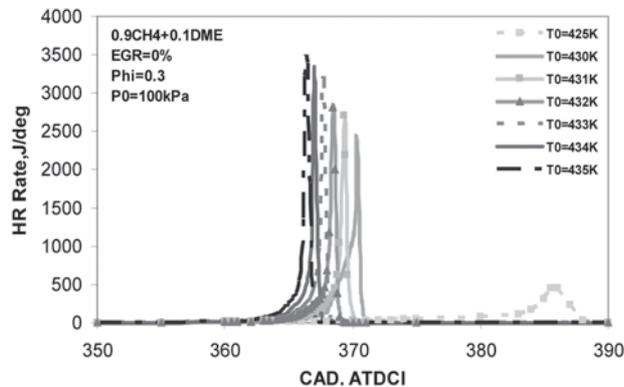
It should be noted that the reactions used for this study do not model LTR reactions well when the volumetric fraction of DME in fuel is increased more than 25%. this is the reason for differences in Figure 4.

In this study, no distinguishable LTR heat release are seen in figures due to small amount of DME in fuel.

Table 3. specification of selected diesel engine

Bore (mm)	Stroke (mm)	Speed (r/min)	CR
150	180	1500	13
IVC (ATDCI)	EVO (ATDCI)	Power (kW/Cyl)	
220	480	78	

The equivalence ratio considered above is overall equivalence ratio related to the blended fuel.

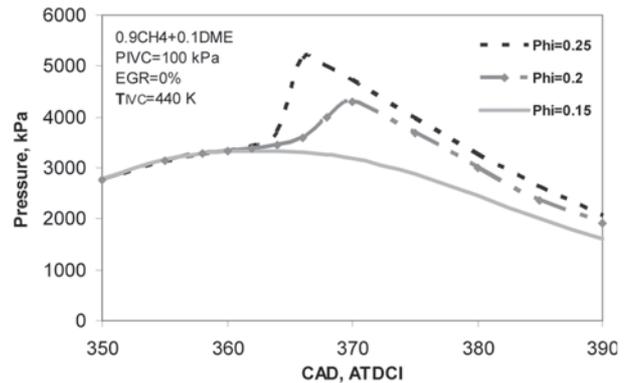
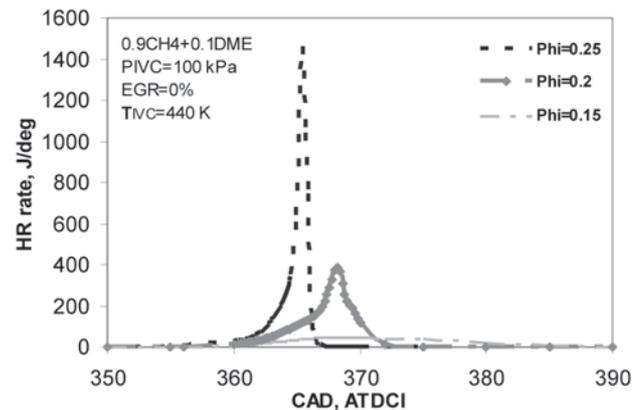
**Fig 5.** In-Cylinder pressure for $P_0 = 100$ kPa with different temperatures at IVC**Fig 6.** heat release rate for $P_0 = 100$ kPa with different temperatures at IVC

Figures 7 and 8 show the cylinder pressure and heat release rate with different equivalence ratio for initial temperature of 440 °K and initial pressure of 100 kPa. Increasing the equivalence ratio has no significant effect on advancing ignition initiation but it advances bulk heat release that results in increasing the heat release and decreasing the duration. At equivalence ratios more than 0.25, the engine tends to knock.

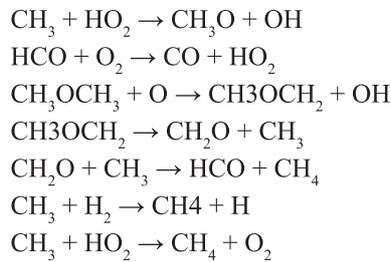
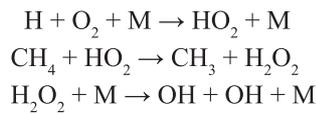
Further decrease in equivalence ratio hampers combus-

tion temperature and this would increase CO and UHC emissions. Therefore, we can define the operating region for each initial condition with limitations for knock and misfire. This issue is explained in detail later.

It should be noted that the start of ignition by definition is when two percent of the fuel is burned.

**Fig 7.** Cylinder pressure at $P_0 = 100$ kPa with different equivalence ratios**Fig 8.** HR rate at $P_0 = 100$ kPa with different equivalence ratios

Another parameter that can affect the combustion phasing is the amount of EGR. Increasing EGR causes the decrease of oxygen amount in the mixture that can lower the rate of the heat release, since the rate of low temperature chain branching reaction becomes lower due to the decrease in oxygen amount. Also, the mean specific heat of the mixture with EGR is higher than no EGR mixture and this has enhanced the above issue and these two factors cause a delay bulk combustion. The main low temperature chain branching for CH₄ and DME includes these two sets of chemical equations [15]:



In homogeneous charge compression ignition (HCCI) engines, H_2O_2 forms at lower temperatures and remains relatively inert until increasing temperature from compression and exothermic reactions reaches a level where it decomposes rapidly via reaction $H_2O_2 + M \rightarrow OH + OH + M$. Due to the activation temperature for set 2 of the above reactions (related to DME oxidization), the low temperature reaction mechanism for DME is similar to set 1 equations.

Figures 9 and 10 show the cylinder pressure and heat release rate with three percentage of EGR. The 20% EGR means that 20 percent of exhaust gases taken as CO_2 and H_2O are trapped in the cylinder after the intake valve closing event. The oxygen amount is considered in equivalence ratio calculation.

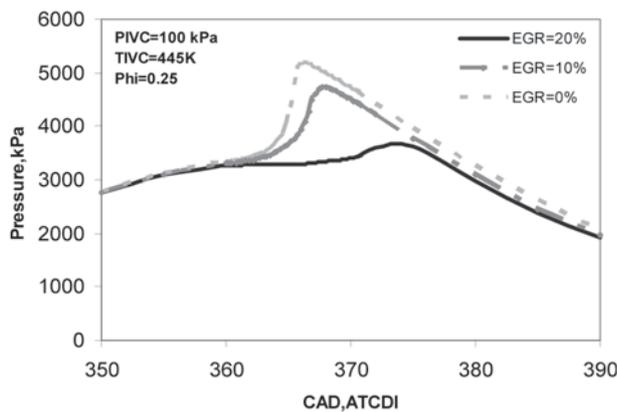


Fig 9. cylinder pressure with different EGR amount

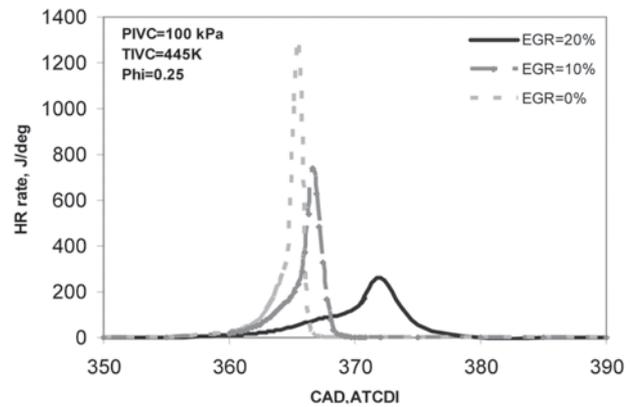


Fig 10. Heat release rate with different amount of EGR

The last parameter that affects the heat release is the initial pressure. Higher initial pressure results in higher compression pressure; therefore, the ignition advances and the main heat release occurs earlier. This results in higher heat release at TDC. Figures 11 and 12 show the effect of the initial pressure on the cylinder pressure and heat release rate.

Extensive work has been carried out to find the working region in which the engine never shall will knock or misfire. In this study, the knock criterion was selected to be the rate of the increasing pressure resulted from combustion. The new revision of the base turbocharged diesel engine has been designed with the stress limit of about 14 bar/deg. But we consider this limit to be 20 bar/deg compared with some new diesel engines [18]. The misfire limit has been selected for the case in which no sensible pressure rise resulted from combustion happens.

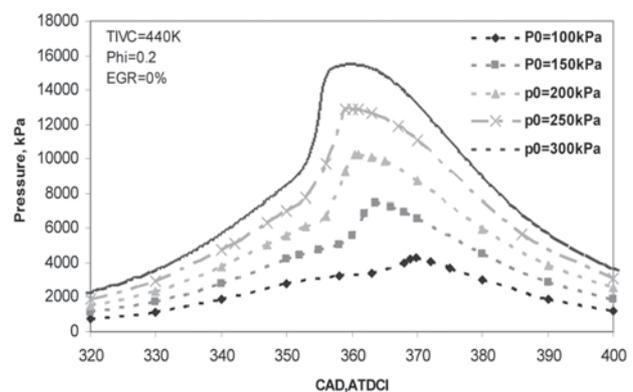


Fig 11. Cylinder pressure with different pressures at IVC

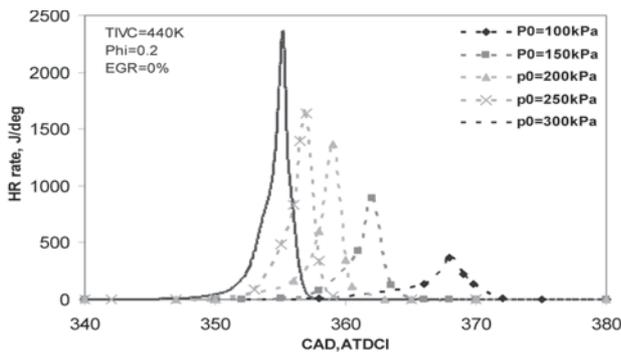


Fig 12. Heat release rate with different pressures at IVC

Figures 13 and 14 show the effect of EGR the intake pressure on knock and misfire limit. The left limit is the one for misfire and the other limit, in the right side, is knock limit. These figures show that increasing the amount of EGR leads to extension of operating regime in which more initial temperature can be used, for a constant equivalence ratio, to achieve controlled auto-ignition. This is because of the role of EGR in decreasing heat release rate. Also, this is clear that increasing equivalence ratio makes the engine go toward the uncontrolled auto-ignition. Uncontrolled auto-ignition refers to both knock and unstable combustion or misfire.

Another parameter that can affect the operating regime is the intake pressure. Increasing the intake pressure makes the operating regime narrower, but it has an advantage that it needs lower temperature.

Working on locations near misfire limit needs lower intake temperature but it leads to CO and UHC emissions. Because in these cases the maximum temperature in the cylinder is low, the combustion efficiency is low, too. On the other hand, although working near the knock limit has the best combustion efficiency, there is a risk of engine knock; also, a higher intake temperature is needed.

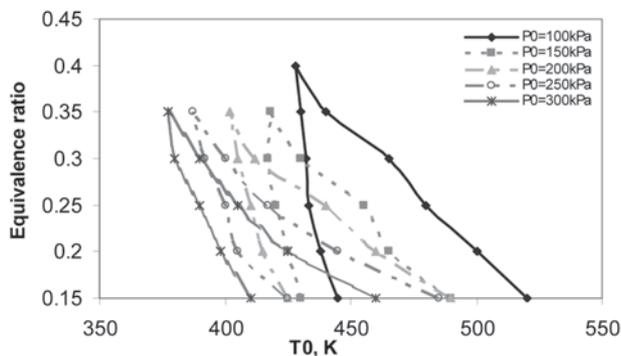


Fig 13. Operating regime of HCCI 0.9CH4 + 0.1DME (EGR = 10% and different intake pressure)

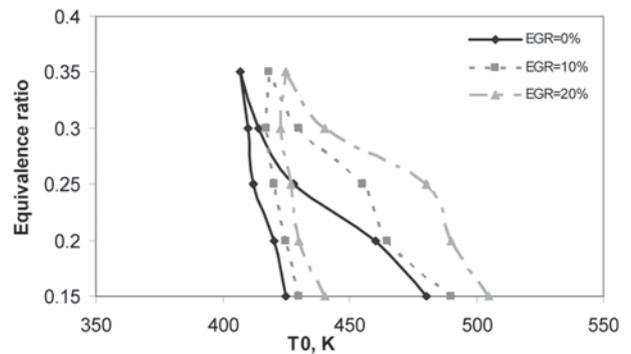


Fig 14. Operating regime of HCCI 0.9CH4 + 0.1DME (P0 = 150 kPa and different EGR rate)

Using the model described in Modeling Section, the in-cylinder NOx emission also has been estimated. The results of the simulation are shown in figures 15- 17 for the initial pressure of 100 kPa.

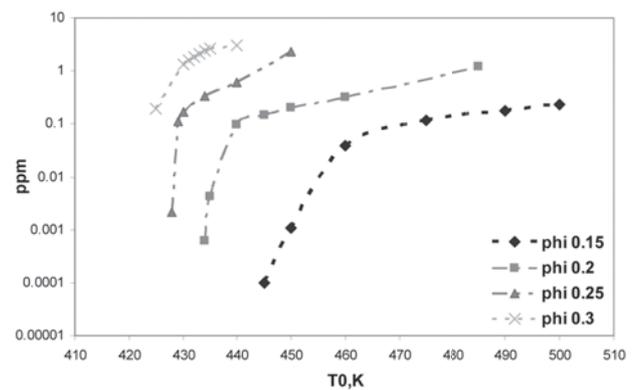


Fig 15. NOx concentration with 100 kPa initial pressure and 0% EGR

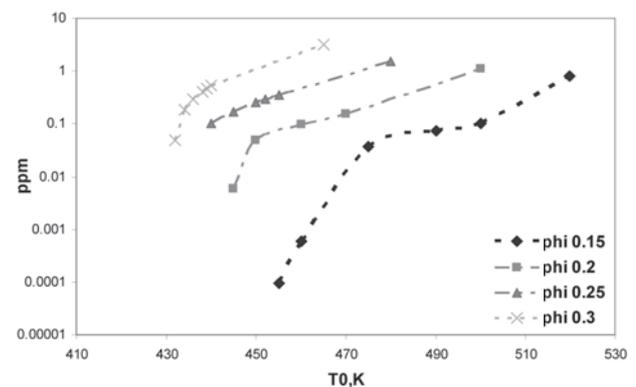


Fig 16. NOx concentration with 100 kPa initial pressure and 10% EGR

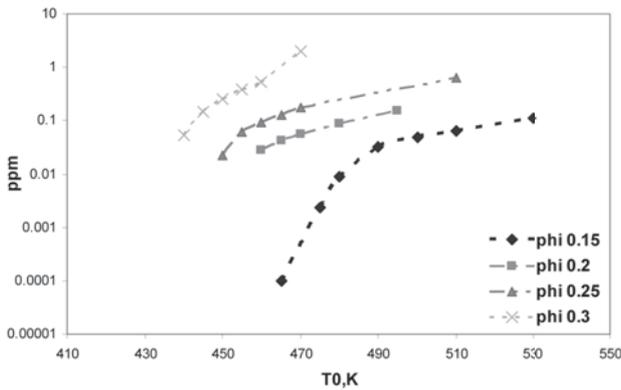


Fig 17. NOx concentration with 100 kPa initial pressure and 20% EGR

Also, figures 18 and 19 represent dry NOx concentration with different initial pressure and 10% EGR rate.

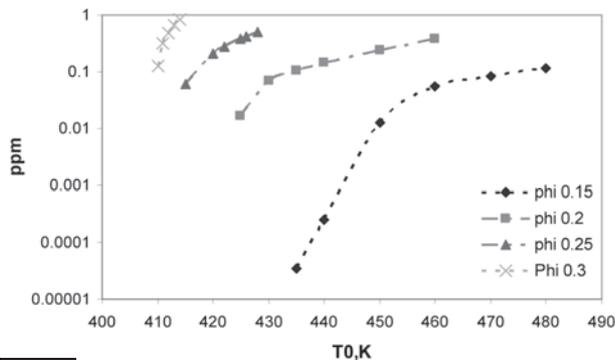


Fig 18. NOx concentration with 150 kPa initial pressure and 10% EGR rate

These figures show that increasing the initial pressure and EGR will decrease the amount of NOx while increasing the initial temperature will increase the amount of NOx, exponentially. Also, the low amount of NOx (in some cases less than 1 ppm) compared with the NOx amount in the current large diesel engines (about 1300 ppm) reveals the importance of HCCI combustion.

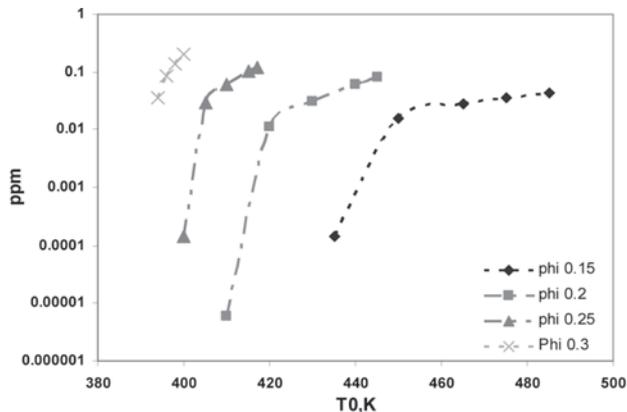


Fig 19. NOx concentration with 250 kPa initial pressure and 10% EGR rate

Standard operating cycle

In order to choose the engine working points for generating electrical power, some considerations should be noted:

- The value of the engine power density should be the maximum possible. The initial pressure is selected to maximum level so that maximum pressure does not exceed design limitations.
- The maximum cylinder temperature has a significant effect on CO and UHC emission levels so that these levels grow considerably when the maximum temperatures are below 1600 K.
- The maximum pressure timing should be located some degrees after TDC to gain the best efficiency.

Figure 20 shows the maximum temperature for 300 kPa initial pressure and 10% EGR rate. Another curve that may be used to find working points is the curve for timing of maximum pressure over initial parameters.

Figure 21-23 shows the output power and maximum pressure timing with different initial pressure and 10% EGR.

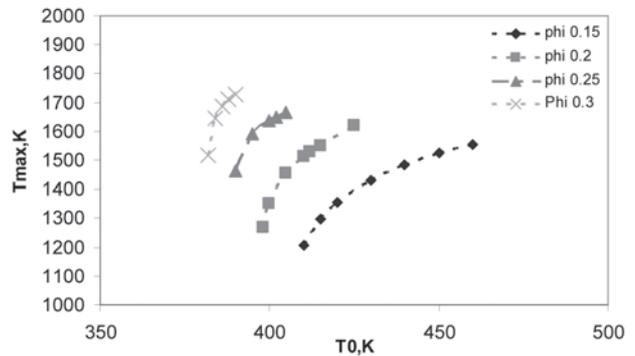


Fig 20. Maximum cylinder temperature for EGR = 10% and 300 kPa initial pressure

Considering all the above points, a set of working points is selected for the engine to work in HCCI mode. Table 4 shows this set of working points and corresponding performance and emission parameters.

The frictional mean effective pressure is selected between 1.4-1.6 bar, corresponding to related maximum pressure and constant speed of the engine from experimental correlation of DESA.

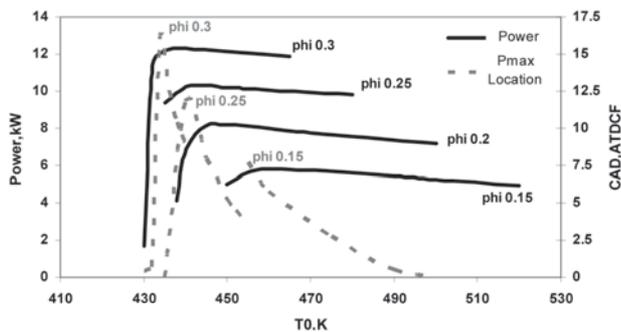
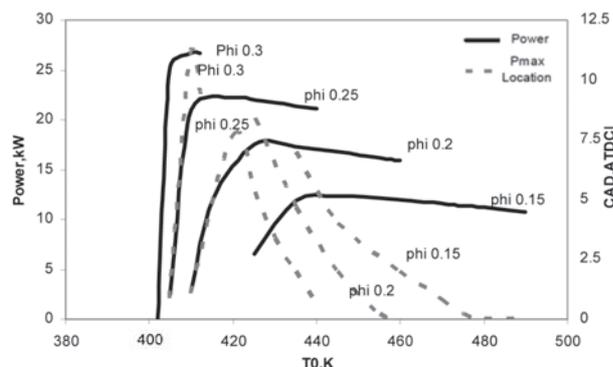
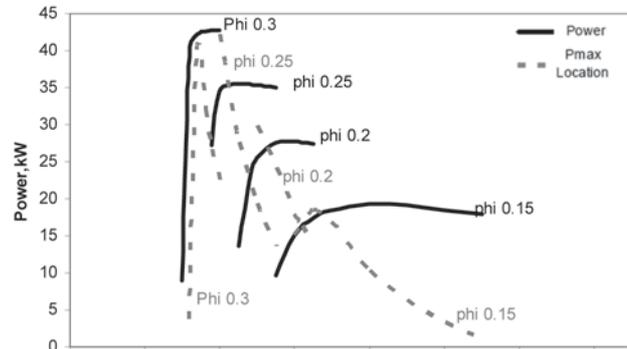
The Diesel mode working point is also shown in table 5.

Table 4. the selected set of working points in HCCI mode

%Load	T0,K	Equivalence ratio	EGR	P0,bar
100.0	398	0.3	0.2	3
72.60	408	0.25	0.1	2.5
52.57	420	0.25	0.1	2
32.71	430	0.25	0.1	1.5
20.19	438	0.3	0.1	1
%Load	bNOx (g/kWh)	bPower (kWb)	Tmax (K)	Pmax (kPa)
100.0	5.0046 E-04	32	1654	14680
75.00	5.7393 E-04	23.8	1640	11950
50.00	1.0806 E-03	16	1670.5	9820
35.00	1.5775 E-03	11.2	1681	7132
20.00	7.0248 E-03	6.4	1793.5	4778

Table 5. Diesel mode specification (closed cycle only)

%Load	Speed	bPower (kWb)	Bmep (bar)	Bsfc (g/kWh)	bNOx (g/kWh)
100.0	1500	75	19	200	7.12
75.00	1500	56.25	14.25	202	7.09
50.00	1500	37.5	9.5	218	7.25
20.00	1500	15	3.8	289	5.48


Fig 21. Indicated power and location of maximum pressure (EGR = 10% and P0 = 100 kPa)

Fig 22. Indicated power and location of maximum pressure (EGR = 10% and P0 = 200kPa)

Fig 23. Indicated power and location of maximum pressure (EGR = 10% and P0 = 300 kPa)

Conclusion

The results could be summarized as follows:

- The results of zero dimensional model shows that this model is suitable for performance analysis and it should be extended to quasi dimensional model in which the emission level and heat release phasing are estimated more accurately.

- By comparing the results of blended fuel of methane and DME with the results for pure methane HCCI in literature [17], it is noted that DME will improve the operating region and will decrease the initial temperature needed for achieving auto-ignition.

- Increasing the amount of EGR leads to extending the operating regime in which for a constant equivalence ratio, more initial temperature can be used to achieve controlled auto-ignition.

- Higher intake pressure is suitable for achieving acceptable power density of the engine but it results in a narrower operating regime and higher NOx level.

- DME is suggested to be introduced through DME injection during intake or compression stroke to reduce the complexity of fueling system and to increase volumetric efficiency meanwhile aid to form a homogenous mixture.

Nomenclature

A	Arrhenius coefficient
a	Third-body coefficient
ATDC	After top dead center
ATDCF	After top dead center (Firing)
ATDCI	After top dead center (Intake)
BDC	Bottom Dead Center
CAD	Crank angle degree
COP	Continuous Power
C_v	Specific heat in constant volume
E	Activation energy
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Opening
HCCI	Homogenous Charge Compression Ignition Engine
HR	Heat release
IVC	Intake Valve Closing
DME	Di-Methyl Ether
GRI	Gas Research Institute
HTR	High temperature reactions
K	Rate constant
LLNL	Lawrence Livermore National Laboratory
LTR	Low temperature reactions
m	Mass
M	Molecular weight
N_s	Total number of chemical species
P	Pressure
PRP	Prime Power
Q	Heat transferred via system boundaries
R _g	Gas constant
R	Ratio of crank radius to con-rod length
RP	Rate of Progress
r_c	Compression ratio
R_u	Global gas constant
q_{ht}	Heat transfer
T	Temperature
t	Time
TDC	Top Dead Center
U	Internal energy of the working fluid
UHC	Unburned Hydrocarbons
u	Specific internal energy
V	Volume

V_c	Clearance volume
Xml	Extensible Markup Language
X	Mole fraction
y	Mass fraction
W	The work done by the system

Greek letters

β	Arrhenius coefficient
γ	Specific heats ratio
θ	Crank angle
ρ	Density
ν'	Forward stoichiometric coefficient
ν''	Reverse stoichiometric coefficient
$\dot{\omega}$	Production rate

Subscripts

f	forward
i	species index
k	reaction index
mot	motoring
r	reverse

References

1. F. Zhao, T. Asmus, D. Assanis, J. Dec, J. Eng, P. Najt, "Homogenous charge compression ignition engines: key research and development issues", PT-94 SAE international
2. M. Keshavarz, "Performance of Homogenous Charge Compression Ignition (HCCI) engine with wide variety of fuels by means of a single zone Thermo-Kinetic simulation", MSc. Thesis, K. N. Toosi University of Technology, November 2006
3. S. Soyulu, "examination of combustion characteristics and phasing strategies of a natural gas HCCI engine", Energy conversion and management, issue 46 (2005), pp 101-119
4. S. Fiveland, R. Agama, M. Christensen, B. Johansson, J. Hiltner, F. Mauss, D. Assanis, "Experimental and simulated results detailing the sensitivity of natural gas HCCI engines to fuel composition", SAE paper No. 2001-01-3609
5. AVL powertrain technologies, inc., "DME as an Automotive Fuel: Technical, Economic & Social Perspectives", Energy Frontiers Conference
6. Z. Chen, M. Konno, M. Oguma, T. Yanai, "Experimental study of CI natural-gas/DME homogenous charge engine", SAE paper No. 2000-01-0329
7. S. Fiveland, D. Assanis, "A four-stroke homogeneous charge compression ignition engine simulation for combustion and performance studies", SAE paper No. 2000-01-0332
8. Scott B. Fiveland, Dennis N. Assanis, "Development of a two-zone HCCI combustion model accounting for boundary layer effects", SAE paper 2001-01-1028
9. S. M. Aceves, D. L. Flowers, C. K. Westbrook, J. R. Smith, W. Pitz, R. Dibble, M. Christensen, B. Johansson, "A multi-zone model for prediction of HCCI combustion and emissions", SAE paper 2000-01-0327
10. R. Ogink and V. Golovitchev, "Gasoline HCCI Modeling: An Engine Cycle Simulation Code with a Multi-Zone Combustion Model", SAE paper 2002-01-1745
11. T. Miyamoto, A. Harada, S. Sasaki, H. Akagawa, K. Tujimura, A. K. Hayashi, "A computational investigation of premixed lean diesel combustion characteristics of fuel-air mixture formation, combustion and emissions", SAE paper No. 1999-01-0229
12. S. Hong, D. Assanis, M. Wooldridge, "Multi-dimensional modeling of NO and Soot emissions with detailed chemistry and mixing in a direct injection natural gas engine", SAE paper No. 2002-01-1112
13. "Chemkin 4.0 theory manual", Reaction Design, May 2004
14. "http://www.me.berkeley.edu/gri_mech/"
15. "<http://www-ms.llnl.gov/combustion/>"
16. "<http://www.cantera.org/ref/languages.html>"
17. S. Fiveland, D. Assanis, "Development and validation of a quasi-dimensional model for HCCI engine performance and emissions studies under turbocharged conditions", SAE paper No. 2002-01-1757
18. M. Christensen, B. Johansson, P. Amnéus, F. Mauss, "Supercharged Homogeneous Charge Compression Ignition", SAE paper No. 980787