ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

SIMULATION AND PERFORMANCE ANALYSIS OF CNG FUELED HCCI ENGINE

P. M. Diaz[1] and B. Durga Prasad[2]
[1]Sathya Bama University, Chennai, India
[2]JNTU College of Engineering, Ananthapur, India
E-Mail: diaz_p_m_27@yahoo.co.in

ABSTRACT

Compressed Natural Gas (CNG) is a difficult fuel to use in a Homogenous Charge Compression Ignition (HCCI) engine because of high octane number, high auto-ignition temperature, and rapid heat release. These properties force CNG HCCI engines to use extreme levels of intake heating. The Homogeneous Charge Compression Ignition concept has the potential to meet the need for a high efficiency and low emission engine. Fluent is one of the promising operating tools in the computational fluid dynamics. In the present study the Computational Fluid dynamics (CFD) code FLUENT is used to model complex combustion phenomenon in homogeneous charge compression Ignition engine. The variation of various properties like the peak cylinder pressure, peak cylinder temperature at various crank angles, at different relative air fuel mixture inlet temperatures and with hemi spherical combustion chamber and toroidal bowl has been studied. It was found that heated inlet air fuel mixture enhance combustion, peak cylinder pressure, peak cylinder temperature and change peak pressure timing.

Keywords: compressed natural gas, homogenous charge compression ignition (HCCI), computational fluid dynamics.

INTRODUCTION

In comparison with gasoline, natural gas has a very high octane number (≈120) and high auto-ignition temperature (≈600℃). Composed mostly of methane, natural gas is the only common fuel to exhibit relatively pure, single-stage combustion. Other fuels have stronger low-temperature reaction and the required enthalpy for main stage combustion can be obtained from low-temperature heat release as a result of compression to moderate pressure and temperature. In contrast, the methane molecule resists destruction by free radicals and produces negligible heat release at low temperature. Hence, in CNG-fueled HCCI engines the activation energy required for auto-ignition must be obtained by extreme levels of charge heating and compression.

LITERATURE REVIEW

The basic idea of homogeneous charge compression ignition HCCI engines is the auto-ignition of a homogeneous mixture that sufficiently lean to keep peak combustion temperatures below 1700 K such that low NO and low particulate matter PM emissions are achieved [1]. Due to the high compression ratio (CR) required initiating the combustion and the rapid heat release process, HCCI engines also achieve high thermal efficiency [2]. As a result, HCCI engines achieve fuel economy levels comparable that of compression ignition [CI] engines, while generating engine-out NOx emissions that are as good as tailpipe NO_x emissions from conventional spark ignition [SI] engines with after treatment [1]. Therefore, the HCCI engine is a promising option for a clean and efficient configuration of internal combustion engine.

One of the key difficulties in the implementation of HCCI technology in production engines is that ignition cannot be directly actuated. The timing of auto-ignition of HCCI combustion is determined by the cylinder charge conditions, rather than the spark timing or the fuel injection timing that are used to initiate combustion in the SI and CI engines, respectively [3]. As demonstrated by many experimental results [4, 5], controlled auto-ignition requires regulation of the charge properties, namely, temperature, pressure, and composition at the intake valve closing [IVC]. Once the valves are closed, there is very little that can be done to affect ignition. Hence, all the controllable engine variables need to be adjusted prior to IVC in a judicious way based on accurate predictions of their influence on combustion timing. To regulate the HCCI combustion phasing, as in references [6] and [7], an accurate model for the start of combustion timing (SOC) is necessary. Analysis in Ref. [8] has shown that the auto-ignition modeling approach has significant potential for gasoline engine application but will require calibration for accuracy over the full range of engine operation. Small temperature differences inside the cylinder have a considerable effect on combustion. Chen *et al.*, [9] performed an investigation of internal EGR. He showed that using internal hot EGR leads to an earlier combustion. Fiveland *et al*. investigated in [10] the influence of initial temperature, initial pressure of mixture, natural gas composition, heat transfer model, equivalence ratio and compression ratio on ignition behavior of an HCCI engine.

In the present study the Computational Fluid Dynamics (CFD) code FLUENT is used to model complex combustion phenomenon in compression ignition HCCI engine. The simulation conducted for combustion chamber with hemispherical bowl-in-shape piston on single cylinder and HCCI engine, for different inlet temperature and at constant speed of 1500 rpm. The numerical modeling is solved by unsteady first order implicit, taking into account the effect of turbulence. For modeling turbulence Renormalization Group Theory (RNG) k-ε model is used. The Multi-zone model can also be used

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

independently, using specified heat-transfer parameters from the start of simulation after intake valve closing model description following the multi-zone model approach reported by Aceves *et al*., [11].

SIMULATION PROCEDURES

The IC Engine model is appropriate for a closed system, representing the time between intake-valve closure and exhaust-valve opening in the engine cycle. The start time (or start crank angle) therefore represents the time of intake-valve closure. As a convention, engine events are expressed in crank rotation angle relative to the top dead center (TDC). The intake valve close (IVC) time of our test engine is 142 degrees (crank angle) before TDC (BTDC). The combustion chamber is bowl-in-piston type, which having a hemispherical groove on piston top. The geometry has been modeled at its -13°, -10°, -6°, -3°, 0°, 3°, 6°, 10°, 13°, 16°, 20° and 23° crank angle position at ATDC. The engine specification used in the simulation is given in Table-1. The fuel composition used in the simulation is given in Table-2.

Table-1. Engine specification.

S. No.	Parameters	Values
1	Compression ratio	17.5
2	Stroke length	110mm
3	Bore diameter	80mm
4	Engine speed	1500 rpm
6	Exhaust valve open	43° BBDC
7	Exhaust valve close	6° ATDC
8	Inlet valve open	8°ATDC
9	Inlet valve close	36° ABDC

Table-2. Composition of initional gas mixture.

S. No.	Species	Mole fraction
1	CH_4	0.0350
2	C_2H_6	0.0018
3	C_3H_8	0.0012
4	O_2	0.1824
5	CO_2	0.0326
6	H_2O	0.0609
7	N_2	0.6861

RESULTS AND DISCUSSIONS

Figure-1 depicts the variation of peak cylinder pressures with respect to crank angles for hemi spherical combustion chamber at different inlet temperatures of air fuel mixtures. It is observed that the peak cylinder pressure is found to be maximum which is 81.9 bar for inlet temperature of air fuel mixture at 170°C. It is seen that the peak cylinder pressure for inlet temperature of air fuel mixture at 130°C shows minimum value and is about 76 bar. The study reveals that the more advanced start of combustion in turn leads to higher peak net heat release rates. With more heat released prior to very or near TDC, the physical volume in which the energy is released becomes smaller, and due to engine geometry does not change as much per degree of crank angle. This causes higher cylinder pressures.

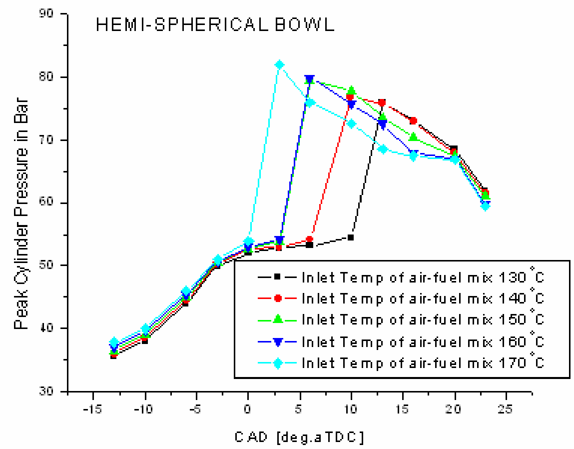

Figure-1. Variation of peak cylinder pressure vs crank angle for the different air-fuel mixture inlet temperatures with HSB.

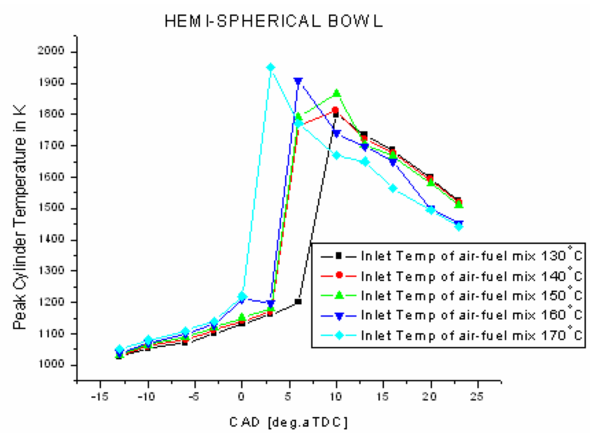

Figure-2. Variation of peak cylinder temperature vs crank angle for the different air-fuel mixture inlet temperatures with HSB.

The variations of peak cylinder temperatures with respect to crank angles for hemi spherical bowl combustion chamber at different inlet temperatures of air fuel mixture are shown in the Figure-2. It is seen that the peak cylinder temperature for inlet temperature of air fuel mixture at 170°C is maximum and is found to be 1950K. The peak cylinder temperature for inlet temperature of air-fuel mixture at 130°C is found to be minimum and is about 1800K. It is found that heat released is higher as the inlet temperature of air-fuel mixture is higher. The heat released is found to be lower with lower inlet temperature of air-fuel mixture. In general, the study reveals that as the inlet temperature of air fuel mixture increases the rate of combustion which results in high peak cylinder temperatures.

Figure-3 depicts the variation of peak cylinder pressures with respect to crank angles for toroidal bowl combustion chamber at different inlet temperatures of air fuel mixtures. It is observed that the peak cylinder pressure is found to be maximum which is 83.2 bar for inlet temperature of air fuel mixture at 170°C. It is seen that the peak cylinder pressure for inlet temperature of air fuel mixture at 130°C shows minimum value and is about 79 bar. Changing inlet temperature of air fuel mixture to the CNG fueled HCCI engine is one of the effective ways to control the ignition timing since inlet temperature able to change ignition delay time effectively. The pressure rise in the case of toroidal bowl is advanced than hemi-spherical bowl because of surface nature.

The variations of peak cylinder temperatures with respect to crank angles at different inlet temperatures of air fuel mixture for toroidal bowl combustion chamber are shown in the Figure-4. It is seen that the peak cylinder temperature for inlet temperature of air fuel mixture at 170°C is maximum and is found to be 2000 K. The peak cylinder temperature for inlet temperature of air-fuel mixture at 130°C is found to be minimum and is about 1880K. It is found that heat released is higher as the inlet temperature of air-fuel mixture is higher. With the same inlet temperature of air fuel mixture for HSB and TB, the toroidal bowl combustion chamber engine performance higher than HSB. The peak cylinder temperature for TB is higher due to enhanced combustion and early completion of combustion.

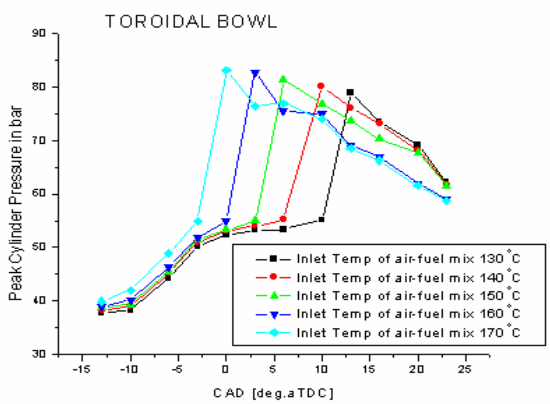

Figure-3. Variation of peak cylinder pressure vs crank angle for the different air-fuel mixture inlet temperatures with TB.

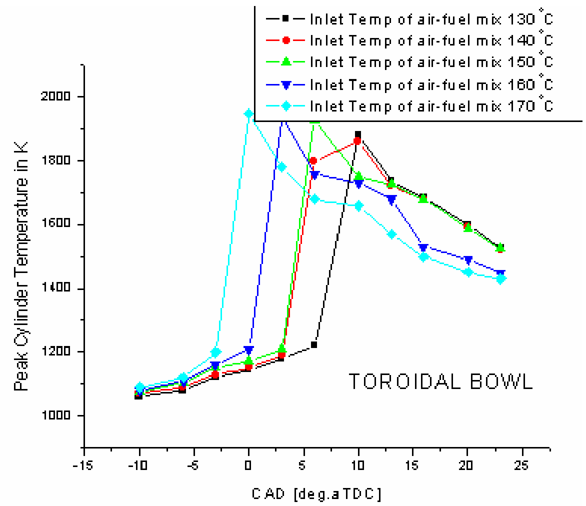

Figure-4. Variation of peak cylinder temperature vs crank angle for the different air-fuel mixture inlet temperatures with TB.

CONCLUSIONS

The CNG fueled HCCI engine with hemi spherical piston bowl configuration for inlet temperature of air fuel mixture at 170°C shows higher peak cylinder pressure is about 81.9 bar which is higher than inlet temperature of air fuel mixture at 130°C and it is 76 bar. It is observed that the maximum peak cylinder temperature for inlet temperature of air fuel mixture at 170°C is 8.3% higher than air fuel mixture inlet temperature at 130°C.

The CNG fueled HCCI engine with hemi spherical piston bowl configuration for inlet temperature of air fuel mixture at 170°C shows higher peak cylinder temperature is about 1950K which is higher than inlet temperature of air fuel mixture at 130°C and it is 1800K. It is observed that the maximum peak cylinder temperature for inlet temperature of air fuel mixture at 170°C is 7.7% higher than air fuel mixture inlet temperature at 130°C.

It is observed for toroidal bowl combustion chamber that the maximum peak cylinder pressure for inlet temperature of air fuel mixture at 170°C is 5.3 % higher than air fuel mixture inlet temperature at 130°C.

It is observed for toroidal bowl combustion chamber that the maximum peak cylinder temperature for inlet temperature of air fuel mixture at 170°C is 6.3% higher than air fuel mixture inlet temperature at 130°C.

REFERENCES

[1] Chia-Jui Chiang and Anna G. Stefanopoulou. 2009. Sensitivity Analysis of Combustion Timing of Homogeneous Charge Compression Ignition Gasoline Engines. Journal of Dynamic, Systems, Measurement and Control, ASME. 131 (JANUARY), 014506-1, DOI: 10.1115/1.2936877.

[2] Semin A.R. Ismail and T.F. Nugroho. 2010. Experimental and Computational of Engine Cylinder

Iteration

Pressure Investigation on the Port Injection Dedicated CNG Engine Development. Journal of Applied Sciences. DOI: 10.3923/jas.2010.107.115. 10: 107-115.

[3] A. Rashid Firmansyah and A. Aziz. 2011. The Combustion Behavior Analysis of Dual Fuel HCCI using the Shell Model. Journal of Applied Sciences. DOI: 10.3923/jas.2011.1559.1565. 11: 1559-1565.

[4] U. Azimov, E. Tomita, N. Kawahara, N. Kawahara and Y. Harada. 2011. Premixed mixture ignition in the end-gas region (PREMIER) combustion in a natural gas dual-fuel engine: operating range and exhaust emissions. International Journal of Engine Research. DOI: 10.1177/1468087411409664. 129(5): 484-497.

[5] A. Birkigt, K. Michels, J. Theobald, T. Seeger, Y. Gao, MC. Weikl, M. Wensing and A. Leipertz. 2011. Investigation of compression temperature in highly charged spark-ignition engines. International Journal of Engine Research. DOI: 10.1177/1468087411399216. 12(3): 282-292.

[6] Taku Tsujimura, William J. Pitz, Yi Yang and John E. Dec. 2011. Detailed Kinetic Modeling of HCCI Combustion with Isopentanol. 10th International Conference on Engines and Vehicles, Session: HCCI Modeling, Naples, ITALY, September, DOI: 10.4271/2011-24-0023.

[7] Kaiyuan He, Ioannis P. Androulakis and Marianthi G. Ierapetritou. 2011. Numerical Investigation of Homogeneous Charge Compression Ignition (HCCI) Combustion with Detailed Chemical Kinetics Using On-the-Fly Reduction. American Chemical Society. DOI: 10.1021/ef200290z. 25(8): 3369-3376.

[8] Ming Jia and Maozhao Xie. 2007. Numerical simulation of homogeneous charge compression ignition combustion using a multi-dimensional model. Journal of Automobile Engineering. DOI: 10.1243/09544070JAUTO263P. 221(4): 465-480.

[9] Chen R. and Milovanovic N. 2002. 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. DOI: 10.1016/s1290-0729(02)01375-3. 41: 805-813.

[10] Fiveland S. B. and Assanis D. 2000. A Four Stroke Homogenous Charge Compression Ignition Engine Simulation for Combustion and Performance Studies. SAE paper 2000-01-0332, DOI: 10.4271/2000-01-0332.

[11] Aceves M. S. and D. L. Flowers *et al*. 2001. A Sequential Fluid-Mechanic Chemical-Kinetic Model of Propane HCCI combustion. SAE Technical Paper 2001-01-1027. DOI: 10.4271/2001-01-1027.