

Experimental investigation on the effect of equivalent ratio and engine speed on cyclic variation of DME HCCI combustion and performance parameters

Kanit Wattanavichien¹, Teerawut Rakyhao²

¹ Internal Combustion Engine Research Laboratory, Department of Mechanical Engineering
Faculty of Engineering, Chulalongkorn University
Phaya-Thai Road, Patumwan, Bangkok 10330, Thailand.

Phone: 0-2218-6610, Fax: 0-2252-2889, Email: wkanit@chula.ac.th¹, t.rakyhao@hotmail.co.th²

Abstract

Combustion in HCCI engines is a controlled autoignition of pre-mixed fuel, air and residual gas. Since there is no direct control on the start of combustion process, the onset of HCCI combustion depends on the auto ignition of fuel/air mixture. This experimental work is conducted to provide insight into HCCI combustion phasing for DME fuel using statistical techniques. In this study, combustion and performance parameters of HCCI engine fueled with DME were investigated using a modified single cylinder, four-stroke engine. Port fumigation mixer is used for preparing homogeneous charge for HCCI engine. The experiments were conducted with ambient intake air temperature and different air–fuel ratios at constant engine speed (1200, 1300 and 1400 rpm). For each test condition, P- θ diagram of 120 consecutive combustion cycles at steady state operation were recorded. Consequently, cycle-to-cycle variations of the main combustion parameters and performance parameters were analyzed. To evaluate the cyclic variations of DME HCCI combustion parameters, coefficient of variation (COV) and standard deviation of parameters were calculated for every engine operating condition. The experimental results show that the duration of low temperature reaction plays an important role on HCCI combustion, particularly at higher engine speeds. The low temperature reaction appearance, θ_{LTR} , and high temperature reaction appearance, θ_{HTR} , as function of the equivalence ratio for different engine speeds versus the crank-angle does not show high sensitivity to the equivalence ratio and to the engine speed. The knowledge of the duration of the different phases of combustion, as well as cycle-to-cycle variations phenomena of the main combustion and performance parameters, is useful for the definition of regulation strategies.

Keywords: HCCI, DME, low temperature reaction, high temperature reaction, cyclic variation.

1. Introduction

Di-methyl ether (DME) is an interesting alternative fuel because of its high Cetane number, low temperature of evaporation and low auto ignition temperature. This fuel is suitable for

homogeneous charge compression ignition engine (HCCI) that mixtures will spontaneous ignition without a spark plug or injection to control ignition angle of HCCI engine. HCCI engine is currently under widespread investigation due to

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its potential to lower NO_x and particulate emissions while maintaining high thermal efficiency. The HCCI engine can also operate with lean-mixture and the turbulence of combustion in each cycle is low. Ignition of homogeneous charge occurs when the temperature of combustion chamber reaches auto-ignition temperature of fuel-air mixture and the ignition delay period has elapsed after this. Therefore in HCCI combustion, the start of combustion is primarily governed by the chemical kinetics of auto-ignition. Thus, controlling the combustion timing requires the tuning of the auto-ignition kinetics, which is affected by in-cylinder charge composition and pressure and temperature histories of the reactants during the compression process [1,2].

In HCCI engine, unstable engine operation has often been observed at some engine operating conditions. At high loads, due to relatively higher temperature of residual gases and cylinder walls, early combustion and faster combustion velocity may lead to pressure oscillation, knock, in the combustion chamber. At light loads, lower initial in-cylinder temperature and leaner fuel/air mixture often results in large cyclic variations leading to misfire and partial combustion [3].

Cyclic variations in combustion process parameters have key role in combustion stability and operating range definition for the HCCI combustion engine [4]. Systematic study of cycle-to-cycle variations of HCCI combustion parameters is then essential for understanding the behavior of HCCI combustion. Therefore, this paper is aimed to demonstrate the result of systematic study of cycle-to-cycle variations of

combustion parameters using Di-methyl ether (DME) in a develop HCCI engine. The understanding the HCCI combustion can widely be supported of HCCI advanced control algorithms development in the future.

2. Combustion of DME fuel in HCCI engine.

The heat release of DME in HCCI engine can be separated into two periods: first is LTR (low temperature reactions) is similar to the knock of the spark ignition engine (SI engine). It has some of the energy release after period of delay time was combustion in combustion chamber in HTR (high temperature reaction). Most of the energy of combustion is released during the combustion of HCCI engine is shown in Fig.1.

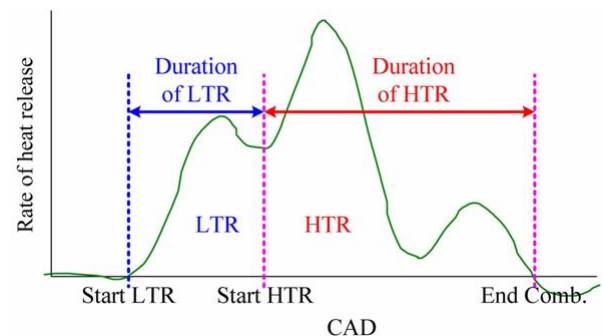


Fig.1 Definitions of LTR and HTR [5].

3. Properties of DME fuel

DME is an ether gas which has chemical formula $\text{C}_2\text{H}_6\text{O}$ or CH_3OCH_3 . The physical property is similar to LPG (Liquefied Petroleum Gas). The combustion of DME fuel has emerged blue flame that can be visible because DME fuel has higher Cetane number than diesel fuel. It is suitable to be used as a fuel in HCCI engine, which can provide a complete combustion and less particulate matter than petroleum diesel. The physical and chemical properties of DME are illustrated in Table 1. [6,7].

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Table.1 the properties of DME fuel [6,7]

Properties	DME
Chemical formula	CH ₃ OCH ₃
Mole weight [g/mol]	46.07
C ratio [%wt]	52.2
H ratio [%wt]	13
O ratio [%wt]	34.8
S ratio [%wt]	0
Stoich. A/F ratio	9 : 1
Boiling Point [°C]	-24.9
Explosion limit in air [vol.%]	3.4~17
Auto Ignition temperature [°C]	235
Liquid Viscosity [cP]	0.15
Liquid density at 20°C [kg/m ³]	668
Lower Heating Value [kJ/kg]	28430
Latent heat of evaporation [kJ/kg]	460
Vapor pressure @ 20°C [MPa]	0.51
Cetane Number	55-60

4. Equations

The coefficient of variation of “x” parameter, COV(x), is calculated for variation of 120 consecutive cycles for each test condition. COV(x) is a ratio between standard deviation of “x” divided by the mean of “x”, and is usually expressed in percent.

$$COV(x) = \frac{\sigma}{\bar{x}} \times 100 \quad (1)$$

Where

COV(x) = the coefficient of variation.

σ = Standard deviation

$$\bar{x} = \sum_{i=1}^n \frac{x_i}{n}$$

5. Experimental setup

The HCCI engine was used in this research. An agricultural engine, 4-stroke direct injection

Kubota model RT140, was converted into DME-HCCI engine by first determined the optimal compression ratio. It is the lowest compression ratio that can provide a high enough in-cylinder temperature to ignite the DME fuel (Start of Ignition) which occurs in the compression stroke at a position angle of the crankshaft. To get the maximum break torque found that the optimal compression ratio is 10.3. This compression ratio have a start of ignition angle is not too fast and temperature inside the cylinder was high enough to compensate for heat transfer to the cylinder wall. The engine in take port is also be modified to reduce a swirl will affect in heat loss is transferred to the cylinder wall and increase the flow around the perpendicular axis to the cylinder axis (Tumble). The technical specifications of the engine are shown in Table. 2. [8, 9]

Table. 2 Technical specifications of the engine. [8,9]

Engine	Kubota	DME-HCCI
Model	RT140	
Type	Single cylinder	
Combustion chamber system	Direct Injection (DI)	Compression ignition
Bore	97 mm. (bore)	
Stroke	96 mm. (stroke)	
Swept volume	709 cc.	
Maximum Power	14 hp / 2400 rpm	N/A
Maximum Torque	4.0 kg-m / 1600 rpm	N/A
Compression ratio	18 : 1	10.3:1
Fuel supply	Single spring Pintle nozzle 140 bar	Gas mixture (Ventury mixer)
Fuel system	Fuel pump Type: independent or piston Pressure: 600 bar	Pressure reducing device
Lubricant system	Rotary	
Cooling system	Natural Convection	
Air filter	Wet	Dry

In this study, the combustion cycle-by-cycle variation of using Di-methyl ether (DME) was

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investigated using a hydraulic dynamometer as a load to engine. The specification of dynamometer is shown in Table 3 [9, 10]. The equipment details and installation of equipment used in this study are shown in Fig 2.

Table. 3 Specification of dynamometer [9, 10]

Dynamometer	Hydraulic Dynamometer
Manufacturer	Redman Heenan International Company, England
Model	Froude Hydraulic Dynamometer (DPX2)
Resolution	0.1 kg
Balance Arm (L_B)	0.3525 m
Peak Power	150/7500 CV/rpm, (1 CV $\cong$ 0.986 hp)

The intake air mass flow rate was calculated, using an air box and orifice, from measuring pressure drop data of air flow across an orifice by a manometer.

The thermocouple type K (Chromel-Alumel, CA), 0.65 mm core diameter, was used to measure the temperature of the lubricant, coolant, intake and exhaust.

The engine speed is measured by OMRON detecting Proximity Sensor E2EY.

The ambient in the test room was measured using a thermometer and barometer.

The mass flow rate of DME fuel was measured by stopwatch and digital scale (a resolution is 0.002 kg). The time for each 0.01 kg of consumed DME fuel was recorded to the compute the rate of DME fuel consumption.

In-cylinder pressure data was measured by piezoelectric pressure transducer AVL model GU12P, which is installed in the combustion chamber. The position of the crankshaft angle measured by crank angle encoder products of Kislter 2613B. The cylinder pressure-crank angle

history were captured using a high-speed data acquisition, DEWE--5000 [11].

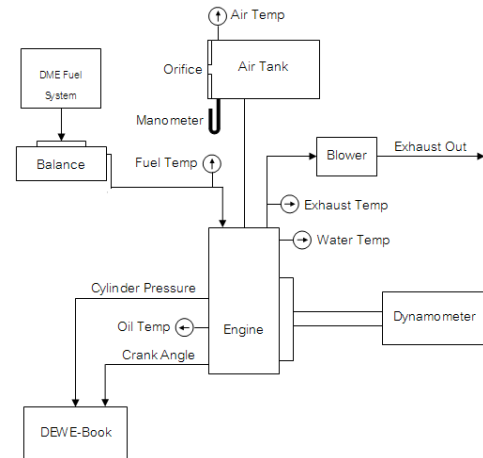


Fig.2 Schematic diagram of experimental system.

6. Experimental procedures

This research was testing at constant speed full load and wide open throttle (WOT) with different equivalent ratios for the engine speed of 1200, 1300 and 1400 rpm. During the test, the DME fuel supply pressure was fixed controlled by open a cylinder gas valve at 25% and set a regulate pressure regulator at 25% of Full Scale for every testing point. The DME fuel volume rate that supply to the engine, was controlled by regulating needle valve at 40%, 50%, 60% and 70% of full scale. The test will start by control DME fuel volume that supply to the engine at 70% Needle Valve and wait until the engine speed reach the maximum. After the engine speed is constant, applied load will be added on the engine bit by bit until the engine speed approach to 1400 rpm. When the engine speed is constant, the engine operating information such as torque, engine speed, engine operating temperatures (coolant, lubrication oil, exhaust, intake air), ambient temperature and pressure, fuel consumption, air flow rate will be recorded, The in-cylinder pressure-crank angle history for

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120 consecutive cycles were recorded. After finished the record at 1400 rpm, we will add more load on the engine for the testing at 1300 and 1200 rpm, respectively. The testing conditions should be operated without knock phenomena radically. Then, the engine performance test with different equivalent ratios by regulate the needle valve at 40%, 50% and 60% needle valve setting, respectively, in the same manner as a test at 70% needle valve were conducted. The recorded data were analyzed to evaluate the combustion cycle-by-cycle variation of using Di-methyl ether (DME). The experimental test matrix is shown in Table. 4

Table.4 The experimental test matrix

No.	Throttle valve (%)	DME Equivalence ratio	Needle valve (%)	Engine speed (RPM)
1	100	0.272	40	1200
2		0.352	50	
3		0.395	60	
4		0.333	50	1300
5		0.373	60	
6		0.415	70	
7		0.313	50	1400
8		0.344	60	
9		0.394	70	

7. Results and Discussions

In this section, the cycle-to-cycle variations of different equivalent ratios are presented and discussed.

It is need to investigate the cycle-to-cycle variations of IMEP as it affects the engine drivability. HCCI combustion engines have lower $COV_{(IMEP)}$ as compared to conventional SI engines. However, $COV_{(IMEP)}$ increases rapidly

with late combustion timings. Window for desired combustion timing is reduced at higher engine operating speeds.

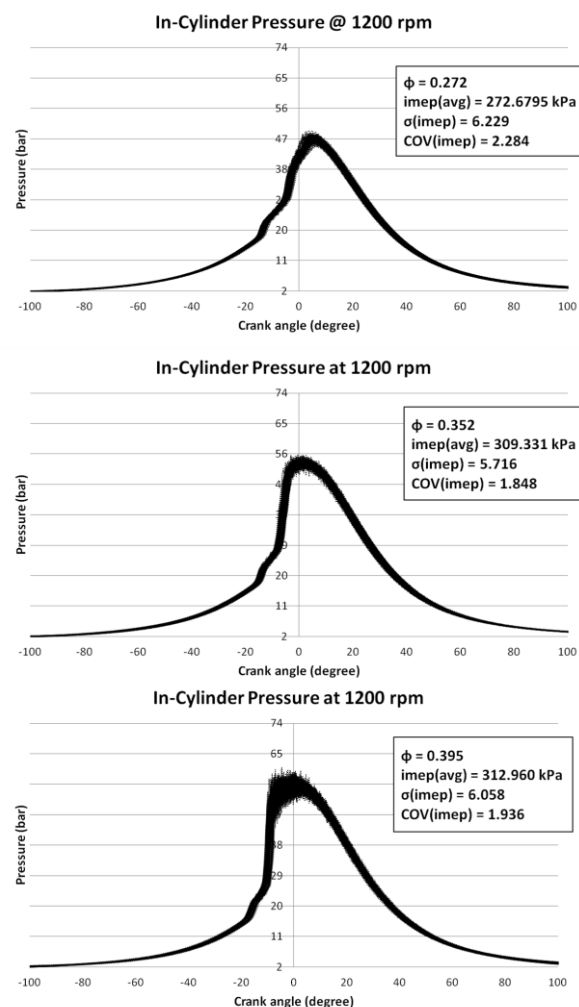


Fig. 3 Comparison of in-cylinder pressure and crank angle of 120 Cycles for different equivalent ratios at 1200 rpm

Fig. 3, 4 and 5 illustrate the cycle-to-cycle variations of IMEP at full load with different equivalent ratios for 120 consecutive combustion cycles at 1200, 1300 and 1400 rpm. It is observed that the IMEP(avg), average value of IMEP, decreases as engine is operated on leaner mixtures because smaller amount of fuel is burnt in the combustion chamber. It can be noticed from Fig. 3 to 5 that $COV_{(IMEP)}$ increases with decrease in equivalent ratio, leaner the mixture.

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The variations in IMEP, $\sigma(\text{imep})$, can also be seen to increase significantly as mixture becomes leaner. As mixture becomes leaner, the combustion is retarded.

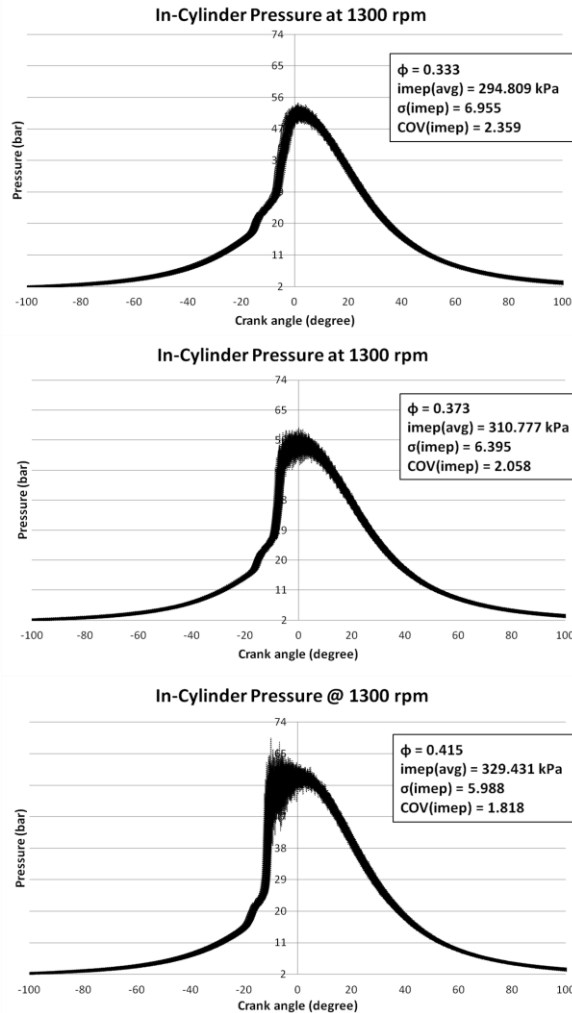


Fig. 4 Comparison of in-cylinder pressure and crank angle of 120 Cycles for different equivalent ratios at 1300 rpm

The mechanism for this increased $\sigma(\text{imep})$ can be explained as follows: as air-fuel mixture is compressed by piston, the mixture temperature increases causing the cylinder contents to react near TDC. As the piston passes TDC, the compressed mixture begins to expand and the temperature begins to decrease. The decreasing temperature slows the rate of the auto-ignition process, escalating the time for the combustible

mixture to release energy during the post combustion stage. As the rate of chemical kinetics declines exponentially with reduced temperature, any cycle-to-cycle variations in the mixture temperature at TDC results in significantly fluctuating reaction rates as the piston descends. A cycle of richer mixture with a relatively high mixture temperature at TDC will be somewhat unaffected by the piston descending. However, a cycle of leaner mixture with a relatively low temperature at TDC will have its rate of reaction significantly lowered and fluctuated by the reduced temperatures after TDC.

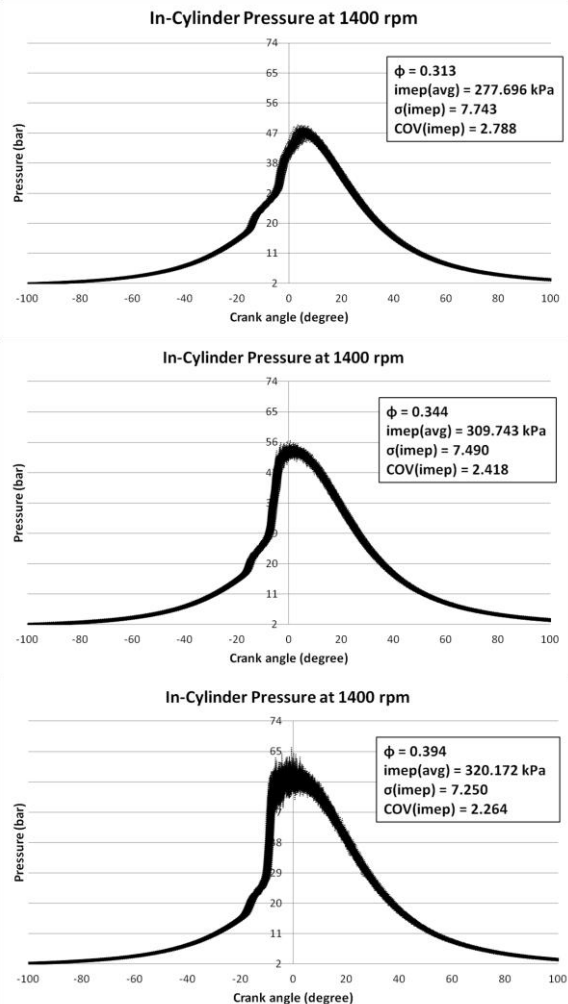


Fig. 5 Comparison of in-cylinder pressure and crank angle of 120 Cycles for different equivalent ratios at 1400 rpm

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Crank angle corresponding to the maximum pressure in the combustion chamber is an important combustion parameter, which can be used for estimation of combustion phasing. Due to high rate of heat release in HCCI combustion, it can be assumed to be a crank angle position corresponding to maximum pressure, which is in the vicinity of middle of the combustion duration. [12]

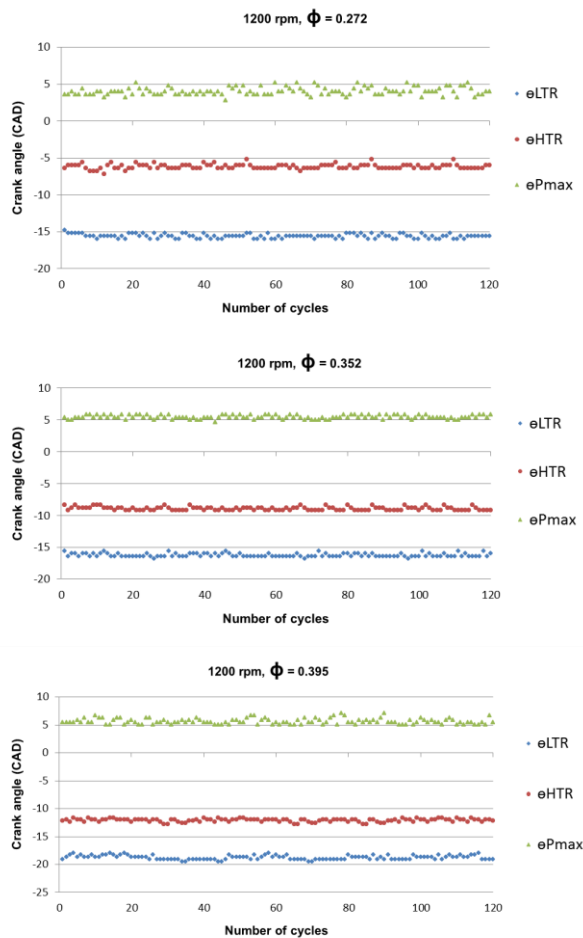
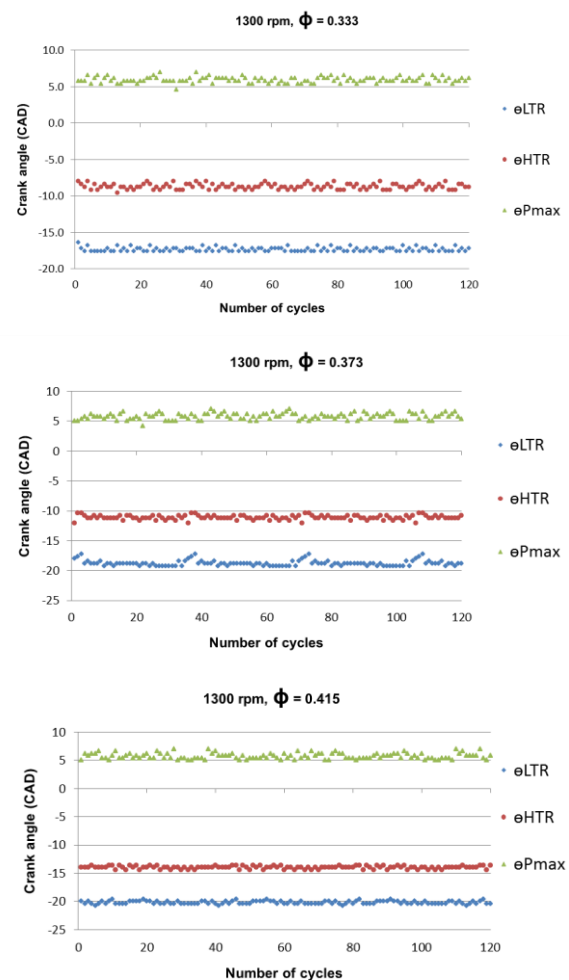


Fig.6 Effect of equivalent ratio on cycle to cycle variation of θ_{LTR} , θ_{HTR} and θ_{Pmax} at 1200 rpm

The cycle-to-cycle variations in ignition timing affect the crank angle position corresponding to maximum pressure. Fig. 6, 7 and 8, shows the full load cycle-to-cycle variations of crank angle corresponding to start of LTR, start of HTR and P_{max} (θ_{LTR} , θ_{HTR} and θ_{Pmax}) for DME with different

equivalent ratios for 120 consecutive combustion cycles at 1200, 1300 and 1400 rpm. DME has good characteristics of auto-ignition and combustion with low flame temperature and show two-stage heat release. Richer the mixture of DME has ability to promote the kinetics of auto ignition chemistry of low temperature heat release (LTR) which has a strong impact on high temperature heat release (HTR). It can be observed in Fig. 6 to 8 that the $\theta_{LTR}(avg)$ and $\theta_{HTR}(avg)$ occurs earlier as engine is operated on richer mixtures. The $COV(\theta_{HTR})$ decreases with increase in equivalent ratio, richer the mixture. The duration of LTR, onset time to HTR, also become shorter with increase in equivalent ratio, but they are longer as engine speed was increased.



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Fig.7 Effect of equivalent ratio on cycle to cycle variation of θ_{LTR} , θ_{HTR} and θ_{Pmax} at 1300 rpm

It can also be noticed from these figures that the crank angle position for Pmax is scattered more for either richer or leaner fuel–air mixtures because value of standard deviation is large for either richer or leaner mixture

In Fig. 6 to 8 also show that cyclic variations of crank angle at Pmax for leaner mixtures is higher. However, with the DME, the θ_{Pmax} (avg) occurred around 5 to 6 CA ATDC for almost all test conditions.

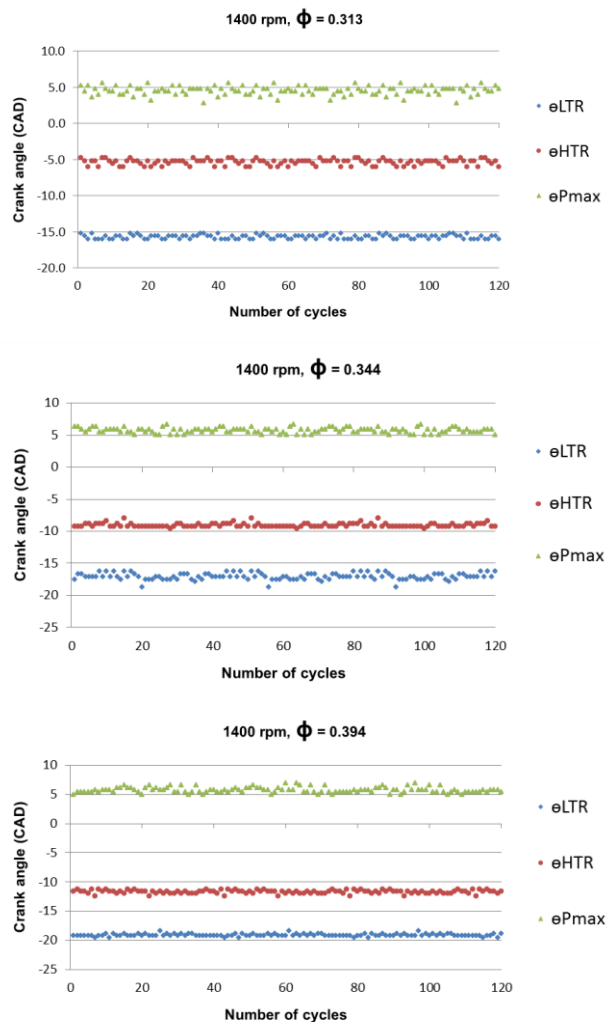


Fig.8 Effect of equivalent ratio on cycle to cycle variation of θ_{LTR} , θ_{HTR} and θ_{Pmax} at 1400 rpm

The combustion parameters in this investigation also extended to the pressure level

due to kinetic reaction in the engine cylinder. To recognize the effects of mixture strength on HCCI combustion process, this study was preferred to identify the outcome in-cylinder pressure due to kinetics of auto ignition chemistry of LTR and HTR (P_{LTR} and P_{HTR}) and their impact on the maximum in-cylinder pressure, P_{max} . This method uses only in-cylinder pressure signal to identify the effects of mixture strength on HCCI combustion process. Therefore, the in-cylinder pressure measured by a piezo-electric sensor can be directly used. The dispersions of P_{LTR} , P_{HTR} and P_{max} are shown in Fig. 9, 10 and 11.

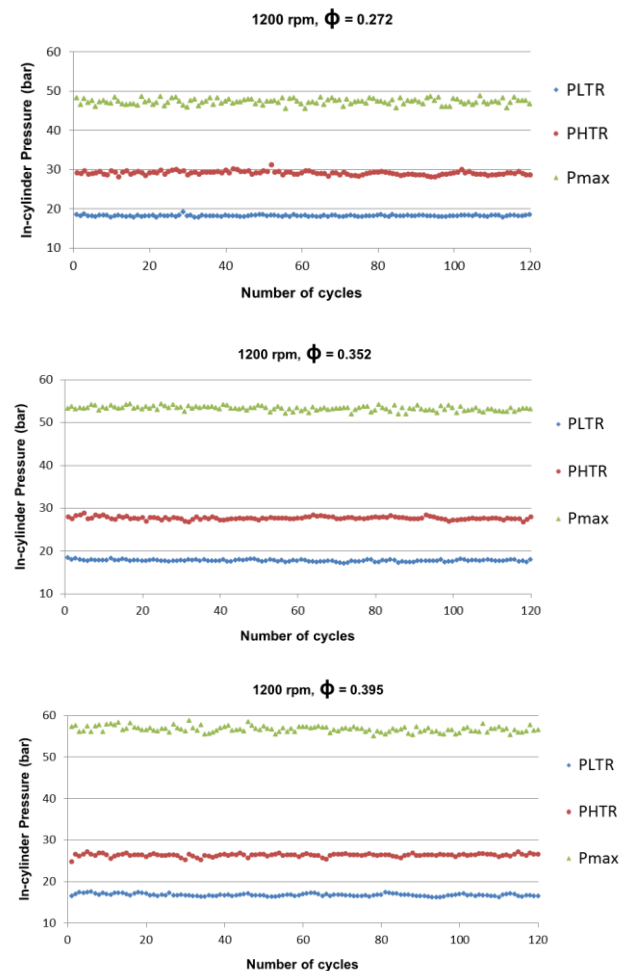


Fig.9 Effect of equivalent ratio on cycle to cycle variation of In-cylinder pressures (P_{LTR} , P_{HTR} and P_{max}) at 1200 rpm

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It can be observed in Fig. 9 to 11 that the richer the DME mixtures, the higher the P_{max} . Richer the mixture of DME has ability to promote the kinetics of LTR which has a strong impact HTR, thus the higher maximum in-cylinder pressure, P_{max} , can be achieved. As it was noted earlier that the start of LTR and start of HTR (θ_{LTR} , and θ_{HTR}) for DME occurs earlier as engine is operated on richer mixtures, therefore, the lower in P_{LTR} and P_{HTR} , respectively, will be. The duration of LTR (shorter onset time to HTR) is also shorter as equivalent ratio is increased, thus, the lower in different pressure between P_{LTR} and P_{HTR} can be observed. The more advance of ignition timing of the second heat release as richer engine DME-air mixture will reduce the $COV_{(IMEP)}$.

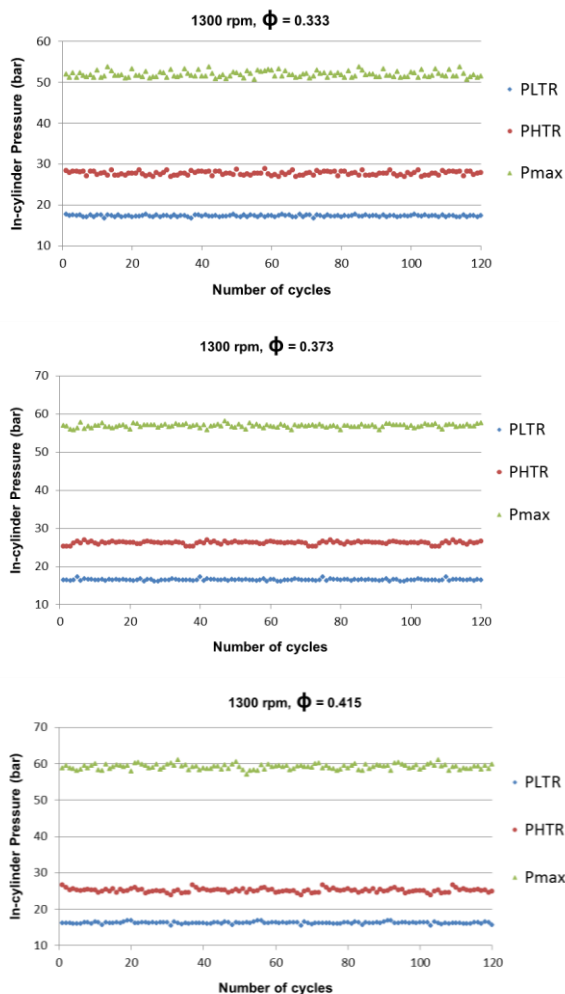


Fig.10 Effect of equivalent ratio on cycle to cycle variation of In-cylinder pressures (P_{LTR} , P_{HTR} and P_{max}) at 1300 rpm

Thus, the governor of DME-air mixture as a mean to control the timing of the second heat release considerably can be used to improve engine performance as well as to reduce the engine cycle to cycle variations.

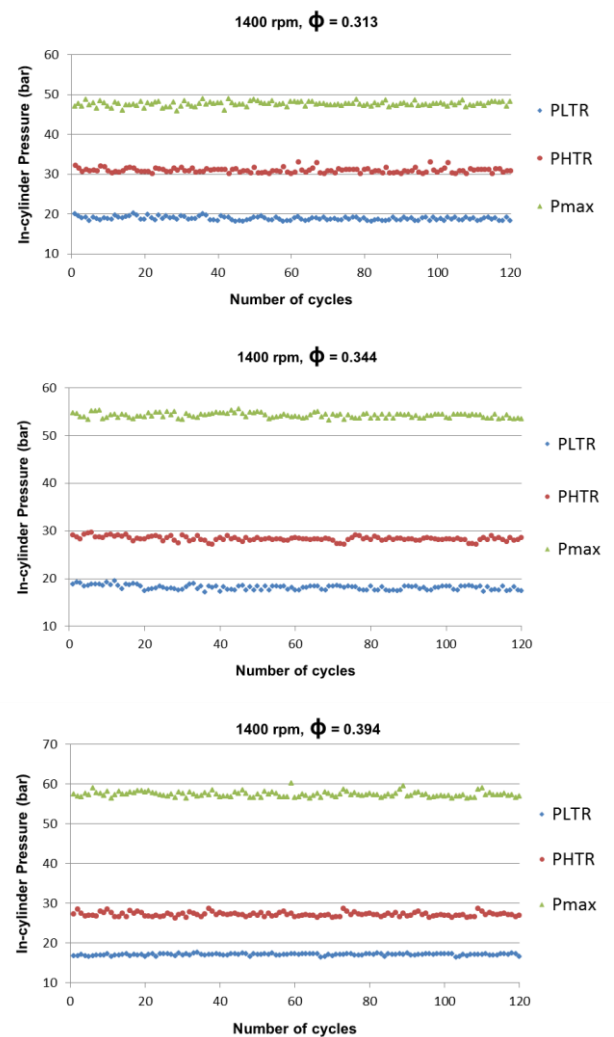


Fig.11 Effect of equivalent ratio on cycle to cycle variation of In-cylinder pressures (P_{LTR} , P_{HTR} and P_{max}) at 1400 rpm

8. Conclusions

As engine is operated on richer mixture:

- The P_{max} and $IMEP_{(avg)}$ increase at the same time as the variations in $IMEP$

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decrease significantly, therefore, the $\sigma(\text{imep})$ and $\text{COV}_{(\text{IMEP})}$ are lower.

- The $\theta_{\text{LTR}}(\text{avg})$ and $\theta_{\text{HTR}}(\text{avg})$ occurs earlier.
- The duration of LTR, onset time to HTR, also become shorter with increase in equivalent ratio, but they are longer as engine speed was increased.
- The $\text{COV}_{(\theta_{\text{HTR}})}$ decreases.
- The start of LTR and start of HTR (θ_{LTR} , and θ_{HTR}) for DME occurs earlier, therefore, the lower in P_{LTR} and P_{HTR} , respectively.

As mixture becomes richer, the combustion is advanced. Richer the mixture of DME has ability to promote the kinetics of auto ignition chemistry of low temperature heat release (LTR) which has a strong impact on high temperature heat release (HTR). Thus, the adjustment of DME-air mixture that is employed to control the timing of the second heat release can be used to improve engine performance as well as to reduce the engine cycle to cycle variations.

8. Acknowledgement

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