SURESH PATIL.G.L, DR.B.ANJANEYA PRASAD, DR. DATTA MAHESWAR / INTERNATIONAL JOURNAL OF ENGINEERING RESEARCH AND APPLICATIONS (IJERA) ISSN: 2248-9622
WWW.IJERA.COM VOL. 3, ISSUE 1, JANUARY -FEBRUARY 2013, PP.1613-1620

Fuel Injector's Stratergies Of Hcci Engine.

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ABSTRACT

The traditional fuel injection systems for diesel engines are designed with the objective to secure acceptable fuel spray characteristics during the combustion process at all load conditions. Incorrect injection causes reduced efficiency and increased emission of harmful species. In the later years, different fuel injection system with electronic controls has been promoted as the future standard in fuel injection systems for diesel engines. Among the advantages claimed with respect to injection rate shaping, variable timing and duration of the injection, in addition to variable injection pressure, enabling high injection pressure even at low engine loads. When the rate of injection is the key to an effective combustion process, it is vital to determine. Alternative methods to change the combustion process, to improve engine efficiency. The combustion of a homogeneous air/fuel mixture in the cylinder of a diesel engine is very efficient way to do this, where the concept is called "Homogeneous Charge Compression Ignition (HCCI) ."The purpose of this paper is to study on different fuel injection and combustion performances in addition to the engine operating zone for HCCI engines with alternative fuels.

Key words : HCCI, fuel injectors, alternative fuels , emission , VVT, VCR, hydrogen

1. INTRODUCTION

The Spark Ignition (SI) and the Compression Ignition (CI) engines development has reached substantial advantages in efficiency and exhaust emissions in the last decades. The CI engine has a fuel efficiency advantage over the SI engine due to higher thermodynamic efficiency and lower pumping losses. In regard to the exhaust emissions the SI engine holds an advantage over the CI engine. In the Diesel engine the combustion is initiated because of some special conditions of pressure and temperature. However, in the petrol engine the combustion is caused by a spark that ignites a mixture that has been premixed before. Due to these different kinds of combustion, the two engines have different characteristics. The CI has a high efficiency, but it is very contaminating. Contrarily, the SI is not very efficient but it has low emissions However, the IC engine is certainly not the best apparatus in every aspect but seems to be a good compromise overall. It is possible to further develop the IC engine to be better in some property but this is usually at the "cost" of another. In the commercial vehicle segment, the diesel engine has always been prevalent due to its robustness and unequalled efficiency. In the years to come, however, future emission limits will require the simultaneous reduction of nitrogen oxides (NOx) and particulate emissions. to extremely low values.

To improve the engine performance one of the alternative method is reducing engine emissions to change the combustion process, and so as to improve engine efficiency. The combustion of a homogeneous air/fuel mixture in the cylinder of a diesel engine is very efficient way to do this. The concept is not new. It provides a very robust combustion system which can be used with a variety of fuels. In a sense, the Homogeneous Charge Compress Ignition (HCCI) combustion system merges the advantages of SI engine combustion using a homogeneous mixture and that of a diesel engine. The blending of these two designs offers diesel-like high efficiency without the difficult--and expensive--to deal with NOx and particulate matter emissions. In its most basic form, it simply means that fuel is homogeneously (thoroughly and completely) mixed with air in the combustion chamber very similar to a regular spark ignited gasoline engine, but with a very high proportion of air to fuel i.e. lean mixture. As the engine's piston reaches its highest point (top dead center) on the compression stroke, the air/fuel mixture auto-ignites from compression heat, much like a diesel engine. The result is the best of both worlds: low fuel usage and low emissions.

2. FUEL SYSTEM DEVELOPMENT

Now a days R&D is carried out to develop a fuel delivery system because it is a key enabling technology to overcome the challenge of maintaining proper ignition timing, smooth combustion rates, and low emissions over the operating range of the engine. Various types of fuel systems have been proposed including port fuel injectors, DI fuel injectors similar to those designed for SI engines, DI diesel engine injectors, and combinations of these injectors. Each type has advantages for different operating regimes and fuel types.

For diesel fuel, the type of injector required depends on the strategy selected for fuel-air mixing and combustion. This is the most promising approach. For this approach, an injector based on a DI diesel injector would be needed. However, modifications would likely be required to achieve the very high mixing rates necessary for HCCI-like combustion.

For gasoline-type fuels, DI injectors for SI engines appear attractive. However, as HCCI research evolves it may indicate that changes in these injectors are required to meet the needs of HCCI engines. For example, partial charge stratification appears to be an attractive method for controlling the combustion rate at high loads and for reducing the HC and CO emissions at light loads . However, a different type of stratification would be required for each of these cases. Achieving the desired stratification under all operating conditions would likely require specialized HCCI injectors with spray characteristics different from those of current DI gasoline injectors. R&D will be required to adapt existing injectors or to develop advanced injectors that provide the spray characteristics desired for HCCI.

3. COMBUSTION ANALYSIS & EFFECT OF FUEL INJECTOR'S PERFORMANCE.

The Homogeneous Charge Compression Ignition (HCCI) engine is a promising alternative to the existing spark ignition (SI) engines and compression ignition (CI) engines. To limit the rate of combustion, much diluted mixtures have to be used. Compared to the diesel engines the HCCI has a nearly homogeneous charge and virtually no problems with soot and NOX formation. On the other hand HC and CO levels are higher than in conventional SI engines. Overall, the HCCI engine shows high efficiency and fewer emissions than conventional internal combustion engines.HCCI combustion was first discovered as an alternative combustion mode for two-stroke IC engines by Onishi et al. [1979]. They successfully utilized a perceived drawback of "run-on" combustion with high level of residuals and high initial temperature at light load condition to achieve a stable lean combustion with lower exhaust emissions, specifically UHC, and fuel consumption. This new combustion mythology was named "Active Thermo-Atmosphere Combustion" (ATAC). HCCI combustion was first applied to two-stroke engines [1], [2] with improvement in fuel efficiency and combustion stability. When HCCI as applied to the four-stroke engine, the fuel efficiency could be improved up to 50 % compared to the SI engine [3].

Despite advantages, HCCI engines produce high HC and CO emissions as the ignition timing and combustion duration is difficult to control. Therefore, the HCCI operating zone is limited between misfire and knocking. Lack of direct control over ignition initiation is one of the obstacles that need to be addressed. The auto-ignition timing relies on indirect ways such as the air-fuel charge, octane number, temperature, and pressure [6]. The problem with the HCCI engines is related to the lean mixtures, the fast combustion, and the high compression ratio (high engine efficiency) that causes the exhaust temperature to become quite low. This can make it difficult to get both turbo charging and oxidizing catalysts to work. The commercialization of the HCCI engine would require overcoming certain challenges. Low combustion temperatures, though conducive for low NOx emissions, lead to high HC and CO emissions. This is because of incomplete conversion of fuel to CO2 [7] Also, it is difficult to control ignition timing and the rate of combustion for a required speed and power range [8]. The control over ignition timing is achieved by a spark plug or fuel spray in gasoline engines and diesel engines, respectively. Absence of such mechanisms makes it difficult to directly control ignition in HCCI and therefore, indirect methods are adopted.

The purpose of this paper is to study on different fuel injection and combustion performances in addition to the engine operating zone for HCCI engines with alternative fuels.In HCCI mode, combustion initiation has to be controlled indirectly, via in-cylinder temperature at the start of compression, the intake air path from the air inlet to engine intake ports in terms of pressure losses. Particular attention to be paid on 1.Optimization of the geometry in respect to the pressure drop . 2. Flow distribution investigation in the port and manifold. 3. Investigation of the air delivery to each cylinder tuning of ports for the required target. 4. Reduction of recirculation and stagnation of air. 5.Negative valve overlap combined with fuel injection – heat supply during gas exchange phase . 6. External Exhaust Gas Recirculation (EGR).

Recent advances in extending the operational range have utilized stratification at all three parameters: fuel, temperature and EGR. Fuel injection system determines mixing effect of fuel, air and EGR. For gasoline a conventional PFI injection system can form a good homogeneous mixture [Kontarakis et al., 2000]. Fuel stratification can extend the HCCI low and high load limit. Additionally, by a direct injection accompanied with exhaust recompression strategy [Willand et al. 1998], the fuel injected into exhaust prior to the intake process will undergo pre-ignition reactions and thus promote whole chemical reaction system. As a consequence, the operational range can be extended toward low load conditions. However, the stratified mixture resulting from late injection leads to more NOx and even PM formation.

Stratification of fuel is absolutely necessary for HCCI using diesel type fuels, at high load conditions. Although the HCCI combustion of diesel type fuels can be more easily achieved than with gasoline type fuels because of the diesel fuels' lower autoignition temperature, overly advanced combustion timing can cause low thermal efficiency and serious knock at high load conditions. In addition, mixture preparation is a critical issue. There is a problem getting diesel fuel to vaporize and premix with the air due to the low volatility of the diesel fuel [Christensen et al., 1999; Peng et al., 2003]. Many of investigators [Ryan and Callahan,1996; Christensen et al., 1999; Helmantel and Denbratt, 2004; Ra and Reitz, 2005] have indicated the potential for HCCI to reduce NOx and PM emissions. However, premixed HCCI is not likely to be developed into a practical technique for production diesel engines due to fuel delivery and mixing problems.

This has led to the consideration of alternative diesel-like fuel delivery and mixing techniques, such as early direct-injection HCCI and late direct-injection HCCI, which produce a stratification of equivalence ratio. Early direct-injection has been perhaps the most commonly investigated approach to diesel-fueled HCCI. By appropriate configuration of the cylinder, fuel mixing with air and EGR can be promoted. However, the injector must be carefully designed to avoid fuel wall wetting, which can result in increased UHC emissions and reduced thermal efficiency [Akagawa et al., 1999]. If mixing is not achieved, NOx and PM formation will be enhanced. Combustion phasing remains a critical issue in this kind of HCCI. The UNIBUS (UNIform BUIky combustion System) using early direct-injection, which was introduced into production in 2000 on selected vehicles for the Japanese market, chose a dual injection strategy [Yanagihara, 2001]. Su et al. [2005] used multi-injection modes. The injection rate pattern, the mass ratios between pulses and the pulse number have been proved to be very important parameters in achieving acceptable results.

One of the most successful systems to date for achieving diesel-fueled HCCI is late-injection DI-HCCI technique known as MK (modulated kinetics) incorporated into their products of the Nissan Motor Company. In the MK system, fuel was injected into the cylinder at about 3 CAD ATDC under the condition of a high swirl in the special combustion chamber. The ignition delay is extended by using high levels of EGR [Mase et al. 1998; Kimura et al., 2001].

The effectiveness of combustion retardation to reduce pressure-rise rates increases rapidly with increasing temperature stratification. With appropriate stratification, even a local stoichiometric charge can be combusted with low pressure-rise rates. Sjöberg et al. [2005] suggested that a combination of enhanced temperature stratification and moderate combustion retardation can allow higher loads to be reached, while maintaining a robust combustion system. The effect of EGR stratification also takes a role in enhancing stability through fuel and temperature stratifications. Controlling the coolant temperature also extends the operational range for a HCCI combustion mode [Milovanovic et al., 2005]. Additionally, Since MTBE and ethanol have low cetane numbers, two additives mixing in diesel fuel could delay overly advanced combustion phasing [Akagawa et al., 1999]. Moreover, water injection also improved combustion phasing and increased the duration of the HCCI, which can be used to extend the high load limit [Nishijima et al., 2002]. However, UHC and CO emissions increased for all of the cases with water injection, over a broad range of water loading and injection

A multi-pulse injection strategy for premixed charge compression ignition (PCCI) combustion was investigated in a four-valve, direct-injection diesel engine by a computational fluid dynamics (CFD) simulation using KIVA-3V code coupled with detailed chemistry[9]. The effects of fuel splitting proportion, injection timing, spray angles, and injection velocity were examined. The mixing process and formation of soot and nitrogen oxide (NOx) emissions were investigated as the focus of the research. The results showed that the fuel splitting proportion and the injection timing impacted the combustion and emissions significantly due to the considerable changes of the mixing process and fuel distribution in the cylinder. While the spray, inclusion angle and injection velocity at the injector exit, can be adjusted to improve mixing, combustion and emissions, appropriate injection timing and fuel splitting proportion must be jointly considered for optimum combustion performance.

The study conducted on simulations using the KIVA-3V code [10], which was improved by introducing several sub models, as shown in Table 1. The sub models introduced have been tested has been suggested that these new sub models are more appropriate for diesel PCCI combustion. For resolving the turbulent flows in cylinder, the Renormalization Group (RNG) k-ε turbulence model [11] was used. The heat transfer from the wall was computed by the model developed by Han and Reitz [12] which counted the variations of both gas density and the turbulent Prandtl number in the boundary layer. The spray process was modeled by a particle method, where the break-up processes of injected droplets were simulated by a Kelvin-Helmholtz Rayleigh-Taylor (KH-RT) model [13]. The collision model used here was one developed by Nordin [14], with improved grid independence. The interaction between spray and wall was represented by the model introduced by Han et al. [15], which considered the effects of gas density variation in simulating the size of secondary droplets in splashing. The CHEMKIN [16] solver was coupled with the KIVA-3V code to compute the chemical reactions. A reduced n-heptane reaction mechanism [17] was used to simulate diesel fuel chemistry, where the soot formation was solved by a phenomenological model, and NOx formation was represented with the extended Zeldovich mechanism. From the simulation results of Jia et al. [17], it has been shown that fair agreement with experimental results could be achieved when the simulation method was used.

Table 1. Computational Sub models.

Turbulent Model	RNG k-ε Model [11]
Break Up Model	KH-RT Model [13]
Collision Model	Nordin Model [14]
Splash Model	Han et al. Model [15]
Heat Transfer (wall)	Han-Reitz Model [12]
Combustion	CHEMKIN [16]
Fuel Chemistry Reduced	n-Heptane Mechanism [18]
Soot Model	Phenomenological Model [17]
NOx Mechanism Extended	Zeldovich Mechanism [17]

A study done on High Speed Direct Injection (HSDI) diesel engine, as experimentally investigated by Lee [19]. The engine has four valves and a displacement of 0.477 L per cylinder. Most of engine parameters were kept the same as in Lee's original experiments for the model Validation, except some modifications made to the injector for further combustion simulations. The original nozzle with eight orifices of 0.133 mm diameter was replaced by a nozzle with six orifices of 0.11 mm diameter. The specifications for the engine are listed in Table 2, while the details for the injection system are given in Table 3.

Table 2. Engine Specifications.

Engine Type	4 valve HSDI diesel
Bore × Stroke	82.0 mm × 90.4 mm
Compression Ratio	16.0:1
Displacement	477 cm3
Combustion Chamber	Open Crater Type Bowl
Intake Ports	1 helical + 1 direct port
Swirl Ratio	(at IVC) 1.83 ~ 3.30 (1.85)
Piston Bowl Diameter	48 mm
Squish Height	1.61 mm
IVO/IVC	350 °CA ATDC/−142 °CA ATDC
EVO/EVC	142 °CA ATDC/368.5 °CA ATDC

Table 3. Injection System Specifications.

Injector Type Controlled	Electro hydraulically
Nozzle Type	Dual guided VCO
Injection Pressure	1,800 bar maximum
Flow Number	400 cm3/30 s @ 100 bar
Included Angle	120°, 130°, 140°
Number of Orifices	6
Orifice Diameter	0.11 mm

Further two more fuel splitting strategies studied (10% and 30% fuel in the first injection pulse) which were simulated, and the comparisons of emissions observed that, at −20° CA ATDC, NOx emission increased with the increase of the fuel amount in the first injection, while soot emissions showed low sensitivity and remained at a very low level (less than 0.1 g/kg-fuel). Later at −20° CA ATDC, the trend switched to an opposite direction with NOx emissions decreasing when the fuel amount increased for the first injection. Meanwhile, soot emissions increased as well. This suggests that, for the cases where the fuel is injected prior to occurrence of combustion in the first injection, the mixture at ignition was richer. This would result in higher temperature during combustion, and thus more NOx formation. For the cases where more fuel was injected in the first injection, it means less fuel is left to be oxidized in the second heat release phase. Lower combustion temperature can be expected for these cases, with lower NOx formation. As the proportion of fuel in the first injection increased, both CO and UHC emissions increased. The range of that increase grew when the start of the main injection was retarded.

The effect of the spray targeting angel, three different spray inclusion angles (120°, 130°, and 140°) were studied. As shown in Figure 1, more fuel will be captured in the squish region with a larger inclusion angle. When the inclusion angle increased from 120° to 130°, no significant differences appear in NOx emission and UHC emission, as shown in Figure 11. Only slight disadvantages were observed in the increase of CO emission for this structural change. Once the inclusion angle was enlarged more to 140°, NOx emissions increased visibly, and CO emissions dropped simultaneously. This demonstrated that higher combustion efficiency and higher in-cylinder temperature resulted from a higher inclusion angle. For this increase in the inclusion angle, soot emissions were reduced when the start of second injection was retarded to later than −15° CA ATDC.

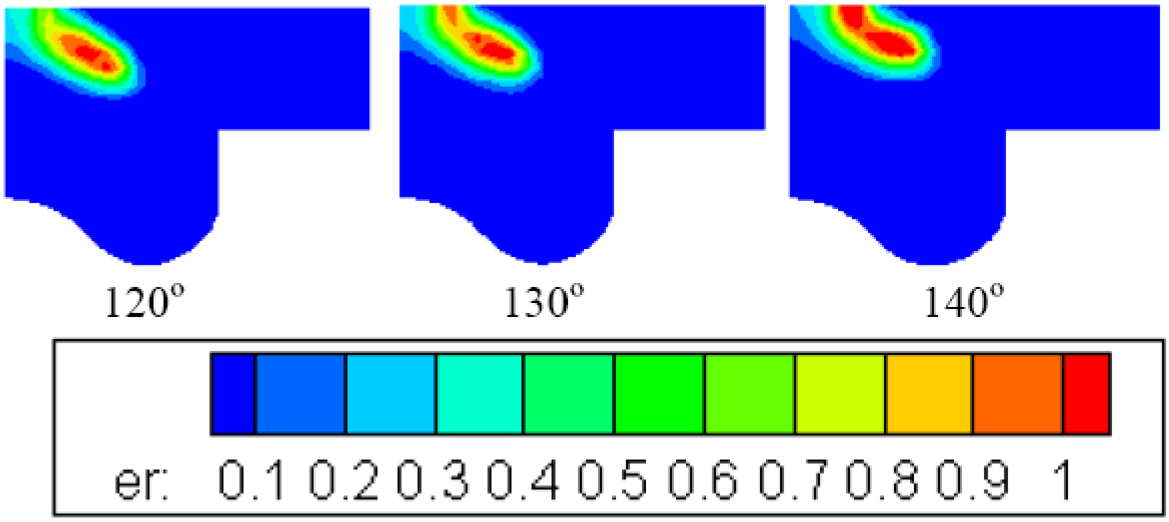

Figure 1. Equivalence ratio distribution at 2° CA after the first injection with different inclusion angles = −30° CA ATDC.

Many Numerical and experimental investigations were presented with regard to homogeneous- charge compression- ignition for different fuels. In one of the dual fuel approach, N-heptane and n-butane were considered for covering an appropriate range of ignition behavior typical for higher hydrocarbons[20].Starting from detailed chemical mechanisms for both fuels, reaction path analysis was used to derive reduced mechanisms, which were validated in homogeneous reactors and showed a good agreement with the detailed mechanism. The reduced chemistry was coupled with multi zone models (reactors network) and 3D-CFD through the Conditional Moment Closure (CMC) approach. In 2002 a study introduces a modeling approach for investigating the effects of valve events In a model based control strategy, to adapt the injection settings according to the air path dynamics on a Diesel HCCI engine, researcher complements existing air path and fuel path controllers, and aims at accurately controlling the start of combustion [21-]. For that purpose, start of injection is adjusted based on a Knock Integral Model and intake manifold conditions Experimental results were presented, which stress the relevance of the approach. Simulations of combustion of direct injection gasoline sprays in a conventional diesel engine were presented and emissions of gasoline fueled engine operation were compared with those of diesel fuel [22]. A multi-dimensional CFD code, KIVA-ERC-Chemkin, that is coupled with Engine Research Center (ERC)-developed sub-models and the Chemkin library, was employed. The oxidation chemistry of the fuels was calculated using a reduced mechanism for primary reference fuel, which was developed at the ERC. The results show that the combustion behavior of DI gasoline sprays and their emission characteristics are successfully predicted and are in good agreement with available experimental measurements for a range of operating conditions. It is seen that gasoline has much longer ignition delay than diesel for the same combustion phasing, thus NOx and particulate emissions are significantly reduced compared to the corresponding diesel cases. The results of parametric study indicate that expansion of the operating conditions of DI compression ignition combustion is possible. Further investigation of gasoline application to compression ignition engines is recommended. Simulations include direct in-cylinder fuel injection, and feature direct coupling between the stochastic Lagrangian fuel-spray model and the gas-phase stochastic Lagrangian PDF method. For the conditions simulated, consideration of turbulence/chemistry interactions is essential. Simulations that ignore these interactions fail to capture global heat release and ignition timing, in addition to emissions. For these lean, low-temperature operating conditions, engine-out NOx levels are low and NOx pathways other than thermal NO are dominant. Engine-out NO2 levels exceed engine-out NO levels in some cases. In-cylinder in homogeneity and unmixedness must be considered for accurate emissions predictions. These findings are consistent with results that have been reported recently in the HCCI engine literature.

4. HYDROGEN FUELED INJECTION SYSTEM - AN ANALYSIS

A fuel injection system performs two basic functions: fuel pressurization and fuel metering. When dealing with gaseous fuels, only the metering function is required to be carried out by the injection system as the pressurization is performed separately. The use of hydrogen direct injection in a diesel engine gave a higher power to weight ratio when compared to conventional diesel fuelled operation, with the peak power being approximately 14% higher[23]. The use of inlet air heating was required for the hydrogen-fuelled engine to ensure satisfactory combustion, and a large increase in the peak in cylinder gas pressure was observed. Reduced nitrogen oxides emission formation and a significant efficiency advantage was found when using hydrogen as opposed to diesel fuel, and it was argued that the latter is mainly due to reduced heat transfer and engine frictional loss. Hydrogen stored at pressures in the order of 200 bar does not require an energy supply (in the form of a fuel pump) to pass the fuel through the nozzle into the combustion chamber. In the experimental set-up, a pressure regulator is fitted in the fuel line, and a needle isolating valve and a flame trap are also inserted before the hydrogen injector. As in conventional direct injection engines, the design of the injection nozzle is critical for efficient engine operation, since by optimization the hydrogen flow characteristics can be obtain the required dynamic response to achieve good injection control, the way the pressurized hydrogen will spread within the combustion chamber and mix with the air.

With the inlet air heating system, stable engine operation was achieved in hydrogen direct injection mode. Despite a reduced air mass flow through the engine due to the inlet air heating, an increase of more than 14% in peak power was achieved for the hydrogen-fuelled engine compared to conventional diesel operation. This is due to the higher heating value of hydrogen per standard kg of charge air, in-cylinder gas pressure for one cycle in hydrogen-fuelled mode at full load. The fast combustion process with a rapid pressure rise can be seen, and a very high peak pressure was obtained, more than 30% higher than when operating in conventional, diesel-fuelled mode, which is due to the high engine power output. the engine efficiency results for the different operating modes. Engine efficiency was significantly higher in hydrogen direct injection (DI) mode, with the engine achieving a brake efficiency of 42.8%, compared with 27.9% when using diesel fuel, this is mainly due to lower losses to the cooling system, which constitute engine frictional losses and heat transfer losses, mainly to the combustion chamber walls. The frictional losses are not heavily influenced by the choice of fuel, but the increased engine power makes the relative influence of the mechanical losses lower in hydrogen-fuelled mode. Reduced in-cylinder heat transfer losses are expected in the hydrogen-fuelled engine due to the properties of the gaseous fuel, leading to enhanced fuel-air mixing, thereby reducing peak gas temperatures, and the lower inertia of the fuel, reducing the problems associated with spray-wall impingement. The direct injection of hydrogen allows much better control of engine operation compared to when operating in port-injected, HCCI mode. Consideration must be given to the control of injection timing and duration, as these variables heavily influence factors such as rate of pressure rise and maximum combustion pressure. Direct injection offers the possibility to control and limit excessive mechanical loads whilst this is virtually uncontrolled in the HCCI mode of operation.

The Concept of Injection system based on assuming that hydrogen behaves as a perfect gas and applying basic compressible flow theory, the flow from the storage tanks to the injector sac volume and nozzle exit can be modeled to obtain initial design guidelines. Since the fuel injection in a direct injection engine occurs in a region of 40 crank angle degrees around TDC, the cylinder pressure against which the injector has to operate may vary from approximately 3MPa at the start of injection to approximately 8MPa at the peak pressure conditions, shortly after TDC. It is interesting to note that due to the low pressure difference between the hydrogen supply and the in cylinder gas pressure, the flow of fuel from the injector will vary as the cylinder pressure changes. This gives a fuel injection rate profile different to that known from conventional engines, and must be taken into account in the design concept of the injection system. To achieve appropriate fast response times required for the hydrogen injector, a fast actuator is required. Also Due to the high self-ignition temperature of hydrogen, heating of the inlet air may be necessary to ensure fuel auto ignition The air heating control system ensures that the correct compression temperature for fuel auto ignition is reached, and allows investigations into the influence of this operational variable on the engine performance. The minimum air inlet temperature was found to be 80 ◦C, since this particular engine compression ratio was limited to 17:1, and this temperature was used in the experiments reported. The Auto-ignition of the hydrogen jet can be obtained based on the work of Tsujimura et al. [24], illustrates the strong dependence of the cylinder charge temperature on the auto ignition delay of the hydrogen jets. It can be seen that for temperatures below approximately 1100K, the auto ignition delay increases rapidly and becomes significantly longer than for higher temperatures. The auto ignition delay is strongly dependent on the ambient gas temperature, and the temperature dependency follows an Arrhenius function .It was found that for temperatures below 1100K, the auto ignition delay is longer than that of conventional diesel fuels, but much shorter delays can be obtained if the cylinder charge temperature is close to or above 1100K.

5. SUMMARY.

While HCCI has been known for some twenty years, it is only with the recent advent of electronic engine controls that HCCI combustion can be considered for application to commercial engines. Even so, several technical barriers must be overcome before HCCI engines will be viable for high-volume production and application to a wide range of vehicles.HCCI amounts to detonation of the entire charge all at once. First is, NOx reduction is usually a driver for HCCI combustion solutions. This can only be achieved with an ultra lean mixture (due to the cooling effect of charge dilution) , Second is, detonation is a violent event. Existing commercial engine structures are incapable of withstanding ongoing detonation of a stoichiometric mixture. True HCCI with early fuel injection is a nightmare to control. The temperature at end of compression affects the moment of ignition, but the temperature at end of compression is different if the outside temp is -30 C compared to if it is +30 C. The effects of fuel splitting proportion, spray inclusion angle, and velocity at the injector were observed and concluded that the fuel velocity at the injector significantly impact the combustion and emissions due to the considerable changes to the mixing process and fuel distribution in the cylinder. For the cases where all the fuel was injected before the start of ignition, reducing the fuel amount in the first injection can achieve lower NOx emissions. To the contrary, when increasing the fuel amount in the first injection one can obtain better NOx emissions when the start of the main injection is retarded after ignition. A prototype high pressure hydraulic unit injector will be implemented to assess the effects of multiple injections on diesel HCCI combustion regimes – Use of cylinder pressure-based sensing for engine control will be analyzed – Ignition characteristics of diesel fuel will be explored in engine experiments – Efficient methods for including detailed kinetics in multidimensional models will be implemented and tested.

Still various studies to be taken in to consideration on control of ignition timing & with EGR effect. The further study has to be done on various injected gas fuel mixtures and intake air mixing of injector nozzle multi holes are very good and may better . In the best investigated results above, the injected fuel mixture and intake air has been mixed spread evenly in combustion chamber for the best trend air-fuel mixing. The mixing has been happen in all the side wall, more turbulence and go to the center of combustion chamber. The mixed of air-fuel is occur in the center of piston surface. Work still has to develop on reliable hydrogen injector so that it may produce more accurate control of the fuel injection. By taking into consideration the properties of hydrogen fuel, still better results can be obtained if an optimized combustion chamber design is developed for various engine operations.

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