International Journal of Automotive Technology, Vol. 8, No. 4, pp. 395–402 (2007)

1229–9138/2007/035–01

COMBUSTION STABILITY OF DIESEL-FUELED HCCI

L. SHI*, K. DENG and Y. CUI

Key Laboratory for Power Machinery and Engineering of Ministry of Education,
Shanghai Jiaotong University, Shanghai 200030, China

(Received 25 August 2005; Revised 12 July 2007)

ABSTRACT–Homogeneous Charge Compression Ignition (HCCI) shows great potential for low NO_x emission but is hampered by the problem of no direct method to control the combustion process. Therefore, HCCI combustion becomes unstable easily, especially at lower and higher engine load. This paper presents a method to achieve diesel-fueled HCCI combustion, which involves directly injecting diesel fuel into the cylinder before the piston arrives at top dead center in the exhaust stroke and adjusting the valve overlap duration to trap more high temperature residual gas in the cylinder. The combustion stability of diesel-fueled HCCI combustion and the effects of engine load, speed, and valve overlap on it are the main points of investigation. The results show that: diesel-fueled HCCI combustion has two-stage heat release rate (low temperature and high temperature heat release) and very low NO_x emission, combustion stability of the HCCI engine is worse at lower load because of misfire and at higher load because of knock, the increase in engine speed aids combustion stability at lower load because the heat loss is reduced, and increasing negative valve overlap can increase in-cylinder temperature which aids combustion stability at lower load but harms it at higher load.

KEY WORDS : HCCI, Diesel fuel, Negative valve overlap, Combustion stability

NOMENCLATURE

IMEP : indicated mean effective pressure
BMEP : brake mean effective pressure
TDC : top dead centre
BTDC : before top dead centre
ATDC : after top dead centre
CoV_{pz} : coefficient of variance of in-cylinder peak pressure
Δp_z : difference between peak pressure and mean peak pressure
p_z : peak pressure
CoV_{imep} : coefficient of variance of IMEP
CAD : crank angle degree
Q_w : energy of heat loss
U : internal energy of mixture
V : cylinder volume at one crank degree
Q_b : energy released by fuel combustion
α_g : heat transfer coefficient
F_c : area
T_w : average temperature of cylinder wall
σ_{pz} : standard deviation of peak pressure
$\bar{p}_z$: mean value of peak pressure

Corresponding author. e-mail: shi_lei@sjtu.edu.cn

1. INTRODUCTION

Homogeneous charge compression ignition (HCCI) engines are being considered as a future alternative for diesel engines and for part-load operation in spark ignition (SI) engines (Ishibashi and Asai, 1996; Kim and Lee, 2006). This is because HCCI combustion has great potential to reduce NOx emission and maintain low PM emission of engines, while achieving higher thermal efficiency as compared with SI engines. HCCI engines operate on the principle of having a dilute, premixed charge that reacts and burns volumetrically throughout the cylinder as it is compressed by the piston. In some regards, it incorporates the best features of both SI and compression ignition (CI). As in an SI engine, the charge is well mixed, which minimizes particulate emission, and as in a CIDI engine, the charge is compression-ignition and has no throttling losses, which leads to high efficiency. However, the combustion of HCCI occurs automatically and simultaneously throughout the volume rather than ignited by a spark in the SI engine or controlled by fuel injection in the CI engine. Therefore, HCCI combustion cannot be controlled directly and the control of ignition timing and combustion rate are critical tasks for HCCI engines.

Several research groups all over the world have investigated HCCI combustion. Najt and Foster were the first to run a four-stroke engine in HCCI mode (Najt and Foster, 1983). The results showed that HCCI is controlled by chemical kinetics, with negligible influence from physical effects (turbulence, mixing). Thring investigated HCCI combustion by varying the inlet air temperature and exhaust gas recirculation (EGR) fraction over a range of equivalence ratios (Thring, 1989). Gao also studied the effects of EGR and split fuel injection on diesel engine emissions (Gao and Schreiber, 2001). Ryan and Callahan performed a more comprehensive study of a premixed diesel-fueled HCCI engine, looking at the operation range of different compression ratios (Ryan and Callahan, 1996). The UNIBUS scheme of early injection of diesel fuel also showed low NO_x and soot emissions compared with a base diesel engine (Yanagihara, 2001); Hu and Takeda *et al.* investigated early injection of diesel fuel using a novel injection scheme (Hu, 1990; Takeda and Keiichi, 1996) and in their studies a longer time ignition delay was achieved at lower and part load. In recent years, Su *et al.* (Su *et al.*, 2003) in China have achieved diesel-fueled HCCI combustion using a multi-pulse fuel injection system (FIRCRI) and a BUMP combustion chamber. Henrik Nordgren *et al.* in the Lund Institute of Technology have investigated the effects of injection timing, fuel pressure, and spray shape on HCCI combustion (Henrik *et al.*, 2004). In Germany, Pöttker *et al.* also achieved HCCI combustion using a multi-pulse fuel injection stratagem, but they controlled the engine load by increasing pulse number (Pöttker *et al.*, 2004).

The aim of this study was to investigate the combustion stability of a diesel-fueled HCCI engine at different loads, especially at lower and higher load. Currently, the narrow load range of the HCCI engine is one of the problems that have restricted the development of HCCI engines. As a result, it is important to study the combustion stability of HCCI engines at the limit load. In this paper, the effects of engine load, speed, and valve overlap on combustion stability were also studied.

Table 1. Specifications of test engine.

Bore×Stroke (mm)	135×150
Combustion chamber type	ω
Compression Ratio	14.8
Number of injection holes	5
Injection nozzle diameter (mm)	0.32
Fuel	Diesel
Fuel injection timing	Fixed at 10 °CA BTDC in exhaust stroke

2. EXPERIMENTAL APPARATUS AND PROCEDURE

In this study, diesel-fueled HCCI was achieved in a single cylinder four-stroke diesel engine (Shi *et al.*, 2005). The specifications of the test diesel engine are shown in Table 1. The fuel was injected directly into the residual in-cylinder gas during the negative valve overlap. A large amount of residual in-cylinder gas with high temperature aided fuel vaporization and mixing, and the homogeneous charge was prepared. When the mixture was compressed by the piston and the in-cylinder temperature increased to a high level, auto-ignition occurred. A schematic of the experimental system is shown in Figure 1. The experimental system mainly includes the exhaust gas recirculation system, inlet air heating system, and valve timing control system.

An auxiliary injector was installed in the cylinder head in addition to the original injector, because in engine cold-start and warm-up stages, the in-cylinder temperature is so low that HCCI combustion cannot be achieved. The auxiliary injector was used to inject pilot diesel fuel before TDC of the compression stroke. The pilot fuel was used to ignite the premixed fuel injected by the original injector at near TDC of the intake stroke. The injection mode is shown in Figure 2.

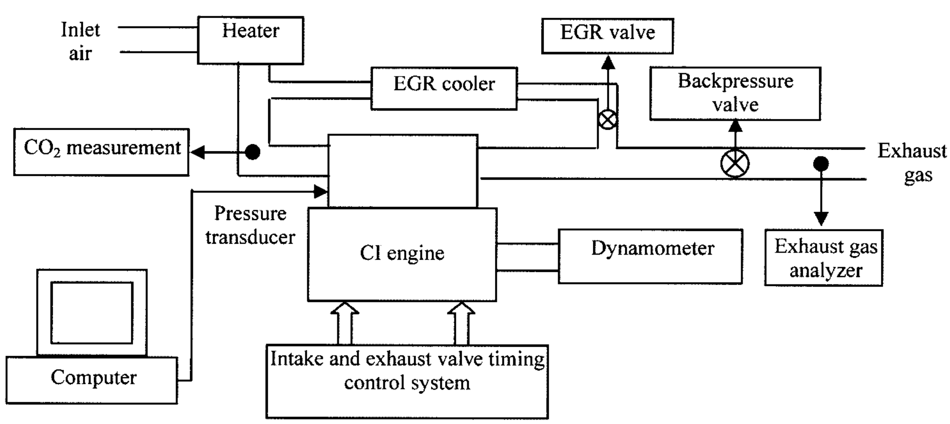


Figure 1. Schematic of the experimental system.

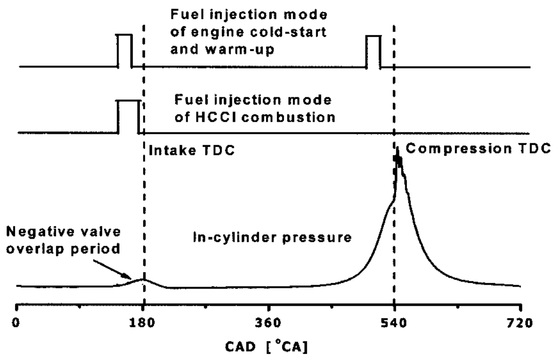


Figure 2. Fuel injection mode.

The negative valve overlap was adjusted by the intake and exhaust valve timing control system, which can change the intake valve open timing and exhaust valve closure timing independently.

In this study, in-cylinder pressure was measured by a PC based data acquisition system and the TDC error was ±0.2 °CA. The NO_x emission was measured by an AVL Digas 4000 exhaust gas analyzer, and the smoke emission was measured by a FBY-200 smoke meter. The IMEP was used to indicate the engine load and was controlled by adjusting the amount of injected fuel at the same engine speed.

3. ESTIMATION OF THE EXPERIMENTAL HEAT RELEASE RATE

In this study, combustion analysis was based on the calculation of the heat release rate using measured in-cylinder pressure data. For each operating condition, the in-cylinder pressure of fifty cycles was recorded, and the mean cylinder pressure trace was estimated. The heat release rate was determined by applying the first thermodynamic law using the following expression (Hountalas and Kouremenos, 1999).

$$\frac{dU_b}{d\varphi}=\frac{dQ_w}{d\varphi}+\frac{dU}{d\varphi}+p\frac{dV}{d\varphi} \tag{1}$$

The following assumptions were made in this calculation:

- Cylinder charge was considered to behave as an ideal gas.
- Composition of the working gas was considered to be variable and was estimated from the trapped mass (air and gaseous fuel) at inlet valve closure and the amount of fuel burned to the current crank angle.
- Distribution of thermodynamic properties inside the combustion chamber was considered to be uniform.
- Dissociation of combustion products was neglected.
- No variation of cylinder mass due to blowby was considered. This assumption did not result in significant error, since the engine used had a low blowby rate on the order of 0.5% of the intake mass flow rate as revealed from measurements taken during normal diesel operation.

The heat loss (negative from gas to walls) was obtained from Equation (2)

$$Q_w=\alpha_g\cdot F_c\cdot(T-T_w) \tag{2}$$

in which, α_g was obtained from the G. Sitkei experimental model.

The previous methodology provided a good estimation of the actual rate of heat release inside the combustion chamber.

4. ESTIMATION METHOD OF COMBUSTION STABILITY

In this study, combustion stability was studied by statistical analysis of cylinder pressure in continuous cycles. Because the peak value cylinder pressure can be easily calculated from the measured cylinder pressure directly, in this paper the CoV_{pz} is used to evaluate combustion stability instead of the CoV_{imep}, and the formula used is as follows:

$$CoV_{p_z}=\frac{\sigma_{p_z}}{p_z}\times 100\% \tag{3}$$

5. EXPERIMENTAL RESULTS AND DISCUSSION

5.1. Heat Release Rate and NO_x Emission of HCCI Combustion

It is well known that diesel-fueled HCCI combustion has a two-stage heat release (low temperature and high temperature heat release) and very low NO_x emission. These two characteristics can prove the success of HCCI combustion.

The heat release rate of diesel-fueled HCCI combustion is different from that of a conventional diesel engine. For fuel with a high cetane number, such as diesel fuel, the HCCI combustion can be divided into a low temperature and a high temperature reaction phase (Helmantel, 2004). Figure 3 shows the in-cylinder pressure and heat release rate diagram for HCCI combustion. It can be seen that a low-temperature heat release exists which cannot be seen in conventional diesel combustion, and when the IMEP is increased from 0.3 to 0.5 MPa, the start of combustion is advanced from 3 °CA BTDC to 9 °CA BTDC. This is because the ignition process in HCCI combustion is very sensitive to temperature and the in-cylinder temperature is increased along with the load. When the load is increased to near limit, the start of combustion is so advanced that the engine runs roughly. This hampers the engine operation range at higher load. When the load is decreased to a low limit, misfire appears

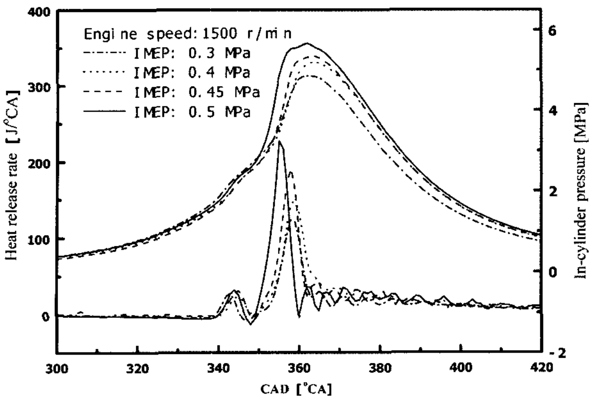


Figure 3. Effects of engine load on heat release rate of the HCCI engine.

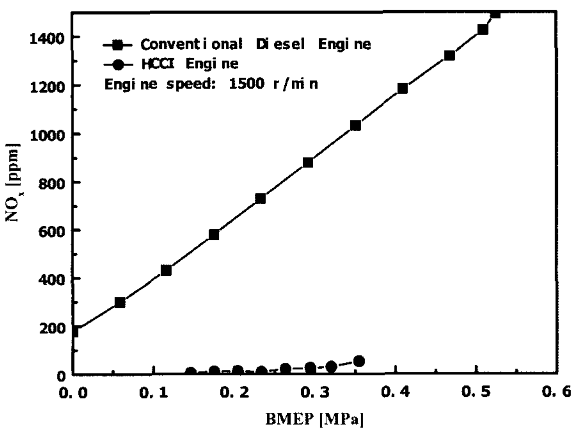


Figure 4. Comparison of NO_x emissions.

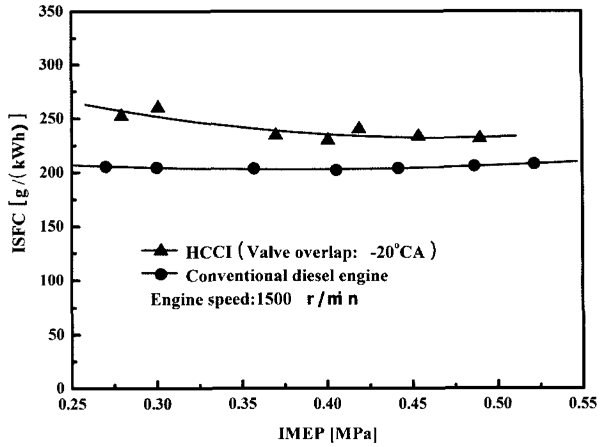


Figure 5. Fuel consumption of HCCI and conventional diesel engine.

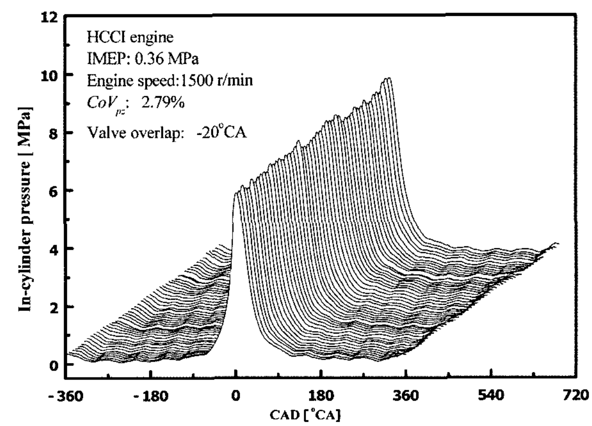


Figure 6. Cycle variation of HCCI engine.

more easily and combustion becomes unstable.

The HCCI engine undergoes an auto-ignition process throughout the entire combustion chamber that can eliminate the high-temperature flames of conventional engine combustion. Therefore the NO_x emission from HCCI engines can be kept very low as compared to that of the conventional engine combustion process (Zhao, 2003). Figure 4 shows the measured NO_x emission of a conventional diesel engine and an HCCI engine with –10 °CA valve overlap. As shown in this figure, the NO_x emission of the conventional diesel engine increases with engine load but the NO_x emission of HCCI engine remains at about 5~30 ppm across the entire stable operation load range. It can be seen that compared with conventional diesel engine, the NO_x emission of HCCI combustion is reduced by nearly 95~98%.

Figure 5 shows the specific fuel consumption of HCCI and conventional diesel engines. Compared with the conventional diesel engine, the indicated specific fuel consumption of HCCI combustion is higher than that of the conventional diesel engine. The main reasons can be illustrated as follows: firstly, the start of combustion is very early in HCCI combustion and the thermal efficiency is decreased; secondly, the fuel spray wets the cylinder and cannot burn completely, so the HC and CO emissions are about ten times higher than those of conventional diesel engine (Bhave *et al.*, 2006).

5.2. Comparison of Combustion Stability between HCCI Engine and Conventional Diesel Engine

The ignition process of HCCI combustion is mainly controlled by mixture chemical kinetics. Therefore, the ignition and combustion is very sensitive to the mixture concentration and in-cylinder temperature history. But in a conventional diesel engine, the combustion is mainly controlled by fuel injection. In-cylinder temperature fluctuation will make the combustion stability of the HCCI engine worse than that of a conventional diesel engine. Figure 6 and Figure 7 show a set of fifty continuous cycles of in-cylinder pressure histories of HCCI combustion and conventional diesel engine combustion. As shown in these figures, at the same load case (IMEP

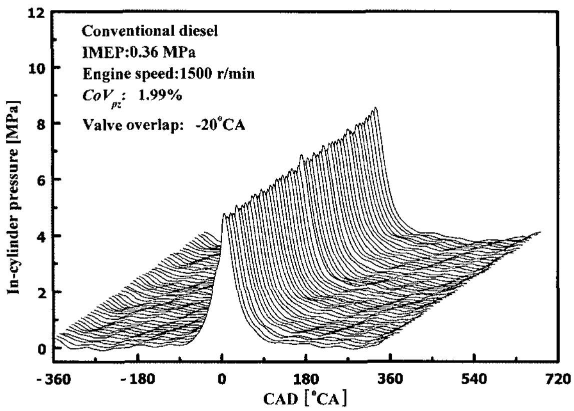


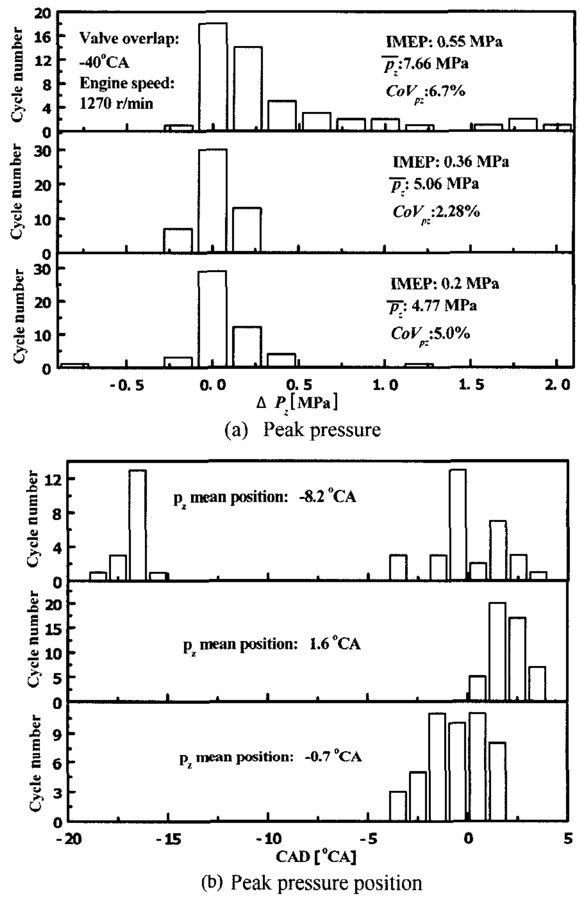
Figure 7. Cycle variation of conventional diesel engine.

=0.36 MPa), the CoV_{pz} of HCCI combustion is 2.79% and it is larger than that of the conventional diesel engine (CoV_{pz} is 1.99%). At the same IMEP shown in these figures, the mean peak pressure value of HCCI combustion (6.25 MPa) is higher than that of the conventional diesel engine (5.33 MPa). This is because the start of HCCI combustion is so early and it harms the engine power-out.

5.3. Effect of Engine Load on Combustion Stability of HCCI Engine

HCCI combustion becomes unstable easily, especially at lower and higher loads, because it is sensitive to the working conditions.

Figure 8 shows the statistical analysis results of peak pressure and peak pressure position at lower load (IMEP =0.2 MPa), middle load (IMEP=0.36 MPa), and higher load (IMEP=0.55 MPa) in diesel-fueled HCCI combustion. It can be seen that at a relatively middle load, Δp_z is in the range of –0.2 to 0.2 MPa and the CoV_{pz} value is the minimum compared with that at lower load and higher load. At lower load, Δp_z is in the range of -0.8 to 1.2 MPa, and that of 92% total cycles are distributed in the range of –0.2 to 0.4 MPa due to misfire. At higher load, Δp_z are distributed in the –0.2 to 2.0 MPa range because the combustion is relatively rough and the in-cylinder pressure shows oscillation. The CoV_{pz} values at lower load and higher load are 5.0% and 6.7%, which are larger than that at relatively middle load (CoV_{pz}=2.28%). The peak pressure position relative to crank angle degree is also studied. At the middle load, peak pressure positions of all cycles are located in the range of 0.5-4.5 °CA ATDC, and at lower load, they are located in the range of –3.5 to 1.5 °CA ATDC, but at higher load, 36% of total cycles are located in the range –15.5 to 18.5 °CA ATDC and 64% are located in the range of –3.5 to 3.5 °CA ATDC. This is because at higher load, in-cylinder temperature is high, and results in advancing the start of combustion in some cycles.



(a) Peak pressure

(b) Peak pressure position

Figure 8. Distribution of peak pressure and position at different loads in HCCI combustion.

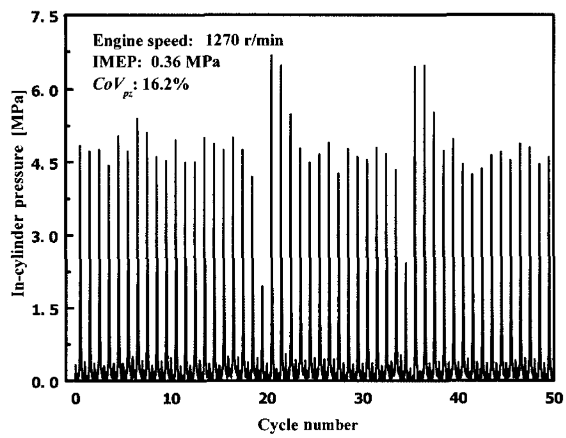


Figure 9. Fifty cycles of pressure at lower load.

When the HCCI engine runs at lower load, the mixture concentration is lean, which lowers the reaction rate. At the same time, the in-cylinder temperature is relatively low. Due to these two reasons, the start of combustion of HCCI is delayed, and misfire even appears. When misfire

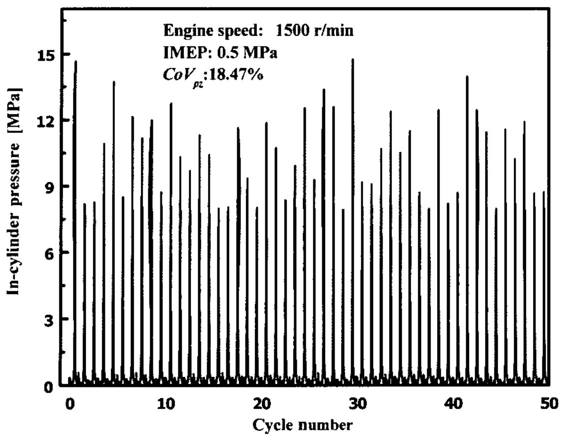


Figure 10. Fifty cycles of pressure at higher load.

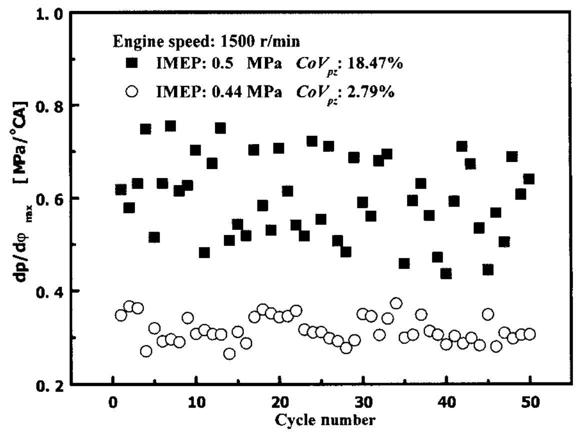


Figure 12. Maximum pressure rise rate comparison of knock and steady combustion.

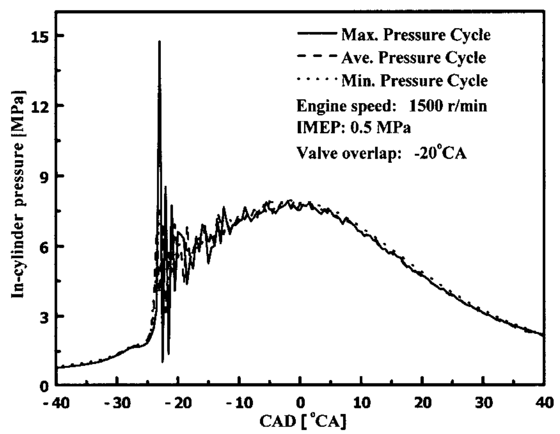


Figure 11. Comparison of in-cylinder pressure at higher load in HCCI combustion.

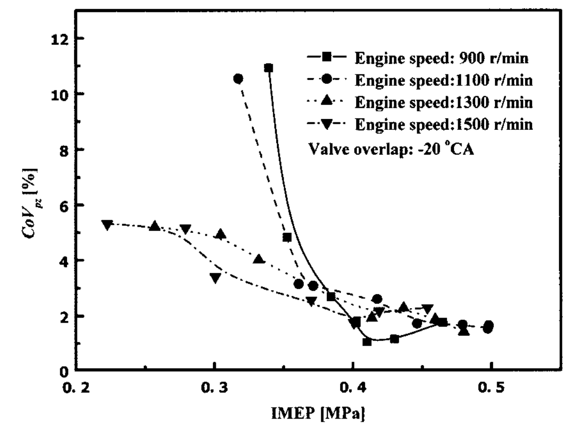


Figure 13. CoV_{pz} of HCCI combustion.

appears in the first cycle, part of the fuel will remain in the combustion chamber and the next cycle will then work more roughly than the other cycles. Figure 9 shows fifty continuous cycles of the in-cylinder pressure history. It can be seen that in these cycles, there are two misfire-cycles and the peak pressure of the cycles immediately after the misfire-cycles is higher than that of the mean value.

When the HCCI engine runs at higher load, due to the high in-cylinder temperature and thick mixture concentration, the reaction rate is very fast and the start of combustion is advanced. Overly fast combustion speed makes the pressure increase quickly and the parts of the engine can be easily damaged. At higher load, due to the advanced start of combustion (about –25 °CA ATDC) and high heat release rate, the in-cylinder pressure shows oscillation. Figure 10 shows fifty continuous cycles of in-cylinder pressure history at higher load. It can be seen that in these cycles, there are many that show knock. Figure 11 shows the pressure oscillation that appeared in the combustion process due to high combustion acceleration. Figure 12 shows fifty cycles of pressure rise rate for relatively rough combustion (higher load) and stable combustion (middle load). It can be seen that during rough combustion, the pressure rise rate is about two times higher than that of stable combustion. Additionally, the difference in pressure rise rate during rough combustion (about 0.35 MPa) is also larger than that of stable combustion (about 0.12 MPa).

5.4. Effect of Engine Speed on Combustion Stability of the HCCI Engine

Figure 13 shows the CoV_{pz} for different speeds and loads. It can be seen that at lower load, the CoV_{pz} of all speeds is higher because misfire appears easily. With the increase of IMEP, the CoV_{pz} decreases, which means the combustion has become stable. The CoV_{pz} at higher loads in this experiment is not very high because the high load is limited by 100 ppm NO_x emission and at that time the combustion is not very rough. However, as seen from the curve, at the high IMEP, the CoV_{pz} is a little higher than

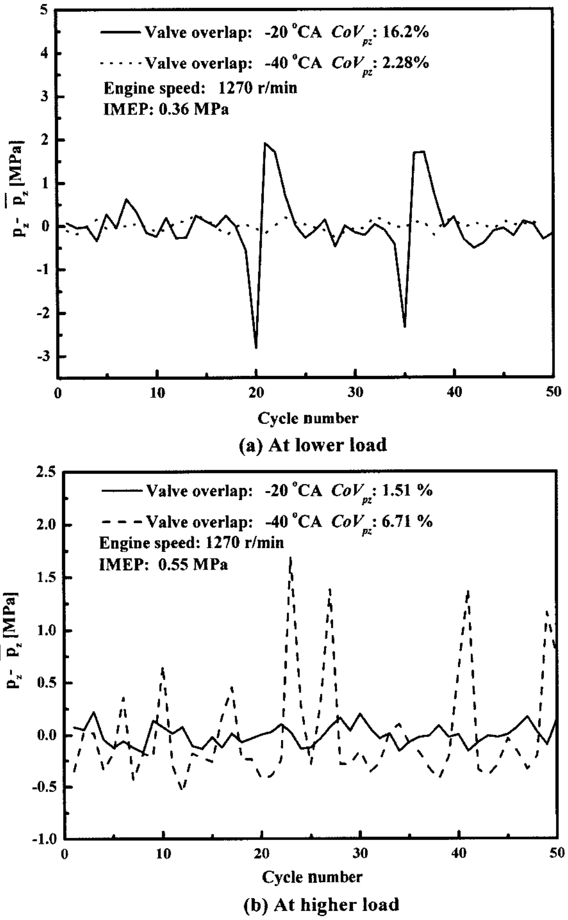


Figure 14. The effects of valve overlap on HCCI combustion stability.

that at the middle load. When the engine speed is reduced from 1500 r/min to 900 r/min, the CoV_{pz} at lower load increases because the in-cylinder temperature decreases due to the increase in amount of heat lost and misfire appears easily.

5.5. Effect of Valve Overlap on the Combustion Stability of HCCI Engine

In this experiment, the valve timing was adjusted in order to achieve negative valve overlap and allow more high temperature residual gas in the cylinder. The high temperature residual gas aided the fuel vaporization and changed the in-cylinder temperature history. Therefore, it also affected the combustion process of the HCCI engine.

Figure 14 shows the effect of different valve overlap durations on the combustion stability at lower and higher loads. It can be seen that at 0.36 MPa IMEP, the CoV_{pz} of –40 °CA valve overlap is 2.28%, which is smaller than that of –20 °CA (16.2%), but at 0.55 MPa IMEP, the CoV_{pz} of –40 °CA valve overlap is 6.71%, which is larger than that of –20 °CA valve overlap (1.51%). It can be concluded that at lower loads, a large negative valve overlap benefits combustion stability, but at higher load, a smaller value benefits stability. The reason is that the high in-cylinder temperature resulting from a large negative valve overlap reduces the misfire tendency at lower load but increases knock tendency at higher load.

6. CONCLUSIONS

Diesel-fueled HCCI was achieved by injecting fuel when the piston moves to the TDC position in the exhaust stroke and by adjusting negative valve overlap. The high-temperature in-cylinder residual gas assisted fuel evaporation and changed the in-cylinder temperature history. Combustion stability was studied and the effects of engine load, speed, and valve overlap on it were also studied. The following conclusions can be drawn from the current work:

(1) Injecting diesel fuel near TDC of the exhaust stroke is one method for forming a homogeneous mixture and achieving HCCI combustion. The high temperature of the in-cylinder residual gas aids fuel vaporization. Diesel-fueled HCCI combustion has a two-stage heat release (low-temperature and high-temperature heat release) and very low NOx emission compared with that of a conventional diesel engine.
(2) Combustion stability of the HCCI engine is worse than the conventional diesel engine at lower and higher loads because the combustion cannot be controlled directly. At lower load, misfire more easily appears and at higher load, knock appears more easily. Combustion in the HCCI engine is more stable at middle loads than that at lower and higher load.
(3) Engine speed can affect the stability of HCCI combustion. When the engine speed was reduced from 1500 r/min to 900 r/min, the CoV_{pz} at lower load increased because heat loss was increased.
(4) Changing the negative valve overlap duration can change the in-cylinder temperature and the mixture composition. Therefore, it has effects on HCCI combustion. In this study, the high in-cylinder temperature resulting from large negative valve overlap reduces the misfire tendency at lower load but increases knock tendency at higher load.

ACKNOWLEDGEMENT–The work in this paper is a part of the item 2001CB209205 of the National Key Fundamental Research Programming 973. The authors are grateful to the National Science and Technology Ministry of China for financial support of this work.

REFERENCES

Bhave, A., Kraft, M. and Montorsi, L. (2006). Sources of

CO emissions in an HCCI engine: A numerical analysis. *Combustion and Flame* **144**, **3**, 634–637.

Gao, Z. and Schreiber, W. (2001). The effects of EGR and split fuel injection on diesel engine emission. *Int. J. Automotive Technology* **2**, **4**, 123–133.

Helmantel, A. (2004). *Reducing Diesel Engine Emissions-An Experimental Investigation.* Licentiate Thesis. Chalmers University of Technology. Gothenburg. Sweden.

Henrik, N., Anders, H. and Bengt, J. (2004). Start of injection strategies for HCCI-combustion. *SAE Paper No.* 2004-01-2990.

Hountalas, D. T. and Kouremenos, A. D. (1999). Development and application of a fully automatic troubleshooting method for large marine diesel engines. *Appl. Therm. Eng.*, **19**, 299–324.

Hu, G. D. (1990). New strategy on diesel combustion development. *SAE Paper No.* 900422.

Ishibashi, Y. and Asai, M. (1996). Improving the exhaust emissions of two-stroke engines by applying the activated radical concept. *SAE Paper No.* 960742.

Kim, D. S. and Lee, C. S. (2006). Improved emission characteristics of HCCI engine by various premixed fuels and cooled EGR. *Fuel* **85**, **5-6**, 695–704.

Najt, P. M. and Foster, D. E. (1983). Compression-ignited homogeneous charge combustion. *SAE Paper No.* 830264.

Pöttker, S., Eckert, P., Delebinski, T., Baumgarten, C., Oehlert, K., Merker, G., Wagner, U. and Spicher, U. (2004). Investigations of HCCI combustion using multi-stage direct-injection with synthetic fuels. *SAE Paper No.* 2004-01-2946.

Ryan, T. W. and Callahan, T. J. (1996). Homogeneous charge compression ignition of diesel fuel. *SAE Paper No.* 961160.

Shi, L., Deng, K.Y. and Cui, Y. (2005). Study of diesel-fuelled homogeneous charge compression ignition combustion by in-cylinder early fuel injection and negative valve overlap. *Proc. Institution of Mechanical Engineers, Part D; J. Automobile Engineering* **219**, **10**, 1193–1201.

Su, W. H., Lin, T. J. and Pei, Y. Q. (2003). A compound technology for HCCI combustion in a DI diesel engine based on the multi-pulse injection and the BUMP combustion chamber. *SAE Paper No.* 2003-01-0741.

Takeda, Y. and Keiichi, N. (1996). Emission characteristics of premixed lean diesel combustion with extremely early staged fuel injection. *SAE Paper No.* 961163.

Thring, R. H. (1989). Homogeneous charge compression-ignition (HCCI) engines. *SAE Paper No.* 892068.

Yanagihara, H. (2001). Ignition timing control at Toyota "UNIBUS" combustion system. *Proc. IFP Int. Cong. a New Generation of Engine Combustion Processes for the Future?*, 34–42.

Zhao, F. Q. (2003). *Homogeneous Charge Compression Ignition (HCCI) Engines: Key Research and Development Issues.* SAE Book. PT-94.