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Studying the Effect of Reformer Gas and Exhaust Gas Recirculation on Homogeneous Charge Compression Ignition Engine Operation

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Combustion in homogeneous charge compression ignition (HCCI) engine is controlled by auto-ignition of well-mixed charge of fuel, air and residual gas. Since onset of HCCI combustion depends on the auto-ignition of fuel/air mixture, there is no direct control on the start of combustion process, therefore, HCCI combustion becomes easily unstable especially at lower and higher engine load. Charge stratification has the potential to extend the load limits of HCCI combustion by improving the control over the combustion phase as well as reducing the maximum pressure rise. In this study, a combination of experiment and numerical simulation is carried out to investigate the effect of fuel stratification using reformer gas and EGR on HCCI natural gas combustion. Results show that fuel stratification in this case increased the auto-ignition property of natural gas. On the lean operation boundary, reformer gas and EGR blending enhanced the auto-ignition by advancing combustion timing at identical initial conditions compared to pure natural gas that expanded the lean boundary of the operating region.

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NOMENCLATURE

$CO_{2,up}$	$CO_{2,up}$	CD	Combustion Duration in CAD	IMEP	Indicated Mean Effective Pressure
$CO_{2,down}$	$CO_{2,down}$	CAD	Crank Angle Degree	BSFC	Brake Specific Fuel Consumption
m_{RG}	m_{RG}	SOC	Start of Combustion (CAD,aTDC)		
m_{fuel}	m_{fuel}	TDC	Top Dead Center		

1. INTRODUCTION

Over the long history of the modern internal combustion engine, two types of internal combustion engine i.e. spark ignition (SI) and compression ignition (CI) have dominated. The SI engine is limited by knock to relatively lower compression ratios than the CI engine, giving lower initial cost and mass, but also lower efficiency. For CI engines, it is difficult to reduce NO_x and particulate matter (PM) emissions simultaneously. A solution that combines low NO_x and PM emission SI engine benefit with high efficiency of CI engine is called homogeneous charge compression ignition (HCCI) combustion engine. Combustion in HCCI engine is a form of simultaneous multi-point autoignition of a lean premixed charge, without any external energy as an ignition source. Although being reported in numerous research papers as a new

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combustion concept for reciprocating internal combustion engines, HCCI also known as controlled auto-ignition (CAI) has been around for over 100 years [1]. The first patent refers to inventing a hot-bulb two stroke oil engine by Weiss in 1897 [2]. Semonov and Gussak [3] built the first CAI engine that controls combustion by using active species which are discharged from partially burned mixture in a separate prechamber. The most recognized and first systematic early investigations on CAI were done by Onishi [4] and Noguchi [5] on two stroke engines in the late nineteen seventies, and then on a four stroke engine by Najt and Foster in 1983 [6]. But it was not until the late 1980s that the terminology homogenous charge compression ignition (HCCI) was introduced by Thring [7]. In his research paper it is focused on studying the effect of external EGR and air-fuel ratio on HCCI.

Since the combustion in HCCI comes from homogenous charge and uniform combustion chamber temperature, auto-ignition occurs in various points of the homogenous mixture of fuel and air, by a sudden extreme increase in pressure. Such high rate of increase in pressure may lead to fluctuation of pressure, vibrations, knock and serious damages to the engines. As the charge in an HCCI engine ignites spontaneity, there is no direct control on start of ignition [8]. Ignition is highly dependable on chemical and thermal condition of the mixture, which may limit the operating range of HCCI engine. Such limitation is defined by knock in rich side and misfiring or partial burning in lean side. Detailed studies in respect to HCCI reveal that the formation process of air/fuel mixture has a significant effect on the combustion. In HCCI engines, start of ignition is controlled by auto-ignition chemical mechanisms. Hence, combustion phasing control needs a configuration mechanism that is influenced by the combination of fuel/air mixture and the pressure of reactants in the process of compression. However, it should be noted that it is impossible to achieve practical combustion of HCCI with fully homogenous charge mixture. Even a low rate of inhomogeneity in the mixing or distribution of charge and the temperature of the mixture, has a significant effect on auto-ignition and the process of combustion. Therefore, solving the issue of HCCI combustion control leads to investigation of different solutions that may deprive the mixture from homogenous combination. Hence, creating desired inhomogeneity in the distribution of temperature, combination and distribution of charge along with the traditional methods of combustion controlling such as the pressure and temperature of charge, exhaust gas recirculation (EGR) make possible the ability to increase the operating region of HCCI engine [9]. Aroonsrisopon et al. [10] showed that charge stratification using two nuzzles, one in the combustion chamber and another in the intake manifold, improves the emission, combustion efficiency, fuel consumption and IMEP in the lean side of operating region. An experimental and numerical study, comparing two methods of fuel injection for creating homogenous and stratified mixture, showed that achieving a sustainable and practical combustion in a fully homogenous mixture is extremely difficult [11]. In another experimental study, the effect of charge mixture inhomogeneity on the combustion of HCCI with methanol as fuel in a single cylinder engine with transparent combustion chamber was investigated. Results indicate that in same equivalence ratio, maximum amount of heat release occurs in less inhomogeneity and the duration of combustion is longer in this method [12]. In an experimental study, Lee et al. [13, 14], expanded the operating region of engine in different work circumstances (different speed and compression ratio) by direct injection of gasoline to the combustion chamber. To control the timing of HCCI combustion in a gasoline engine and the increase of operating region, Wang et al. [15, 16] proposed the utilization of multiple injections. Inagaki et al. [17], has controlled the ignition timing with direct injection of diesel fuel into the combustion chamber in early stage of intake cycle, and the injection of iso-octane to intake manifold in late stage of intake cycle.

Considering the technologies that provide simultaneous availability of two fuels, using double fuel HCCI combustion is a practical solution. But amore interesting solution for controlling combustion in HCCI engines and internal combustion engines as a whole would be to utilize a basic fuel and reforming a part of it to an auxiliary fuel with different auto-ignition specification than the basic fuel. The derivative of fuel reforming process is called reformer gas (RG). Reformer gas is a combination of light gases with higher ratios of CO and H_2 and other ineffective gases that is formed using a fuel reactant, with or without a catalyst. Researchers in Birmingham University have studied different aspects of creating reduced gas and its application in internal combustion engines. An in-depth study on reforming technologies in a wide variety of fuels and the application of H_2 in internal combustion engines has been done by Jamal and Wyszynski [18]. Quader and colleagues attempted to expand the operating region of a lean SI engine by using reforming fuel techniques. Shudo et al. [19, 20] first examined the effect of methanol reformer gas and DME. They found that both H_2 and CO retarded the second stage of combustion of DME considerably, and thus shifted operating range toward richer mixtures. Tsolakis et al. [21] examined REGR in a partially premixed compression ignition engine. The reformer gas was mixed with the air in the intake and the diesel engine was fueled with regular diesel and biodiesel. REGR enrichment of the intake mixture improved NO_x and smoke emissions and the effect of REGR varied based on the engine operating mode but basically it retarded combustion timing. Hosseini [22] indicated that reformer gas in natural gas HCCI combustion reduce intake heating requirement and expands the lean boundary of the operating region.

Thus, it is clear that the ideal HCCI combustion which incorporates a completely homogeneous fuel-air mixture cannot be applied to an internal combustion engine. This study aimed to investigate the effect of charge stratification by adding EGR and reformer gas on performance in a HCCI combustion engine. Various mixtures of H_2 and CO with different ratios are applied as simulated Reformer gas and experimental tests of a single cylinder engine with this method for validating the results of the simulation are carried out.

2. EQUIPMENT AND TEST CONDITIONS

Experimental study was conducted on a diesel engine converted to run at HCCI mode. Main features of the engine are shown in Table 1. Engine modifications for the HCCI combustion include:

1. The inlet air heating system: by installing an electric heater with variable heating and the temperature sensor before and after the heater so that variable intake temperature was achieved.
2. Exhaust gas recirculation (EGR) system: gas recirculation system was installed on exhaust manifold and the system was calibrated to control gas mass fraction that was returned to intake system.
3. The fuel supply system: according to HCCI combustion mode of natural gas fueled engine, intake path is included a gas mixer and throttle valve to let addition of natural gas and reformer gas. Also for controlling the engine in natural gas mode, an intelligent electronic system was designed to regulate the amount of fuel used.
4. Measuring instruments: various sensors for measuring and displaying the combustion chamber pressures, exhaust manifold pressure, intake and exhaust temperature, engine speed and air and natural gas mass flow is placed on the engine. Figure 1 indicates the experimental set-up and the location of the sensors. As the engine speed is constant at 1500 RPM, the engine was linked directly to a 10 kW AC electric dynamometer. The in-cylinder pressure was measured by a Kistler-6052C piezoelectric sensor with an accuracy of ±0.5%.

The engine setup was modified to work with EGR system. In EGR system exhaust gas is circulated through an external piping which is connecting exhaust manifold to intake manifold using control valves. The quantity of EGR was controlled manually by adjusting the EGR throttle valve. This issue increases the exhaust back pressure and hence influences engine performance. Provisions were made on both the intake and exhaust manifolds to measure the percentages of CO_2 in the intake charge and exhaust gases. Percentage of recirculated exhaust gas is calculated using Equation (1):

$$EGR = \frac{CO_{2,up}}{CO_{2,down}} \times 100 \tag{1}$$

Reformer gas blend fraction is calculated using mass rate of the base fuel and mass of reformer gas using Equation (2):

$$RG_{mass,\,frac} = \frac{m_{RG}}{m_{RG} + m_{fuel}} \times 100 \tag{2}$$

where m_{RG} and m_{fuel} are reformer gas and fuel mass, respectively.

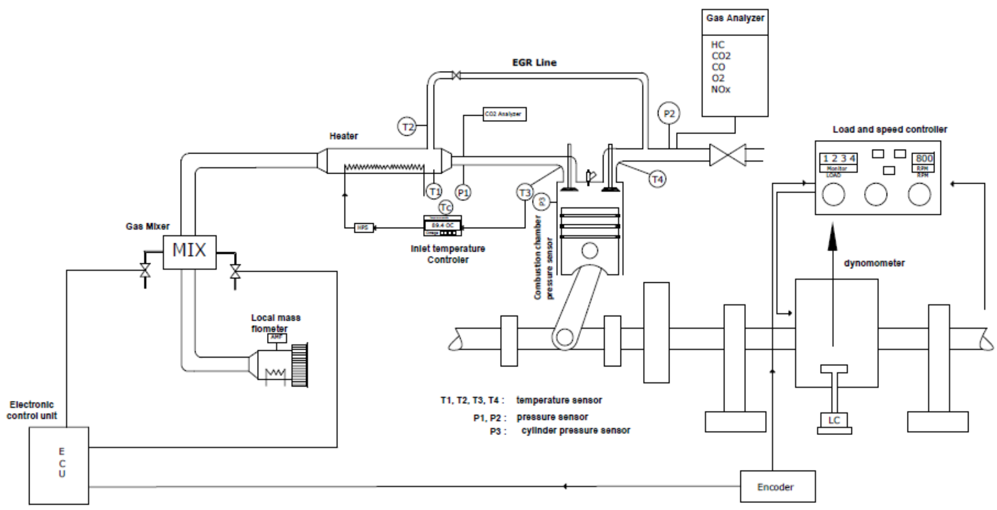


Figure 1. Schematic of experimental setup

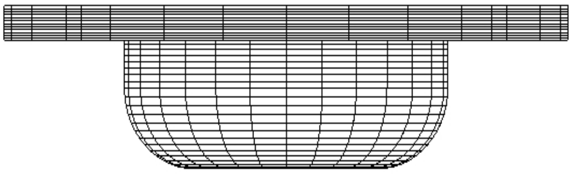


Figure 2. Computational grid at TDC used for engine calculations

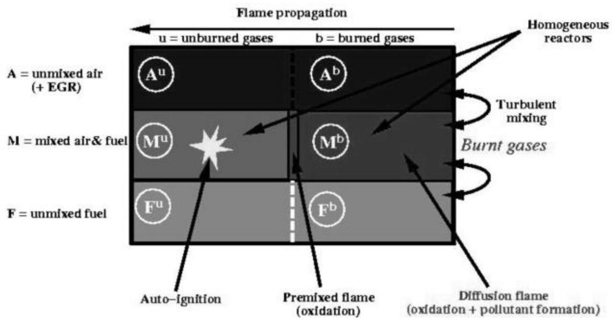


Figure 3. Zones in ECFM-3Z Model

TABLE 1. Engine Specification

Engine Type	**Diesel – Air cooled**
Engine Model	**Kirloskar- DAF10**
Bore	102 (mm)
Stroke	110 (mm)
Displacement	0.948 (lit)
Compression Ratio	18:1
Engine Speed	1500 (rpm)
Maximum Power in Diesel Mode	7.5 (kW)

3. SIMULATION OF COMBUSTION IN HCCI ENGINES

The commercial CFD software AVL-FIRE and GT power are used to perform the numerical simulation of engine performance, combustion and emission formation in HCCI engine. Combustion characteristics and its influence on flow field and emission formation is investigated in three-dimensional CFD modeling using FIRE code. Related three-dimensional combustion chamber model is shown in Figure 2. At TDC, the whole mesh number is 24200 cells. This mesh size will be able to provide good spatial resolution for the distribution of most variables in the combustion chamber. Calculations are carried out on the closed system from IVC to EVO. The generated model in FIRE code is a close cycle in compression and expansion strokes of engine. Therefore, EGR and turbo charging influence cannot be simulated on the initial condition of engine such as boost pressure and initial temperature. Also, engine performance parameters such as IMEP, BSFC and etc. cannot be calculated in closed cycle simulation. GT power software is a one dimensional CFD model that can simulate four strokes of engine operation. This software is used in this work to determine initial condition in different cases beside the prediction of engine performance. In order to simulate the in cylinder combustion process in one dimensional CFD model, calculated heat release rate in three-dimensional simulation is imported as a burn rate profile. So that, the heat release rate is imported from Fire to the GT-Power. Results of one dimensional simulation including inlet pressure, inlet temperature and equivalence ratio is imported to the three-dimensional model and simulation is repeated with new condition. This procedure was continued to reach a small residual between two simulations. Combustion process, emission formation and engine performance variation was investigated in different conditions in parallel simulation of one and three-dimensional CFD software.

4. COMBUSTION MODEL

ECFM-3Z model, presented in AVL-FIRE software, is utilized in combustion simulation. This combustion model belongs to the Coherent Flame Model that has been developed for inhomogeneous turbulence premixed combustion, un-premixed or diffusion, and auto ignition. Pre-reactions of auto-ignition in premixed charge of fuel and air are calculated and auto-ignition delay time in premixed mixture is determined by local temperature and pressure, equivalence ratio and amount of residual gas. After local auto-ignition, premixed combustion in fuel occurs, air and burned gas followed by diffusion combustion in an area containing fuel and oxidizer. Figure 3 shows the combustion zones, mixture and combustion components, and the reaction process of components [23].

5. VALIDATION OF SIMULATION RESULTS

To validate the results obtained from simulations, intake temperature, equivalence ratio, EGR ratio, engine speed and output power are considered as engine performance variables according to the pattern provided in Table 2. Figure 4 indicates the comparison of simulated and experimental results in-cylinder pressure against the crank angle for tests number 1 and 3 which are introduced in Table 2. The good agreement of predicted in-cylinder pressure with the experimental data can be observed. Maximum pressure and maximum rate of pressure rise of experimental results are lower compared to simulation results.

TABLE 2. Experimental test condition

Test No.	Speed (RPM)	Pressure in IVC (bar)	Intake Temperature (°C)	Equivalence Ratio (-)	EGR (%)
1	1500	1.1124	150	0,3125	20
2	1500	1.1124	150	0,3334	10
3	1500	1.1124	120	0,32	0
4	1500	1.1124	120	0,3125	20

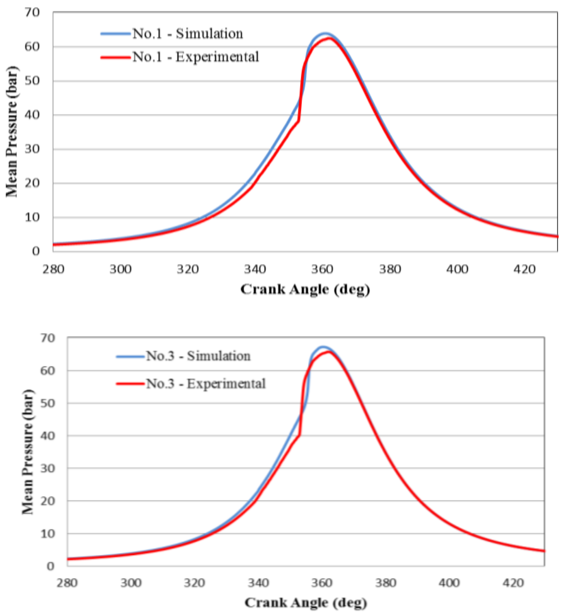


Figure 4. Cylinder pressure alteration vs. crank angle for test No. 1 and 3

This difference is mainly due to the inevitable inhomogeneity of the in-cylinder charge in terms of fuel density and charge temperature distribution in test engine. Inhomogeneity present in intake mixture, fresh charge and residual gas within the combustion chamber together with heat transfer and turbulence during compression stroke are main reasons for this natural inhomogeneity.

5. EXHAUST GAS RECIRCULATION (EGR) EFFECTS

The exhaust gas recirculation line, as shown in Figure 1, connects exhaust manifold to intake manifold. Amount of recirculation gas flow depends on EGR valve state, outlet manifold pressure, inlet pressure, engine speed and valve timing. In lower speed range of engine the outlet pressure is reduced and even in full open state of EGR valve, the increment of recirculation gases are limited. In present work, engine speed is considered constant at 1500 rpm and thus there is no limitation on maximum amount of outgoing recirculation gases. At high EGR fractions, thermodynamic effects (higher heat capacity and lower ratio of specific heat in comparison with air) causes low after-compression temperature resulting in later less intense combustion. In addition, a dilution effect of EGR reduces the chance of fuel molecules to react with oxidizer. All of these effects cause combustion inhibition even in the flameless HCCI combustion. Reduction in cylinder pressure after compression cycle leads to delayed combustion. As shown in Figure 5, increases of EGR ratio causes retard start of main combustion (SOC) and prolong combustion duration. This increase in time has considerable effects on uniform and noiseless combustion. In general, the effects of recycled exhaust on HCCI combustion behavior can be summarized into the following four categories: charge heating, dilution of inlet mixture, thermodynamic properties change and stratification of mixture in the cylinder. Charge heating independently advances combustion time and ignition timing, increases combustion stability, makes combustion duration shorter and enhances heat release rate. Dilution does not have direct effect on auto auto-ignition, but reduces combustion rate and therefore provides smoother and quieter operation. Change in thermodynamic properties, as mentioned earlier, retards ignition timing and extends combustion duration due to lower ratio of specific heat [24, 25]. As stated earlier, maximum cylinder pressure and maximum rate of pressure rise in HCCI engines are higher than conventional engines. Sudden multi-site combustion of air/fuel mixture at relatively high compression ratio and high intake temperature causes high maximum cylinder pressure, which is a limiting factor for HCCI combustion engine. Figure 6 shows the effect of recycled exhaust ratio and air/fuel ratio on maximum in cylinder pressure and maximum rate of pressure rise. Increasing EGR considerably decreased maximum pressure and maximum rate of pressure rise at any constant air/fuel ratio with slower combustion. Combination of EGR and air/fuel ratio determines the power output of an HCCI engine. Figure 7 indicates that despite of replacing some portion of the intake air by EGR, IMEP increased considerably. The highest IMEP was achieved at maximum EGR fraction. Increasing EGR shifted the operating region toward richer intake mixture. At any constant EGR rate, increasing air/fuel ratio decreased indicated power due to reduction of the energy content inside the cylinder. On the other hand, keeping air/fuel ratio constant and increasing EGR would reduce the indicated power, but increasing EGR to achieve lower air/fuel ratio increased indicated power. Higher indicated power is associated with higher thermal efficiency as indicated in Figure 7. While indicated power is a function of EGR and air/fuel ratio, it is observed that indicated thermal efficiency is directly a function of EGR and air/fuel ratio. Thermal efficiency is to be found maximum near the knock boundary and minimum near the low load boundary.

Increasing air/fuel ratio by replacing fuel molecules with air molecules leads to lower heat capacity of mixture and higher temperature after compression stroke, but on the other hand lean mixture decreases combustion temperature. Reducing mixture temperature and subsequently expanding cycle causes incomplete oxidation of CO and THC and increases the above-mentioned emissions in outlet. As expected, quantity of CO and THC were increased with increment of recycled exhaust ratio at any constant air/fuel ratio. However, increasing EGR in the test engine leads to a reduction in air-fuel ratio. The quantity of these emissions is lowest at maximum EGR and near-to-stoichiometric mixture condition.

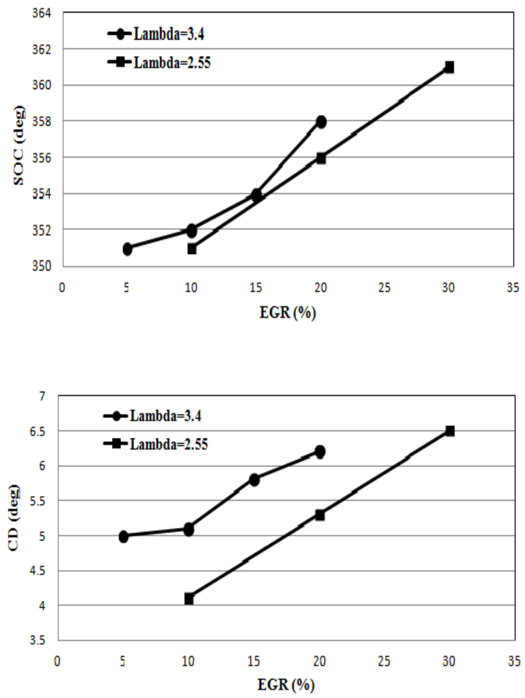
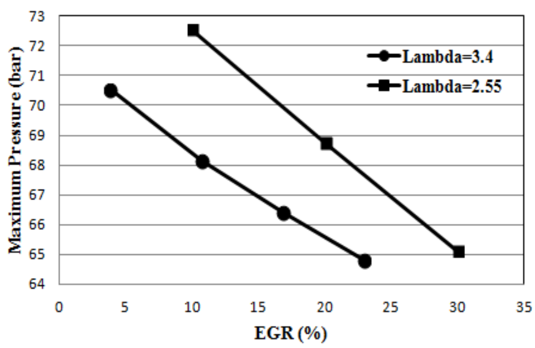


Figure 5. The effect of EGR ratio on combustion duration and start of combustion



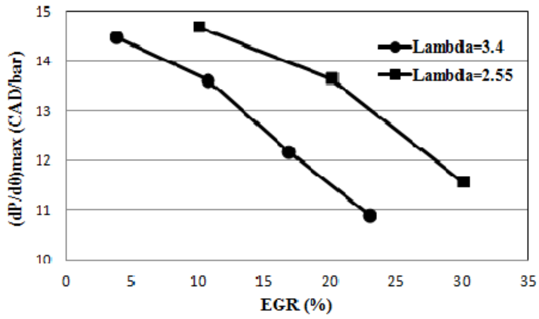


Figure 6. The effect of EGR ratio on maximum pressure and maximum rate of pressure rise

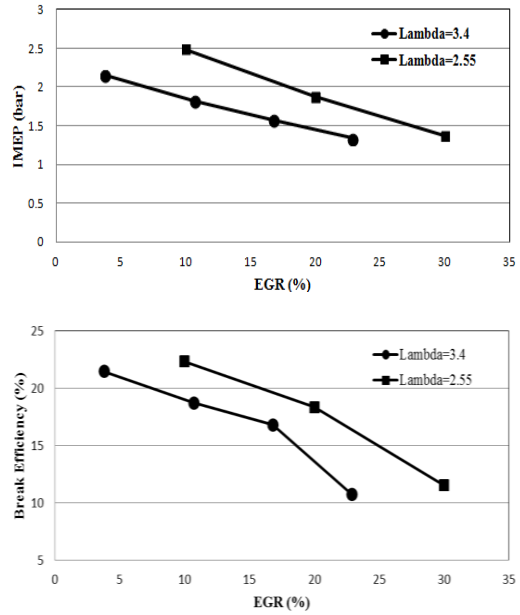


Figure 7. The effect of EGR ratio on IMEP and thermal efficiency

6. REFORMER GAS TO NATURAL GAS RATIO EFFECTS

Dual fuel HCCI combustion engine is a practical solution for combustion timing control with current technologies, when infrastructure is provided and carrying two fuels onboard is feasible. Other solution is to carry the base fuel onboard and partially convert it to the gaseous secondary fuel with alternated auto-ignition quality from the base fuel. This process is called fuel reforming and the produced fuel is called reformer gas (RG) or synthesis gas (syngas). Reformer gas is a mixture of light gases dominated by H_2 and CO and some inert gases. It can be produced with or without catalyst using a fuel processor [22].

In the present paper, a combination of H_2 and CO is considered to investigate the effect of reformer gas on natural gas HCCI combustion engine. All the implementations in this section is done with normally aspirated inlet temperature of 150°C and without recirculation of exhaust gas in order to assess only the effects of adding transform gas on natural gas in HCCI engines. In this section a combination of 75% of H_2 and 25% CO is utilized and in the next section the same effects with 50% of each will be investigated.

Figure 8 shows the effect of reformer gas blend fraction on maximum pressure and maximum pressure rise rate. As increment reformer gas blend fraction at any constant air/fuel ratio line increased maximum cylinder pressure and maximum pressure rise rate dramatically. Increasing reformer gas blend fraction by 10% increased maximum pressure by an average of 0.4 to 1.8 bar and maximum pressure rise rate by an average 0.8 to 1.8 bar/CAD. Maximum pressure and maximum pressure rise rate increase is caused by earlier combustion timing. Compression of more combustion product by early combustion caused more intense combustion. Rate of pressure rise for all of the points are high. For most cases, a rate of pressure rise of 10 bar/CAD is considered as the border line for maximum rate of pressure rise. In this case, almost all selected points exhibited higher maximum pressure rise rate than the conventional limit. Hence expanding the operating region toward leaner mixture reduced maximum pressure and maximum rate of pressure rise. For a given air/fuel ratio, increasing reformer gas blend fraction increased maximum pressure and maximum pressure rise rate. Also, increasing reformer gas blend fraction changed the timing of these two parameters substantially. As indicated in Figure 9, timing of both maximum pressure and maximum rate of pressure rise are advanced considerably as the result of reformer gas blend fraction increase at constant air/fuel lines. The tendency of RG to shift the allowable operating range towards leaner mixtures while advancing the peak pressure timing explains the significant drop in exhaust temperature because of longer time for expansion of combustion products.

Figure 10 shows the effect of increasing reformer gas blend fraction on IMEP reduction. Keeping air/fuel ratio constant and increasing reformer gas blend fraction decreased IMEP. As air/fuel ratio was constant, the flow of energy into the engine was constant. Hence, IMEP reduction was a result of less efficient combustion. The efficiency drop result from early combustion timing by reformer gas blend fraction increase. Higher cyclic variation is an indication of less robust combustion by reformer gas blend fraction increase. The operating points in this case are at the edge of knocking and misfiring points. Increasing RG blend fraction advance combustion timing and cause increment instability in combustion by increasing knocking intensity. Keeping air/fuel ratio constant and replacing natural gas with RG does not change energy flow to the engine considerably as both fuels are in gaseous state and air flow to the engine is not drastically changed. IMEP reduction could be due to less efficient fuel utilization with increased RG blend fraction. Thermal efficiency is decreased because of increase in RG blend fraction. This is not the result of combustion efficiency reduction or emission increase, as indicated in Figure 10. At this condition, the engine converts more fuel to combustion product as combustion efficiency increase by increasing RG blend fraction, but the fuel utilization suffered from non-optimized combustion timing and combustion duration.

Table 3 shows the relation of engine emissions to reformer gas blend fraction at constant air/fuel ratio. Keeping air/fuel ratio constant and increasing RG blend fraction decreased HC and increased CO emissions considerably. Decreasing HC emissions is a function of both air/fuel ratio and RG blend fraction, while increasing CO emissions is independent of air/fuel ratio and related to RG blend fraction. Natural gas HCCI combustion at high compression ratio and intake temperature exhibits a narrow operating range between knock and low load. Most of the operating points had knock and high rate of pressure rise, which could not be considered a practical HCCI engine. In most of the fuels and operating conditions during this study, NO_x is almost zero; however, in this specific case extreme amount of NO_x is observed because of knocking. Knock causes the disturbance of thermal boundary layer resulting less heat transfer to the cylinder wall and considerable increase in mixture temperature during and after combustion. In addition, knock produces hot spots inside the cylinder that cause advanced combustion timing those results in high rate of pressure rise and high knock intensity.

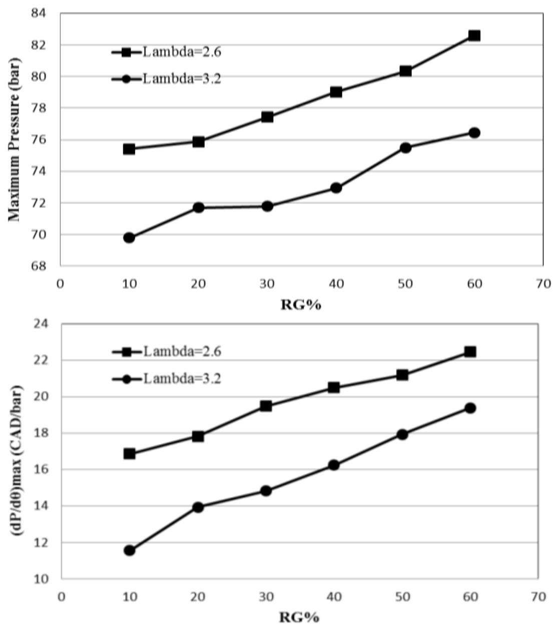


Figure 8. The effect of RG ratio on maximum pressure and maximum rate of pressure rise

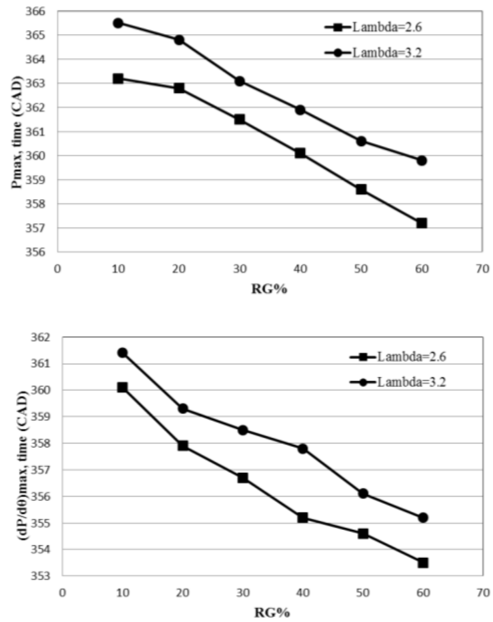


Figure 9. The effect of RG ratio on maximum pressure and maximum rate of pressure rise time

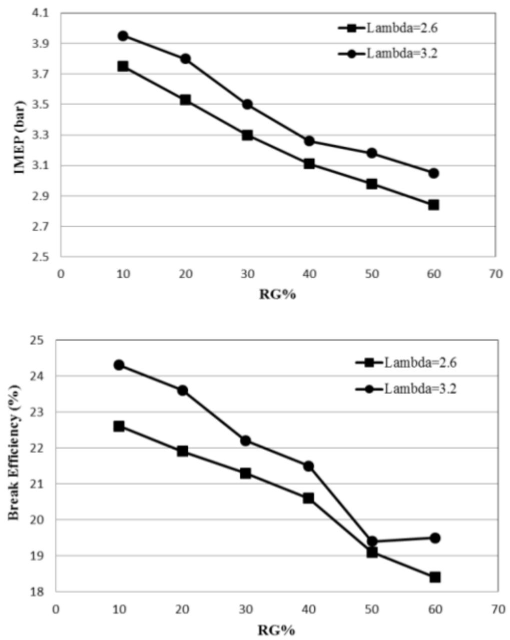


Figure 10. The effect of RG ratio on IMEP and thermal efficiency

TABLE 3. The effect of RG ratio on the mass fraction of CO, THC and NO_x

RG Ratio	10%	20%	30%	40%	50%	60%
CO mass (10^{-6}) fraction	104	127	136	131	150	160
THC mass (10^{-6}) fraction	93	85	83	81	67	33
NOx mass (10^{-6}) fraction	1309	1623	2360	3447	5340	7930

As shows in Table 3, NO_x increase with increasing reformer gas bland fraction. To obtain an optimal value for NO_x in acceptable performance condition, air/fuel ratio should be increased while reformer gas blend fraction kept constant.

7. REFORMER GAS COMPOSITION EFFECTS

Details of reformer gas composition selection have been discussed in the previous section. Two reformer gas compositions of RG 75/25 and RG 50/50 were used for natural gas HCCI combustion. As previous results were based on reformer gas 75/25, series of simulations were conducted on natural gas HCCI combustion using reformer gas 50/50. Simulations were conducted on the similar initial and operating conditions for both combination of reformer gas. The migration from reformer gas 75/25 at constant fueling rate requires increasing reformer gas mass to keep air/fuel ratio constant. Although mass of reformer gas 50/50 is higher than that of 75/25, the energy flow to the engine is slightly lower because of the reduced LHV of reformer gas 50/50 on mass basis. It is the same for the cases of constant air/fuel ratio and reformer gas blend fraction with two different RG compositions. If the natural gas fueling rate be constant, fewer reformers 50/50 are needed than reformer gas 75/25 to keep air/fuel ratio constant. Hence, the energy flow to the engine is lower in the reformer 50/50 case at constant air/fuel ratio and reformer gas blend fraction.

In this study, effect of RG (both composition) on natural gas HCCI combustion was advancing combustion timing. Advanced combustion timing on the lean side of operating region compensated the retarded combustion timing by high air/fuel ratio values and the lean side of the operating region pushed back further. It was more possible to operate the engine with high reformer gas blend ratio using reformer gas 50/50 than with reformer gas 75/25. Both H_2 and CO have similar octane numbers, although no certain octane numbers is reported. From auto-ignition point of view, H_2 has more auto-ignition resistance than CO, hence replacing reformer gas 50/50 with reformer gas 75/25 increased the resistance of the mixture to auto-ignition and limited the maximum reformer gas blend fraction. Figure 11 indicates the effect of changing reformer gas blend fraction on natural gas HCCI combustion timing at any constant air/fuel ratio. Despite having the same reformer gas blend fraction for each pair of data and identical change of air/fuel ratio, advancing combustion timing by reformer gas increase was slightly sharper for the case of reformer gas 75/25 than for reformer gas 50/50. Considering this issue for the case of reformer gas 75/25, air/fuel ratio is leaner than the reformer gas 50/50 case and leaner air/fuel ratio result in later combustion.

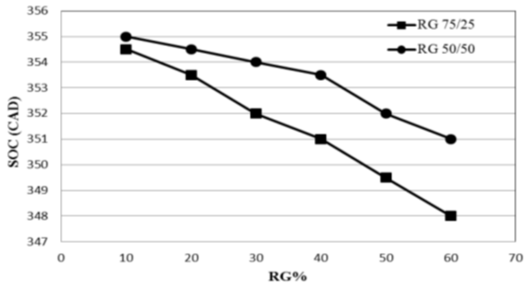


Figure 11. The effect of RG composition on start of combustion

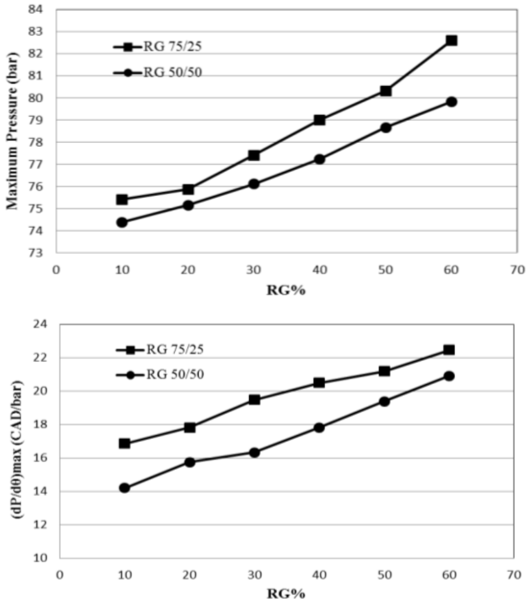


Figure 12. The effect of RG composition on maximum pressure and maximum rate of pressure rise

Results show that reformer gas 75/25 is more efficient on advancing combustion timing. As indicated in Figure 12, combustion timing is influenced by the maximum pressure and maximum rate of pressure rise.

8. CONCLUSION

In this study, effects of applying EGR and RG on HCCI combustion characteristics were investigated. Results of this work can be summarized as follows:

- Increasing EGR at constant excess air delays SOC and prolongs burn duration.
- In-cylinder peak pressure and maximum pressure rise rate decrease with EGR.
- Increasing EGR at constant excess air decreases IMEP and thermal efficiency.
- Increasing RG to natural gas ratio greatly increases in-cylinder peak pressure and maximum pressure rise rate. The increase is more at 50/50 RG ratio.
- The location of in-cylinder peak pressure and maximum pressure rise rate are advanced by increasing RG to natural gas ratio at constant excess air.
- Increasing RG to natural gas ratio at constant excess air reduces IMEP and thermal efficiency.
- Increasing RG to natural gas ratio at constant excess air reduces THC, but increases CO and NO_x.
- Applying RG generally delays SOC the delay is more at 75/25 RG ratio than 50/50.
- In terms of the performance range of HCCI engine using RG, the case using a 75/25 RG ratio can operate at leaner mixtures than the one using 50/50 due to lower octane number of RG compared to natural gas.

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10. REFERENCES

1. Shahbakhti, M., "Modeling and experimental study of an HCCI engine for combustion timing control", University of Alberta, (2009).
2. Erlandsson, O., Early swedish hot-bulb engines-efficiency and performance compared to contemporary gasoline and diesel engines,(2002), SAE Technical Paper.
3. Gussak, L., Turkish, M.C. and Siegla, D.C., High chemical activity of incomplete combustion products and a method of prechamber torch ignition for avalanche activation of combustion in internal combustion engines, (1975), SAE Technical Paper.
4. Onishi, S., Jo, S.H., Shoda, K., Do Jo, P. and Kato, S., Active thermo-atmosphere combustion (atac)-a new combustion process for internal combustion engines,(1979), SAE Technical paper.
5. Noguchi, M., Tanaka, Y., Tanaka, T. and Takeuchi, Y., A study on gasoline engine combustion by observation of intermediate reactive products during combustion,(1979), SAE Technical Paper.
6. Najt, P.M. and Foster, D.E., Compression-ignited homogeneous charge combustion,(1983), SAE Technical paper.
7. Thring, R.H., Homogeneous-charge compression-ignition (HCCI) engines,(1989), SAE Technical paper.
8. Nazoktabar, M. and Jazayeri, S., "Controlling the power output and combustion phasing in an HCCI engine", ***International Journal of Engineering-Transactions B: Applications***, Vol. 27, No. 11, (2014), 1757-1766.
9. Nobakht, A.Y., Saray, R.K. and Astiar, G.S., "Optimization of gri-mech 3.0 mechanism using hcci combustion models and genetic algorithm", ***International Journal of Engineering, Transactions A: Basics***, Vol. 24, No. 2, (2011), 155-168.
10. Aroonsrisopon, T., Werner, P., Waldman, J.O., Sohm, V., Foster, D.E., Morikawa, T. and Iida, M., Expanding the HCCI

operation with the charge stratification,(2004), SAE Technical Paper.

11. Dec, J.E. and Sjöberg ,M., A parametric study of HCCI combustion-the sources of emissions at low loads and the effects of gdi fuel injection,(2003), SAE Technical paper.
12. Kumano, K. and Iida, N., Analysis of the effect of charge inhomogeneity on hcci combustion by chemiluminescence measurement,(2004), SAE technical paper.
13. Lee, C. and Lee, K., "An experimental study on the combustion and emission characteristics of a stratified charge compression ignition (SCCI) engine", ***Energy &Fuels***, Vol. 21, No. 4, (2007), 1901-1907.
14. Lee, K. and Lee, C., "An experimental study of the extent of the operating region and emission characteristics of stratified combustion using the controlled auto-ignition method", ***Energy &Fuels***, Vol. 20, No. 5, (2006), 1862-1869.
15. Wang, Z., Shuai ,S.-J., Wang, J.-X. and Tian, G.-H., "A computational study of direct injection gasoline hcci engine with secondary injection", ***Fuel***, Vol. 85, No. 12, (2006), 1831-1841.
16. Wang, Z., Wang, J., Shuai, S. and Ma, Q., "New gasoline homogeneous charge compression ignition combustion system using two-state direct injection and assisted spark ignition", ***Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering***, Vol. 220, No. 3, (2006), 367-378.
17. Inagaki, K., Fuyuto ,T., Nishikawa, K., Nakakita, K. and Sakata, I., Dual-fuel PCI combustion controlled by in-cylinder stratification of ignitability,(2006), SAE Technical Paper.
18. Jamal, Y. and Wyszynski, M., "On-board generation of hydrogen-rich gaseous fuels—a review ,"***International Journal of Hydrogen Energy***, Vol. 19, No. 7, (1994), 557-572.
19. Shudo, T. and Ono, Y., HCCI combustion of hydrogen, carbon monoxide and dimethyl ether,(2002), SAE Technical Paper.
20. Shudo, T., Ono, Y. and Takahashi, T., Influence of hydrogen and carbon monoxide on hcci combustion of dimethyl ether,(2002), SAE Technical Paper.
21. Tsolakis, A. and Megaritis, A., "Partially premixed charge compression ignition engine with on-board H_2 production by exhaust gas fuel reforming of diesel and biodiesel", ***International Journal of Hydrogen Energy***, Vol. 30, No. 7, (2005), 731-745.
22. Hosseini, V., "Reformer gas application in hcci combustion engine, ProQuest, (2008).
23. Colin, O. and Benkenida, A., "The 3-zones extended coherent flame model (ECFM3Z) for computing premixed/diffusion combustion", ***Oil & Gas Science and Technology***, Vol. 59, No. 6, (2004), 593-609.
24. Zhao, H., Peng, Z., Williams, J. and Ladommatos, N., Understanding the effects of recycled burnt gases on the controlled autoignition (CAI) combustion in four-stroke gasoline engines,(2001), SAE Technical Paper.
25. Zhao, H., Peng, Z. and Ladommatos, N., "Understanding of controlled autoignition combustion in a four-stroke gasoline engine", ***Proceedings of the Institution of MechanicalEngineers, Part D: Journal of Automobile Engineering***, Vol. 215, No. 12, (2001), 1297-1310.

Studying the Effect of Reformer Gas and Exhaust Gas Recirculation on Homogeneous Charge Compression Ignition Engine Operation

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چکیده

در موتورهای اشتعال تراکمی مخلوط همگن (HCCI)، احتراق به واسطهٔ اشتعال خودبخودی مخلوط کاملی از هوا، سوخت و گازهای باقیمانده شکل می‌گیرد. با توجه به اینکه شروع احتراق در این گونه موتورها به خاصیت خوداشتعالی مخلوط هوا و سوخت بستگی دارد، روش پایش مستقیمی برای شروع احتراق وجود ندارد. بنابراین، احتراق موتورهای اشتعال تراکمی مخلوط همگن، به‌ویژه در توان‌های کم و زیاد، بسیار ناپایدار خواهد بود. با استفاده از چینه‌بندی مخلوط ورودی می‌توان پایش مراحل احتراق را بهبود بخشید.همچنین می‌توان بیشینه افزایش فشار داخل محفظه را کاهش داد، که در نتیجه بازهٔ عملکردی گسترش پیدا خواهد کرد. در این مقاله، با ترکیبی از کار تجربی و عددی تاثیر چینه‌بندی مخلوط با استفاده از گاز تبدیلی و گازهای بازگردانی شده بر احتراق HCCI گاز طبیعی مورد بررسی قرار گرفت. نتایج نشان می-دهد که استفاده از چینه‌بندی سوخت موجب افزایش خاصیت خوداشتعالی گاز طبیعی می‌شود. ترکیب گاز تبدیل و گازهای بازگردانی شده در شرایط ثابت ورودی سبب پیش انداختن احتراق شده و بازهٔ عملکردی را، به‌ویژه در محدودهٔ توان کم، افزایش می‌دهد.

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