Proceedings of the 2009 International Conference on

Chemical, Biological and Environmental Engineering

Singapore 9 – 11 October 2009

CBEE 2009

editor

Li Kai

Guizhou University, China

World Scientific

NEW JERSEY • LONDON • SINGAPORE • BEIJING • SHANGHAI • HONG KONG • TAIPEI • CHENNAI

AN EXPERIMENTAL STUDY ON THE EFFECT OF ENGINE SPEED VARIATION ON PREMIXED RATIO, EQUIVALENCE RATIO AND EMISSIONS OF DUAL FUEL HCCI ENGINE

M. R. KALATEH and M. GHAZIKHANI

Department of Mechanical Engineering, Ferdowsi University of Mashhad, P.O. Box No. 917751111, Mashhad, Iran

In this study, effect of engine speed variation on premixed ratio, equivalence ratio and emissions of a dual fuel HCCI engine are investigated. The experiments were conducted on a variable compression ratio (VCR) single-cylinder research engine with compression ratio of 17.5:1. Premixed gasoline was introduced to inlet manifold through a carburetor which is equipped with a needle screw to adjust the gasoline mass flow rate. Results show that the CO, HC and soot emissions increases by increasing the engine speed due to less homogeneity in the mixture and decreasing the overall reactivity, also the premixed fuel and therefore the equivalence ratio increases as a result of engine speed increase in HCCI combustion.

Keywords: Dual fuel HCCI engine, engine speed, equivalence and premixed ratios, emissions.

1. Introduction

Reducing exhaust emissions and increasing the fuel efficiency of internal combustion engines are of global importance. In this regard, diesel engines are attractive because of their relatively high efficiency. Unfortunately, they produce greater NO_X emissions and particulate matter (PM) than spark-ignition engines. However, because diesel combustion involves turbulent diffusion flames that produce locally rich low-temperature and locally lean high-temperature regions, it is difficult to reduce both NO_X and PM simultaneously and still maintain high efficiency. One way to achieve both of these objectives would be to use homogeneous charge compression ignition (HCCI) combustion [1]. HCCI was identified as a distinct combustion phenomenon about 30 years ago. The initial papers of what may be considered as the modern era of HCCI research was issued by Onishi et al. and subsequently by Noguchi et al. These researches first successfully applied HCCI concept to a two-stroke gasoline engine with high levels of recycled or trapped residual gas [2].The main concepts of HCCI are breathing premixed air/fuel mixture, as in conventional spark ignition (SI) engines, and ignition without a spark plug, as in conventional compression ignition (CI) engines [3]. Ideal HCCI combustion is characterized by the lean and low-temperature reactions that are initiated at multiple sites simultaneously without any flame propagation [4]. HCCI combustion is the process in which a homogeneous mixture is auto-ignited by the compression from the piston motion, so the fuel chemical kinetics plays a dominate role during the whole combustion process. Also, auto-ignition occurs at various locations in the combustion chamber (multi-point ignition), which may cause extremely high rates of heat release, and consequently, high rates of pressurization [5, 6]. Depending on the fuel, the auto-ignition can proceed in one-stage, two-stage or multi-stage ignitions. This confirms that the fuel chosen is crucial for the combustion process, since the chemical kinetic pathways can be considerably different [7]. Since the HCCI combustion is lean burn and occurs without flame propagation, it provides much lower combustion temperature than that of conventional SI and CI engines. As a result, HCCI combustion produces very low levels of NOx and particulate matter (PM) while maintaining high thermal efficiency. However, greater amounts of hydrocarbon (HC) and carbon monoxide (CO) emissions are released in HCCI engine relative to conventional SI and CI engines [3, 8-11].

HCCI combustion has been considered as a potential substitute for traditional engines. However, the HCCI combustion system has several challenges, including the controlling auto-ignition timing over a wide range of speeds and loads, the improvement of cold-start capability (especially for the HCCI engine with a low compression ratio or with high octane rating fuel), the expansion of operating range and meeting emission standards [12-14]. The lack of a well-defined ignition timing control has led researchers to explore a

AN EXPERIMENTAL STUDY ON THE EFFECT OF ENGINE SPEED VARIATION ON PREMIXED RATIO, EQUIVALENCE RATIO AND EMISSIONS OF DUAL FUEL HCCI ENGINE

M. R. KALATEH[1], M. GHAZIKHANI[1]

[1]*Department of Mechanical Engineering, Ferdowsi University of Mashhad, P.O. Box No. 917751111, Mashhad, Iran*

Abstract: In this study, effect of engine speed variation on premixed ratio, equivalence ratio and emissions of a dual fuel HCCI engine are investigated. The experiments were conducted on a variable compression ratio (VCR) single-cylinder research engine with compression ratio of 17.5:1. Premixed gasoline was introduced to inlet manifold through a carburetor which is equipped with a needle screw to adjust the gasoline mass flow rate. Results show that the CO, HC and soot emissions increases by increasing the engine speed due to less homogeneity in the mixture and decreasing the overall reactivity, also the premixed fuel and therefore the equivalence ratio increases as a result of engine speed increase in HCCI combustion.

Keywords: Dual fuel HCCI engine, engine speed, equivalence and premixed ratios, emissions

1. Introduction

Reducing exhaust emissions and increasing the fuel efficiency of internal combustion engines are of global importance. In this regard, diesel engines are attractive because of their relatively high efficiency. Unfortunately, they produce greater NO_X emissions and particulate matter (PM) than spark-ignition engines. However, because diesel combustion involves turbulent diffusion flames that produce locally rich low-temperature and locally lean high-temperature regions, it is difficult to reduce both NO_X and PM simultaneously and still maintain high efficiency. One way to achieve both of these objectives would be to use homogeneous charge compression ignition (HCCI) combustion [1]. HCCI was identified as a distinct combustion phenomenon about 30 years ago. The initial papers of what may be considered as the modern era of HCCI research was issued by Onishi et al. and subsequently by Noguchi et al. These researches first successfully applied HCCI concept to a two-stroke gasoline engine with high levels of recycled or trapped residual gas [2].The main concepts of HCCI are breathing premixed air/fuel mixture, as in conventional spark ignition (SI) engines, and ignition without a spark plug, as in conventional compression ignition (CI) engines [3]. Ideal HCCI combustion is characterized by the lean and low-temperature reactions that are initiated at multiple sites simultaneously without any flame propagation [4]. HCCI combustion is the process in which a homogeneous mixture is auto-ignited by the compression from the piston motion, so the fuel chemical kinetics plays a dominate role during the whole combustion process. Also, auto-ignition occurs at various locations in the combustion chamber (multi-point ignition), which may cause extremely high rates of heat release, and consequently, high rates of pressurization [5, 6]. Depending on the fuel, the auto-ignition can proceed in one-stage, two-stage or multi-stage ignitions. This confirms that the fuel chosen is crucial for the combustion process, since the chemical kinetic pathways can be considerably different [7]. Since the HCCI combustion is lean burn and occurs without flame propagation, it provides much lower combustion temperature than that of conventional SI and CI engines. As a result, HCCI combustion produces very low levels of NOx and particulate matter (PM) while maintaining high thermal efficiency. However, greater amounts of hydrocarbon (HC) and carbon monoxide (CO) emissions are released in HCCI engine relative to conventional SI and CI engines [3, 8-11].

HCCI combustion has been considered as a potential substitute for traditional engines. However, the HCCI combustion system has several challenges, including the controlling auto-ignition timing over a wide range of speeds and loads, the improvement of cold-start capability (especially for the HCCI engine with a low compression ratio or with high octane rating fuel), the expansion of operating range and meeting emission standards [12-14]. The lack of a well-defined ignition timing control has led researchers to explore a range of control strategies, such as variable valve timing (VVT), variable compression ratio (VCR), EGR and multi-pulse injection [12,15]. Now, it is generally accepted that the

Nomenclature	
HCCI	Homogeneous Charge Compression Ignition
EGR	Exhaust Gas Recirculation
BTDC	Before Top Dead Center

DI	Direct injection
CI	Compression Ignition
SI	Spark Ignition
r_p	Premixed ratio
h_{up}	Heating value of premixed fuel (kJ/kg)
$\dot{m}_p$	Mass flow rate of premixed gasoline (kg/s))
$\dot{m}_d$	Mass flow rate of injected fuel (kg/s)
h_{ud}	Heating value of diesel fuel (kJ/kg)
$\dot{m}_{aEGR}$	Mass flow rate of intake air with EGR (kg/s)
$\dot{m}_{EGR}$	Mass flow rate of EGR gases (kg/s)
ϕ	Equivalence ratio
MON	Measured octane number
$(A/F)_s$	Stoichiometric air-fuel ratio
UHC	Unburned hydrocarbon
CO	Carbon monoxide
CO_2	Carbon dioxide
NO_X	Oxides of nitrogen
PM	Particulate matter

EGR technique is an efficient way to control HCCI combustion. HCCI combustion can occur in internal combustion engines by varying the inlet air temperature and exhaust gas recirculation (EGR) fraction over a range of equivalence ratios [15,16].

EGR is widely used as the main method to depress the NO_X emission from diesel engines. Currently, EGR is also used as the basic method to control the ignition timing and burn rate of HCCI combustion [16]. EGR consists of many gaseous chemical species, which includes the main components of burned gases, CO_2, H_2O, N_2 and O_2, partial burned gases such as CO, particular matters, HCs and high temperature combustion products NO_X. Different species has different heat capacity and chemical reactivity, therefore has different effect towards ignition timing and heat release rate of HCCI combustion [17]. The application of EGR on HCCI combustion engine has a number of effects on the combustion process and emissions. The effect of EGR on HCCI combustion can be divided into three parts: a dilution effect (inert gasses present in the EGR), a thermal effect (heat exchange, thermal loss to the wall, EGR ratio mixture quality, EGR temperature) and a chemical effect. The diluting effect influences the auto-ignition process, by influencing directly the concentrations along with the kinetics. With the thermal effect one can influence the overall kinetics directly by the reaction rates. The chemical effect, however, is more complex. It not only influences the overall kinetics, but it also can change a specific reaction path [6, 16-18].

The objective of this study is to investigate the effect of engine speed variations on premixed ratio, equivalence ratio and emissions of a dual fuel HCCI engine using premixed gasoline.

2. Experimental setup and procedure

2.1. *Experimental setup*

In this study all experiments were conducted on a four stroke VCR single-cylinder naturally-aspired research engine with a displacement volume of 582 cm^3, which the test rig is TD43 model equipped by Techquipment Co. The engine has a bowel type piston with a bowel diameter of 41.5 mm. The base engine can operate as SI engine with the compression ratio in the range of 7 to 11, and also can convert to diesel type (spark plug is replaced by an injector) in the range of 14 to 18 for the compression ratio. Premixed gasoline is introduced to the inlet manifold by means of a carburetor which is mounted on 180 mm from intake valve upstream and equipped with a fuel adjustment needle screw. Diesel fuel is injected into the cylinder through an injector. The specifications of the test engine are listed in the table1.

The experimental apparatus is composed of electrical dynamometer, AVL-415 smoke meter, RE 205 Plint exhaust analyzer, which measures HC (as C_6), K-type thermocouples, 220 V single-phase 2kW intake air heater which placed in the air flow stream, external EGR system, adjustable coolant system and air mass flow meter (surge tank and orifice system). Fig. 1 shows the schematic diagram of the experimental apparatus.

The electrical dynamometer which is connected directly to the engine plays the role of rotating initiator in starting mode, and when running the engine, consumes the power output for generating electricity. The stator of dynamometer can rotate freely around its shaft and as a result of engine torque during power generation; it is forced out from the horizontal equilibrium position. Using a Newton-meter on a known-length beam, for

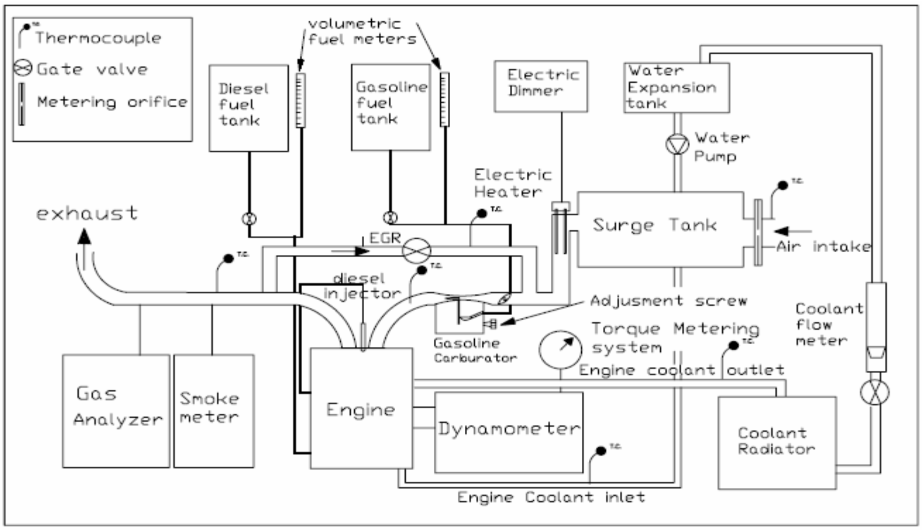

Figure 1: Schematic diagram of experimental apparatus

Table 1: Engine specifications

Engine	Four stroke
Number of cylinders	1
Compression ratio	17.5:1
Displacement volume	582 cm^3
Bore	82 mm
Stroke	95 mm
Injection mode	DI
Number of injection holes	4
Injection pressure (bar)	250
Gasoline fueling	Carburetor

bringing the dynamometer back to the horizontal, the engine torque is measured.

2.2. *Experimental procedure*

In this work, the diesel engine was firstly started at idle position in the compression ratio of 17.5:1. Then intake charge temperature was adjusted on 115 ^{0}C. After a few minutes to achieve steady condition at constant cooling temperature of 50 ^{0}C, premixed gasoline was introduced to engine and torque increased gradually. Then by using an external EGR system different rates of EGR were introduced to get HCCI auto-ignition close to TDC. Table 2 shows the specifications of the fuels used in dual fuel HCCI engine. It should be noted that using early injection (35BTDC) in HCCI engine in compare with conventional CI engine was the necessity of initiating the auto-ignition of dual fuel HCCI engine.

To show the effect of fuels in dual fuel HCCI engine, the premixed ratio (r_p) is defined as a ratio of premixed fuel energy (Q_p) to total energy (Q_t). It can be obtained from the following equation [1]:

$$r_p = \frac{\dot{Q}_p}{\dot{Q}_t} = \frac{\dot{m}_p h_{up}}{\dot{m}_p h_{up} + \dot{m}_d h_{ud}} \qquad (1)$$

In the equation (1), $\dot{m}_P$ represents mass flow rate of premixed gasoline, $\dot{m}_d$ is mass flow rate of injected fuel, hup is heating value of premixed fuel and hud is heating value of diesel fuel. Therefore, rp=1 corresponds to single fuel HCCI combustion and rp =0 corresponds to typical CIDI combustion.

EGR rate also is calculated as follow [25]:

$$EGR(\%) = \frac{\dot{m}_{EGR}}{\dot{m}_{EGR} + \dot{m}_{aEGR}} \times 100 \qquad (2)$$

Where, $\dot{m}_{aEGR}$ is mass flow rate of intake air with EGR, and $\dot{m}_{EGR}$ is mass flow rate of EGR.

Different EGR rates were applied in each test by using a simple computer code. The code can estimate the orifice pressure drop at specific EGR rate in term of engine speed, ambient conditions and intake air properties at orifice.

In this study the intake charge temperature was increased up to 115 ^{0}C, which helps the fuel to overcome its activation energy and improves the pre-ignition chemical reactions. Altering the coolant temperature at the 40 to 70 ^{0}C, HCCI-DI combustion showed better results at 50 ^{0}C. Therefore, coolant temperature was maintained 50 ^{0}C throughout all the tests.

Engine tests were carried out at the operating conditions shown in Table 3, trying to modify every one of the main parameters affecting the oxidation kinetics of any fuel, such as the engine speed, injection timing and the charge composition (quantified by the EGR percentage). All these parameter combinations provide different levels of pressure and temperature in the cylinder. As it can be observed in Table 3, HCCI conditions were achieved by an early start of injection (SOI) diesel fuel, which promote the auto-ignition of gasoline premixed fuel due to heat releases by diesel fuel before HCCI combustion of premixed fuel. Having reached HCCI combustion, emissions and other data were recorded.

Table 2: Fuel specifications [19].

Fuel type	Gasoline	Diesel fuel
MON	87	-
Cetane number	-	54
Higher Heating Value (kJ/kg)	47300	46100
Lower Heating Value (kJ/kg)	44000	43200
Heat of vaporization (kJ/kg), at 1 atm, 25 ^{0}C)	305	270
Density (kg/m^3)	720	780
$(A/F)_S$	14.6	14.5

Table 3:Test conditions

Speed (rpm)	1200-1700
Intake charge temperature (^{0}C)	110-115
Coolant temperature (^{0}C)	50
EGR rate (based on mass flow rate of intake air)	0-15%
Injection timing	35 BTDC
Premixed ratio (r_p)	0-1

3. Results

3.1. *Variation of equivalence ratio with engine speed variation*

The variation of equivalence ratio due to engine speed variation in different EGR rates of a dual fuel HCCI engine is shown in Fig. 2. As it can be seen, the equivalence ratio has increased with increasing the engine speed. Increasing the engine speed reduces the time needed to prepare a homogeneous mixture, also the probability of mixing the EGR species with the air-fuel mixture reduces. Therefore the reactivity of low temperature reactions is reduced and retarded, which causes lower advanced premixed gasoline HCCI combustion. In this situation, for the main HCCI combustion to take place, the premixed fuel or the equivalence ratio has been increased, which increases the total reactivity and energy produced by the low temperature reactions, though the main HCCI combustion is advanced.

The important point in Fig. 2 is the comparison between the range of engine speed and equivalence ratio of a dual-fuel HCCI engine, in presence of EGR and the case of no EGR used. Increasing the EGR and air dilution causes the maximum temperature of the cylinder to decrease. In comparison with the case of no EGR used, the knock happens in higher speeds and equivalence ratio, so the EGR expands the speed range of dual fuel HCCI engine. Also EGR expands the range of equivalence ratio in dual fuel HCCI engine.

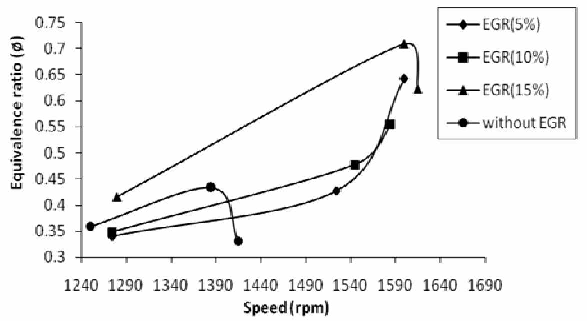

Figure 2: Variation of equivalence ratio with variation of engine speed in different EGR rates

3.2. *Variation of premixed ratio with engine speed variation*

In Fig. 3 the variation of premixed ratio in a dual fuel HCCI engine is shown on the basis of engine speed variations for different EGR rates. It can be seen that by increasing the engine speed, the premixed ratio and therefore the premixed fuel is increased to catch up the target speed. Higher premixed ratio provides more homogeneous charge before the premixed auto-ignition happens. The upper limit of premixed ratio is restricted by knock phenomenon. As it is shown in Fig. 3 by introducing EGR higher premixed ratios is obtained, due to lower maximum temperature of in cylinder charge. In other words, by increasing the premixed ratio higher homogeneity and better HCCI combustion is obtained.

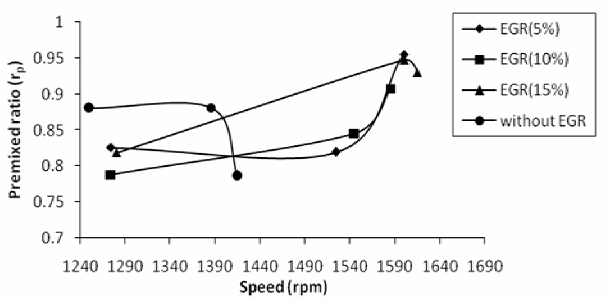

Figure 3: Variation of premixed ratio with variation of engine speed in different EGR rates

3.3. *CO emission*

Fig. 4 shows the variations of equivalence ratio and CO emission due to speed variations, in a dual fuel HCCI engine. Fig. 4 shows that increasing the engine speed provides less time for the homogeneous mixture to form, so at first the CO emission increases a bit due to incomplete combustion. But increasing the equivalence ratio causes lower CO emission levels due to more radicals produced (including OH radicals) and also more reactivity of the system.

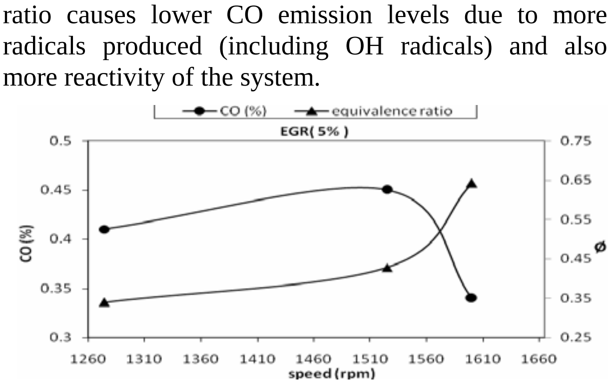

Figure 4: variations of equivalence ratio and CO emission with engine speed variation

3.4. *HC emission*

The variations of equivalence ratio and HC emission is shown in Fig. 5 on the basis of engine speed variations. Is can be seen that, increasing the engine speed, due to less homogeneity in the mixture and incomplete combustion, the HC emission increases. Also increasing the equivalence ratio provides more HC emission, because of more premixed fuel trapped in boundary layers and crevice which is hardly oxidized in low temperature HCCI combustion.

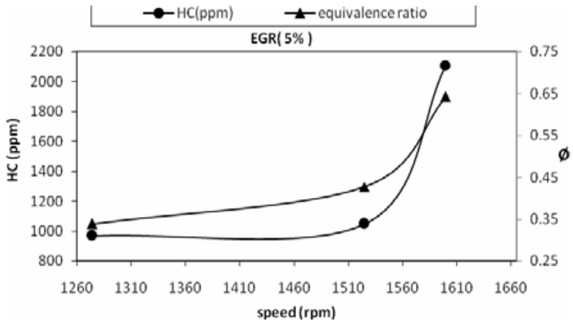

Figure 5: variations of equivalence ratio and HC emissions with engine speed variation

3.5. *Soot emission*

Fig. 6 shows the variations of equivalence ratio and soot emission on the basis of engine speed. It is shown that increasing the engine speed; the soot emission is increased due to less homogeneity and incomplete combustion.

In dual fuel HCCI engine the soot emission is produced in the first stage of the combustion, which is the diesel fuel auto-ignition. Increasing the engine speed provides less time for this product of diesel combustion to re-burn or participate in the premixed gasoline auto-ignition. So the soot emission of the engine increases.

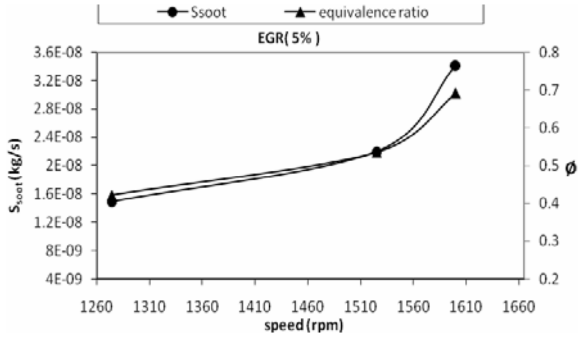

Figure 6: variations of equivalence ratio and soot emission with engine speed variation

4. Conclusions

In this study, the effect of engine speed variation on premixed ratio, equivalence ratio and emissions of dual fuel HCCI engine were investigated. Results can be summarized as:

1)Increasing EGR rate expands the range of speed and equivalence ratio in dual fuel HCCI engine due to dilution effect and lower maximum cylinder temperature.

2) The premixed fuel and therefore the equivalence ratio increases as a result of engine speed increase in HCCI combustion.

3) Increasing EGR rate causes higher premixed ratio and more homogeneity of the mixture, and improvement in HCCI combustion is obtained.

4) Increasing engine speed causes higher level of HC, CO and Soot emissions, due to less homogeneity in the mixture.

5) Increasing the equivalence ratio in parallel with the engine speed, causes lower level of CO emission as a result of increasing reactivity of the in-cylinder mixture, but HC emissions increased due to more premixed fuel trapped in crevice and boundary layers.

5. References

1. Sh. Tanaka, F. Ayala, J. C. Keck, J. B. Heywood, Two-stage ignition in HCCI combustion and HCCI control by fuels and additives, Combustion and Flame 132 (2003) 219–239.
2. X. C. Lu¨, W. Chen, Z. Huang, A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR. Part 1. The basic characteristics of HCCI combustion, International Journal of Fuel, 84, 1074-1083, 2005.
3. K. Yeom, J. Jang, Ch. Bae, Homogeneous charge compression ignition of LPG and gasoline using variable valve timing in an engine, Fuel 86 (2007) 494–503.
4. D. S. Kim, Ch. S. Lee, improved emission characteristics of HCCI engine by various premixed fuels and cooled EGR, International Journal of Fuel, 85, 695–704, 2006.
5. M. Jia , M. Xie, A chemical kinetics model of iso-octane oxidation for HCCI engines, Fuel 85 (2006) 2593–2604.
6. A. C. Alkidas, Combustion advancements in gasoline engines, Energy Conversion and Management 48 (2007) 2751–2761.
7. H. Machrafi, S. Cavadias, Ph. Gilbert, An experimental and numerical analysis of the HCCI auto-ignition process of primary reference fuels, toluene reference fuels and diesel fuel in an engine, varying the engine parameters, Fuel processing technology 89 (2008) 1007-1016.
8. D. Ganesh, G. Nagarajan, M. M. Ibrahim, Study of performance, combustion and emission characteristics of diesel homogeneous charge compression ignition (HCCI) combustion with external mixture formation, Fuel 87 (2008) 3497–3503.
9. J. Andrae, D. Johansson, P. Bjrnbom, P. Risberg ,G. Kalghatgi, Co-oxidation in the auto-ignition of primary reference fuels and n-heptane/toluene blends, Combustion and Flame 140 (2005) 267–286.
10. H. Machrafi, K. Lombaert, S. Cavadias, P. Guibert, J. Amouroux, Reduced chemical reaction mechanisms: experimental and HCCI modelling investigations of autoignition processes of iso-octane in internal combustion engines, Fuel 84 (2005) 2330–2340.
11. J. Ma, X. Lü, L. Ji, Z. Huang, an experimental study of HCCI-DI combustion and emissions in a diesel engine with dual fuel, International Journal of Thermal Sciences, 47, 1235-1242, 2008.
12. M. Yao, Z. Chen, Z. Zheng, B. Zhang, Y. Xing, Study on the controlling strategies of homogeneous charge compression ignition combustion with fuel of dimethyl ether and methanol, Fuel 85 (2006) 2046–2056.
13. S. Soylu, Examination of combustion characteristics and phasing strategies of a natural gas HCCI engine, Energy Conversion and Management 46 (2005) 101–119.
14. W. Zeng, M. Xie, M. Jia, Numerical investigation on the application of catalytic combustion to HCCI engines, Chemical Engineering Journal 127 (2007) 81–93.
15. L. Shi, Y. Cui, K. Deng, H. Peng, Y. Chen., study of low emission homogeneous charge compression ignition (HCCI) engine using combined internal and external exhaust gas recirculation (EGR), Energy 31 (2006) 2665–2676.
16. X. C. Lu¨, W. Chen, Z. Huang, a fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR. Part 2. Effect of operating conditions and EGR on HCCI combustion, International Journal of Fuel, 84, 1084-1092, 2005.
17. R. Chen, N. Milovanovic, A computational study into the effect of exhaust gas recycling on homogeneous charge compression ignition combustion in internal combustion engines fuelled with methane, International Journal of Thermal Sciences 41 (2002) 805–813.
18. H. Machrafi,“ Experimental validation of a kinetic multi-component mechanism in a wide HCCI engine operating range for mixtures of n-heptane, iso-octane and toluene: Influence of EGR parameters,’’ Energy Conversion and Management 49, 2008, 2956–2965.
19. J.B. Heywood, Fundamentals of Internal Combustion engine,’’ McGraw Hill, New York, 1988.