DEVELOPMENT OF PRE-MIXED CHARGE COMPRESSION IGNITION ENGINE (PCCI) OPERATING ON VARIABLE COMPRESSION RATIO WITH VAPORIZED DIESEL

A Thesis Submitted To The
University of Petroleum and Energy Studies

For the award of
Doctor of Philosophy
in
Mechanical Engineering

BY
Swapnil Sureshchandra Bhurat

October 2020

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Dehradun-248007; Uttarakhand

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May 2021

DECLARATION

I declare that the thesis entitled **"Development of Pre-mixed Charge Compression Ignition Engine (PCCI) Operating on Variable Compression Ratio with Vaporized Diesel"** has been prepared by me under the guidance of Dr. Shyam Pandey, Professor of Mechanical engineering, University of Petroleum and Energy Studies, Dehradun and Dr. Venkateswarlu Chintala, Founding Associate Professor, School of Engineering and Applied Science, NRTI, Vadodara. No part of this thesis has formed the basis for the award of any degree or fellowship previously.

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ABSTRACT

The present experimental work is mainly based on developing a partially pre-mixed charge compression ignition (PCCI) engine operated with the fuel vapouriser and exhaust gas recirculation (EGR) system. The heterogeneous air-fuel mixture is the main drawback of the naturally aspirated conventional diesel engine. An attempt is made to overcome this drawback by developing the PCCI engine set up. This technology is producing lower Nitrogen Oxide (NO_x) and smoke emissions. However, PCCI combustion also has the limitation of producing higher hydrocarbon (HC) and carbon monoxide (CO) emissions compared to a conventional diesel engine. To overcome this limitation, an attempt has been made by replacing the default, hemispherical combustion chamber (HCC) with the modified, toroidal combustion chamber (TCC). Therefore, the main objective of this research work was to run the conventional direct injection diesel engine in PCCI mode with a toroidal combustion chamber to reduce the harmful HC, CO, NO_x, and smoke emissions.

In this research work, the conventional, single-cylinder, 1500 rpm, direct injection (DI), water-cooled variable compression ratio engine was modified to the PCCI engine. For transforming into a PCCI engine, a few modifications were required to be carried out on the conventional test rig. Firstly, the intake manifold was modified to allow fuel vapour entry and exhaust gas through two different ports. Secondly, the HCC piston was replaced with TCC geometry for the enhancement of engine

characteristics. External power supplied fuel vapouriser was also developed to convert the conventional fuel (diesel) into vapour.

The experiments were conducted on the PCCI engine set up in three phases. In all three phases, the engine was tested for the same engine characteristics. The performance characteristics of an engine were evaluated by knowing the brake thermal efficiency (BTE), volumetric efficiency, and exhaust gas temperature (EGT). The combustion characteristics of an engine were evaluated by knowing the combustion pressure, rate of pressure rise, ignition delay, net heat release rate with the help of a data acquisition system (DAQ). The emissions characteristics of an engine were evaluated by testing the various emissions hydrocarbon (HC), Carbon monoxide (CO), oxides of nitrogen (NO_x), and smoke opacity from the engine with the help of a gas analyser and smoke meter. In the first experimental condition, the conventional engine was run at three compression ratios. This experimental study explores the effects of compression ratio (16:1, 17:1 and 18:1) and piston geometry, namely hemispherical combustion chamber (HCC) and toroidal combustion chamber (TCC), on a direct injection compression ignition engine behavior. As mentioned earlier, the test engine was operated with exhaust gas recirculation (EGR) at a constant speed of 1500 rpm. With the increased compression ratio of the engine, TCC piston geometry has shown better improvement in brake thermal efficiency, carbon monoxide (CO) and hydrocarbon (HC) emissions than HCC. However, a slight penalty in NO_x emission was observed with increasing compression ratio and TCC piston geometry. In-cylinder peak pressure, net heat release rate (NHRR), and rate of pressure rise (RoPR) were increased significantly with increasing compression ratio and the use of TCC geometry. It is concluded that simultaneous modification of piston geometry and compression ratio is a promising option for better performance and lower emissions behavior of the engine.

In the second and third experimentation, the engine was operated in PCCI mode. In the second experiment, experiments were carried out to understand the comparison of HCC and TCC geometry in the PCCI engine operation. In this investigation, the engine was run on a fixed compression ratio (18:1) at all the engine loads. The tests were conducted with 2 ml/min and 4 ml/min diesel vapor for both HCC and TCC piston geometries and compared it with the conventional DI diesel engine results. As mentioned earlier, the required engine characteristics were investigated. It has been found that the BTE at full engine load was increased by 3.83% by using TCC geometry as compared to HCC in PCCI mode with 2 ml fuel vapour. Compared to a conventional engine, BTE was slightly lower in PCCI mode with both the piston geometries. TCC piston is providing an advantage by lowering the percentage decrement in BTE in PCCI mode compared to a conventional engine. Due to better combustion in TCC geometry, HC and CO emissions were found to be decreased in PCCI mode compared to HCC. However, increment in combustion temperature and marginal pressure increment in NO_x and smoke opacity was observed with TCC geometry. The increment in NO_x emissions was compensated by using the limited quantity (10%) of EGR without much affecting the brake thermal efficiency of the PCCI engine.

Many pre and post-treatment of exhaust gas strategies have been instigated in compression ignition engines to meet the stringent vehicular emissions norms. The (PCCI) combustion technique has shown a significant reduction in emissions without much affecting the efficiency of the conventional diesel engine by emitting low nitrogen oxides and smoke emissions simultaneously. However, PCCI has also shown some limitations by producing higher hydrocarbon carbon monoxide emissions from the engines. Toroidal combustion geometry has proved its potential in reducing the penalty in HC and CO emissions from the conventional engine. Therefore in the current experimental study, PCCI engine characteristics were investigated with toroidal piston geometry. The homogeneous mixture for PCCI combustion was obtained with an external mixture formation technique with the help of a fuel vaporizer (retrofit device). The experiments were performed with an 18:1 compression ratio on PCCI mode. The tests were conducted with 2 ml/min and 4 ml/min diesel vapor without and with 10% exhaust gas recirculation and then compared with the conventional engine. The investigation results show that due to better air-fuel mixing results in low engine emissions. NO_x emissions were reduced by 29% and 31%, and smoke opacity was decreased by 33% and 38% with 2 ml/min and 4 ml/min diesel vapor with EGR with a marginal decrement in thermal efficiency. Hence the recommendation was to use the PCCI engine set up equipped with fuel vapouriser, with low fuel vapour quantity to enhance the engine emission characteristics over the conventional CI engine.

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LIST OF SYMBOL

A-F	Air-fuel ratio	BTE	Brake thermal efficiency
BSFC	Brake Specific Fuel Consumption	CA	Crank Angle
CI	Compression ignition	CO	Carbon monoxide
CR	Compression Ratio	DI	Direct injection
DAQ	Data Acquisition System	EGR	Exhaust gas recirculation
CC	Combustion chamber	FV	Fuel vapor
EGT	Exhaust gas temperature	HCC	Hemispherical Combustion Chamber
HC	Hydrocarbon	IC	Internal Combustion
HCCI	Homogeneous charge compression ignition	MF	Mass Fraction
ID	Ignition Delay	NO_x	Oxides of nitrogen
NHRR	Net heat release rate	PM	Particulate Matter
NA	Naturally aspirated	PCCI	Partially pre-mixed charge compression ignition
ROPR	Rate of pressure Rise	SI	Spark Ignition
TCC	Toroidal Combustion Chamber		

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CHAPTER 1 INTRODUCTION

The diesel engines are the most consistently used chemical conversion device for enormous power and transportation applications because of their better combustion efficiency and economics (Challen B, 1999). However, the adverse effect of compression ignition (CI) engines on the environment has led many country's attention to imposing limits on increasing exhaust emissions (Johnson, 2010). The annual energy outlook revealed that the transportation sector's total energy consumption was 38%, and global liquid fuel consumption was 68% in 2018. If it continues in the same way, then liquid fuel usage for the transportation sector may rise to 72% in 2035 (Outlook, 2001). In the Indian transportation sector, CI engines contribute significant share because the Original Equipment Manufacturers (OEMs) strive to achieve high performance and low emissions to meet Bharat stage-VI emission norms that are implemented in India w.e.f. 1st April 2020. The existing after-treatment technologies like diesel particulate filter (DPF), selective catalytic reduction (SCR) providing encouraging solutions to reduce the harmful emissions like nitrogen oxides (NO_x), hydrocarbon (HC), carbon monoxide (CO), and soot emissions (Adler, 2005). However, due to engine performance limitations, clogging of the filter generates backpressure; nevertheless, it requires frequent preventive maintenance. There are two possible ways of reducing emissions from the CI engines. Firstly, to use alternate fuels like biodiesel (Kinnal, Sujaykumar, D'costa, & Girishkumar, 2018; Piloto-Rodríguez et al., 2019), plastic oil (Damodharan, Sathiyagnanam, Rana, Kumar, & Saravanan, 2018), and hydrogen (Chaichan, 2018). However, due to the high production cost, the fuel mentioned above could not be used as an alternate to diesel fuel in IC engines. Secondly, it necessitates modifying engine technology to reduce engine emissions (Heywood, 1988). Another way seems to be more viable if the newer advanced engine technology can be made so that it can be retrofitted on the existing engines and future engines.

Research studies conclude that emission norms have always challenged engine design (Roy, Banerjee, & Bose, 2014). It is becoming unavoidable to ignore the requirement of cutting down the emissions from internal combustion engines since it is also hazardous to human health, particularly the soot formation and NO_x emission in diesel engines (Sydbom et al., 2001). The exhaust gas recirculation (EGR) has led to a considerable decrease in NO_x emissions in direct injection (DI) diesel engines due to its effectiveness in diluting excess oxygen in the intake air (Dürnholz, Eifler, & Endres, 1992).

In order to meet the mandated emission norms, it is essential to provide alteration in the piston bowl geometry for better air-fuel (A-F) mixing throughout the combustion chamber, thereby reducing emissions as well as high engine performance (De Risi, Donateo, & Laforgia, 2003; Saito, Daisho, Uchida, & Ikeya, 1986). Jaichandar et al. (S Jaichandar & Annamalai, 2012) investigated the effect of the Toroidal re-entrant combustion chamber (TRCC) and hemispherical combustion chamber (HCC) on brake thermal efficiency (BTE) and emission characteristics of a single-cylinder direct injection CI engine. It was found that the BTE was 33.07% with TRCC and 31.48% with HCC. Also, 20.7% in hydrocarbon (HC) emissions reduction were noted with TRCC. However, NO_x was increased by 9% in TRCC, i.e., 784 ppm and 712 ppm for TRCC and HCC, respectively. Jyothi et al. had investigated the effect of toroidal combustion chamber (TCC) geometry on a single cylinder CI engine (Jyothi & Reddy, 2017). It was found that BTE increased by 2.94%, BSFC decreased by 1.3% as compared to HCC geometry. HC and CO emissions were decreased by 2% and 3.5% respectively. However, NO_x emissions increased by 3.5% by using TCC geometry (Jyothi & Reddy, 2017).

The heterogeneous mixture of diesel-air is another critical parameter, atomization/vapourization of injected fuel to form a combustible mixture formation. Subsequently, this phenomemnon has considerable effects on ignition delay as NO_x, PM, and HC emission.

Homogeneous Charge Compression Ignition (HCCI) technology is a promising one that targets both performance and emissions of CI engines. The HCCI combustion technique reduce the NO_x and PM emissions without much compromise in thermal efficiency (A. K. Agarwal, Singh, Lukose, & Gupta, 2013; Maurya & Agarwal, 2014; G. Singh, Singh, & Agarwal, 2014). However, it has a major drawback of producing high HC and CO emissions, and uncontrolled combustion when fueled with high cetane rating fuel like diesel (Najt & Foster, 1983). Little modification on naturally aspirated CI engines makes it possible to modify its combustion phenomenon, and it can operate on HCCI mode. In this context, the combustion chamber (CC) plays a vital role in achieving better performance and emissions characteristics of direct injection CI engines. On the other hand, engine characteristics of CI engines can be better only when a high degree of air swirl and turbulence occurs in the combustion chamber during first two strokes of four-stroke engine. The swirl-squish interaction in the combustion chamber produces the turbulent flow field when the piston moves towards the top dead center (TDC), which could be obtained by changing the default piston geometry of the HCC (Arcoumanis, Bicen, & Whitelaw, 1983; Ogawa, Matsui, Kimura, & Kawashima, 1996; B. Prasad, Sharma, Anand, & Ravikrishna, 2011). The HCCI engine has claimed to be the counter for such mixture by minimizing NO_x and particulate emission and enhancing thermal efficiency (Zhao et al., 2003). However, the HCCI is limited by lean mixture formation at all engine loads. There is no control over the combustion. HCCI engines have shown the limitation at full engine load due to uncontrolled combustion (G. Bhiogade & Suryawanshi, 2016; D. Ganesh & Nagarajan, 2010).

Partially Pre-mixed Charge Compression Ignition (PCCI) engine (G. E. Bhiogade, Sunheriya, & Suryawanshi, 2017) could be another alternative to CI engines. Achieving the low emissions without affecting the engine thermal performance compared with CI engines is the prime target of this technology and it can attain through the homogeneous mixture. PCCI combustion can be achieved in two ways, either by in-cylinder (internal) or external mixture preparation strategy. The possible ways of internal mixture formation are either through early or late fuel injection in a cycle. Takeda et al. implemented the early injection technology in the single-cylinder direct injection (DI) diesel engine and noticed ten times decrement in NO_x emissions as compared to traditional engine. However, HC and CO emissions were increased drastically due to the lean air-fuel (A-F) mixture. It also showed the limitations of using the engine over a wide range of operations due to uncontrolled combustion at higher engine loads (Takeda, Keiichi, & Keiichi, 1996). In external mixture preparation strategy, A-F mixture will be formed by mounting low presure fuel injection system with air intake manifold, fumigation (Bhurat, Pandey, Chintala, & Ranjit, 2019; Pandey, Diwan, Sahoo, & Thipse, 2015; Telli, Altafini, Rosa, & Costa, 2018) or by injecting fuel via port fuel injection technique. Low volatile fuel like diesel can also be used through a fuel vaporization technique (G. Bhiogade & Suryawanshi, 2016). The complete induction or injection of fuel through an external mixture formation technique has shown a significant decrement in NO_x and smoke emissions in CI engines. However, HC and CO emissions were increased in the same manner. To cope up with this limitation, the fuel must be inducted partially through manifold induction and remaining through conventional DI technique along with a limited quantity of exhaust gas recirculation (EGR). The use of a limited proportion of EGR resolving two main issues of PCCI engines, decreasing the NO_x emissions and decreasing the net heat release rate by dilution of charge. A limited proportion of EGR can also support decreasing the HC emissions by resending the unburned gases into the combustion chamber (D. Agarwal, Singh, & Agarwal, 2011). The air intake manifold geometry is critical for supplying fresh air to the combustion chamber at maximum possible velocity concerning its volumetric efficiency (C. L. Lee, 1997). Partially premixed charge compression ignition (PCCI) compensates HCCI technology's drawbacks by altering the homogeneous air-fuel mixture and conventional direct injection at varying engine loads, hence providing control over the combustion. After scrutiny, it is clear about piston bowl geometry's effect on the swirl velocity as the piston moves towards TDC. The swirl-squish interaction in the combustion chamber produces the turbulent flow field when the piston moves towards TDC which could be obtained by changing the default piston geometry (B. Prasad et al., 2011). To minimize the emissions from PCCI engine, the alteration in the piston geometry for better A-F mixing throughout the combustion chamber would aid in minimizing the reduction in BTE in the PCCI technology along with reduced HC and CO emissions.

The current research work focused on analysing the PCCI engine's characteristics with cooled-EGR and TCC piston bowl geometry. The naturally aspirated CI engine was made to operate on PCCI mode using external mixture formation technique through fuel vaporizer. The PCCI engine operated at all the engine loads with various fuel vapour induction rates with 10% EGR and without EGR. Engine characteristics have been analyzed under various engine loads conditions along with the modified TCC and default HCC piston geometry separately.

Externally power supplied fuel vaporizer (D. Ganesh & Nagarajan, 2010; G. Singh et al., 2014) was used to obtained the homogeneous A-F mixture. The fixed (10%) proportion of EGR is used in the present study; researchers conducted experimentations with high proportions of EGR and observed that it results in a significant rise in HC and smoke emissions with increasing percentage of EGR (D. Agarwal et al., 2011; D. Ganesh & Nagarajan, 2010). The fixed amount of fuel vapor (2 ml/min and 4 ml/min) was inducted at all the engine loading conditions, and diesel is supplied by using DI injection into the comustion chamber to maintain the constant load.

1.1 Research Gaps

- A conventional diesel engine has more NO_x emission and significantly less CO and HC emissions. On the other side, PCCI has more CO and HC and fewer NO_x emissions. Research needs to be done to improve the CO and HC emissions.

- There was no attempt has been made to reduce HC and CO through changes in the piston geometry in the PCCI engine set up in the literature review.

- Researchers have reported lower BTE compared to conventional CI. An investigation needs to be done on the enhancement of brake thermal efficiency.

- Researchers have used alternative fuels in the PCCI engine set up and then investigated the engine characteristics. There could be a scope to use conventional fuel (diesel) instead of alternate fuel in PCCI mode. Fuel can be used through various techniques like fumigation of fuel in induction manifold, vapourised fuel and mixing it with intake air, port fuel injection or early injection by checking its feasibility in respective mode.

1.2 Problem Statement

The pre-mixed charge compression ignition engine suffers from the problem of higher HC and CO emissions. It can be minimized if the charge quality is improved in the PCCI engine. To overcome the same, implementing the retrofitting of fuel vaporizer and modifying existing combustion geometry (hemispherical bowl) to the toroidal combustion chamber. Thus, it improves the air-fuel mixture's homogeneity, resulting in decreased HC and CO emissions than a conventional diesel engine. PCCI combustion also minimizing the NO_x and smoke emissions by lowering the in-cylinder pressure and temperature of combustion. Therefore, the overall aim was to decrease the conventional engine's harmful emissions by using the PCCI technique.

1.3 Motivation

The new engine technology HCCI, also known as PCCI, has again motivated the researcher’s enthusiasm to work further on the internal combustion engines. The unstable fuel costs, and stricter norms have encouraged we researcher to focus on such a worthy and youthful innovation. As Prime Minister Mr. Modi, India, has already declared to switch from BS-IV to BS-VI in 2020, there will be a need to decrease NO_x emission by 76% to fulfill BS-VI emission norms (Source - Indian Emissions Regulations/ARAI). In such a case, the HCCI engine dominates conventional diesel engines. Types of HCCI have been investigated by automobile manufactures like Honda (Ishibashi, 2000), Nissan (Kimura, Aoki, Ogawa, Muranaka, & Enomoto, 1999), and Toyota (Hasegawa & Yanagihara, 2003).

To improve the blending of A/F, Nissan and Toyota have injected fuel in their Diesel variants at modified timings. The Japanese automaker Mazda Motor has presented the HCCI engine technology in the year of 2018 that offers 30% better fuel proficiency by using injection pressure, not spark plugs, to burn the fuel.

According to General Motors, an HCCI engine, when equipped with other advanced technologies, delivers 15% greater fuel economy than the conventional CI engine by radically changing the combustion process. It saves fuel and reduces the harmful emissions of CI engines (Rex-Roy, 2009).

Even though the automobile market is growing towards electric vehicles, Mazda Motor Corporation has predicted from their studies that the IC engines will keep on accounting for the huge share of new-vehicle deals in the future. With any innovation, it is essential to build up an exhaustive comprehension of the components' operation and the results of control.

The advantages of HCCI are close to diesel efficiencies with negligible emissions of NO_x and PM, which is one of the significant concerns associated with CI engines. The PCCI engine concept can effortlessly be adjusted to current engine generation. HCCI is appropriate for running at an enhanced, consistent load and speed with the developing prominence of electric vehicles in shopper markets. Due to low NO_x and PM emissions, PCCI engines could soon discover their large-scale manufacturing approach.

Regardless of PCCI benefits, higher HC and CO emissions at the start of combustion (SOC), and the span of combustion is hard to control. The absence of direct control on SOC is one of the obstructions of this technology.

1.4 Objectives

- Design and development of fuel vaporizer for the PCCI engine operating on variable compression ratios.
- Modification in the existing engine to operate on PCCI combustion mode includes modifying piston geometry and developing a mechanical EGR system.
- Comparative analysis of performance and emission characteristics of the PCCI engine with conventional diesel engine.

CHAPTER 2 LITERATURE REVIEW

2.1 Combustion in Compression Ignition engine

The auto-ignition of the air-fuel mixture initiates combustion in a CI engine. The auto-ignition occurs due to high in-cylinder pressure during compression, triggers the vaporization of diesel droplets followed by combustion. The entire combustion process complete in four stages. These phases are illustrated in Figure 2.1 with reference to the crank angle (CA).

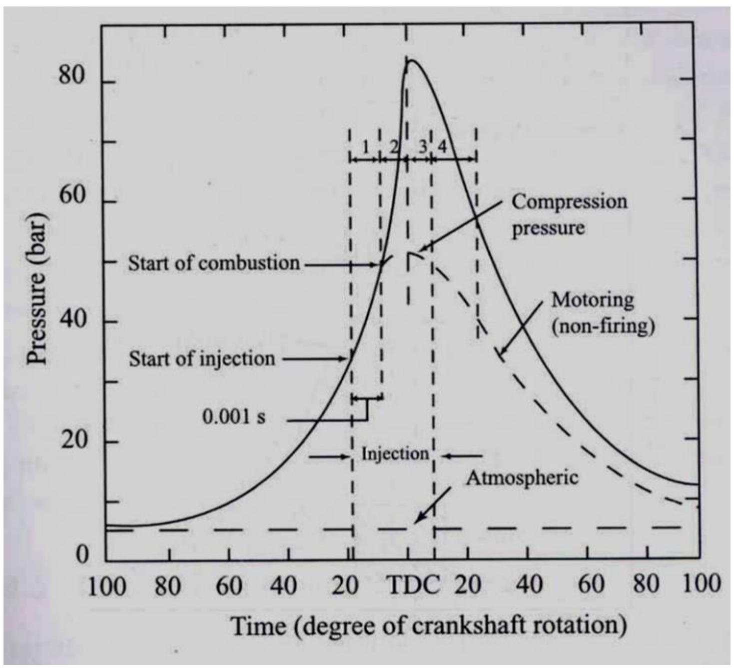


Figure 2.1: Pressure Vs. crank angle graph for combustion in CI engine

2.1.1 Stages of combustion in CI engine

Pre-flame Combustion

In a conventional CI engine, actual fuel injection begins at 15° before top dead center (bTdc). The high pressure inside the combustion chamber causes the atomization of the fuel droplet and the A-F mixture formation. The atomized fuel then starts absorbing heat from the surrounding air. During this pre-combustion state, the energy released by the process is significantly less than the heat absorption. This results in pressure decline as the vaporization progresses. This phase attains an equilibrium when the total energy involved in pre-flame combustion becomes equal to that of the total heat absorbed in vaporization. The increase in energy released continues, finally causing self-ignition at almost 8-10° bTdc. After this, the temperature and pressure inside the cylinder increase continuously, and this gap between the fuel injection and self-ignition is called Delay Period. The delay can be further classified into Physical and Chemical delay, where the physical delay is the time before the chemical process starts and chemical delay is the low chemical process rate at the beginning of combustion depending upon the surrounding temperature and other factors (Yoon & Lee, 2011).

Uncontrolled Combustion

This is the period between combustion and the crank angle at which the peak pressure is achieved. During this stage, the air-fuel mixture is heterogeneous, causing fuel accumulation during the delay period (Lujaji, Kristóf, Bereczky, & Mbarawa, 2011). This fuel burns instantaneously at the beginning of combustion, causing a precipitous rise in the in-cylinder pressure. After this, flame propagation occurs through multi-point ignition, inducing scorching in the nearby mixtures through radiation. Increase in delay period led to increase in fuel accumulation.

Controlled Combustion

Generally, the accumulated fuel gets consumed in the uncontrolled phase itself. The combustion after this stage occurs with negligible delay, which means the atomization, vaporization, air-fuel mixing, and burning occur as soon as the fuel droplets leave the injector nozzle. There is sufficient temperature, pressure, and turbulence inside the cylinder, which results in no delay. This is the prime reason why this phase is termed as the controlled combustion phase, as it can alter the combustion by changing the amount of fuel injected (Shahir et al., 2014)

Afterburning

After the decomposition of the controlled combustion phase, which contains a significant number of unburnt hydrocarbons and carbon-monoxide molecules, coming in contact with the oxygen at high temperature gets oxidized further, and the entire process is called Afterburning. This cycle terminates at the beginning of the exhaust cycle.

2.1.2 Emissions and its formation from CI engine

Oxides of Nitrogen Formation

Combustion in CI engine occurs through highly compressed hot air with the temperature ranging from 900°-1400° K. The compressed air inside the combustion chamber through the air intake manifold, and fuel is injected right before the end of the compression stroke. Generally, nitrogen present in the air sustains naturally and gets directed through exhaust and other gases. However, suppose the combustion temperature goes above 1600° K. In that case, nitrogen starts reacting with the available oxygen to form Nitrogen Oxide (NO) and Nitrogen Dioxide (NO_2), with NO comprising around 85% of the total NO_x emission. Both the gases have distinct physical and chemical properties with NO being colorless & odorless while NO_2 being reddish-brown with a pungent smell. Regardless of their color and scent, both gases are hazardous to human health. It caused 13 lakh heart and lung issues every year in India (X. Shi et al., 2005; Silitonga et al., 2013).

Particulate matter (PM)

These are generated during the combustion phase due to the accumulation of minute particles which are harmful for human respiratory system and lungs. PM formation's prime cause is the incomplete combustion of the fuel and lubricating oil. Agarwal et al. (A. K. Agarwal, 2007) concluded that PM consists of carbon-31%, unburned fuel-7%, oil-25%, sulfate-14% and water, 13% ash, and other entities. PM emission from a CI engine is 8-10 times higher than that of an SI engine. Further PM from a CI engine can be classified into three types: Inorganic fraction, soot, and Soluble organic fraction, with soot comprising 85-90% of the total emission. Inhaling these can lead to asthma, lung cancer, and other cardiovascular issues, even death if not handled properly (Abdul-Khalek, Kittelson, Graskow, Wei, & Brear, 1998; Daniyan, Bello, Ogedengbe, & Mogaji; Guttikunda & Goel, 2013).

Carbon monoxide (CO) emissions

CO emission is formed due to incomplete oxidation of carbon molecule with its amount in the exhaust gases, mainly depending on the factors like air-fuel mixture, swirl & turbulence, and delay. It is primarily formed at the high load or instantaneous acceleration, resulting in rich A-F mixture formation. However, a small amount of CO is also emitted in a lean mixture due to the chemical kinetic effect. The deficiency of oxygen cause CO to sustain in that form preventing further oxidation to CO_2. CO has a high affinity to the blood, thus gets absorbed quickly by the hemoglobin leading to asphyxiation (Dhariwal, 1997; DIAT-SANCHEZ, 1997; Faiz, Weaver, & Walsh, 1996).

Hydrocarbon emissions

Hydrocarbon emission occurs due to the insufficiency of oxygen and low temperature for Hydrocarbons to get converted into H_2O and CO_2. It occurs when the air-fuel mixture is lean, especially near the wall of the combustion chamber. These consist of Alkane, Alkene, and aromatic compounds. Although the conventional diesel engine tends to have lower HC emission due to high-temperature combustion, still some traces of HC can be found in CI at low load when the mixture is lean. Another factor contributing to this is low flame propagation speed causing incomplete combustion or a misfire. HC emission also takes place in the crankcase, venting system, and fuel system. Like other emissions, HC also harms human health, which contributes to the ground level Ozon formation, which is toxic (Demers & Walters, 1999; Yamada et al., 2011; M. Zheng & Banerjee, 2009).

2.2 Control of Emissions in diesel engines

In the CI engines, the exhaust gas is a composed of huge number of organic and inorganic solid, liquid and gaseous compounds. However, the emissions of NO_x and PM is a main concern. Attempts were made to control the emissions of NO_x and PM from the CI engines with the modifications in engine design (pre-treatment Techniques). However, it was noticed that with the engine modification approach only, emissions of NO_x and PM, cannot be reduced to the required level. Therefore, the engineers were bound to switch over to different alternatives. The techniques and devices were developed to minimise the NO_x and PM are shown in the

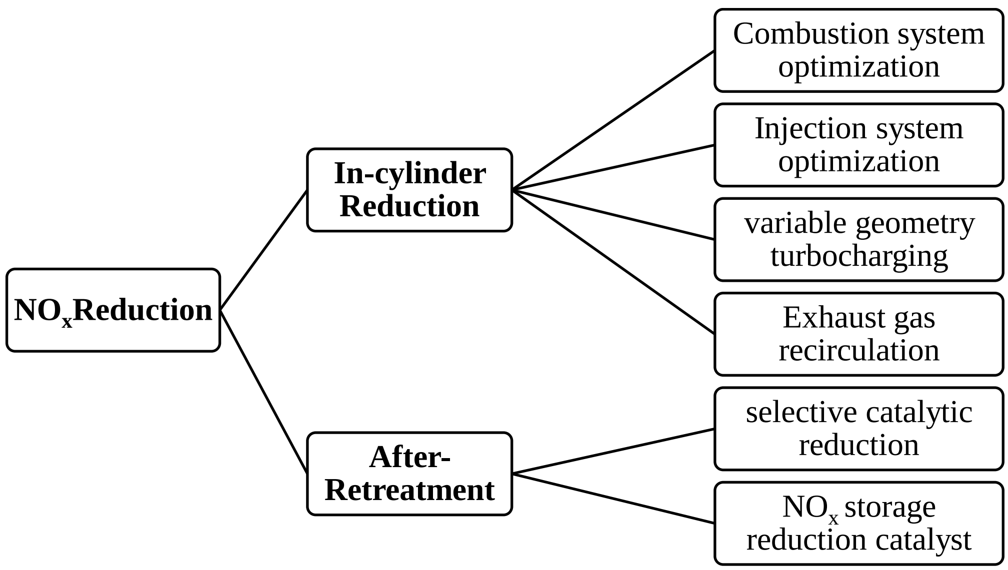


Figure 2.2: Technologies used for NO_x reduction in diesel engines

2.2.1 Turbocharging

Turbocharging provides the high-density air, greater than 50% at the direct injection engines' rated power conditions. High density and air temperature during fuel injection are achieved in turbocharged engines compared to conventional naturally aspirated engines, which makes the ignition delay period shorter, thereby decreasing the amount of fuel burned in premixed phase. With turbocharging, the fuel injection timing can be retarded, which can reduce the NO_x emissions without affecting the fuel efficiency. Conventional turbochargers show the limitations at the low engine speeds with its pressure ratio characteristics. However, it can be resolved by using the variable geometry turbocharger.

2.2.2 De-NO$_x$ catalyst

A large quantity of oxygen is available in a diesel engine exhaust system. It operates with an excess quantity of air during their operation and caters to an oxidizing atmosphere for exhaust pollutants. NO$_x$ and CO pollutants require reducing and oxidizing atmosphere, respectively, to convert itself into harmless products. The SI engine operating with a stoichiometric A-F ratio will cater to both oxidizing and reducing atmosphere in the exhaust system. In contrast, the diesel engine is only rich in an oxidizing atmosphere. Hence, a new reducing catalyst, different from conventional, is required to reduce the exhaust system's atmosphere. This reducing catalyst is known as reductants. Ammonia and hydrocarbons are used as a reducing agent or reductant in diesel engine exhaust systems. There are two ways for NO$_x$ reduction, which are explained below.

2.2.2.1 NO$_x$ Storage – Reduction (NSR) catalyst or NO$_x$ trap

NO$_x$ Storage – Reduction (NSR) catalyst or NO$_x$ trap in this method, NO is reduced to NO$_2$ in the presence of Platinum (Pt) reductant.

$$NO + O_2 \xrightarrow{Pt} NO_2$$

This NO$_2$ reacts with alkali (BaO) and form barium nitrate.

$$NO_2 + BaO \rightarrow BaO.No_2$$

This barium nitrate requires a large quantity of HC to convert itself into harmless products. For which diesel engine injection system caters spike of fuel after top dead center, while in SI engine, it is governed by a synchronized fuel injection system or made to run it over rich air-fuel ratio for a period of one second only.

$$BaONO_2 + HC \rightarrow BaO + N_2 + H_2O + CO_2$$

2.2.2.2 Selective catalytic reduction (SCR)

In this method, ammonia is used as a reducing agent to convert NO_x into a harmless product. Since the source of ammonia is urea, available in abundant, safer to handle and store, it is used in the practical application of the SCR method. This method completes in two processes as listed below.

Hydrolysis of Urea

$$\left(NH_2\right)_2 CO + H_2O \rightarrow CO_2 + 2\ NH_3$$

Conversion of NO_x

$$4\ NO + 4\ NH_3 + O_2 \rightarrow 4\ N_2 + 6\ H_2O$$

$$6\ NO_2 + 8\ NH_3 \ \rightarrow 7\ N_2 + 12\ H_2O$$

A 30 to 40% concentrated urea solution in water is used. Vanadium and titanium oxide mixture is SCR catalyst and coated on the ceramic honeycomb structure. Variation in NO_x level during the engine's operation is usual. Hence a continuous variation in ammonia (urea) injection rate is needed. About 10% of NO_2 exist in exhaust NO_x emission. A molar ratio of 0.9 of NH_x/NO_x is needed for the successful conversion of NO_x. For accomplishing this task, an excess quantity of ammonia is needed to be supplied, however, some quantity remains unused, called ammonia slip. To minimize this ammonia slip, a dynamically controlled ammonia dosage system is employed. Figure 2.3 shows the advanced SCR catalyst system for a heavy-duty diesel engine operation.

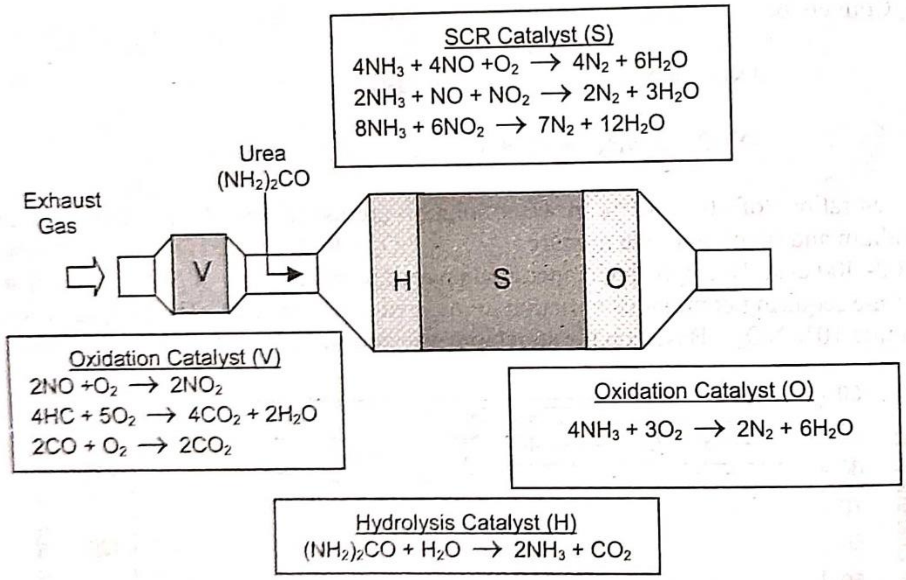


Figure 2.3: Advanced SCR catalyst system using pre-oxidation catalyst

To minimize this secondary emission of ammonia, a oxidation catalyst is employed in the SCR system.

2.2.3 Plasma catalyst system for NO_x reduction

This concept is adopted form power generation industries in which flue gases containing NO_x are treated with plasma to convert it into acid and combine with ammonia to form ammonium nitrate. The scrubber removes this. Now, this technology is adopted for automotive applications (Chun et al., 2000).

In this reduction method, plasma is used to convert NO component in NO_2, and it will be reduced to N_2 in the presence of catalyst by the hydrocarbon present in exhaust gases. This benefit of this system is that it performs NO_x conversion without depilation of hydrocarbons by oxidation as in the NSR catalyst. Secondly, it does not contribute to the conversion of SO_2 to SO_3 and sulphuric acid, responsible for the catalyst's poisoning catalyst. The best conversion efficiency is obtained with the HC/NO_x ratio of 3:1 to 6:1 over a temperature range of 300 ºC.

2.2.4 Diesel Particulate filter

This method of emission control is in use since 1980. In the DPF system, different types of filters like alumina coated wire mesh, ceramic fiber, and ceramic monoliths are employed in the exhaust system. The honeycomb structure is the most commonly used filter for filtration of exhaust gases in which gases flow through porous wall cells. It is also known as a ceramic wall-flow filter. Figure 2.4 depicts the structure of the ceramic wall filter.

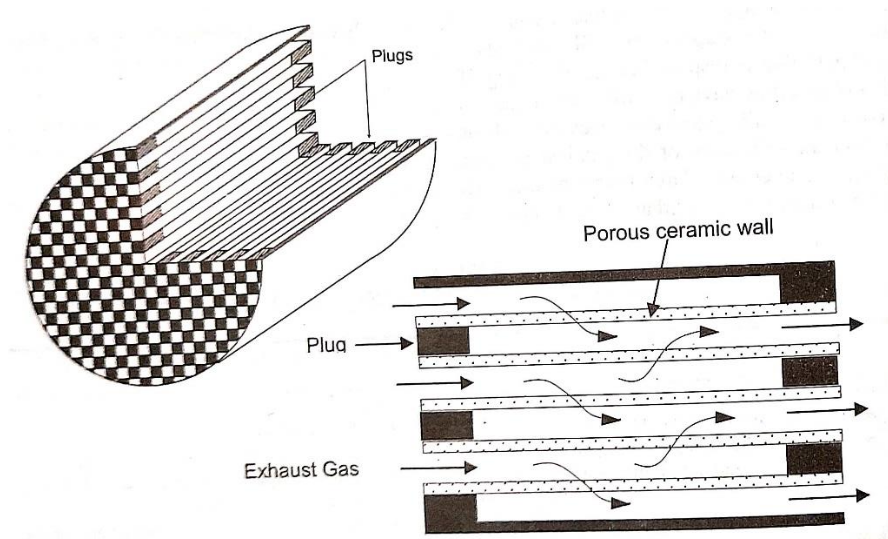


Figure 2.4: Ceramic wall-flow filter for diesel particulate and exhaust gas flow during filtration

Alternate rows (cells) are plugged at one end while the same are open at another end. Gases enter in open cells in upstream and change the path flow by entering adjacent cells during the filtration process, are exhausted through open cells of downstream. The efficiency of filtration depends on porosity (pore size) and density of the filter. Hence, it needs to be optimized pores size and density to achieve the best efficiency point. A less dense filter will not perform as expected, while a highly dense filter causes pressure drops of exhaust gases during filtration. Efficiency close to 98% is achievable. Porous cordierite ceramic ($2MgO_2.Al_2O_3.5SiO_2$) is a commonly used material for the DPF because of its chemical inertness, low coefficient of thermal expansion, and high melting points (1400 ºC).

2.2.5 Regeneration of DPF

As listed above, there are various methods available for exhaust emission filtration. However, the problem is concerning with high-pressure drop in the filtration process due to the collection of soot particles, affecting fuel economy. According to design consideration and study, soot loading of 10g/l of filter volume increases the pressure drop 3-4 times, which results in an increment in backpressure by 350 mm of water. It reduces fuel economy by 1% at 65 Km/h vehicle speed. Hence regeneration of the DPF became a necessity.

The regeneration process is to bring the filter in its original clean state after the filtration process. In the regeneration process, soot collected in the filter is oxidized to carbon dioxide without hampering the filter, mainly cracking and melting under high temperature. This regeneration process has two methods, namely the active system and passive system. An additional heat source engine over fueling is employed in an active system to increase exhaust gas temperature for starting soot oxidation. This increase in temperature is achieved by engine throttling and electric heater or burner in the filter's upstream. The passive regeneration system catalyst is incorporated to decrease soot particle combustion temperature to average exhaust gas temperature. This catalyst is incorporated by adding it in fuel as an additive or coated on the filter substrate's surface.

2.3 The need for HCCI/PCCI engine

Despite the booming scenario of e-vehicles, diesel engines are still extensively used for mass transportation, public transportation, ships, small-scale power generation, and power generation in remote areas. This could be due to their significant advantages such as higher torque, higher fuel economy, higher efficiency, low-cost diesel fuel, and low carbon-based emissions. However, these diesel engines face challenges in terms of NO_x and particulate matter pollutants due to the combustion of heterogeneous A-F charge (H.-W. Wu, Wang, Ou, Chen, & Chen, 2011).

Heterogeneous charge in diesel engines is formed mainly due to the short residence of fuel to mix with air and poor fuel injection characteristics. The HCCI technology is one of the promising options to address heterogeneous charge formation problems and, thus, NO_x and PM pollutants formation. On the other hand, in many countries, stringent diesel emission legislations are gaining the focus of Original Engine Manufacturers (OEMs) towards emission reduction technologies (Hoekman & Robbins, 2012). For example, NO_x emission was targeted to reduce 3.6 g/kWh under Euro-IV to 0.5 g/kWh under Euro-VI norms, as seen in Figure 2.5. In the same direction, India has also declared to hop from Bharat stage IV (BS-IV) to BS-VI, so it requires NO_x reduction about 76% to achieve BS-VI norms. In order to deal with future emissions, researchers are now concentrating on in-cylinder combustion technologies such as low combustion temperature, homogeneous charge combustion, and EGR. In HCCI engines, combustion occurs at a low temperature that can reduce NO_x and reduce soot formation because of the low equivalence ratio. Soot mainly forms at the start of combustion in the rich zone of the heterogeneous mixture; however, in HCCI technology, as the mixture is homogeneous, there is no scope for soot.

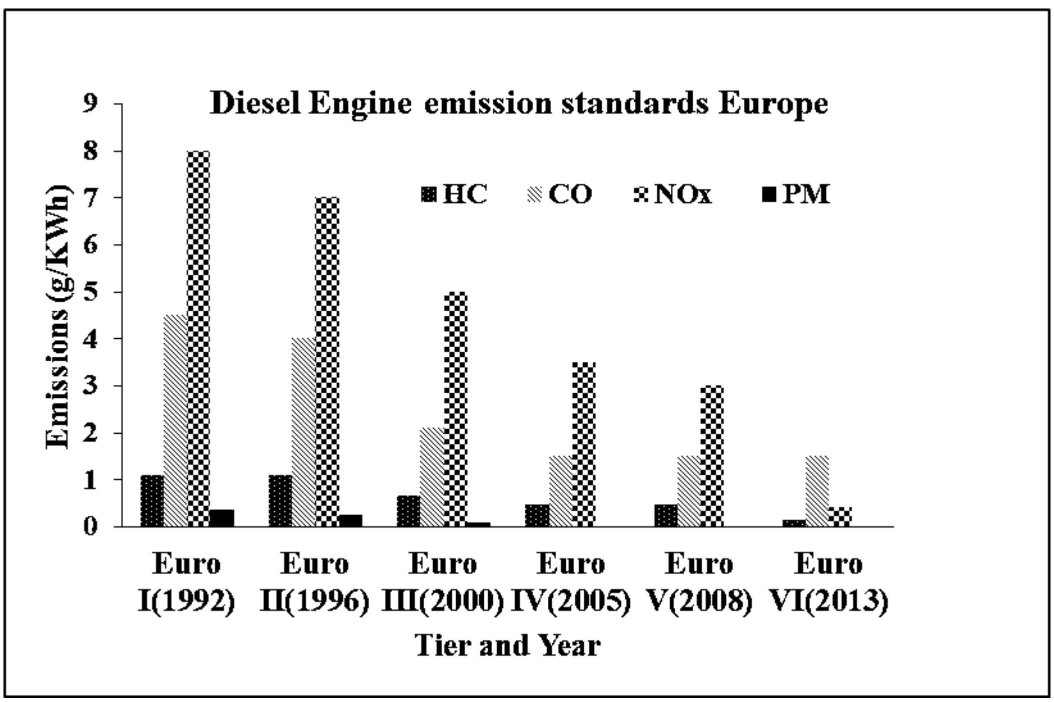


Figure 2.5: Euro emission standards for diesel engines (Hoekman & Robbins, 2012)

Though there are several benefits with HCCI technology, it suffers significant obstacles, including controlled auto-ignition, abrupt pressure rise during combustion, cold start, and higher carbon-based emissions. Hence, an attempt has been made to assess the barriers to HCCI technology and viable solutions.

2.4 Comparison of HCCI Engine with SI and CI engines:

The HCCI combustion is a combination of both spark ignition SI and CI combustion processes (Saxena, Schneider, Aceves, & Dibble, 2012). The A-F charge mix homogeneously before entering the cylinder as it occurs in the SI engine, and the mixture gets auto-ignited as it occurs in the CI engine. HCCI combustion can improve the performance of the engine as compared to other combustion modes, as illustrated in

Table 2.2. The key benefit of the HCCI engine is that it may not require a new engine production line. However, with some retro-fitments or minor modifications in the existing SI or CI engine production lines, the technology can be employed (Epping, Aceves, Bechtold, & Dec, 2002).

Table 2.2: Comparative assessment among SI, CI, and HCCI engines (Bendu & Murugan, 2014; Hasan & Rahman, 2016; Sharma, Rao, & Murthy, 2015)

Particulars	SI (gasoline) engine	CI (Diesel) engine	HCCI engine
Cycle	Otto	Diesel	Otto
Efficiency	Low (30%)	High (40%)	Higher (>40%)
Throttle Loss	High	------------	Negligible
Combustion Length	More time	More time	Less time
Emissions	NO_x from SI engine lies between HCCI and diesel engines		
	Particulate matter (PM) from HCCI engine lies between SI and CI engines		

	Hydrocarbons (HC) from CI engine lies between SI and HCCI engines		
Mixture before ignition	Homogeneous	Heterogeneous	Homogeneous
Fuel Economy	Better in SI engine than CI and HCCI engines		
Ignition starts at	Single point	Multiple points	Single point
Injection method	Port injection	Both port and DI	Direct injection
Equivalence ratio	Almost one	Less than one	----------
Ignition method	Spark	Auto	Auto
Air fuel charge preparation	External	Internal	External
Combustion temperature	Moderate	HCCI engine is lower than CI engine	
Power control through	Airflow	Fuel flow	Fuel flow

It could be observed from

Table 2.2 that CI engines emit high levels of NO_x emission due to high-temperature combustion. At the same time, HCCI engines undergo low-temperature combustion phenomenon so that NO_x formation will be lesser than the CI engine. In HCCI, auto ignition starts at multiple points with unseen flame front propagation (Kong & Reitz, 2002). Combustion completes in a shorter period, and it is completely controlled through chemical kinetics instead spark and injection timing in SI and CI engine, respectively (Najt & Foster, 1983).

Table 2.2: Performance and emissions characteristics of HCCI engines

Description	Performance	Emissions		Ref.
Diesel vapour and 10% EGR	Decrease in BTE, ID, EGT	Decrease in NO_x and Smoke	Increase in HC and CO	(D. Ganesh, Nagarajan, & Mohamed Ibrahim, 2008)
Diesel vapour with 20% and 30% EGR	Decrease in BTE, ID, EGT			(D. Ganesh & Nagarajan, 2010)
Varying loads and EGR (0, 10%) with constant speed	Decrease in BTE			(D. Ganesh et al., 2008)
Varying loads and EGR (0, 10, 20, 30%) with constant speed	Decrease in BTE, IP			(H. Guo, Hosseini, Neill, Chippior, & Dumitrescu, 2011)
Variable speed and loads	Decrease in BTE, Increase in Cylinder pressure and HRR			(Ma, Lü, Ji, & Huang, 2008)

Different speeds and EGR percentages (0, 29, 42, 50%)	Increase in SFC			(L. Shi, Cui, Deng, Peng, & Chen, 2006)
Different speeds and different loads	Decrease in SFC			(M. Y. Kim & Lee, 2007)
Early injection between 70° bTDC and 50° bTDC. With increase in injection pressure from 500 bar to 1500 bar	Decrease in BTE			(Wåhlin, Cronhjort, Olofsson, & Ångström, 2004)

IP: Injection Pressure, ID: Ignition Delay, SFC: Specific Fuel Consumption, HRR: Heat Release Rate, EGT: Exhaust Gas temperature, bTDC: before top dead center

In SI engines, NO_x formation occurs due to pre-ignition of end-charge with an increment in combustion chamber (CC) temperature. Whereas, in CI engines NO_x formation occurs at starting of the combustion phenomenon.

Typically, carbon dioxide (CO_2) and water should be the final products of complete combustion of hydrocarbon fuels in Internal Combustion (IC) engines (R. Prasad & Bella, 2010). However, improper air to fuel (A/F) ratio, early/retard fuel injection timings, and higher in-cylinder temperature etc. causes the production of harmful byproducts (pollutants) of combustion (Venkateswarlu Chintala, Kumar, & Pandey, 2017; Venkateswarlu Chintala & K. A. Subramanian, 2017). The most noteworthy pollutants are CO, HC, NO_x, and PM. Another problem in CI engines is higher combustion temperature due to fuel accumulation and its explosion in specific zones of the CC (Venkateswarlu Chintala et al., 2018). As a result, the fuel-rich zone generates CO, HC, and PM/soot emissions, while the oxygen-rich zone produces NO_x at high temperature (Venkateswarlu Chintala & K. A. Subramanian, 2016; Venkateswarlu Chintala et al., 2018). Usually, CO generates wherever the excess air factor (λ) is less than one due to incomplete oxidation of the fuel (C.-W. Wu, Chen, Pu, & Lin, 2004). However, in CI engines, as the combustion takes place at $\lambda > 1$ (lean mixture), it produces a low amount of CO emission. In some other studies, it was reported that CO might produce due to the large droplet size of fuel and low swirl ratio (Demers & Walters, 1999). Also, as the in-cylinder temperature varies from the central axis to the walls of the CC, the hydrocarbon fuel remains unburned (Demers & Walters, 1999). In SI engines, CO formation occurs mainly due to improper air-fuel mixture preparations. Typically, during cold start and acceleration conditions, a rich mixture causes a high CO formation amount. If the mixture is non-homogeneous, it leads to more CO emissions (Heywood, 1988).

In CI engines, HC emission forms because of A-F mixture's lean nature at low loads. In lean combustion, flame speed becomes slow and causes high HC emission levels (M. Zheng et al., 2008). It is to be noted that HC emissions from CI engines may be low at the tailpipe compare to the emission at the exhaust port. Due to the continuous reaction of HC with oxygen present in exhaust gas wherever the temperature is above 600 ºC (Faiz, Weaver, Walsh, Gautam, & Chan, 1997). In SI engines, HC emission forms mainly because of the lubricating film in CC. Fuel hydrocarbons get absorbed into the cylinder wall's oil film during suction and compression strokes and get desorbed back into the burned gases during the expansion stroke. Desorbed fuel vapours from oil film may get oxidized depending on the temperature of the burned gases (Stone, 1999).

Particulate matter (PM), which is less than one micrometer in diameter, is a composition of soluble and insoluble fractions. The insoluble fraction is typically termed as soot, which has a significant share in the overall PM composition. In CI engines, rich mixture zones with high temperatures cause PM emissions (Epa, 2004). According to Burtscher, several variables are responsible for PM emissions like fuel quality, lubricant use, combustion temperature, load, and combustion process (Burtscher, 2005). Dhananjay et al. reported that if the engine load increases, the soot particles' size increases (Srivastava, Agarwal, & Gupta, 2011). Kittleson concluded that the PM emissions mainly caused because of incomplete combustion of fuel and lube oil (Kittelson, 1998). High temperature and air proportion are the two main factors in NO_x emissions formation in CI engines. It is well known that below 1600 ºC, nitrogen does not react with oxygen; however, it starts to react with oxygen and forms NO_x emission. The flame temperature at the start of combustion is quite high so high amounts of NO_x formation occur at the start of combustion (T. Lee, Park, Kwon, Lee, & Kim, 2013). Whereas in SI engines, it may occur mainly due to the longer residence time of reactants in the CC (Stone, 1999).

Among all aforementioned pollutants, NO_x and PM emissions dominate CI engines, whereas HC and CO emissions are dominant in SI engines. In the case of HCCI mode, HC and CO emissions are dominant compared to NO_x and PM emissions. A comparison of pollutants from HCCI, SI, and CI engines are summarized in Table 2.2.1.

Table 2.2.1 Pollutants from HCCI, SI and CI engines

Pollutant	From HCCI Engines	From SI Engines	From CI Engines
HC	Higher than CI and SI engines	Insignificant	Higher than CI engines
CO	Higher than CI and SI engines	Insignificant	Higher than CI engines
NO_x	Lower than CI and SI engines	Higher than SI engines	Lower than CI engines
Smoke / PM	Lower than CI engines	High amounts	Insignificant

It could be observed from the table that the use of HCCI technology reduces PM and NOx emissions with an increase in HC and CO emissions. In addition to the increase of these emissions, other significant barriers are associated with HCCI operation, which is addressed in detail in forthcoming sections.

2.5 Significant barriers to HCCI Engines:

Despite various benefits in terms of performance and emission characteristics of the HCCI engines, they suffer some technical problems such as control of the combustion process, cold start, and a limited band of operation. Significant technical barriers are illustrated in Figure 2.6.

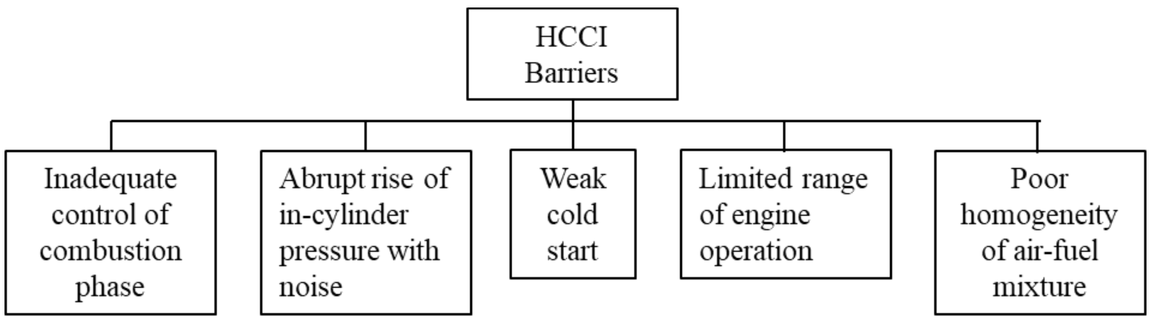


Figure 2.6 Significant barriers to employment of HCCI technologies

2.5.1 Inadequate control of combustion phase

In HCCI engines, auto ignition of homogeneous air-fuel charge plays a vital role. In HCCI engine, ignition of the A-F mixture occurs at several points simultaneously and spontaneously. Accurate control of auto-ignition timing is a tedious task and involves a wide range of engine operating conditions. The utmost severe problem in the HCCI engine is to control the timing of auto-ignition and, subsequently, the heat release rate (HRR) (Angelos et al., 2008; Flowers, Aceves, Westbrook, Smith, & Dibble, 2001). Hence, an adequate understanding and assessment of the ignition timing and combustion processes are must for the smooth operation of HCCI engines. HCCI does not have direct control over the combustion of charge, as it is auto-ignited. Auto-ignition of the charge depends on numerous parameters such as cylinder walls surface temperature, exhaust gas recirculation ratio, engine speed, fuel properties, and a compression ratio of the engine (V. Chintala & K. A. Subramanian, 2017; Dec & Sjöberg, 2004; Yao, Zheng, & Liu, 2009). It is noted that inadequate control of combustion phenomenon affects the engine behaviour substantially (Venkateswarlu Chintala & K. A. Subramanian, 2016). For example, too early combustion leads to a decrease in thermal efficiency along with NOx emission increment. Too late combustion leads to misfiring the engine, and continuous repetition of misfiring leads to engine stall. Too late HCCI combustion increases the fuel consumption with the exponential rise in HC and CO emissions.

2.5.2 Abrupt rise of in-cylinder pressure with noise

In HCCI engines, combustion starts simultaneously at multi-points. It leads to a sudden rise in pressure and temperature in the CC. This abrupt increase of in-cylinder pressure is another major problem, because of which the probability for knocking increases exceedingly. Sheppard et al. also confirmed that the auto-ignition of air-fuel charge at the end of the compression stroke is the main reason for knocking in HCCI engines (Sheppard, Tolegano, & Woolley, 2002). Typically, knock limited rate of pressure rise (RPR) is about 8-10 bar/^{o}CA for CI engines (V. Chintala & Subramanian, 2015b). Saravanan et al. (S. Saravanan, Pitchandi, & Suresh, 2015) carried out the experiments on a single-cylinder HCCI engine fuelled with ethanol and gasohol. They found that gasohol and ethanol showed higher peak pressure than base diesel fuel. The lower calorific value of ethanol (26 MJ/Kg) than gasohol (44 MJ/Kg) leads to lower the peak value of pressure than gasohol. Peak pressure values achieved were 72.6, 68.8, 68.2 bars for 20% premixed gasohol, 20% premixed ethanol and neat diesel. Abrupt pressure rise causes higher heat release rates (HRR) and subsequently leads to mechanical damage of the engine components, including head gasket, piston bowl, and cylinder liner.

2.5.3 Weak cold start

Cold start is another major limitation of the HCCI engine. While starting the engine, heat transfer occurs from the compressed A-F mixture to the cylinder walls. The A-F charge does not receive any heat from the intake manifold. In such a situation, it is difficult to start the HCCI engine. Emissions during the cold start have acquired more attention in the last few years. Bielaczyc et al. (Bielaczyc, Merkisz, &

Pielecha, 2001a, 2001b) examined the emissions from a DI diesel engine during cold and hot start conditions. Their outcomes revealed that CO and HC emissions in the first one minute of the start are greater than 40% of the emissions emit in first three minutes of the cold start, and PM emissions are even more significant than 50%. It has also seen that all the significant emissions CO, HC, NO_x, PM emitted during cold start were almost more than two times that of the hot start. Thus, the cold start problem affects the emissions and creates problems such as long cranking duration, unsteady combustion, and misfiring. To get full advantage of the HCCI engine, cold start operation at the part, and high load conditions should happen without failure (Mack, 2007).

2.5.4 Poor homogeneity of air-fuel mixture

The degree of homogeneity of A-F charge is a crucial parameter for accurate and effective control of the combustion phenomenon inside the CC (V. Chintala & Subramanian, 2014b). Because of the low volatility of fuel (Ra, Reitz, McFarlane, & Daw, 2009), it becomes challenging to vaporize the fuel. It may be noted that thermodynamic cycle time for all engine processes is minimal (in terms of milliseconds), out of which the time available for the A-F mixture preparation is still smaller (V. Chintala & Subramanian, 2015a). Therefore, the charge's homogeneity percentage can only be improved with the provision of high mixing time. It is essential to have a homogeneous mixture for achieving a better fuel economy and low emissions. Christensen et al. (Christensen, Hultqvist, & Johansson, 1999) have investigated the HCCI engine's capability with gasoline and diesel blends on a single-cylinder engine with a constant A/F equivalence ratio of 3 and with different inlet air temperatures and variable compression ratio. It was explored that if the blend contains more diesel in the diesel-gasoline blend, then the mixture remained inhomogeneous before the start of combustion due to low volatility of diesel fuel and increased emissions of HC, CO and soot.

2.5.5 Limited range of engine operation

A significant drawback of HCCI engines is that it operates efficiently only at limited loading conditions. This barrier mainly depends on auto-ignition characteristics of fuel and engine geometry. At lower loads, the self-ignition of the charge is challenging due to the low energy released per cycle, making it challenging to sustain a high temperature in CC. At higher load, HRR becomes very high, with more than half of the charge burnt before the piston reaches TDC. Such conditions could easily prone to knocking and higher NO_x emissions due to a sudden rise in pressure and temperatures in the CC (Yao et al., 2009). Experimental investigations of Olsson et al. also revealed that HCCI engines performed well at moderate load conditions. In contrast, at the part and high load conditions, its thermal efficiency decreased significantly and increased CO and HC emissions (Olsson, Tunestål, Haraldsson, & Johansson, 2001). Thus, the limited operation range might be the foremost hurdle to penetrate the HCCI engine technology in the market. However, it offers low NO_x and soot emissions at moderate loads (Olsson et al., 2001).

2.5.6 Higher HC and CO emissions

In HCCI engines, HC emission levels increase drastically because of lean mixture formation, valve overlapping (Kraft, Maigaard, Mauss, Christensen, & Johansson, 2000). Also, as the HCCI engines operate at low temperatures (i.e., low-temperature combustion process), the after-treatment device (catalytic converter) could also fail to convert HC to water vapor and CO to CO2 (V. Chintala & Subramanian, 2014a). Typically, HC and CO emissions formation increases if the A/F charge preparation fluctuates from the stoichiometric ratio. If the A/F ratio is lean, it leads to inflammability, and if it is rich, it leads to incomplete combustion (Heywood, 1988). The reasons for all the aforementioned barriers and their effect on the engine behavior/components are summarized below in table 2.2.

Table 2.2.2 The problems and their reasons and effects on the engine behavior/components

Significant barriers in HCCI Engine	Cause of barriers	Effect on engine behaviour/ components	References
Inadequate control of combustion phase	• Higher cylinder walls surface temperature • Fuel properties-higher cetane value. • High calorific value of fuel • Higher compression ratio	• Decrease in engine BTE and increase in NO_x emissions due to too early combustion. • Misfire and increase in HC and CO emissions due to too late combustion	(Angelos et al., 2008; Dec & Sjöberg, 2004; Flowers et al., 2001; Yao et al., 2009)
Abrupt rise of in-cylinder pressure with noise	Charge auto ignites at multi-points with high HRR simultaneously at the end of compression stroke	• Knocking • Higher HRR • Damage to engine components like head gasket, piston bowl and cylinder liner	(S. Saravanan et al., 2015; Sheppard et al., 2002)
Weak Cold start	Charge gain no heat from the air induction manifold and the highly compressed charge transferred the heat to the walls of the CC at low temperature.	• CO and HC emissions during cold start are 40% and PM emissions are 50% more than the hot start. • Long cranking duration	(Bielaczyc et al., 2001a, 2001b)

		• Unsteady combustion	
Poor homogeneity of air-fuel mixture	• Low volatile fuel • Less mixture preparation time	• HC, CO and soot formation increase with decrease in fuel volatility	(Christensen et al., 1999; Ra et al., 2009)
Limited range of engine operation	• Auto-ignition characteristics • Engine geometry	At high load • It prone to knocking • Decrease in engine thermal efficiency • Increase in CO and HC emissions	(Olsson et al., 2001)
Higher HC and CO emissions	• Lean air fuel mixture • Low temperature of combustion • Higher HC due to crevice volumes	• Increase in CO and HC emissions	(Christensen et al., 1999; Heywood, 1988; Kraft et al., 2000)

2.6 Proposed solutions to the barriers of HCCI Engines

To market the HCCI engines, its barriers, as mentioned in section 2.4, may be controlled by employing different technologies. All the possible solutions to the barriers in HCCI engines have been discussed in detail in this section. Table 2.2.3 gives highlights of all the barriers and its possible solutions.

Table 2.2.3 Barriers of HCCI engines and their proposed solutions

HCCI Barriers	Proposed Solutions	References

Inadequate control of combustion phase and abrupt rise of in-cylinder pressure with noise	Compression ratio reduction	(Christensen et al., 1999; Kimura et al., 1999; Olsson et al., 2002; Z. Peng, Zhao, Ma, & Ladommatos, 2005)
	Intake temperature	(Akagawa et al., 1999; Choi, Miles, Yun, & Reitz, 2005; Haraldsson, Tunestål, Johansson, & Hyvönen, 2004; Maurya & Agarwal, 2011)
	Exhaust gas recirculation	(R. Chen, Milovanovic, Turner, & Blundell, 2003; Yamaoka et al., 2005)
	Exhaust gas trapping through variable valve timing	(Jennische, 2003; Law, Kemp, Allen, Kirkpatrick, & Copland, 2001)
	Water injection	(Chistensen & Johansson, 1999)
	Variable coolant temperature	(Milovanovic, Blundell, Pearson, Turner, & Chen, 2005)
	Fuel stratification	(Liu et al., 2012; Y. Yang, Dec, Dronniou, & Sjöberg, 2011)
	Fuel additives and reforming	(Hosseini, Neill, & Checkel, 2009; Xu et al., 2004)
Limited range of engine operation	Delayed combustion timing	(Z. Peng et al., 2005)
	Temperature stratification	(Herold et al., 2009; Liu et al., 2012; Sjöberg & Dec, 2005; Yu et al., 2006; Yu, Joelsson, Bai, & Johansson, 2008)

	Boosting intake pressure of air through • Supercharging • Turbocharging	(Christensen, Johansson, Amnéus, & Mauss, 1998; Hyvönen, Haraldsson, & Johansson, 2003; Sun, Thomas, & Gray, 2004; Yoshioka, Matsuoka, Hamada, & Hinatase, 1987) (Olsson et al., 2001; Sun et al., 2004; J. Yang, 2004)
	Dual-mode diesel-HCCI engine	(Burton et al., 2009; Canova et al., 2007)
Poor homogeneity of air-fuel mixture	Fuel injection in a highly turbulent port flow for gaseous and highly volatile fuels	(Aoyama, Hattori, Mizuta, & Sato, 1996; Christensen, Johansson, & Einewall, 1997)
	Early injection with sophisticated fuel injectors for diesel fuels	(Harada et al., 1998)
	Narrow angle fuel spray with dual injection strategy	(M. Y. Kim & Lee, 2007)
Weak cold start	Mixed-mode concept Run the engine on CAI mode at low and moderate loads, and SI mode at idle, cold start at high load.	(Kimura et al., 1999; Sato, Yanagihara, & Mizuta, 1996)
	Use of auxiliary injector for the pilot injection of fuel	(L. Shi et al., 2006)

	Partially closed choke valve and fully open EGR valve during first few firing cycles of an engine	(H. Peng, Cui, Shi, & Deng, 2008)
Higher HC and CO emissions	Modified piston geometry and Swirl Ratio	(Abdul Gafoor & Gupta, 2015; Goryntsev, 2008; S Jaichandar & Annamalai, 2012; Kakaee, Nasiri-Toosi, Partovi, & Paykani, 2016; Yun, Sellnau, Milovanovic, & Zuelch, 2008)
	Steam injection upto 20%, reduced CO emissions	(Gonca, 2014; Hadia, Wadhah, Ammar, & Ahmed, 2017; Kökkülünk, Parlak, Bağci, & Aydin, 2014)
	High injection Pressure	(Wåhlin et al., 2004)
	Low cetane rating fuel	(Starck, Lecointe, Forti, & Jeuland, 2010)

2.6.1 Uncontrolled combustion phase

HCCI engine has no control over its combustion phase, as it has not provided any actuators like a spark plug and fuel injector in SI and CI engines. The combustion phase in HCCI engines can be controlled by controlling the various properties of charge like temperature, pressure, and mixture composition, which enters into the CC. Several approaches are available to control the combustion phase in HCCI engines. A few of them, such as compression ratio (CR) reduction, intake charge temperature, and exhaust gas recirculation, are discussed below.

2.6.1.1 **Compression ratio reduction**

Lowering the CR is the best approach to control the combustion phase in HCCI engines. It is known that with CR reduction, the start of ignition timing delays, which makes it possible to start the combustion after TDC (V. Chintala & Subramanian, 2015b). Due to the late start of combustion, HRR decreases, and it averts the explosive self-ignition of charge, and thus uncontrolled combustion can be avoided. Peng et al. investigated a single-cylinder variable compression ratio engine with n-heptane fuel to determine the influence of CR on HCCI combustion (Z. Peng et al., 2005). It was observed that the knocking tendency of the engine decreased by lowering the compression ratio and indicated mean effective pressure (IMEP) improved from 2.7 bar to 3.5 bar (Z. Peng et al., 2005). Due to low CR, ignition delay (ID) was increased, which could provide sufficient time for fuel injection before combustion (Z. Peng et al., 2005).Similarly, Kimura et al. obtained a significant improvement in engine performance by lowering the CR from 18:1 to 16:1 and better results at higher load conditions (Kimura et al., 1999). In addition to the benefit of performance enhancement, the researchers also significantly revealed the emissions reductions with CR reduction. Experimental results of Laguitton et al. revealed that NO_x and soot emissions were decreased due to the low temperature combustion phenomenon in partial HCCI engine by decreasing the compression ratio from 18.4:1 to 16.1:1 (Laguitton, Crua, Cowell, Heikal, & Gold, 2007). Thus lowering the CR of HCCI engine has proved to improve the engine performance as well as emission characteristics.

2.6.1.2 **Intake charge temperature**

It may be noted that the combustion and emissions from HCCI engines are affected by a variation of the intake charge temperature. For example, pre-heating of the intake air before sending it to the CC decreases the ID, and thus, ignition timing could be controlled (Antunes, Mikalsen, & Roskilly, 2008). Lu et al., in their research, observed the proportional relation between these two parameters and concluded that the higher the intake temperature, the prolonged is the ID that leads to better mixing of air with fuel (Lü, Chen, & Huang, 2005). Two ways have been observed for pre-heating the intake air, using the external power supply and the other through exhaust gas recirculation (EGR). The use of an external power supply (electric heater) to heat the air is the most convenient way to add extra weight and cost (Antunes et al., 2008; Jun, Ishii, & Iida, 2003). EGR may be the other option to cut the necessity of high intake temperature (Nathan, Mallikarjuna, & Ramesh, 2010; Peucheret, Wyszyński, Lehrle, Golunski, & Xu, 2005).

Gowthaman et al. experimented on the HCCI engine to study the effect of the inlet temperature of air ranges from 90°C-150°C to analyse an effect on engine performance and emissions (Gowthaman & Sathiyagnanam, 2017). It was found that CO and HC emissions decreased with an increase in intake air temperature. At low load, HC emissions were not affected at any temperature range of air (Gowthaman & Sathiyagnanam, 2017). Research works carried out on a single-cylinder engine by using n-heptane fuel with 30% EGR revealed that NO_x and soot emissions were increased linearly from 10 ppm to 50 ppm and HC and CO were unaffected when intake temperature was increased from 31°C to 54°C. Similarly, another study showed the increment in NO_x when the intake air temperature was raised from 35°C to 80°C (Akagawa et al., 1999). On the other hand, soot formation was found to be decreased with a decrement in intake temperature in the HCCI engine (Choi et al., 2005).

2.6.1.3 **Exhaust gas recirculation (EGR)**

The Engine-out exhaust gas drastically increases the combustion temperature as the exhaust gas contains more CO_2 than O_2. Therefore, EGR's concept essentially necessitates the mixing of exhaust gases of the earlier cycle with the fresh A-F charge or air in the intake manifold to decrease the NO_x emissions. EGR helps in enhancing the auto-ignition capability of charge by reducing the peak pressure during combustion (N. Saravanan & Nagarajan, 2010). Kanda et al. (Kanda et al.,

2005) explained that EGR is used in the HCCI diesel engine to dilute the charge to retard the ignition timing. A high EGR percentage (up to 68%) can be used for the better start of combustion, but it has also shown some drawbacks in the case of temperature stability (Boyarski & Reitz, 2006; Iwabuchi, Kawai, Shoji, & Takeda, 1999). Internal EGR, which can be controlled through variable valve timing (VVT), helps improve cold start and warm up timing (Schwoerer, Dodi, Fox, Huang, & Yang, 2004). Shi et al. conducted an experiment on single-cylinder naturally aspirated HCCI engine with both internal and external EGR. It was observed that with an increase in negative valve overlapping, NO_x was increased due to the high temperature of the gases, and with external EGR, no effect on NO_x emissions was observed (L. Shi et al., 2006). It is recommended that EGR should club with some technology like alternate fuel or by any chemical approach for better results. Up to 40%, EGR can reduced NO_x emission in case of late injection in HCCI diesel engines (Wimmer, Eichlseder, Klell, & Figer, 2006). Saravanan et al. (N. Saravanan & Nagarajan, 2010) reported that EGR improves the BTE and NO_x formation in the engine during combustion.

2.6.2 Limited range of engine operation

Typically, engine load is denoted in terms of mean effective pressures, i.e., IMEP or BMEP (Brake Mean Effective Pressure). Knock limited BMEP for light-duty SI and CI engines could be about 12 bar and 18 bar, respectively (Kasseris, 2006; Weiss, Heywood, Drake, Schafer, & AuYeung). However, in the case of naturally aspirated HCCI engines, knock limited BMEP is about 5 bar at the speed range of 1000-1500 rpm (Christensen & Johansson, 1998; Dec & Yang, 2010; Wang et al., 2006; D.-b. Yang, Wang, Wang, & Shuai, 2011). Hence, the operation of HCCI engines at higher loads where the BMEP exceeds 5 bar may prone to severe knock and restricts its operation at higher loads. The main reasons for the problem could be the ringing limit.

Ringing is the undesired noise that occurs due to pressure waves in the cylinder during combustion in HCCI engines (Westbrook, Pitz, & Leppard, 1991). These Pressure waves can severely harm the engine operation (Vavra, Bohac, Manofsky, Lavoie, & Assanis, 2012; Vressner, Lundin, Christensen, Tunestål, & Johansson, 2003). It is to be noted that ringing and knocking are two different phenomena. Knocking occurs in SI engines due to the flame propagation, which makes the end-gas charge auto-ignited. However, HCCI combustion occurs because of self-ignition of fuel, which occurs spontaneously throughout the CC rather than flame propagation as it would occur in SI engines (Noguchi, Tanaka, Tanaka, & Takeuchi, 1979). In the HCCI engine, auto-ignition of charge is desired, and auto-ignition occurs uniformly throughout the CC. It is to be understood that both ringing and knocking occur due to pressure waves but with different frequency levels. Pressure waves frequency for ringing and knocking are 5 to 6 kHz and 8 to 25 kHz range, respectively (Mashkournia, Audet, & Koch, 2011; Vavra et al., 2012). Saxena et al. (Saxena & Bedoya, 2013) reported that the ringing pressure waves propagate during the crank angle rotation from -2° bTDC to +10° aTDC. In another study, it was confirmed that there was no problem of such ringing at low load conditions; however, it occurred at high load conditions (Griffiths & Whitaker, 2002; Saxena, Chen, & Dibble, 2011). The problem of ringing could be avoided by using diluted A/F mixtures that are achieved through EGR and lean equivalence ratios (Kook, Bae, Miles, Choi, & Pickett, 2005; Wildman, Scaringe, & Cheng, 2009). Bahri et al. (Bahri, Shahbakhti, & Aziz, 2017) and Maurya et al. (Maurya & Saxena, 2018) found that an artificial neural network is the best tool to predict ringing intensity in HCCI engine. Maurya et al. (Maurya & Saxena, 2018) investigated the effect of inlet temperature, equivalence ratio, engine speed and CR on ringing intensity of hydrogen (H_2) fuelled HCCI engine with the use of chemical kinetics and artificial neural network. Results proved that the ringing intensity increased with increase in equivalence ratio and compression ratio. However, sensitivity of ringing intensity increases with an increase in engine speed.

To maintain the required power output of the engine, the employment of diluted mixture cannot solve the purpose. However, other strategies may also be employed. For example, delayed combustion timing is another strategy for controlling the ringing intensity (Z. Peng et al., 2005). If the combustion initiates at post TDC (top dead center) to counteract HRR and pressure rise, it controls the ringing. Temperature stratification is another approach to control the HRR and so ringing (Herold et al., 2009; Sjöberg & Dec, 2005). Temperature stratification makes the uniform variations in the rates of chemical reaction to permit hotter zones to auto-ignite before colder zones (Krasselt et al., 2009; Sjöberg, Dec, Babajimopoulos, & Assanis, 2004). Liu et al. (Liu et al., 2012) studied the effect of temperature stratification on HCCI combustion through CFD analysis. They concluded that temperature stratification can decrease the pressure rise rate by smooth out the reaction rates widen the operating range. With the use of large eddy simulation, Yu et al. (Yu et al., 2006; Yu et al., 2008) investigated the impact of temperature stratification on auto-ignition in the HCCI engine. Their results demonstrated that temperature stratification tends to increase the combustion duration and decrease the pressure rise rate. The required ignition timing can be obtained by controlling the temperature in homogeneity (Yu et al., 2006; Yu et al., 2008). To widen the range of HCCI operation, another approach had been used by boosting the pressure of intake air. Boosting the intake air pressure can be achieved either by turbocharger (J. Yang, 2004) or supercharger (Yoshioka et al., 1987). In this regard, Hyvönen et al. (Hyvönen et al., 2003) reported that the turbocharger showed 32 to 97% of the increase in BMEP as compared to the supercharger. Similarly, Olsson et al. (Olsson, TuneståI, & Johansson, 2004) found that a supercharger increased the intake air pressure from 1 to 3 bar. However, the penalty on HCCI engine BMEP may be more eminent than the actual pressure rise due to parasitic losses.

2.6.3 Homogeneity of mixture preparation

2.6.3.1 Fuel injection strategies

Homogeneity of the mixture mainly depends on two parameters; (i) fuel injection timing and (ii) ID period. Authors have illustrated this problem by introducing a term called the degree of homogeneity (β). The degree of homogeneity could be defined as the ratio of the extent of homogeneity to the conventional homogeneity. It is evident from Figure 2.7 that the homogeneity of the mixture goes on decreasing if fuel is injected near to the TDC zone. The figure indicates that HCCI with port fuel injection (PFI) technique gives a better homogeneous mixture and late fuel injection shows more heterogeneity than conventional CI injection. However, PFI does not impact the starting of combustion in the engine (Nakagome, Shimazaki, Niimura, & Kobayashi, 1997). If the intake charge temperature is high, more amount of fuel will be evaporated and subsequently enhances the amount of homogeneous mixture (Gray & Ryan, 1997). In the early direct injection (DI) practice, fuel is sprayed much ahead of the predefined timing, i.e., at mid of the compression stroke. Hence, the injected fuel is exposed to high-temperature air, which enhances the fuel evaporate rate as compared to PFI strategy (Harada et al., 1998).

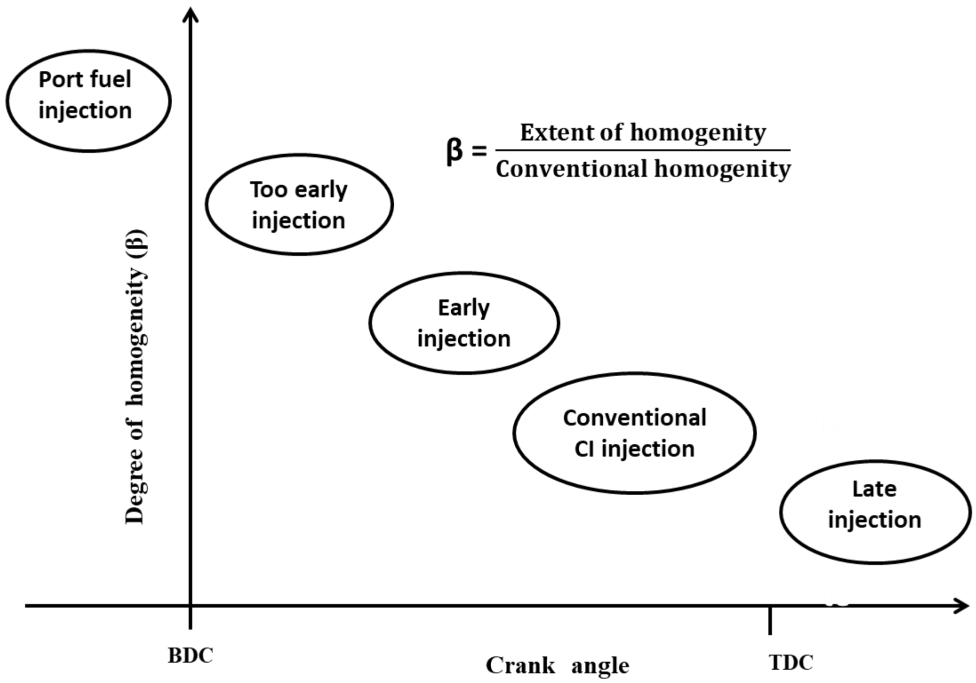


Figure 2.7 Variation of homogeneity with different injection strategies

Sugihara et al. observed the weakening of lubricant film and wall wetting phenomenon with early injection strategy (Sugihara, Nakagawa, Shouyama, & Yamamoto, 1999). It is to be noted that the wall wetting problem is more in the case of an early injection strategy than in the PFI technique. Hence, the more heterogeneous mixture will be formed in early injection strategy, which further leads to high NO_x levels and soot formation compared to PFI strategy (Sugihara et al., 1999). To address the wall wetting problem in early injection timing, Kim et al. (M. Y. Kim & Lee, 2007) were used a narrowed spray angle from 156° to 60° along with compression ratio reduction and dual injection strategy. The first (early) spray injection shown better HCCI combustion due to better mixing of charge, and the second (late) spray injection was sufficient to decrease the NO_x emission. It was also proved that a narrow spray angle concept with a dual injection technique can substantially improve IMEP (M. Y. Kim & Lee, 2007).

In another study, Mathivanan et al. (Mathivanan, Mallikarjuna, & Ramesh, 2016) analysed the impact of multiple pulses (MP) of fuel in three stages in a diesel HCCI engine. It was found that MP injection resulted in improving HRR and combustion phasing. The engine's thermal efficiency increased with MP injection compared to Single Pulse (SP) injection. Emission levels of HC and smoke are found lower with MP than SP injection by 56% and 24%, respectively; however, NO_x emission increased slightly with MP injection. In the case of late injection, the thermal efficiency of the HCCI engine decreased due to late combustion (Iwabuchi et al., 1999). Yoon et al. (Yoon, Kim, & Park, 2018) experimented on a single-cylinder diesel HCCI engine with early and multiple injections to decrease the wall wetting. Further decrease in wall wetting reduction in spray angle from 156° to 60° was also carried out. Increase in ignition delay achieved by decreasing the CR from 17.8:1 to 15:1 by modifying combustion chamber geometry. Results proved that the BTE and combustion performance of an engine was increased.

2.6.3.2 **Stratification of charge**

The stratification ignition's fundamental thought is to upgrade the air-fuel mixing quality and evaporation by a first fuel injection during the compression stroke. The techniques/procedures utilized for mixing A/F has a significant effect on HCCI combustion. Practically, a hundred percent homogeneous air-fuel mixture cannot be attained in actual HCCI operation. Aroonsrisopon et al. (Aroonsrisopon et al., 2004) attained stratification of the charge through PFI and DI strategies. Richter et al. (Richter et al., 2000) experimented with evaluating the effect of inhomogeneity in HCCI operation. It was presumed that charge inhomogeneity was possibly noteworthy and assumed an essential part of the HCCI combustion. Kumano et al. (Kumano & Iida, 2004) reported the impact of inhomogeneity in fuel conveyance in the pre-mixture on HCCI combustion. The injection and blend formation arrangements were intended to control the emissions of HCCI engines (Peters & Weber, 2006).

2.6.4 High HC and CO emissions

In HCCI engines, as combustion occurs at lower temperatures than conventional CI engines, the after-treatment devices (catalytic converters) could also unable to convert HC and CO into water vapour and CO2, respectively. However, some other strategies, including changing the piston bowl geometry, swirl ratio, and higher injection pressure, are discussed below to control these harmful emissions from HCCI engines.

2.6.4.1 Piston bowl geometry and swirl ratio

Combustion chamber geometry and turbulence have a dominant effect on the performance and emission characteristics of HCCI engines (Christensen, Johansson, & Hultqvist, 2002; Vressner, Hultqvist, & Johansson, 2007). Some investigations were carried out with computer modeling tools for designing better configurations of piston bowl to achieve better combustion and emissions performance of HCCI engines (Boyarski & Reitz, 2006; Cao et al., 2009). Goryntsev and Dmitry (Goryntsev, 2008) concluded that piston geometry and swirl have a positive influence on the emissions and fuel economy of the HCCI engine. Many other researchers focused on numerical simulation studies to analyze the combined effect of swirl ratio and piston geometry on engine performance and emission characteristics (Abdul Gafoor & Gupta, 2015; Lim & Min, 2005; B. Prasad et al., 2011). Swirl induction piston geometries and variables like injection pressure and injection timing have a noticeable effect on combustion phenomenon and so emissions (B. Prasad et al., 2011). Similarly, Jaichandar and Annamalai (S Jaichandar & Annamalai, 2012) examined piston bowl geometry's impact on emissions of a diesel engine fuelled with biodiesel. Three different piston shapes, HCC, TCC and SCC (Shallow profundity Combustion Chamber), were examined. TCC has shown better brake thermal efficiency and lower CO, HC and PM emission levels as compared to HCC and SCC bowl geometries (S Jaichandar & Annamalai, 2012).

Kakaee et al. (Kakaee et al., 2016) examined the effect of piston bowl depth and chamfered ring land size with three different piston geometries stock, bathtub and cylindrical in a natural gas/diesel fuel operated engine. Cylindrical piston bowl height up to 1 cm showed a decrease in HC and CO emissions up to 3 g/kWh and 1 g/kWh, respectively. The marginal increase in CO emissions was found with the chamfer sizes of more than 3 mm. Some other researchers also found an enhancement in CO, HC, and soot emissions with shallow dish type piston bowl (Kanda et al., 2005; Nakagome et al., 1997). un et al. reported the influence of the swirl ratio effect (Yun et al., 2008) on emissions of a high-speed diesel engine, and it was observed that the soot emissions were decreased with an increase in swirl ratio from 1 to 4. It also has a minimal effect on NOx and CO emissions (Yun et al., 2008).

Some studies have reported the effect of piston geometry and swirl on combustion characteristics. Saito et al. (Saito et al., 1986) investigated the impacts of piston bowl design on diesel combustion and reported that re-entrant cylinder bowl geometries were met the condition for swirl and turbulent kinetic energy strengthening around TDC. The kinetic energy of turbulent A/F mass can be controlled through swirl in the CC, which leads to improving flame front propagation during combustion (Heywood, 1988). Abdul and Rajesh (Abdul Gafoor & Gupta, 2015) investigated in their numerical study that swirling and turbulent kinetic energy increases as the piston approach TDC during compression and decreases aTDC rapidly during expansion. Overall, the research concludes that decrement in any of the two factors, turbulent kinetic energy and swirl, leads to poor combustion and vice versa.

2.6.4.2 **High injection Pressure**

More the injection pressure better will be the mixing of fuel with air, especially when smaller orifice of the nozzle is used (Okude, Mori, Shiino, & Moriya, 2004). An increase in injection pressure creates a proper spray pattern and leads to better combustion and lower emissions (Suzuki, Kakegawa, Hikino, & Obata, 1997). According to experiments conducted by Wahlin et al. on a single-cylinder DI engine with a high-pressure injection system equipped with Bosch injectors, CO and HC emissions were observed to be low with higher injection pressure by keeping the same NO_x emissions (Wåhlin et al., 2004). Engine speed has no effect on HC emissions with higher injection pressure, but below 1000 bar rising trends in HC and CO emissions were observed with increasing engine speed (Wåhlin et al., 2004). Hassan et al. (Hassan, Abu-jrai, Al-Muhatseb, & Jamil, 2017) performed experiments on HCCI engine with variable speed (rpm) and boosted pressure to study the effects on emission characteristics. It was concluded from the study that at lower rpm with 10% boosted pressure, CO emissions were more whereas HC emissions were gradually decreased with an increase in speed (Hassan et al., 2017).

2.6.4.3 Low cetane rating fuels

Cetane number (CN) of fuel plays a vital role in the combustion performance of HCCI engines, which further influences its emission behaviour. In conventional CI engines, ignition delay and the time lapse between fuel injection and first peak pressure during combustion indirectly rely on fuel quality. So, the higher the CN smoother, the engine operation and low knocking probability. However, low cetane value is preferable to control the combustion and emissions in HCCI engines. Starck et al. (Starck et al., 2010) used a single-cylinder DI engine for their research to study the effect of fuel quality on HCCI combustion. At low speed (1500 rpm), increment in CO emission was very low, but at high speed (2500 rpm), the emission increased up to 55 g/kW-hr as depicted in Figure 2.8. It is evident from the figure that CO emission increased significantly with an increasing cetane number of the fuel.

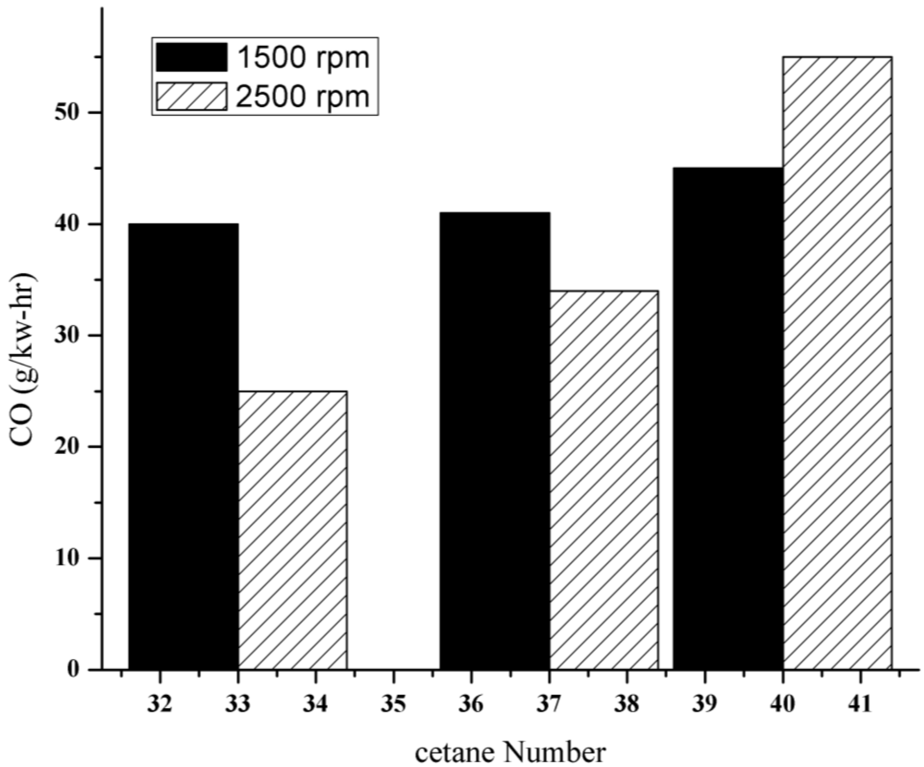


Figure 2.8: Effect of cetane number on CO emissions (Starck et al., 2010)

2.6.4.4 **Steam injection**

Employment of steam in the combustion chamber of HCCI engines is another effective strategy to control the emissions. Hadia et al. (Hadia et al., 2017) investigated the effect of steam injection on HCCI engine behavior. With 10% and 20% steam injection, the CO emission level was decreased by 40% and 70% (Hadia et al., 2017). Similarly, Parlak et al. (Gonca, 2014; Kökkülünk et al., 2014) injected steam in a diesel engine and achieved a 33% decrement in NO_x and marginal decrement in CO emissions. Murthy et al. was used the solar energy to produce steam and injected it into a diesel engine (Murthy, Sastry, & Satyanaryana, 2011). They obtained a significant reduction in CO and NO_x emissions and exhaust temperature. However, at the maximum load, soot emissions, BTE, power, and fuel consumption increased (Murthy et al., 2011).

2.6.4.5 **Dual fuel strategy**

A considerable amount of work was carried out on dual fuel strategy such as biogas-diesel (Feroskhan, Ismail, Reddy, & Sai Teja, 2018; Qian, Sun, Ju, Shan, & Lu, 2017), H2-diesel (V. Chintala & Subramanian, 2014b; Dimitriou, Kumar, Tsujimura, & Suzuki, 2018; M. Kumar, Tsujimura, & Suzuki, 2018), natural gas-diesel (J. Zheng, Wang, Zhao, Wang, & Huang, 2019) in order to obtain a homogenous charge for better emission and performance characteristics than conventional CI engines. Shim et al. (Shim, Park, & Bae, 2018) researched single-cylinder heavy-duty dual fuel (compressed natural gas-diesel) under partially homogeneous charge compression ignition mode. It was found that HC and CO emissions decreased by 31.5% and 35.5%, respectively, with the combination of intake throttle and hot EGR (Shim et al., 2018). It was also observed a 20% reduction in CO_2 emission compared to conventional engines (Shim et al., 2018). Rosha et al. (Rosha, Dhir, & Mohapatra, 2018) examined in their review on the effect of gaseous fuel induction on engine performance characteristics with dual fuel mode. Engine with H2-assisted dual fuel mode produced low carbon emissions as H_2 doesn't contain carbon atoms in it (V. Chintala & K. A. Subramanian, 2016; Rosha et al., 2018). They reported a decrement in CO, CO_2 and HC emissions were observed with marginal increment in NO_x emissions (V. Chintala & K. A. Subramanian, 2016; Rosha et al., 2018). Chen et al. (H. Chen, He, & Zhong, 2018) showed in their review that H2 assisted natural gas engine can increase BTE with a reduction in HC, CO, and PM emissions. Yilmaz and Gumus (Yilmaz & Gumus, 2018) were carried out the experimentation on a multi-cylinder diesel engine and proved that hydrogen augmentation increases the BTE and reduces carbon-based emissions. A recent study by Paykani et al. (Paykani, Saray, Shervani-Tabar, & Mohammadi-Kousha, 2012) achieved decrement in NO_x, CO, and HC emissions with an increase in BTE at medium loads of dual-fuel engine by sending pre-heated intake air with EGR to the cylinder.

2.6.5 Weak cold start

HCCI engines are prone to a weak cold start, as already discussed in earlier section 4.3. Emissions due to a cold start in HCCI engines have also acquired more attention of the researchers (Bielaczyc et al., 2001a, 2001b). However, it can be solved by using several approaches. Researchers (Kimura et al., 1999; Sato et al., 1996) have used the concept of 'mixed mode' to run their engines. In this approach, the engine was run in control auto-ignition (CAI) mode at low and medium engine loads and SI mode at the idle, cold start, and high loads. To solve the cold start problem in the HCCI engine, Shi et al. (L. Shi et al., 2006) used the auxiliary injector that was mounted over the cylinder head and default diesel injector to achieve the pilot injection. A pilot injection of fuel was done to ignite pre-mixed fuel injected by the default injector. Once the engine warmed up, it was switched to HCCI mode. Ignition of the first few firing cycles in the cold start was obtained by Peng et al. (H. Peng et al., 2008) using the choke valve and EGR. The peak in-cylinder pressure was achieved with a partially closed choke valve, and 45% opened EGR valve when the engine was running under normal conditions without EGR and due to this SOC was achieved much earlier and hence overcome the issue of cold start. The glow plug is another solution to the cold starting barrier in the HCCI engine, as it will assist in heating the A/F mixture during cold starting. Increasing CR at the start of operation is another solution (Manofsky, Vavra, Assanis, & Babajimopoulos, 2011).

2.7 Alternative fuels used in diesel engines

The use of alternative fuels such as biodiesel in compression ignition engines have significantly contributed to NO_x reduction. The common composition of biodiesel has 97% tri-glycerides while the rest are mono and di-glycerides. The actual fuel is obtained after trans-esterification of vegetable oil in methanol along with the catalyst potassium hydroxide. Unblended bio-diesel (vegetable oil) can reduce NO_x up to 50% with minor decrease of 10% in fuel economy at full engine load due to lower calorific value of vegetable oils (Murugesan, Umarani, Subramanian, &

Nedunchezhian, 2009). Moreover, considerable fall in CO is also observed especially at full engine load. Increasing injection timing of bio-diesel powered engine by 2 degree CAD bTDC further enhances the engine performance (M. S. Kumar, Ramesh, & Nagalingam, 2001) however, lower volatility, excess deposition of carbon, stickiness, higher viscosity, etc. limits the application of vegetable derived fuels. In addition to properties, domestic utilization of the oil makes it a costly as well as a questionable option despite the fact, oil emissions reduces cancer risks by 90%. In order to counter the expenses waste vegetable oils and plastic oils have been utilized for logistics services of giant food chains such as Mac Donald's (Vivaldini & Pires, 2016). Abundance of used frying oil provides the company a sustainable fuel source especially in Brazil with supply strength of 30,000 tonnes of waste oil per year (Do Biodiesel, 2010). However, diversity in sources of edible and non-edible oils at different demographic distributions leads to variation in viscosity, density as well as calorific values up to an extent that may seriously compromise BTE. For example, Karanja oil has 20% lesser calorific value, 10 times higher viscosity and 4 times higher flash point than diesel making it an inappropriate alternative fuel source for Indian and Malaysian regions.

Fuel emulsification technique as energy source in vehicles and railway engine has enough potential to dominate diesel fuel (Kadota & Yamasaki, 2002). Injection of water-diesel emulsion in the cylinder leads to spontaneous conversion of water droplets into burst of steam. Force generated via steam adds additional pressure for effective BTHE. Certain studies have concluded improvement in combustion efficiency due to emulsified fuel mixture capable of much more refined atomization accompanied by steam clouds (Sawa & Kajitani, 1992). Also the vaporization of fine droplets lessens high temperature combustion mitigating the release of NO_x. Furthermore, the ionization of water, formation of OH radical utilizes free oxygen molecules limiting NO release. The iterative occurrence of both the processes associated with NO_x reduction is termed as heat sink.

Bioethanol, as automobile fuel, has significant benefits that can decrease the dependency on oil and enhance national energy demand. Among all the available renewable sources, ethanol, due to its extensive history, use, and characteristics, such as minimum toxicity to the atmosphere, low boiling point, high research octane number (RON), and calorific value (energy), is considered a primary fuel contender for the future perspective. Although ethanol's energy content is estimated to be 2/3rd of gasoline and butanol, it has a higher RON of 105 compared to butanol (96) and gasoline (93-99) (Lynd, 1996). Research has proved that ethanol can be used up to 85% (v/v) in automobiles without significant modifications (Balat, Balat, & Öz, 2008). The innovative hybrid paths are being established to produce drop-in fuels and fuel additives to mitigate the infrastructure and other needs (Harvey & Meylemans, 2014). Being an oxygenated compound, it produces low emissions and reduces greenhouse gas emissions (GHG) significantly. Ethanol increases the fuel's oxygen content, burning it entirely and increasing its thermal efficiency. It can help directly in the gasoline engine as a knock improver (Thakur, Kaviti, Mehra, & Mer, 2017). In addition, India is an agro-based country. It has suitable resources available for producing a ethanol.

Ethanol blending with diesel is yet another counter for emission reduction especially when particulate emissions are in consideration. Research conducted by Spreen (1999) showed 20-41% decline in PM emissions and 4-5% decline in NOx emission for blending of 10% and 15% ethanol in Diesel (Spreen, 1999). However, the fall in NOx was not consistent and at various points was equivalent to pure diesel emissions. Moreover, up to 10% power decrease was observed when 15% Ethanol was blended in 82.65% Diesel and 2.35% PEC additive (Hansen, Mendoza, Zhang, & Reid, 2000). Increased fuel pump leakage due to lower viscosity of ethanol is one of the major reason of power loss. Furthermore, corrosive nature of ethanol is another challenge for the blend as longer the residence of blended fuel, higher is corrosion to powertrain components due to prolonged absorption of moisture by the alcohol (De la Harpe, 1988). Corrosion due to ethanol molecules is generally dry corrosion with maximum susceptibility to lead, magnesium and aluminum. Hence, the use of anti-corrosive solvent in additives is needed to neutralize the effect.

In the last decades, most of the work has already been done in the technology aspect to increase power output and thermal efficiency by incorporating a common rail system, fuel injection, and EGR. Furthermore, the researcher starts working on fuels to reduce pollution caused by emission. Blending butanol with other fuels is one of the techniques by which we can reduce emissions from CI engines. Bio-butanol is a biofuel produced by the fermentation of biomass. However, biomass can be considered as a renewable and dispatchable form of energy.

The performance and emission characteristics of butanol-diesel blended fuel were studied by a different author to investigate the effect of the various parameters. Li et al. (Li, Chen, & Wu, 2020) proposed a mechanism of diesel-butane blend combustion in the engine. A comparison of ID times of n-butanol is predicted by detailed mechanism and reduced mechanism. The author has also compared laminar flame speed measured by experiment and reduced mechanism for n-butanol. In the 3-D engine simulation, the multi-component evaporation and combustion model was used. Rakopoulos et al. (Rakopoulos, Rakopoulos, Giakoumis, Dimaratos, & Kyritsis, 2010) studied the effect of butanol-diesel fuel blend on the thermal performance of high-speed DI Engine. The analysis shows that soot formation (mg/m3) decreases with an increase in the fraction of butanol in diesel. However, an increase in load soot formation increases, and an increase without blend is higher for the DI engine. Also, NOx emission increases with an increase in load. Although NOx emission is less for the Butanol-Diesel blend, the effect of butanol mass fraction on NOx emission is very less. The impact of the butanol-diesel blend has a negligible impact on brake thermal efficiency. However, emission can be reduced by increasing the butanol fraction in diesel. Wu et al. (F. Wu & Law, 2013) did an experimental and mechanistic study on the laminar flame speed of different butanol isomers. The isomers used were 1-butanol, 2-butanol, iso-butanol, and tert-butanol. The flame speed at 1 atm and equivalence ration of

1.1 is 7% and 22% higher than s-butanol and t-butanol, respectively. However, flame speed decreases with an increase in the pressure from 1-5 bar. The trend of flame speed variation with equivalence ration is the same for the entire configuration. Wu et al. (F. Wu & Law, 2013) studied flame speed versus flame radius for different isomers of the air-butanol mixture. The trend observed by authors in (Gu, Huang, Wu, & Li, 2010) is the same for flame speed variation at different equivalence ratios. The study at equivalence ration of one shows that flame speed decrease with an increase in flame radius up to a certain limit and thereafter increases with increases in flame radius. Miers et al. (Miers et al., 2008) studied butanol-diesel blend on emission and performance of light-duty vehicles. The HC emission increase with increase in butanol fraction. However, NOx emission for Bu-40 is less than the Bu-20 during cold start. Researchers have performed several interesting investigations on engine performance using butanol as an alternate fuel. Most of the current research on performance and emission characteristic are shown in Table 3. However, kinetic modeling of different isomers of butanol is studied by the various author (Grana et al., 2010; Vranckx et al., 2011). San chez et al. (Velázquez-Sánchez & Aguilar-López, 2018) ATCC 824 (Clostridium acetobutylicum) kinetic model to evaluate the fermentation behavior of Acetone-Butanol-Ethanol. Cai et al. (Cai et al., 2014) studied detailed kinetic modeling of i-butanol pyrolysis. The kinetic model was further validated by the species profile and good agreements were obtained between predicted and measured values. Swamy et al. (Rl, 2015) studied combustion characteristic of diesel engine at different butanol percentage, i.e. B0 (Diesel), B5(5% Butanol), B10 (10% Butanol), B15 (15% Butanol) and B20 (20% Butanol). The author studied brake thermal efficicency, Peak pressure and combustion duration at different load varies from 0-5.20 kW (BP). The BTE is higher at higher percentage of butanol and this improves BTE due to lower interfacial surface tension due to butanol. In contrast to BTE the peak in-cylinder pressure is higher for neat diesel and reduces with increase in percentage of butanol. The higher peak in-cylinder pressure of neat diesel is due to cooling effect produced by butanol while the blends of butanol-diesel is being combusted in the combustion chamber. The result also shows that combustion duration of diesel/ butanol blend decreases with increase in the percentage of butanol. The reduction in combustion duration is due to low autoignition temperature and higher latent heat of vaporization. However, a lower cetane number of butanol causes a rapid heat release rate due to long ignition delay.

Although many alternatives have been tried and tested on compression ignition engines, the commercial application require engine modifications as well as demand a new infrastructure for proper production. Moreover, wide variety and variation in properties leads to design changes, which can be cumbersome as well as adds in more investment for the automobile industry.

2.8 Summary of Literature

The following conclusions were drawn based on the detailed assessment of the main advantages, problems, and solutions associated with HCCI engine technology.

- HCCI engines have favorable benefits such as low specific fuel consumption (better fuel economy), lower NO_x, and PM emissions. The slight penalty in CO and HC emissions were reported with the HCCI engines.
- Inadequate homogeneity of air-fuel charge, inadequate control of the combustion phase, abrupt pressure rise in the combustion chamber are the major technical challenges that are to be addressed for the effective implementation of HCCI technology.
- Due to the uncontrolled auto-ignition of the premixed A-F mixture, the HCCI engines would not perform well at higher engine loads.
- To overcome HCCI engine technology limitations, strategies such as early injection, pulse injection, fuel stratification, and compression ratio reduction

were used in the literature. Boosting air intake by supercharging or turbocharging could enable the HCCI engine to be operated at higher loads.

- Water injection strategy, variable coolant temperature, and fuel additives may solve the problems associated with controlling combustion in HCCI engines.
- Piston geometry modification, improved swirl ratio, and higher fuel injection pressure strategies could resolve high HC and CO emissions from HCCI engines. The utilization of low cetane number fuels in HCCI engines could resolve the auto-ignition problems.

Overall, it could be recommended that the implementation of combined strategies of compression ratio with toroidal piston geometry, high fuel injection pressure, and boosted intake air and EGR would provide better emissions behaviour of HCCI engines along with the comparable engine efficiency.

CHAPTER 3 DEVELOPMENT OF EXPERIMENTAL SETUP

3.1 Experimental Setup

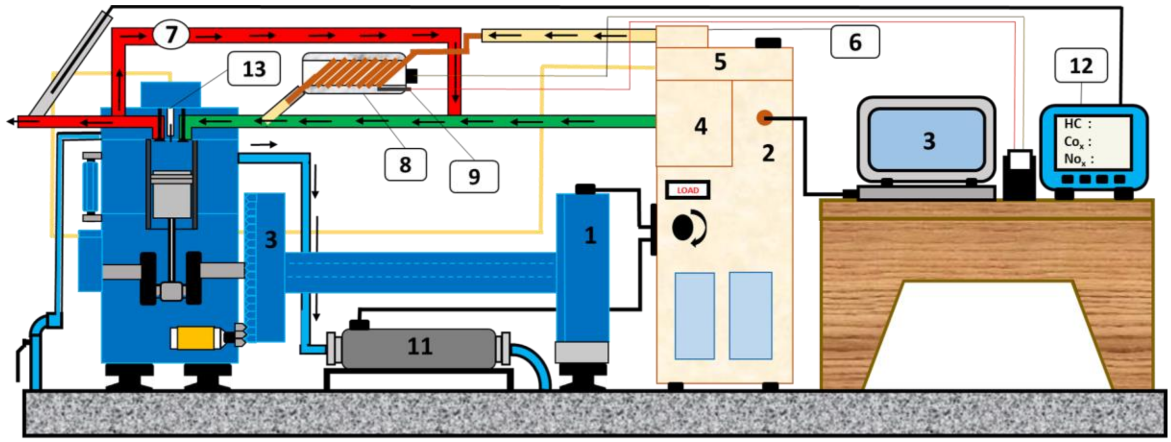
All the experiments were performed on the test rig comprising a single-cylinder, four strokes, water-cooled, variable compression ratio, diesel-powered engine manufactured by Kirlosker and with various instruments and sensors measuring the engine characteristics. The technical specification of the engine is provided in

Table 3.1 and the schematics of engine setup is shown in Figure 3.1

Table 3.1: Technical specification of Kirloskar diesel engine

Parameters	Dimensions
Make	Kirloskar Oil Engines Ltd
Cylinder bore diameter	87.5 (mm)
Piston stroke	110 (mm)
Cubic Capacity	661 cc
Compression ratio (Range)	12:1-18 :1
No. of cylinder/injector	One/One
Engine speed	1500 (RPM)
Type of Injection	Direct injection
Cooling mode	Water-cooled
Power rating	7 HP

The whole test setup consists of a conventional engine, eddy current dynamometer, rotameter, water pump, and data acquisition system (DAQ) and various sensors for measuring the engine characteristics. DAQ consists of load indicator, air surge tank, RPM indicator, crank angle sensor, and load cells. All the parameters are controlled, and the results are extracted to the computer system through DAQ. The test rig helps to study in-cylinder pressure, rate of pressure rises, net heat release rate, and performance characteristics. These parameters enable us to do the comparative analysis of engine characteristics at different loads and operating conditions. The existing conventional engine set up was then modified to operate in PCCI mode. The schematic line diagram of the PCCI engine setup is shown in Figure 3.1. Few modifications were carried out to convert conventional engine into PCCI set up. The existing air intake manifold allowed only air and exhaust gas from the EGR system inside the intake manifold. In PCCI operation, there was a necessity of providing an extra inlet on the intake manifold to allow the entry of fuel vapour coming from the fuel vapouriser. Therefore, at the first, intake manifold was modified to allow the entry of fuel vapour and get it mixed with intake air. This partially Pre-mixed A-F and exhaust gas mixture were then sent to the engine cylinder. The modified intake manifold is shown in Figure 3.2. The fuel vaporizer unit has an inlet and outlet port. The inlet is connected with a separate fuel (diesel) tank, and the outlet was connected to the modified air intake manifold. Further optimization in the PCCI setup is done by changing the default piston used in the conventional engine with the TCC piston.



01 - Dynamometer	05 - Fuel tank	09 - Thermocouple
02 - Data acquisition system	06 - Fuel tank(vaporizer)	10 - PID module
03 - Data output monitor	07 - EGR	11 - Calorimeter
04 - Air intake tank	08 - Fuel Vaporizer	12 - Gas analyzer
		13 - Injector

Figure 3.1: Block diagram of the experimental setup.

As a partial amount of vapour was used in the PCCI operation, the remaining fuel was provided conventionally (DI) through the governor's operation.

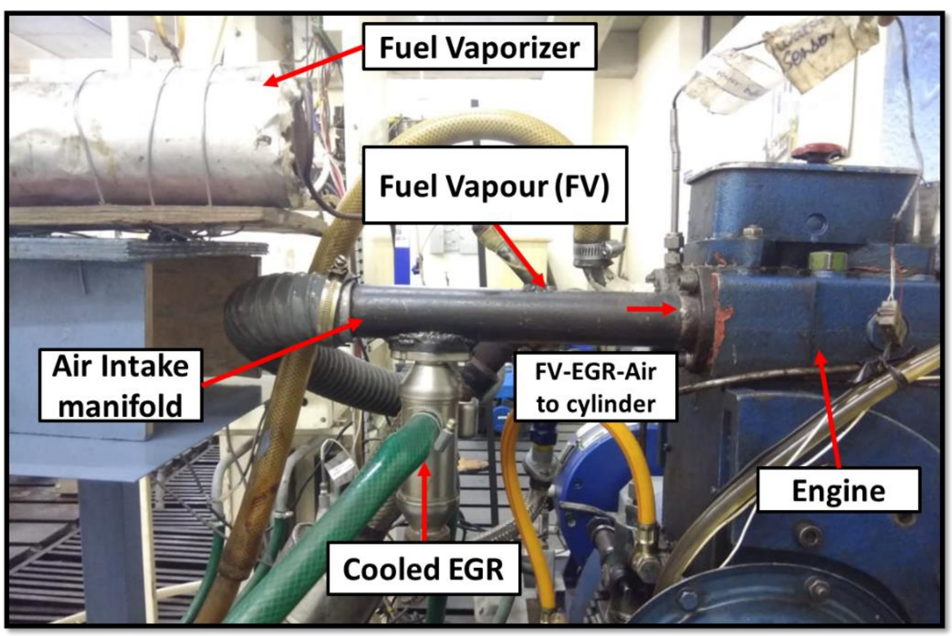


Figure 3.2: Experimental set up PCCI engine

The engine's load was controlled and measured by an AG Series eddy current dynamometer manufactured by Saj Test Plant Pvt. Ltd. The dynamometer load was measured through a strain gauge load cell, and speed was measured from a shaft mounted with a rotary encoder. Senserotronics developed, load cell model 60001 used in the experiment. It is an S-beam universal type load cell with a load range between 0 kg to 50 kg.

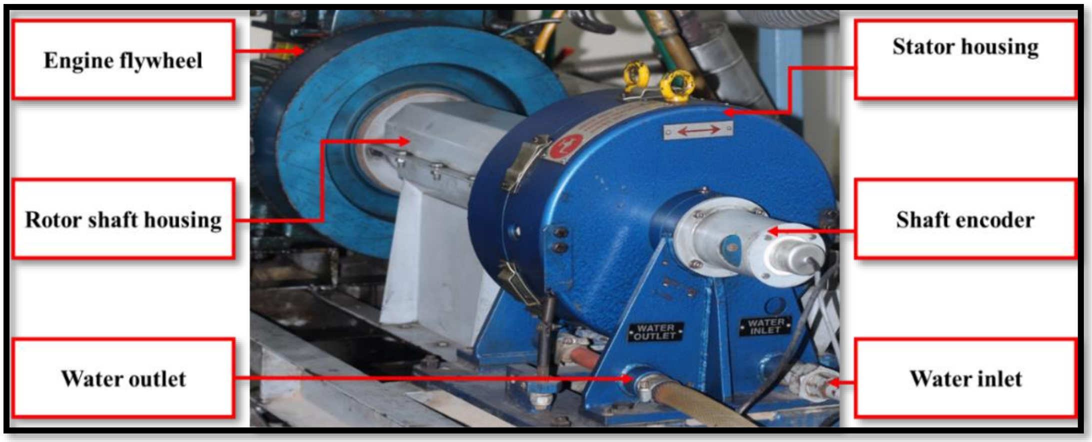


Figure 3.3: Eddy current dynamometer, shaft encoder

The dynamometer consists of a rotor connected to the driveshaft and a stator with two coil windings. When direct current is applied through these coils magnetic field is generated in the area where the shaft is rotating, the shaft rotation produces Eddy current, which creates a braking effect. The load induced by the dynamometer was observed on a load indicator. The load indicator used was the SV-8 series of Universal Process Indicators. A labeled illustration of the dynamometer is given in Figure 3.3.

The engine and dynamometer require liquid cooling. Hence the required rate of water through the calorimeter and engine was necessary to ensure optimum operating temperature. In the experimental setup, the rotameter was used to measure the flow rate of water. It works on the principle of variable area meter, which works on the buoyancy force principle, i.e., upthrust force and the float's weight, i.e., the force of gravity. The Annular space increases due to an increase in the area of the tube.

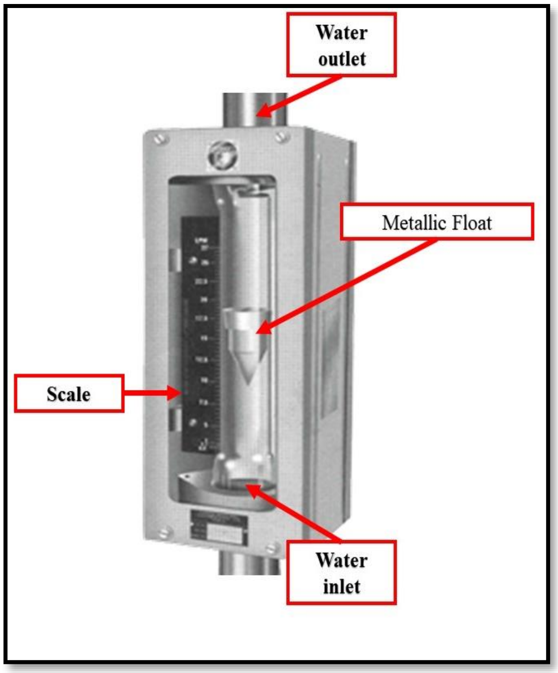


Figure 3.4: Rotameter

A metallic float is present in the setup, which is allowed to move up and down inside a tapered tube through which the water flow is taking place. The water's buoyant force causes the float to rise until the equilibrium is attained between the water and the float's weight. As indicated by scale, the float's vertical position is a measurement of the instantaneous flow rate. A centrifugal pump is used to maintain the flow rate. A labeled image in Figure 3.4 illustrates the functioning of the rotameter. The engine's rotameter water flow was set at 100 litre per hour (lph), and calorimeter water flows at 250 lph as per the standard instructions given in the engine manual. A decrease in water level can increase the risk of engine operation at higher loads with an increase in temperature and even lead to engine seizure.

3.2 Results Extraction through data acquisition system

The DAQ system is the control panel of the engine parameters. It is fitted with a fuel pipe, manometer, orifice, temperature sensors, load indicator, rotameter, dynamometer load unit, ignition switch, and fuel tank. The DAQ system helps to extract data to the computer system and analyse various parameters with graphical representation. The dimmer stat used to vary the engine load through the dynamometer.

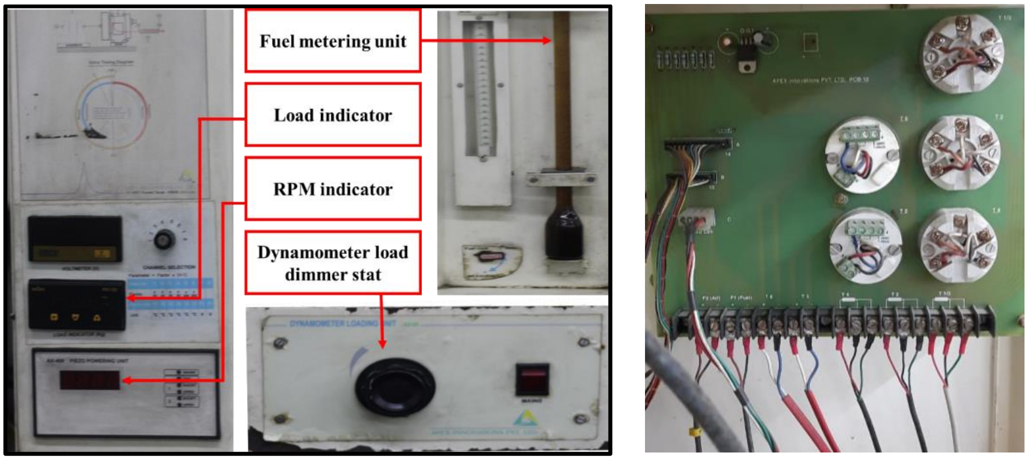


Figure 3.5: Data Acquisition System, transducer and display interface

The fuel tank is placed at the top of the DAQ system, and a manometer was used to measure the amount of fuel consumption. The DAQ system also consists of various sensors for analyzing the engine parameters such as the piezoelectric sensors, engine/calorimeter inlet temperature sensor, engine /calorimeter outlet temperature sensor, engine exhaust temperature sensor, and load cell sensor and differential air pressure transducers. A piezoelectric sensor is a type of transducer designed to measure variation in pressure during compression and combustion. The sensor helps in determining the knock in the engine.

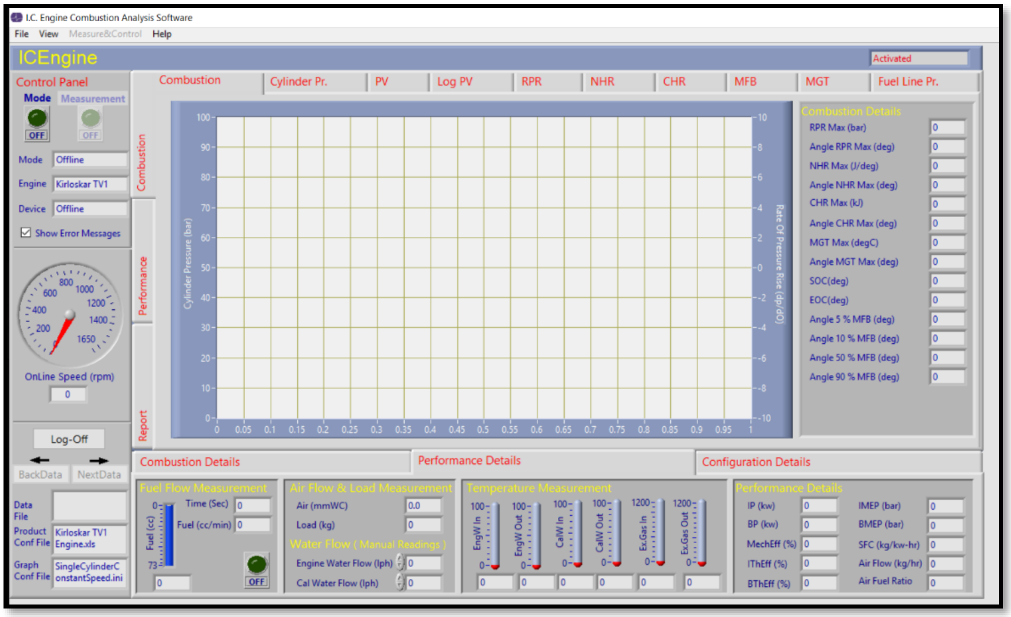


Figure 3.6: Graphic user interface of IC Engine Combustion Analysis Software

The engine/calorimeter inlet/outlet temperature sensor informs about the engine temperature during its operation. The increase in temperature increases with the load, so an increase in load should be gradual. Before increasing the load further, the temperature needs to be stabilized for the engine's efficient operation. Differential air pressure transmitter determines the airflow rates through the intake manifold. All combined data are used for the engine's proper functioning by providing input to the DAQ, against which we get the combustion and performance parameters of the engine. A labeled image of the DAQ is given in Figure 3.5. IC engine analysis software was used to analyse the combustion and performance parameters retrieved through the information from DAQ. Parameters were varied, and data was extracted as per the experimental requirements. The graphic user interface of the software is shown in Figure 3.6.

3.3 Emission Measurement

3.3.1 Gas Analyser

It is a device used to analyses the quality of exhaust gas. It can efficiently determine the quantity of CO, NO_x, CO_2, and HC in the exhaust gases depending upon the working conditions. The purpose of adopting PCCI was to get lower NO_x and smoke in the exhaust gas. Thus, proper assessment of exhaust gases is necessary. The AVL engineering made gas analyser (AVL DIGAS 444) was used in the experiment to measure the emissions. The analyser needs to follow some initial standard operating process before it starts measuring the emissions.

Figure 3.7: AVL DIGAS 444 gas analyzer

The gas analyzer has two main components, the probe through which the sample exhaust gas passes and the analyzer itself, which consists of various sensors to get the required results. The sample cell unit in the analyzer receives the exhaust gas through the cell inlet. The cell unit is placed in between the infrared source and dedicated infrared detector for each gas except NO_x and O_2. Reduction in IR energy due to obstruction offered by a certain gas waveband is measured for emission. NO_x and O_2 measurements depend on the electrochemical sensor output concerning chemical reaction between test gases and analyser chemicals. The probe is connected to the analyzer through a hollow rubber pipe. Before measuring the emissions, a leak test is performed to check if there is some leak in the pipeconnecting probe. After the leak test, the analyser takes some time to calibrate. After calibration, the measurement can start by inserting it in the exhaust outlet. Exhaust gas was analysed for the various engine load loads.

3.3.2 Smoke meter

Smoke opacity is an essential factor concerning engine operation and carcinogenic particulate matter (PM) emission. Higher smoke opacity always lowers engine performance and increase PM emission. AVL engineering made smoke meter, model 437c was used in the experiment to measure the smoke opacity. The exhaust gas opacity interferes between the light emission from an incandescent bulb and photosensitive cell installed in the meter. The decrease in the intensity of light energy received by the photosensitive cell provides a measure for smoke opacity.

Figure 3.8: AVL-437c smoke meter

3.4 Compression ratio adjustment

The variable compression ratio (VCR) engine test rig allows changing the CR within a range from 12:1 to 18:1. The CR was varied by changing the clearance volume and keeping the engine's constant swept volume. The test setup is equipped with a suitable mechanism to increase/decrease the clearance volume. Thus, in turn, it will change the CR within the range of 12:1 to 18:1.

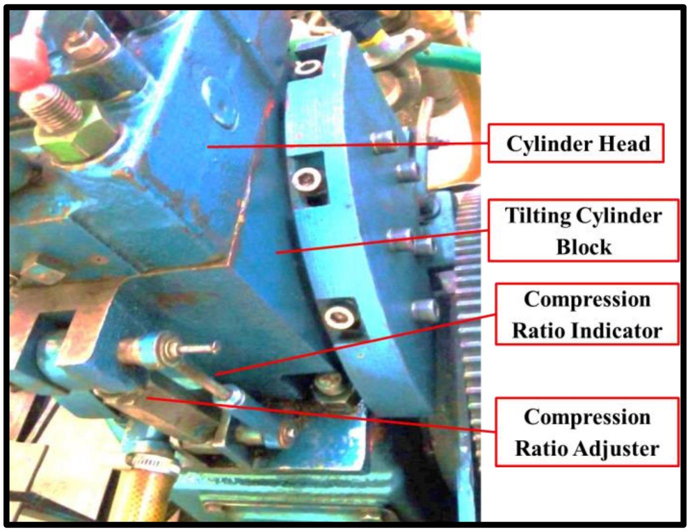


Figure 3.9: Compression ratio adjuster arrangement with the tilting cylinder block of engine

The tilting cylinder block method was used to vary the CR without changing the piston bowl geometry. The arrangement of the variable compression ratio set up is depicted in Figure 3.9. One of the essential goals of this research was to reduce the engine's HC and CO emissions with low NOx emissions. Therefore, experiments were performed at three different compression ratios ranging from 16:1 to 18:1. With the earlier experimental results, it was observed that the engine-out HC emissions were increased drastically at below 16:1 CR due to poor combustion characteristics. Hence the present study was restricted at CR > 16:1.

3.5 Development of Test Rig to Achieve PCCI Combustion

3.5.1 Fuel Vaporizer Unit (Retro fitment device)

3.5.1.1 **Design and development of Fuel Vaporizer**

The fuel vaporizer is a device, which vaporizes fuel (diesel) to form a homogeneous mixture with air to attain better combustion characteristics and thermal efficiency for the HCCI engine. The vaporizer consists of a heater to vaporize the liquid fuel (Prabhudev, Umesh, Kamesh, & Madhu, 2006). Ganesh et al. had successfully used a fuel vaporizer for external A-F mixture preparation in the HCCI engine. The study states that with a little bit modification in the CI engine's inlet manifold, it is possible to mount the fuel vaporizer on intake manifold (D. Ganesh & Nagarajan, 2010; D. Ganesh et al., 2008). Agarwal et al. had also used an electrically heated fuel vaporizer in which the fuel injector sprays the atomized fuel in it, and then diesel vapor from the outlet is mixed with air to form an external homogeneous A-F mixture (A. K. Agarwal et al., 2013; A. P. Singh & Agarwal, 2012). The fuels, which are low volatility and high boiling point, are more preferable in the vaporization technique of external mixture formation. For the effective external mixture formation, the fuel vaporizer's operating temperature kept above the boiling point of fuel (S. Ganesan, 2012). Various approaches have been used to vaporize the fuel like electrical heating (D. Ganesh & Nagarajan, 2010; D Ganesh, Nagarajan, & Ibrahim, 2008; A. P. Singh & Agarwal, 2012) . The high intake air temperature with EGR can also be used to vaporize the fuel in the intake manifold (D. S. Kim & Lee, 2006; Liu et al., 2011; Liu et al., 2012). Many researchers (A. K. Agarwal et al., 2013; D. Ganesh & Nagarajan, 2010; D Ganesh et al., 2008; Maurya & Agarwal, 2014) had design the electrically heated vaporiser in which heating of the main vaporizing chamber was done externally with the help of band heaters. In this vaporiser the high thermal conductivity material containing the liquid fuel is inserted into the ceramic pipe and the heating element, i.e. nichrome was wounded over the ceramic pipe which conducts the heat alone. This vaporizer also uses a port fuel injector mounted at the top of the vaporizer to supply the suitable quantity of atomized fuel to the vaporizer. Here the port fuel injector was controlled by a costly electronic control unit (ECU) to control the quantity and mass flow rate of fuel to vaporizer. The drawback associated with this approach was, the vapor residence time was not as per the expected value and the time to vaporize the liquid fuel is not sufficient as the path for the liquid fuel flow is straight giving very less time to liquid fuel to vaporized, as a result higher quantity of liquid diesel was obtained at the outlet of the vaporizer.

Two approaches were adapted to develop the fuel vapouriser, the retro fitment for the PCCI engine set up. In the first approach, the heating element was used as a band heater. The band heaters were mounted over the vapouriser pipe, as shown in Figure 3.10. The output (fuel vapour) achieved in this approach was not beneficial to use in the PCCI engine set up. The output obtained contained a nearly equal amount of fuel vapour, and liquid quantity as low surface area was available in the vapourising pipe. Overcoming this drawback was the prime target to succeed in this project. To overcome this limitation, the second approach was adopted. In this approach, vapourising pipe use was smaller in diameter, and its length was increased.

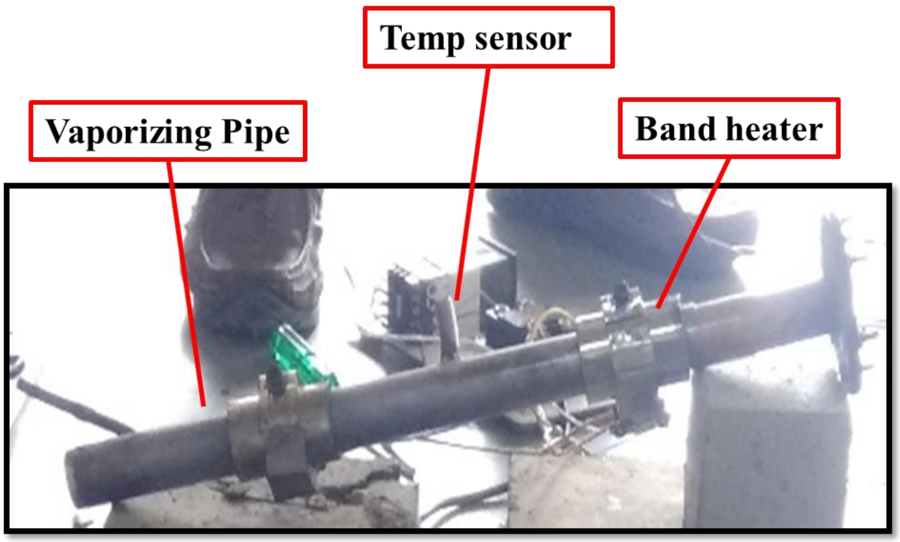


Figure 3.10: First Approach: Heating Element over the Pipe

Also, the significant difference in the first and second approach was, in the second approach, vapourising tube was wounded over the heating element, as depicted in the Figure 3.11. The advantage of this new design was providing complete conversion of liquid fuel into vapour at the outlet section.

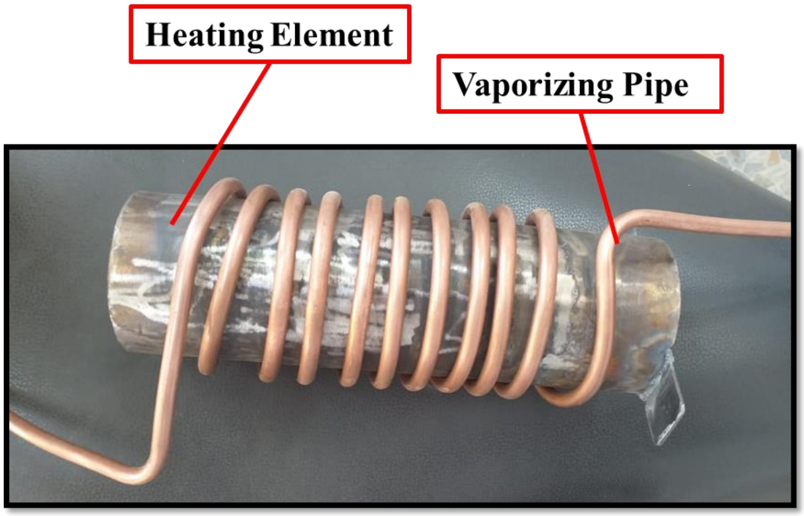


Figure 3.11: Second Approach: Vaporizing Pipe over the heating element

In this research study, the second approach for fuel vaporizer was used. The high thermal conductivity, copper was selected for the pipe material to carry the fuel.

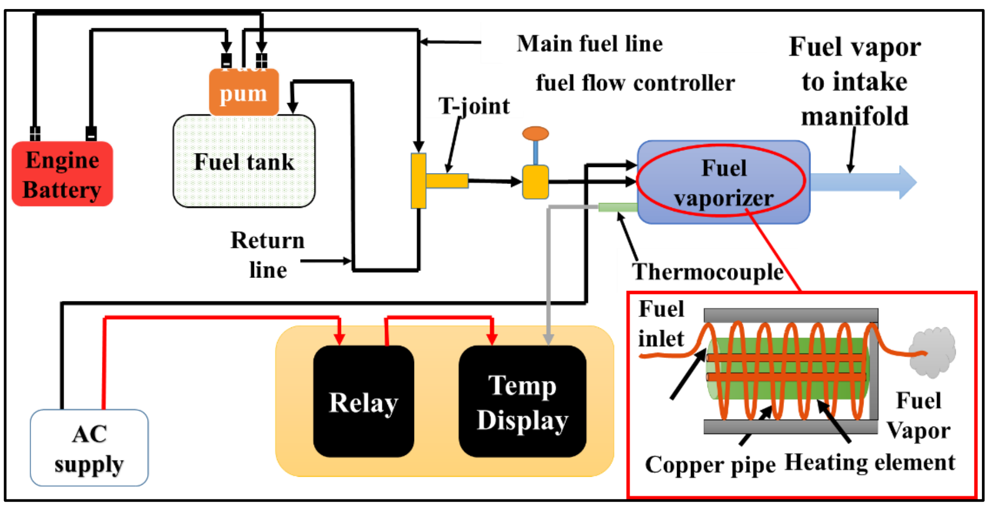


Figure 3.12 Schematic line diagram of the experimental setup of fuel vaporizer

As shown in Figure 3.12, the copper pipe wounded over the heating element, providing a circular path to liquid fuel instead of several researchers' straight paths (D. Ganesh & Nagarajan, 2010; Maurya & Agarwal, 2014). In this design, the vapour residence time was improved. A very negligible amount of liquid fuel was obtained at the fuel vaporizer outlet, further vapourised entirely once it was entered into the combustion chamber. In this experiment, a mechanically operated nozzle was used to control the fuel's mass flow rate as shown in Figure 3.13.

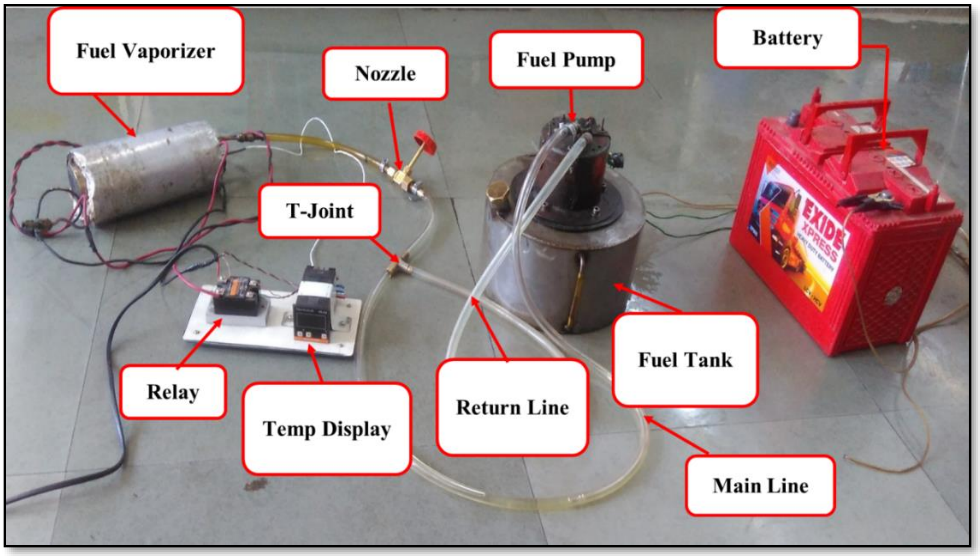


Figure 3.13: Fuel vaporizer arrangements

The external power supplied fuel vaporizer was controlled by a temperature controller to maintain a constant temperature and supply diesel vapour at all engine loads. The diesel fuel temperature must reach above 75-80 ºC (flash point of diesel) to convert the diesel fuel into vapor form. Therefore, the main vaporizing chamber's material was copper due to its better thermal conductivity (384 W/m-k). To avoid auto ignition of diesel in vaporizing chamber, the copper tube temperature was kept 205 ºC, which was between the boiling point temperature (170 °C) and the auto-ignition temperature of diesel (210 °C).

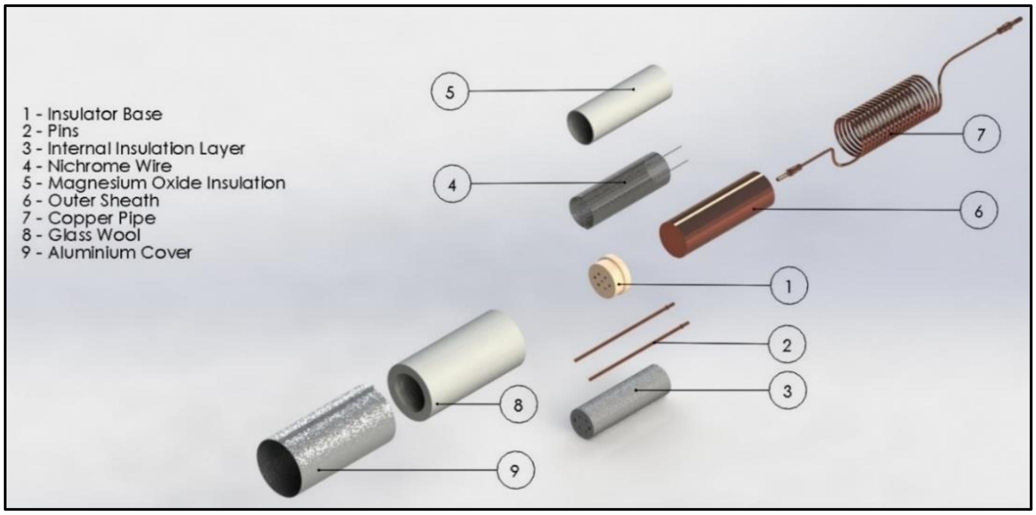


Figure 3.14: Diesel fuel vaporizer (exploded view)

To maintain the temperature of the copper tube, the temperature controller was used along with the relay. The ni-Chrome heating element was used to heat the copper tube. Glass wool was used as a thermal insulator over the copper tube to reduce the heat loss to the environment. The fuel vaporizer contains a ceramic heating element at the center at which the Ni-chrome did primary heating. The copper pipe was wounded over the ceramic pipe. The detailed exploded view of the fuel vaporiser is depicted in Figure 3.14, along with its sectional view in Figure 3.15. The glass wool was used to insulate the fuel vaporizer. The diameter of the copper pipe used was 6 mm, and length was 2400 mm. The detailed design approach of the length calculation of copper pipe has given ahead in this discussion. For the vapouriser's compelling performance, the parameters required to be controlled are cut-off temperature and warm-up time. These two parameters are precisely controlled by using Thermotech PID-41S, the temperature controller.

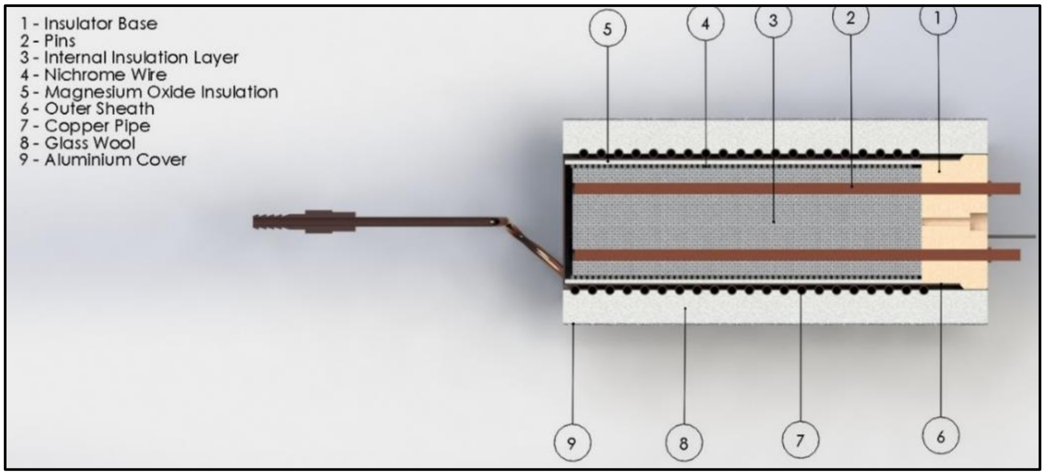


Figure 3.15: Diesel fuel vaporizer (cut-section view)

The warm up-time of the PID controller was 12 minutes to reach 205 °C temperature. The addition of a T-joint effectively solved fuel backflow. A T-joint was added on the main fuel line such that one joint was directly attached to the nozzle and the other joint was attached to the engine return line so that the extra fuel was sent back to the fuel tank. The fuel vapour injection pattern is, as shown in Figure 3.16.

Figure 3.16: Injection of vapor pattern

3.5.1.2 Analytical design approach to calculate the required length of fuel vaporizer tube

The mass flow rate of diesel fuel was the necessary parameter to design the fuel vaporizer. It was necessary to know the fuel vapour's mass flow rate to calculate the fuel vapouriser tube's length. The four-stroke engine was running at a constant

1500 rpm. Therefore, it performed 12.5 cycles/ sec. The maximum mass flow rate of diesel fuel through the direct injection was 16 ml/minute at full engine load, which implies that the fuel flow rate for effective diesel vapor generation is 0.2667 ml/sec. The fuel vapouriser's schematic to calculate the length is given in Figure 3.17: Analytical design of fuel vaporizer Figure 3.17.

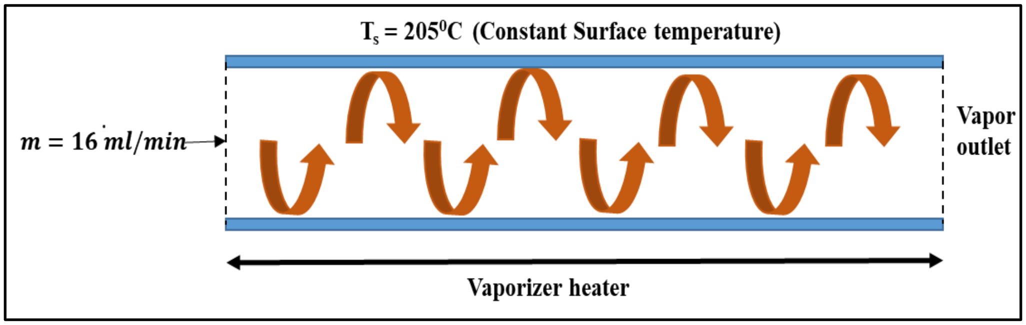


Figure 3.17: Analytical design of fuel vaporizer

The properties of diesel fuel that were considered to design the fuel vaporizer are mentioned in table 3.2.

Table 3.2: Properties of Diesel at 30° C as per ASTM standard

Density (kg/m^3)	835
Kinematic Viscosity @ 30 °C (m^2/S)	2.5×10^{-6}
Calorific value (MJ/kg)	42
Flash point and Boiling point of Diesel (°C)	75 and 180
Thermal Conductivity of diesel (W/m-k)	0.13
Specific heat of Diesel (J/kg-K)	1750
Latent heat of Vapourisation (kJ/kg)	232.6
Thermal conductivity of copper (W/m-k)	401
Wire diameter (mm)	6
Fuel mass flow rate (kg/sec)	222.695×10^{-6}

The thermo-physical properties of diesel fuel used to calculate the vaporizer heater's length is mentioned in table 3.2.

The fuel vaporizer's thermal modeling consists of a 500 Watt heater to calculate the copper pipe length of diameter 6 mm.

Table 3.3: Detailed specifications of fuel vaporizer

Characteristics	Specifications
Heater power	500 Watt
Heating element diameter	60 mm
Heating element length	160 mm
Copper pipe diameter	6 mm
Copper pipe length	2400 mm

The vapouriser heater's required length can be calculated from the basic equation of heat transfer through convection.

$$Q = h.\pi D\,\mathrm{L}.(\mathrm{T}_w - \mathrm{T}_b) \tag{3.1}$$

The copper pipe's required length can be calculated by calculating the unknown parameters mentioned in equation 3.1.

Here, T_w = 230 °C

T_{in} = 25 °C and T_{out} = 205 °C

$$T_b = \frac{T_{out} - T_{in}}{2} \tag{3.2}$$

To calculate the heat transfer coefficient, it was necessary to know the type of flow inside the pipe, which can be evaluated by Reynolds number, Prandtl number, and Nusselt number. To calculate the convective heat transfer coefficient, Nusselt number is required, which is a function of Reynold number and Prandtl number.

$Nu = f(Re, Pr)$ Where,

$$R_e = \frac{\rho.v.D}{\mu} = \frac{\mathrm{v}.D}{\nu} \qquad (3.3)$$

The velocity of the fuel flow can be calculated from the mass flow rate

$$\dot{m} = \rho.A.\mathrm{V}$$

$$V = \frac{\dot{m}\, x\, 4}{\rho.\pi.d^2}$$

$$= \frac{(222.69 x 10^{-6} x 4)}{(835 x \pi x 6^2 x 10^{-6})}$$

$$V = 9.43 x 10^{-3} m/s$$

$$R_e = \frac{(9.43 x 10^{-3} x 6 x 10^{-3})}{2.5 x 10^{-6}}$$

$$R_e = 22.36$$

$$p_r = \frac{\mu.C_P}{k} \qquad (3.4)$$

$$= \frac{\nu.C_p.\rho}{K}$$

$$= \frac{2.5 x 10^{-6} x 1750 x 835}{0.13}$$

$$= 28.1$$

To calculate the heat transfer coefficient, the following equation can be used.

$$Nu = \frac{h.\mathrm{D}}{k} \qquad (3.5)$$

As per the theory of heat transfer, for the laminar flow and for the constant wall temperature, the Nusselt number should be equal to 3.65

$$h = \frac{N_u.K}{D}$$

$$h = \frac{3.65 x 0.13}{0.006}$$

$$= 79.083 w / m^2 K$$

Case 1: Considering phase transformation only

From the eq. (3.1), the length of the required copper pipe can be calculated as below.

$$L = \frac{Q}{(h.\pi.D.(\mathrm{T}_w - \mathrm{T}_b))}$$

$$L = \frac{500}{(79.083 x \pi x 0.006(230-90))}$$

$$L = 2395.83 \approx 2400 mm$$

Case 2: Considering Latent heat and Sensible heat

By considering, the latent and sensible heat require to vapourise the diesel. Length of the pipe was calculated by log mean temperature difference as mentioned below.

As shown in figure the mean temperature differences (LMTD) were calculated by using the standard equation as below.

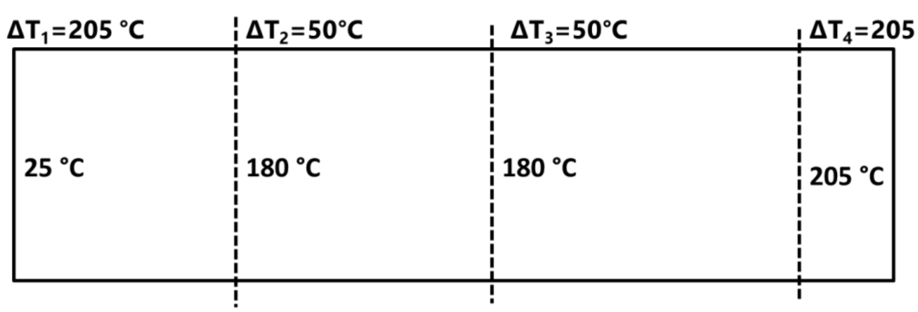


Figure 3.18: Temperature difference at various section of the vapourising chamber

The LMTD were calculated at the first section

$$\Delta T_{m1} = \frac{205 - 50}{\ln \frac{205}{50}}$$

$$\Delta T_{m1} = 109.85$$

Similarly,

$$\Delta T_{m2} = 50°C$$

$$\Delta T_{m3} = 36.06°C$$

$$Q = hA\Delta T_{m1} + hA\Delta T_{m2} + hA\Delta T_{m3}$$

$$Q = hA[109.85 + 50 + 36.06]$$

$$500 = 79.083 \times \pi \times 0.006 \times L \times 195.91$$

$$L = 1.81m$$

Therefore, by calculating the length of vapouriser by both ways as mentioned in case-I and Case-II, the maximum length of vapouriser was selected for the experimentation.

3.5.1.3 Vapor residence time

The vapor residence time of the fuel vaporizer was calculated by using a conical flask setup. A 100 ml capacity conical flask was selected. A wooden cork lid of the same size diameter was machined to close the flask's opening and then covered by the aluminum foil to seal any leakages. The vaporizer was then operated at the most suitable temperature ranges, and the mass flow rate of fuel was adjusted to 0.2667 ml/sec by using the nozzle. Then the vapor coming out from the vaporizer was collected in a beaker for 15 seconds, and then by using a digital stopwatch, the vapor residence time was observed. The Figure 3.19 shows the state of diesel vapor in the beaker at 10 sec of time interval.

Figure 3.19: Diesel vapor at various time intervals of 10 seconds

So the vapor residence time observed during the experiment at the most optimum temperature and mass flow rate of fuel was 120 sec. The obtained residence time of diesel vapor, i.e., 120 sec, is sufficient for efficient engine operation. After entering into the engine inlet manifold, the diesel vapor gets mixed with the cooled exhaust gases from EGR and air. 120 sec vapor residence time of diesel fuel shows that only the vapourised diesel will enter into the PCCI engine combustion chamber.

3.5.2 Modification in combustion chamber of conventional engine

The in-cylinder air motion, fuel injection timing, injection pressure, and bowl dimensions are essential parameters that govern the performance, combustion and emission characteristics of engines (V. Ganesan, 2012). The A-F mixture's quality is the most important controlling parameter to improve performance and low emissions from the CI engine. Quality of air-fuel mixing can be achieved either by increasing fuel injection pressure or compression ratio. Jaichandar and Annamalai investigated the combined effect of high injection pressure and toroidal combustion chamber (TCC) on CI engine. They found that engine performance and emissions were improved by the combined effect of TCC and higher fuel injection pressure. However, an increase in fuel injection pressure was the main cause of increment in NOx emissions due to higher HRR and increased combustion pressure (S. Jaichandar & Annamalai, 2013).

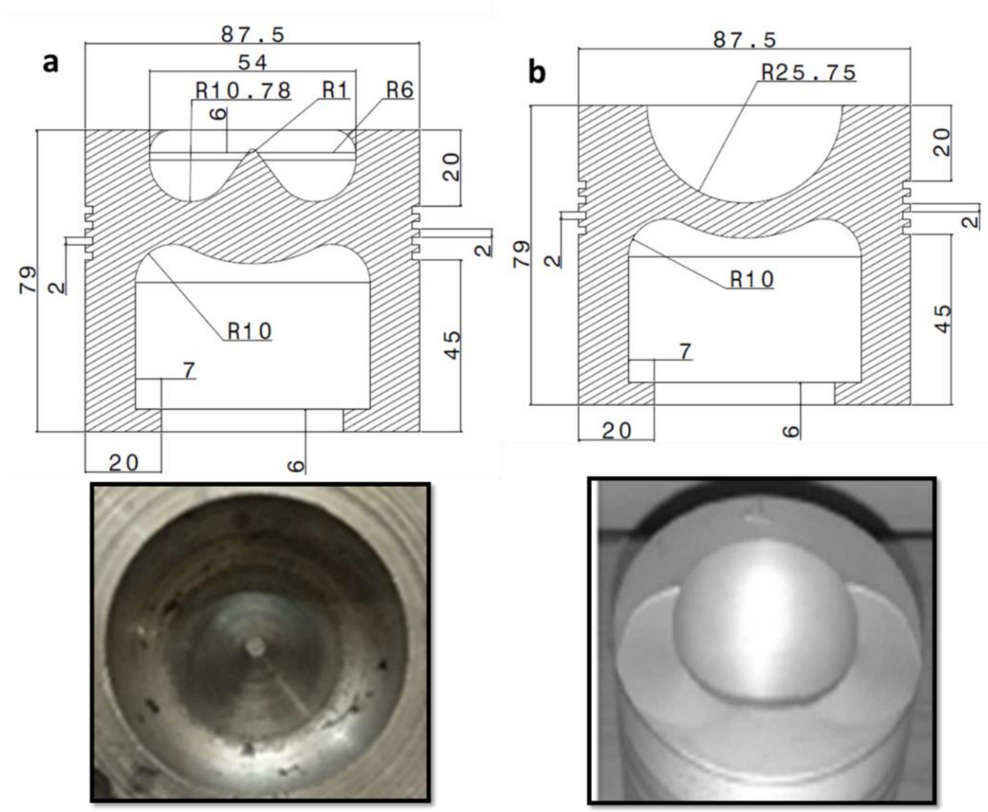


Figure 3.20: a) Toroidal (modified) Combustion chamber b) Hemispherical (default) combustion chamber (All dimensions are in mm)

Therefore, one of the key engine design parameters, combustion chamber geometry, is modified in the current research. Sagaya Raj et al. (Raj, Mallikarjuna, & Ganesan, 2013) studied the air motions inside the CI engine's engine cylinder for four different geometries. It was concluded that the combustion chamber is playing an essential role in air-fuel mixing. Performance, combustion, and emission characteristics of a diesel engine depend on operating parameters and fuel properties (Challen B, 1999). Therefore, it is necessary to achieve better air movement squish, swirl, and turbulence in the CI engine. To optimize the performance, combustion and emissions characteristics of an engine, the hemispherical combustion chamber (HCC) geometry (Figure 3.20 b) was replaced with a toroidal combustion chamber (TCC) (Figure 3.20 a). The volumes of both the piston cavities were kept the same. The default (HCC) piston cavity volume provided by the original equipment manufacturer (OEM) was 34 cubic centimeters, and for the modified (TCC) geometry, it was almost kept the same. The simulations were carried out with CATIA V5-R20 to measure the piston cavity's volume and surface area. The surface area obtained with HCC and TCC piston cavity was 508.95 cm^2 and 517.55 cm^2.

To ensure the same volume for both the pistons, physical measurements using an Isopropyl alcohol (liquid) were carried out. It can easily reach to crevices of the cavity due to its low surface tension property. A flat glass plate with a small hole was kept on the piston head. Isopropyl was poured into the piston cavity through the burette from the glass hole. The volume of the piston cavity was measured from the amount of liquid poured from the burette. Various test results have shown that TCC geometry produces high NO_x emissions (Arumugam, Kasivisvanathan, Arventh, & Maheshkumar, 2015; S Jaichandar & Annamalai, 2012; Jyothi & Reddy, 2017).

3.5.3 Intake manifold modification

As discussed earlier, the conventional intake manifold was modified to mount the PCCI engine's fuel vaporizer. As shown in Figure 3.21, conventional manifold allows only intake air and exhaust gas, and the modified intake manifold was made to allow fuel vapour along with air and exhaust gas, as depicted in Figure 3.21 b. To validate the mounting of fuel vaporiser location, two simulations were performed on ANSYS fluid flow (CFX). Two simulations were performed on the conventional and the modified intake manifold respectively.

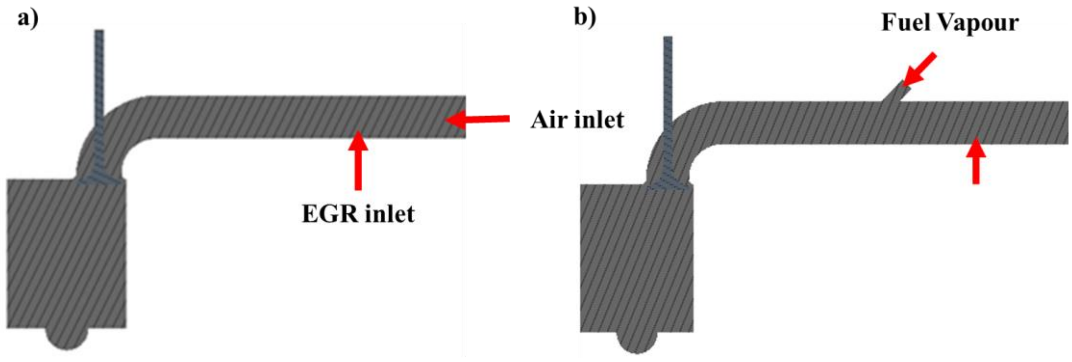


Figure 3.21 CAD model a) Conventional intake manifold b) Partial PCCI modified manifold

The standard k-Epsilon model, suitable for a wide range of flow simulation, was used with total energy heat transfer function.

Table 3.4: Input boundary conditions used for the simulation

Features		Details	
Analysis Type		Transient	
Fluid Type		Variable Composition Mixture	
Domain Type		Single Domain	
Turbulence Model		k-Epsilon	
Heat Transfer		Thermal Energy	
Particle Tracking		Component Source	
Boundary Conditions	Velocity	Air Inlet	10.665 m/s
		EGR Inlet	1.911 m/s
		Diesel vapor inlet	3.807 m/s
	Temperature	Air	300 K
		EGR	330 K
		Diesel vapor	390 K
Time Steps		0.02 sec	

The simulation was based on a steady flow with the reference pressure of 1 atm. on the turbulent model of non-stationary 3D flow. The analysis mainly deals with the multi component mixture formation, and its properties were considered from those of its constituent components. Turbulence plays a vital role in proper mixture formation for the required combustion. The air swirl inside the combustion chamber enhances the mixing with direct-injected fuel, which is expected to have squish flow. Swirl and squish are essential characteristics of turbulence. Hence, the K-epsilon turbulence model is opted to analyse the difference between the mixture's turbulent flow in the conventional intake manifold and the turbulent flow of the mixture in the intake manifold modified for PCCI engine setup. The input boundary conditions used for the analysis areas shown in the Table 3.4.

3.5.3.1 **Analysis of modified intake manifold of PCCI engine**

Mixture flow is analyzed in the combustion chamber and the manifold for the turbulence model and mixture velocity. The conventional setup uses manifold with inlets for intake air and EGR while the modified intake manifold permits entry of diesel vapor along with air and EGR. Both the piston geometries, HCC and TCC were analysed and compared briefly for their turbulence velocity variation.

Turbulence variation of Air-fuel mixture in the intake manifold and combustion chamber

Several techniques are being used to optimize the air-fuel turbulence in the engine cylinder. It is the factor that affects both air swirl and fuel squish (S. Jaichandar & Annamalai, 2013). Piston bowl geometry alterations have been majorly used to enhance air-fuel turbulence (Yadav, Saravanan, Edward, & Perumal, 2015). An increase in DI injection pressure has also favored turbulence (Lalvani, Parthasarathy, Dhinesh, & Annamalai, 2016). Turbulence is categorized by two components, turbulent kinetic energy (k) and turbulent dissipation (ϵ). The k, as shown in Figure 3.22 a and b in the modified intake manifold with TCC piston geometry, was increased by 7.7% near the intake valve. The high turbulence zone around the valve promotes homogenization and enhances the dispersion of charge within the cylinder.

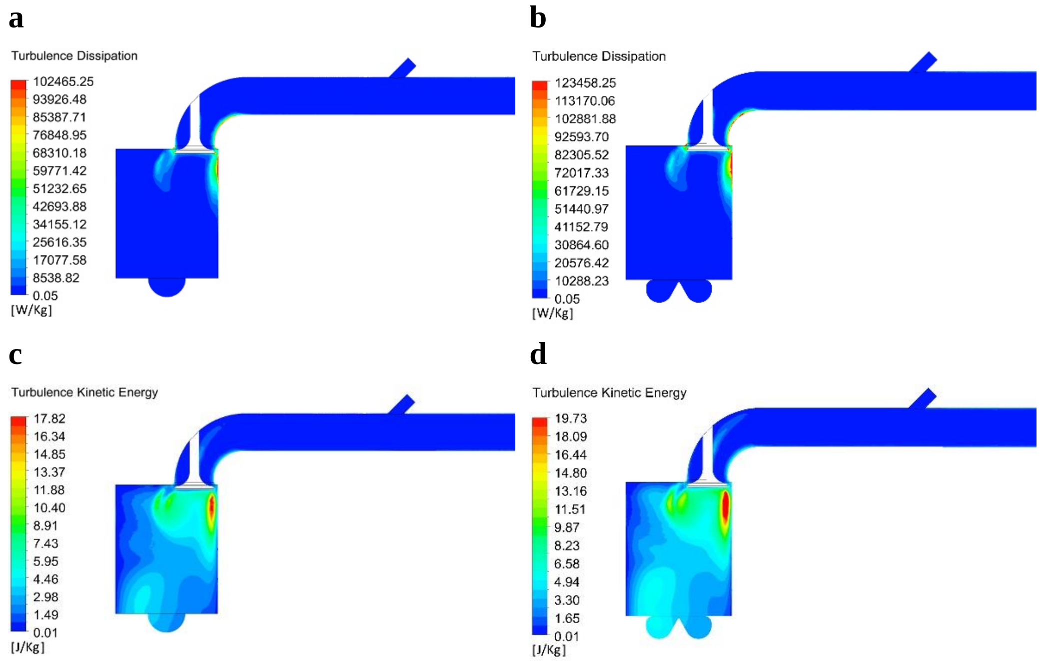


Figure 3.22: Turbulence variation a) Turbulence kinetic energy (conventional manifold), b) Turbulence kinetic energy (modified manifold), c) Turbulence dissipation (conventional manifold), d) Turbulence dissipation (modified manifold)

The conventional manifold lacks the diesel vapour mass in the mixture flow, which is a prominent factor when kinetic energy is considered as discharge rate of diesel vapour directly affects the mixture velocity which is further increased by reduction in flow area near the valve. Increase in both mass and velocity favors high turbulence kinetic energy. Turbulence dissipation as mentioned in Figure 3.22c and d resemble in trend with the turbulence kinetic energy around the valves. The charge entry in the combustion chamber creates the most critical zone for the turbulence to occur. Turbulence dissipation increases to 117939 W/kg in the modified manifold, 44% more than around the conventional manifold valve with.

Velocity variation air-fuel mixture in the intake manifold and combustion chamber

Induction swirl and compression swirl are the primary air motion mechanism in the engine cylinder. Air motion due to induction swirl is affected by piston bowl geometry, masking of valves, etc. (Dent & Derham, 1974). The air motion is analysed with the velocity vector's help. The velocity vector Figure 3.23 a and b shows vortex generated around the intake valve. The vortex flow initiates swirl of the mixture in the combustion chamber, which is essential for obtaining homogeneity and flame distribution. The combustion chamber's vorticity curve linked with the modified intake manifold is much larger than the combustion chamber linked with the conventional manifold. Mixture velocity is elevated by 6.6% in the modified manifold and by 4.3% around the PCCI setup valve.

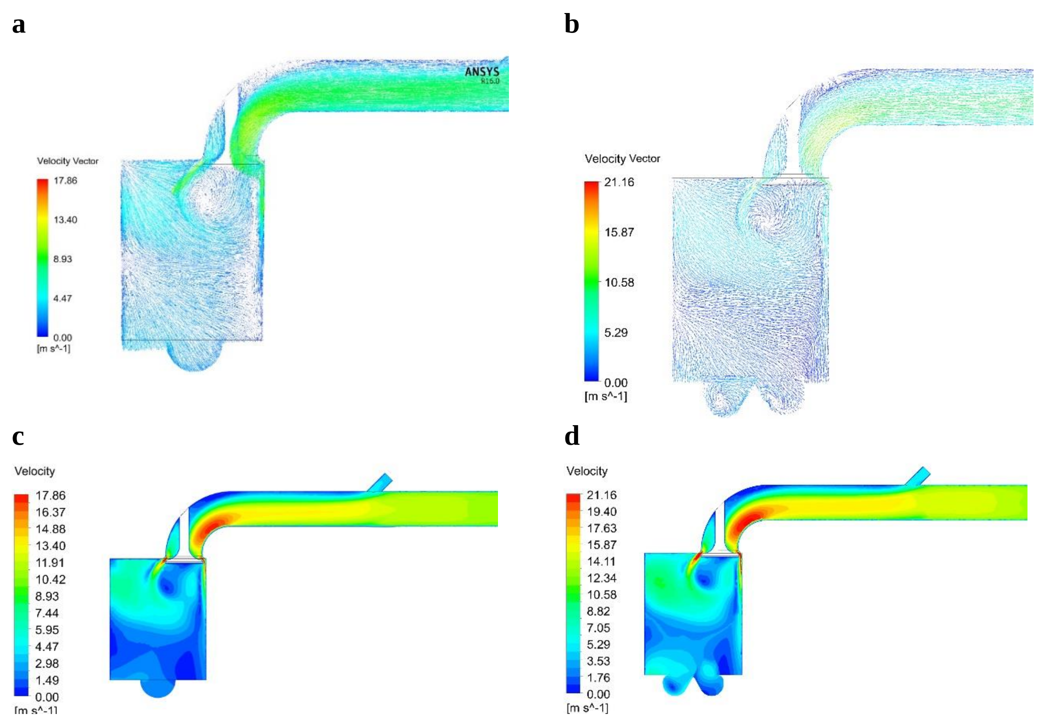


Figure 3.23: Velocity vector distribution a) Velocity vector (conventional manifold), b) Velocity vector (modified manifold), c) Velocity variation (conventional manifold), d) Velocity variation (modified manifold)

Mixture velocity directly affects the turbulence model. Diesel vapour flow in the modified manifold intensifies the mixture velocity, which spreads the charge throughout the chamber, favoring appRoPRiate combustion conditions. The conventional setup relies on direct injection of fuel that is restricted by momentum loss due to atomization restricting the charge spread in the chamber. The addition of diesel vapour adds the mass to the existing air-EGR mixture leading to higher mixture accumulation at the manifold base close to cylinder head mainly due to gravity. The restricted area can be compared with the orifice but with a circumferential opening instead of a small opening. Though the air-EGR mixture in the conventional setup also experiences the restriction, it is less effective due to a lack of diesel vapour.

3.6 Data Analysis

In this section, the processing of combustion and performance parameters data received from the engine is presented. The parameters, in-cylinder pressure, ignition delay, net heat release rate, and brake thermal efficiency, are explained in detail. The PCCI engine was operated with EGR and fuel vapouriser. Therefore the parameters associated with it like EGR ratio and premixed ratio, are also explained in the section.

3.6.1 Theoretical Analysis of In-cylinder Pressure

In-cylinder pressure is the vital parameter for an engine combustion diagnosis and control as it covers useful information on the combustion that takes place in the combustion chamber. The combustion characteristics of an engine were studied during the compression and expansion stroke of the engine cycle. The variation in the in-cylinder pressure during combustion depends on the change in cylinder volume, combustion, heat transfer to the cylinder walls.

To obtain accurate in-cylinder pressure, it is necessary to meet the following requirements.

- Accurate Mounting of the probe in the cylinder-head
- The pressure versus volume phasing is accurate within 0.2°.

- The temperature variation of transducer which affect the calibration.
- The compression and expansion process must be well fitted by a polytrophic relation, PV^n = Constant

Where, n = 1.3 $\pm$ 0.5 for the conventional fuels.

3.6.2 Net heat release rate

In diesel engines, the cylinder content is a single open system. When the intake and exhaust valves are closed, the system boundary's mass flows include the fuel-air mixture into the cylinder. For the calculation of the heat release rate, a single zone model approach was considered. Referring to the first law of thermodynamics, the energy equation for the open system may be written as

$$dQ + \sum \dot{m}_i h_i = dW + dU$$

By neglecting the flow in a crevice the equation can be written with respect to crank angle Θ,

$$\frac{dQ}{d\theta} - \frac{dQ_W}{d\theta} + \dot{m}_{inj}\,.h_f = P\frac{dV}{d\theta} + \frac{dQ}{d\theta}$$

Taking U= sensible internal energy of cylinder charge, h_f as sensible enthalpy of the injected fuel, Q is heat release during the combustion of fuel and Q_w heat rejected out of the cylinder. As h_f is negligible, it can be neglected

Therefore the same equation for the net heat release rate can be written as,

$$\frac{dQ}{d\theta} = P\frac{dV}{d\theta} + \frac{C_v}{R}\left[P\frac{dV}{d\theta} + V\frac{dP}{d\theta}\right] + \frac{dQ_W}{d\theta}$$

Or

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1}\left[P\frac{dV}{d\theta}\right] + \frac{1}{\gamma-1}\left[V\frac{dP}{d\theta}\right] + \frac{dQ_W}{d\theta}$$

For the air temperature at the end of compression when combustion starts, gamma value can be approximately 1.35.

3.6.3 Ignition Delay

Ignition delay of fuel is an essential parameter in determining the knocking behavior of CI engines. Also, it plays a vital role in achieving low smoke and NO_x emissions. As represented in Figure 3.24 period, a-b is an ignition delay. No heat is released during this period. More the ignition delay, better A-F mixing can be possible.

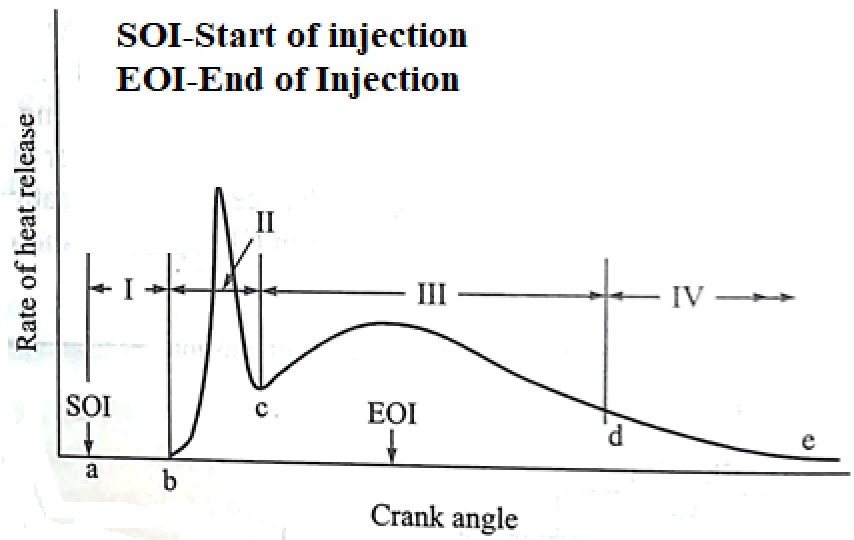


Figure 3.24: Heat release rate and Ignition delay period

Ignition delay period (crank angle) for all the operating conditions is converted to time (ms). Ignition delay period was calculated from the heat release rate diagram by using the following equation.

$$T(ms)=\left[\frac{\text{delay period (CA)}}{((rpm/60)*360)}*1000\right]$$

3.6.4 Mechanical Exhaust Gas Recirculation System

Fuel vaporizer can reduce the NO_x emission up to a certain extent, but it is necessary to decrease further to meet the stringent emission norms. Therefore, to reduce NO_x emissions in the exhaust, it is necessary to keep the peak combustion temperatures under control. One efficient way to ensure this is by Exhaust Gas Re-circulation (EGR).

NO_x formation is a highly temperature-dependent phenomenon, and it takes place when the temperature in the combustion chamber exceeds 2000 K and also when the oxygen concentration in the inlet charge is high (Robert & Shahed, 1981).

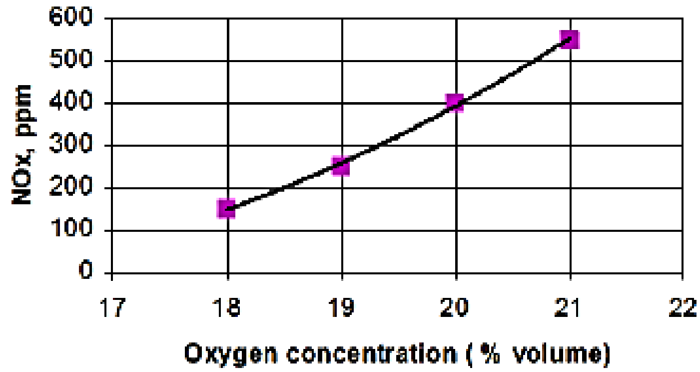


Figure 3.25 Effects of intake oxygen concentration on NO_x formation (Robert & Shahed, 1981)

When EGR is mixed with inlet air supplied to a diesel engine, the inlet charge's temperature increases; this can significantly affect the compressed-charge temperature and the combustion process. The higher inlet charge temperature also reduces the charge density and the inlet charge's mass inlet charge admitted to the engine cylinders.

The exhaust gas consists of CO_2, N_2, and water vapor mainly. When a part of this exhaust gas is re-circulated to the combustion cylinder, it acts as diluents. This also reduces the O_2 concentration in the combustion chamber. The specific heat of exhaust gas is much higher than fresh air; hence, EGR increases the intake charge's heat capacity, thus decreasing the temperature rise for the same heat release.

$$\% \, EGR = \frac{Volume\ of\ \ EGR}{Total\ ch\text{arge intake into}\ Cylinder} \times 100 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Another way to define the EGR ratio is by the use of CO_2 concentration (Baert, Beckman, & Veen, 1999)

$$EGR\ Ratio = \frac{[CO_2]_{\text{int}ake} - [CO_2]_{Ambient}}{[CO_2]_{Exhaust} - [CO_2]_{Ambient}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. \ldots\ldots(2)$$

3.6.5 Standard error of instruments

All instruments have finite precision that limits the ability to resolve small measurement differences. By considering all the possible causes of error, the experiments were performed. The list of types of equipment and their range and accuracy are mentioned in Table 3.5.

Table 3.5: List of equipment and their make, range, accuracy

Equipment	Range	Accuracy
Gas Analyzer (AVL)	CO: 0–10% HC: 0–20,000 NO_x: 0–5000	± 0.01 ±1ppm ±1ppm
Crank angle sensor (Kubler)	0-360°	± 1 °CA
Load cell (Sensortronics)	0 – 50 Kg	± 0.1 kg
Load Indicator (ABUS Technologies)	0-100 kg	± 0.2%
RPM indicator with Speed sensor (Selection process control)	4-9999 rpm	± 0.05%
Pressure transducer (PCB Piezotronics)	5000 psi	± 0.1 psi

CHAPTER 4 EXPERIMENTAL METHODOLOGY

To achieve the research's objectives, various measurement techniques and experimental setup are presented in chapter 3. The Experimental methodology used to conduct the experiments is presented in this chapter. Experiments were performed in three different experimental conditions, as mentioned in

Figure 4.1. In the first experiment, the engine was run on the three compression ratios, one at a time. As discussed in the earlier section, the engine was assessed with two-piston geometry hemispherical combustion chamber (HCC) and a toroidal combustion chamber (TCC).

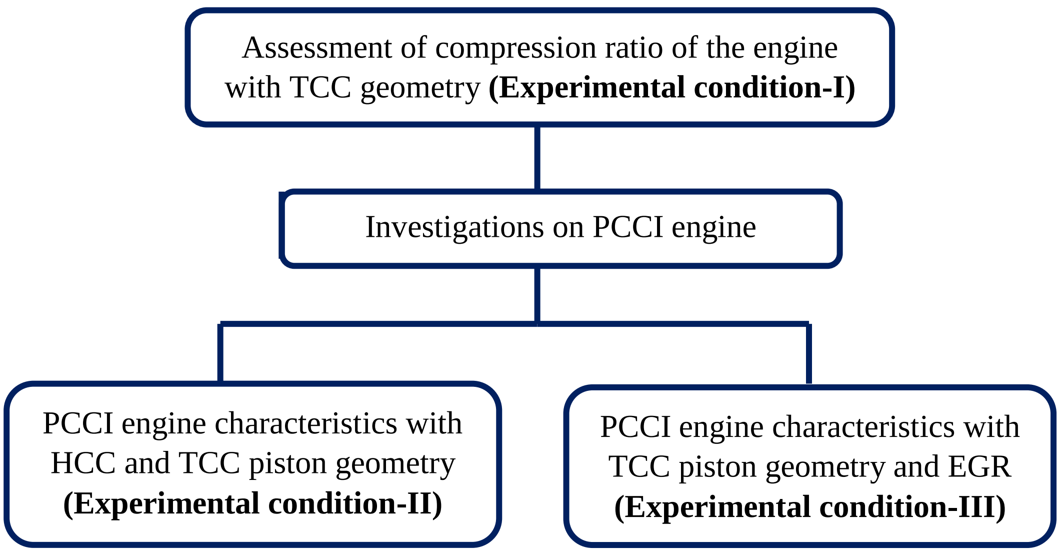


Figure 4.1: Overall Experimental Methodology

The best suitable compression ratio was selected to conduct the experiments on the PCCI engine through experimentation. In second and third experimental condition, the engine was operated on PCCI mode. In the PCCI mode, the engine was operated in mixed mode (Partial vapor and partially conventional, i.e., direct injection). PCCI engine was for the fixed compression ratio of 18:1 (Assessed through experimental condition-I). PCCI engine was investigated with two different vapour quantity. The detailed experimental matrix of the PCCI engine is as depicted in Figure 4.2.

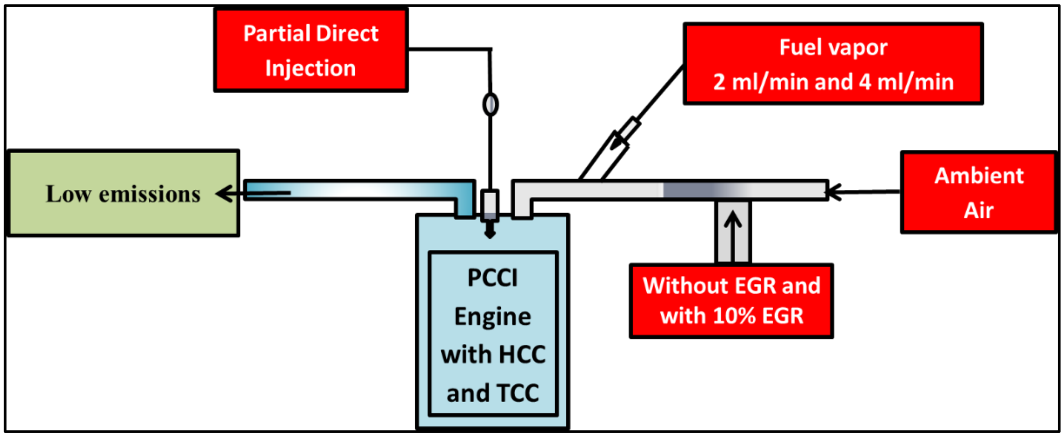


Figure 4.2: Experimental Matrix of PCCI Engine Set Up

The experiments were conducted with 1, 2, 4 and 6 ml in PCCI mode out of which no significant changes in the characteristics were observed at 1 ml of fuel vapor. At 6 ml fuel vapour, engine was completely unstable, may be due to pre-combustion of charge. Therefore, decision were made to operate the PCCI engine on 2 and 4 ml of fuel vapor quantity. Preliminary experiments were carried out with variation of percentage EGR and found that 10% EGR was the optimum based on lower emissions (smoke and NOx) and better thermal efficiency of the test engine.

In **Experimental condition I**, the engine was operated in three different compression ratios, 16:1, 17:1, and 18:1 without and with 10% EGR. In this experiment, the engine was run with two piston geometries HCC and TCC. The comparative analysis was carried out, and its detailed discussion is given in chapter 5, section 5.1. Output parameters evaluated from the experiment were mentioned in Table 4.1. This study's prime motive was to find out the best combination of compression ratio and piston geometry suitable for a PCCI engine. Once the optimized combination was achieved, the engine was operated in PCCI mode to find out the best possible results.

Table 4.1: Experimental Matrix of Experimental Condition-I

Piston	Compression ratios	EGR	Output parameters
HCC piston (Baseline)	16:1, 17:1, 18:1	Without EGR	**Engine Performance** Brake thermal efficiency Exhaust gas temperature Volumetric Efficiency **Engine Combustion** Net heat release rate Pressure-Crank angle Ignition Delay Rate of pressure rise **Engine Emissions** Carbon monoxide Hydrocarbon Oxides of Nitrogen Smoke Emissions
		10% EGR	
TCC piston	16:1, 17:1, 18:1	Without EGR	
		10% EGR	

In **Experimental condition-II**, the engine was operated in PCCI mode. In this experimental condition, the experimentation was performed with varying fuel quantity of 2 ml/min and 4 ml/min. Fuel quantity (e.g., 2 ml/min) was kept constant throughout the engine loads once it was set. Both piston geometries were tested to set fuel vapour quantity to compare all the output parameters mentioned in Table 4.2. The detailed discussion of the experimentation is explained in chapter 5, section 5.2.

Table 4.2: Experimental Matrix of Experimental Condition-II

Piston	Constant Fuel Vapour supply	Output parameters

HCC piston Compression Ratio 18:1	2 ml/min	**Engine Performance** Brake thermal efficiency Exhaust gas temperature Volumetric Efficiency **Engine Combustion** Net heat release rate Pressure-Crank angle Rate of pressure rise **Engine Emissions** Carbon monoxide Hydrocarbon Oxides of Nitrogen Smoke Emissions
	4 ml/min	
TCC piston, Compression Ratio 18:1	2 ml/min	
	4 ml/min	

In the **Experimental condition-III**, the PCCI engine was operated with varying fuel vapour quantity as well as exhaust gas recirculation. In this mode, the engine was operated with TCC piston geometry with 18:1 compression ratio. The detailed discussion of the experimental condition-III results are discussed in chapter 5, section 5.3. The primary aim of this experiment was to analyse the TCC piston performance in PCCI engine along with 10%EGR. Through this experiment it was possible to analyse the NO_x emissions with and without EGR in PCCI engine set up.

Table 4.3: Experimental Matrix of Experimental Condition -III

Piston	Constant Fuel Vapour	EGR	Output parameters
TCC piston,	2 ml/min	Without EGR	**Engine Performance** Brake thermal efficiency

<table>
<tr><td rowspan="3">Compression Ratio 18:1</td><td></td><td>10% EGR</td><td rowspan="3">Exhaust gas temperature
Volumetric Efficiency
<u>Engine Combustion</u>
Net heat release rate
Pressure-Crank angle
Rate of pressure rise
<u>Engine Emissions</u>
Carbon monoxide
Hydrocarbon
Oxides of Nitrogen
Smoke Emissions</td></tr>
<tr><td rowspan="2">4 ml/min</td><td>Without EGR</td></tr>
<tr><td>10% EGR</td></tr>
</table>

CHAPTER 5 RESULTS AND DISCUSSION

5.1 Piston modification and compression ratio increment to enhance Engine Characteristics of CI engine

Many researchers have obtained improved engine performance results, mainly brake thermal efficiency and emission characteristics such as HC and CO from CI engines when TCC has been used (Arumugam et al., 2015; S Jaichandar & Annamalai, 2012; Jyothi & Reddy, 2017). However, the marginal increment was found in NO_x emissions due to high-temperature rise during combustion. To control the increased NO_x emissions in CI engine, two best possible ways can be adopted, either the use of exhaust gas recirculation (EGR) (D. Agarwal et al., 2011; Ladommatos, Abdelhalim, & Zhao, 2000) or reduction in compression ratio (Ozawa, 1997). The Use of the above two solutions to control NO_x emissions, the uncontrolled combustion limitation of HCCI, can also be resolved. The current investigation's main motive was to enhance performance and reduce a conventional CI engine's emissions by incorporating two methodologies, i.e., compression ratio increment and piston modification to TCC geometry. The experiment was performed to find the optimum compression ratio at which TCC geometry provides better engine characteristics. Results obtained from the experiments on the test engine with two different piston geometries HCC (default) and TCC (modified), three different compression ratios 16:1, 17:1, and 18:1 without EGR (Base engine) and 10% EGR are discussed below. Abbreviations used for the discussion are hemispherical combustion chamber without EGR: base-HCC; Toroidal combustion chamber without EGR: base-TCC; hemispherical combustion chamber with 10% EGR: EGR-HCC; and Toroidal combustion chamber with 10% EGR; EGR-TCC.

5.1.1 Engine Performance

5.1.1.1 Brake thermal efficiency

Brake thermal efficiency (BTE) was increased with increasing CR for both the piston geometries. However, TCC had shown higher BTE than HCC due to better combustion and rapid evaporation rate of fuel, as observed in Figure 5.1. At higher compression ratio and full engine load, BTE was decreased by 3% with EGR for both the piston geometries.

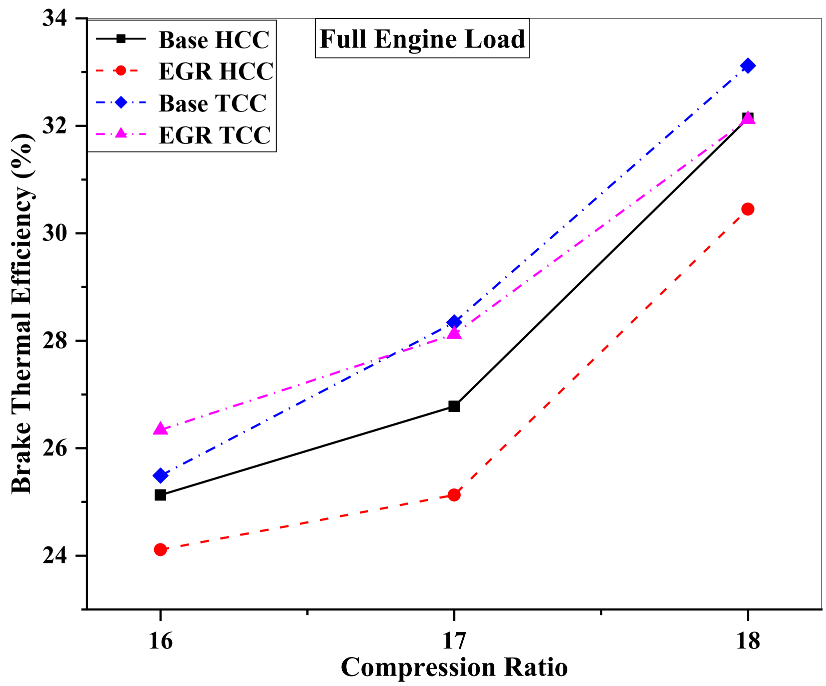


Figure 5.1: Brake thermal efficiency at full engine load at various compression ratios without EGR and 10% EGR

The decrement in BTE with EGR was observed due to the dilution of the A-F mixture. The maximum BTE of 33.12% was achieved with base-TCC, which was higher, by 5.67% than base-HCC. This may be due to better air swirling and turbulence in TCC, which led to better combustion of diesel (Brijesh, Abhishek, & Sreedhara, 2015). Vedhraj et al. observed similar trends with different blends of biodiesel in HCC and TCC engine (Vedharaj, Vallinayagam, Yang, Saravanan, & Lee, 2015).

5.1.1.2 **Volumetric efficiency**

A significant reduction in the amount of intake air to the cylinder was found with exhaust gas addition, as shown in Figure 5.2. These deviations were observed because of the change in the intake air temperature.

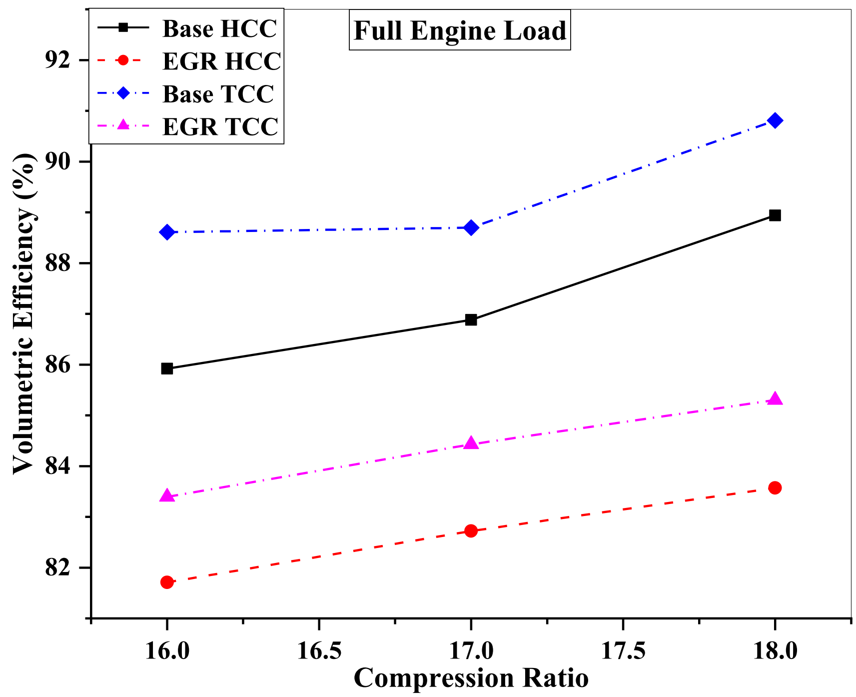


Figure 5.2: Volumetric efficiency at full engine load at various compression ratios without EGR and 10% EGR

Volumetric efficiency was increased with an increase in the compression ratio from 16:1 to 18:1 due to the engine's increase in breathing capacity. The maximum volumetric efficiency was obtained 91% with base-TCC. There was a marginal enhancement in volumetric efficiency with TCC geometry at all specified conditions than HCC.

5.1.1.3 **Exhaust gas temperature**

Variations of exhaust gas temperature (EGT) with the change in compression ratios and piston geometries are depicted in Figure 5.3. EGT decreased with an increase in CR because of better combustion and ignition delay reduction, as observed in Figure 5.6.

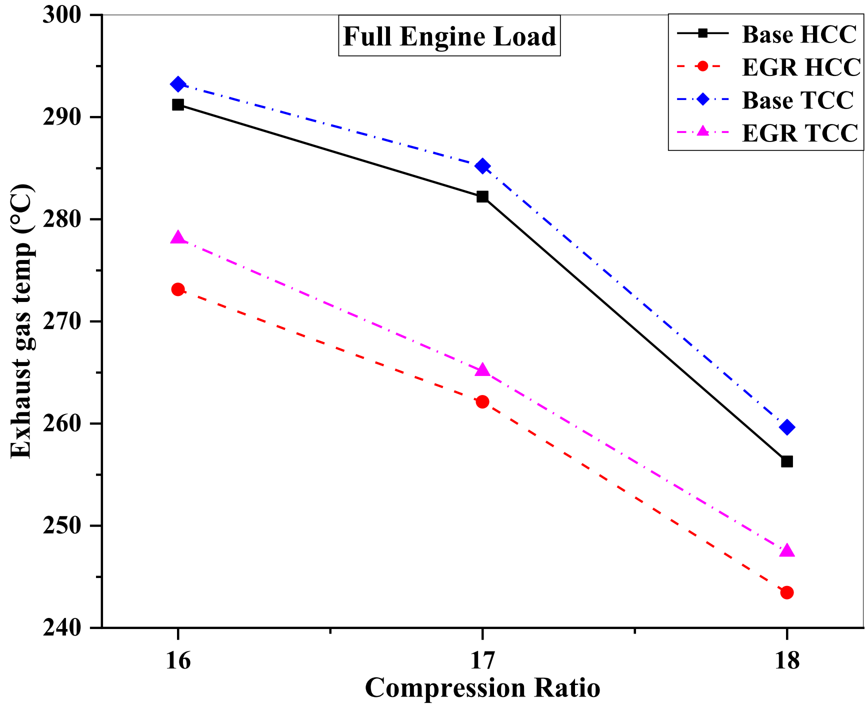


Figure 5.3: Exhaust gas temperature at full engine load at various compression ratios without EGR and 10% EGR

The use of EGR shows a decrement in EGT due to the dilution of the A-F mixture. The heat release rate during combustion decreases with EGR addition and causes the EGT reduction. At full engine load, EGT was decreased by almost 9% at 18:1 CR compared between base-HCC and EGR-HCC. The maximum EGT was achieved about 292 °C with base-TCC. Rapid combustion led to the shorter duration of burning the A-F mixture, which increases overall combustion temperature. The same trend of decreasing EGT with an increase in CR had been obtained by Hariram and Vagesh (Hariram & Vagesh Shangar, 2015).

5.1.2 Engine combustion characteristics

5.1.2.1 In-cylinder pressure and net heat release rate

Cylinder pressure for full engine load at compression ratios 16:1 to 18:1 with and without EGR were plotted and analyzed, as shown in Figure 5.4. The peak in combustion pressure occurred slightly near TDC as the compression ratio increased from 16:1 to 18:1 for both the HCC and TCC geometries. At the same engine load, the peak pressure for base-TCC (60.82 bar) at 18:1 CR is higher than HCC (56.31 bar) by 7.45%, as shown in Figure 5.4 b.

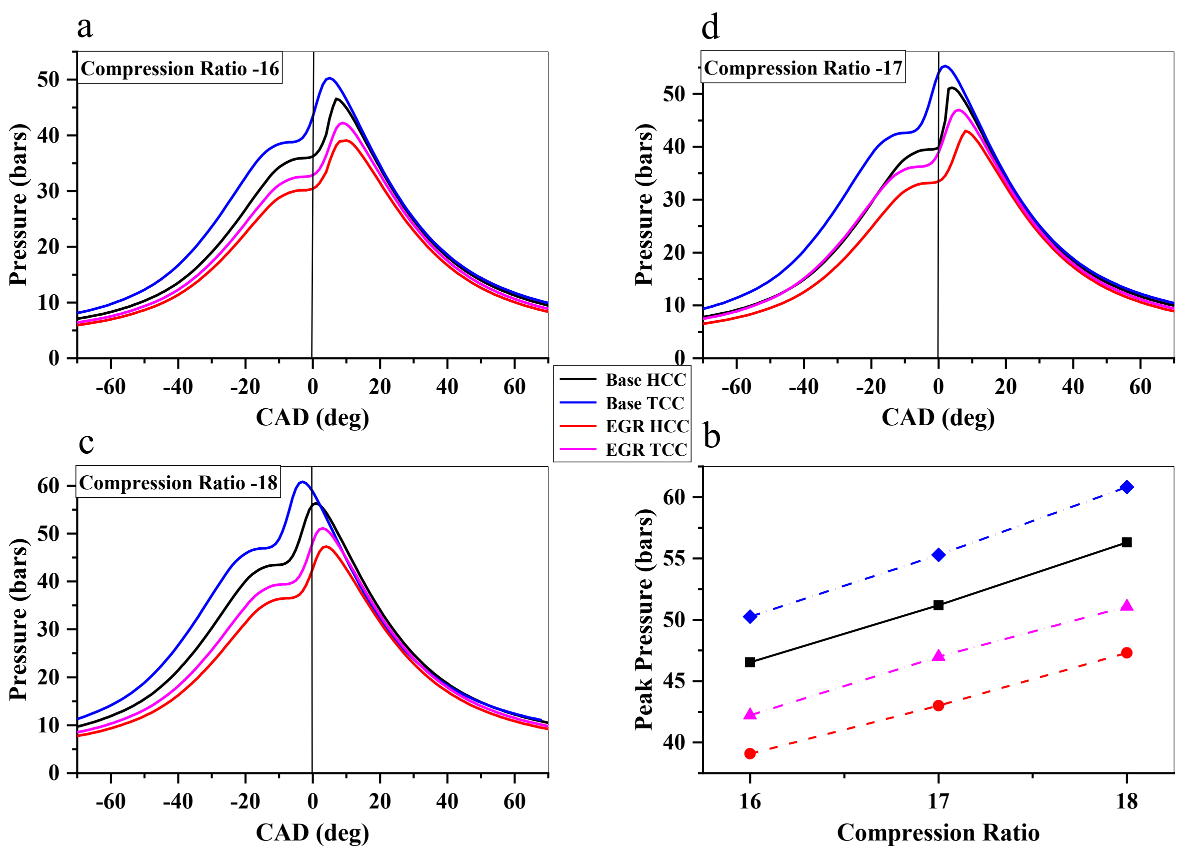


Figure 5.4: In-cylinder pressure at various compression ratios and 10% EGR at full engine load

The use of EGR showed a negative effect on the combustion characteristics of an engine. It is because of increment in the intake charge specific heat capacity and reduction in O_2 availability. EGR led to a decrease in the in-cylinder pressure during combustion and combustion temperature, as depicted in Figure 5.5. At full engine load, the maximum heat release was observed with base-TCC (47.06 J/°CA) at 18:1 CR and also found that the peak point of heat release rate was approaching near TDC with an increase in CR. The negative heat release was observed at all engine loads because of the heat transfer to the cylinder surfaces.

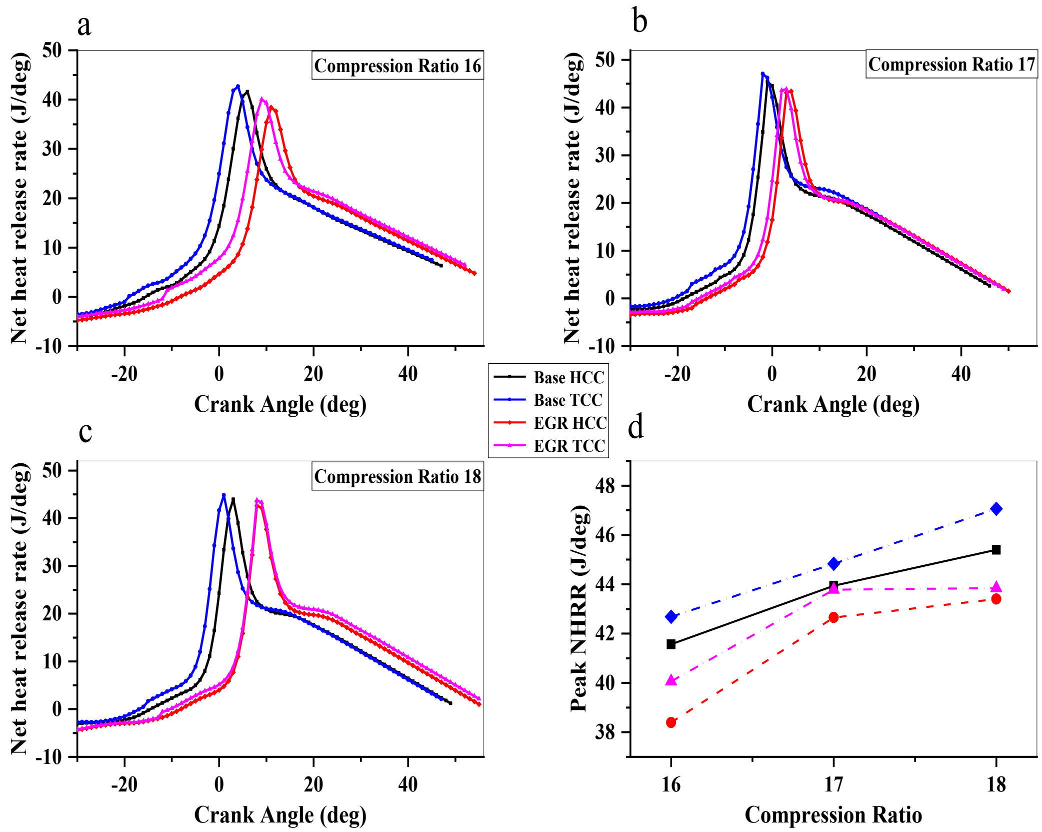


Figure 5.5: Net heat release rate at various compression ratios and 10% EGR at full engine load

5.1.2.2 **Ignition delay**

Ignition delay of fuel is an essential parameter in determining the knocking behavior of CI engines. Figure 5.6 shows the ignition delay variation for various compression ratios and 10% EGR at full engine load. It has been observed that the ignition delay periods for TCC are lower than HCC at all specified engine operating conditions. Due to the rapid mixing of the A-F mixture in TCC, fuel attains the self-ignition temperature in a short period. Hence the delay period decreases in such a combustion chamber (S Jaichandar & Annamalai, 2012). The ignition delay period was

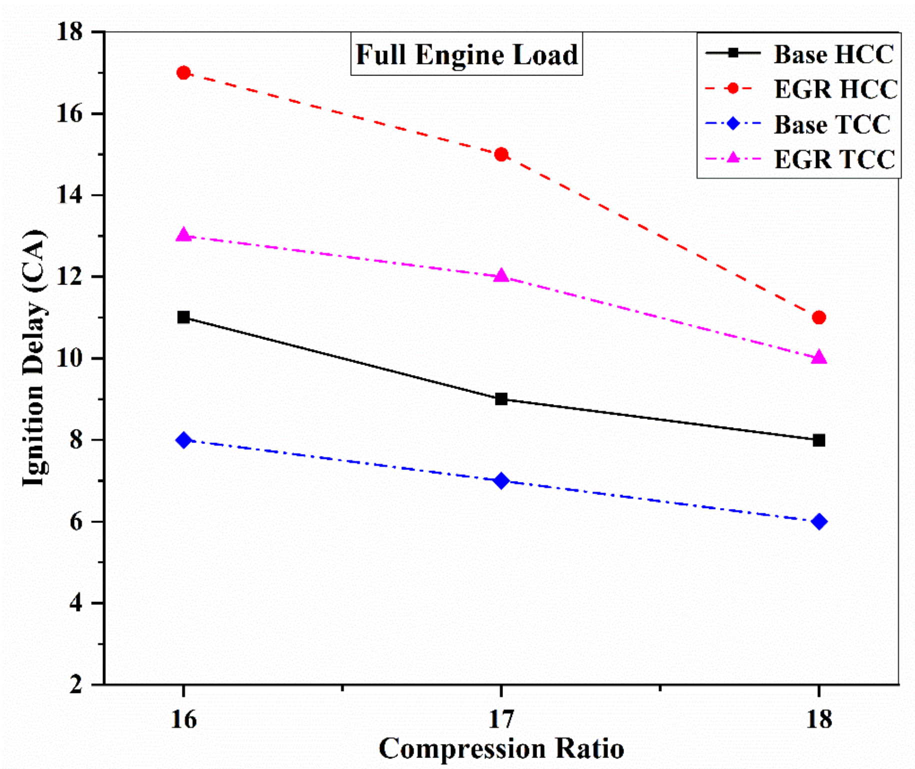


Figure 5.6: Ignition Delay at various compression ratios and 10% EGR at full engine load

increased with EGR with HCC and TCC, and it can be noticed in

Table 5.1. The ignition delay period (crank angle) for all the operating conditions are converted to time (ms) by Equation 1. At 18:1 CR, ignition delay was decreased by almost 24% with base-TCC than base-HCC.

$$T(ms)=\left[\frac{\text{delay period (CA)}}{((rpm/60)*360)}*1000\right] \quad \ldots\ldots (1)$$

Table 5.1: Ignition delay period at various compression ratios and 10% EGR in terms of crank angle and time

Compression Ratio	Base-HCC		Base-TCC		EGR-HCC		EGR-TCC	
	CA (deg)	Time (ms)	CA (deg)	Time (ms)	CA (deg)	Time (ms)	CA (deg)	Time (ms)
16:1	11	1.22	8	0.88	17	1.88	13	1.44
17:1	9	1.00	7	0.77	15	1.66	12	1.33
18:1	8	0.88	6	0.67	11	1.23	10	1.11

5.1.2.3 **Rate of pressure rise**

The RoPR indicates combustion roughness, and it is a crucial parameter in the entire engine operation. A higher the RoPR means the maximum amount of injected fuel is burnt during the premixed combustion phase (Selim, Radwan, & Elfeky, 2003). Figure 5.7 compares RoPR at various compression ratios and 10% EGR at full engine load. The maximum peak in RoPR was found to be 3.47 bar/°CA 2° bTDC with base-TCC, whereas, for base-HCC, it was 3.36 bar/°CA 3° aTDC at 18:1 CR, which is lower than base-TCC with a small margin. It shows that the combustion phenomenon in both the piston geometries is almost the same. On the other side, the minimum peak in RoPR was 1.73 bar/°CA occurred at 16:1 CR with EGR-HCC. The density of A-F mixing decreases as the compression ratio reduces and further decreases with EGR, decreasing the RoPR during combustion.

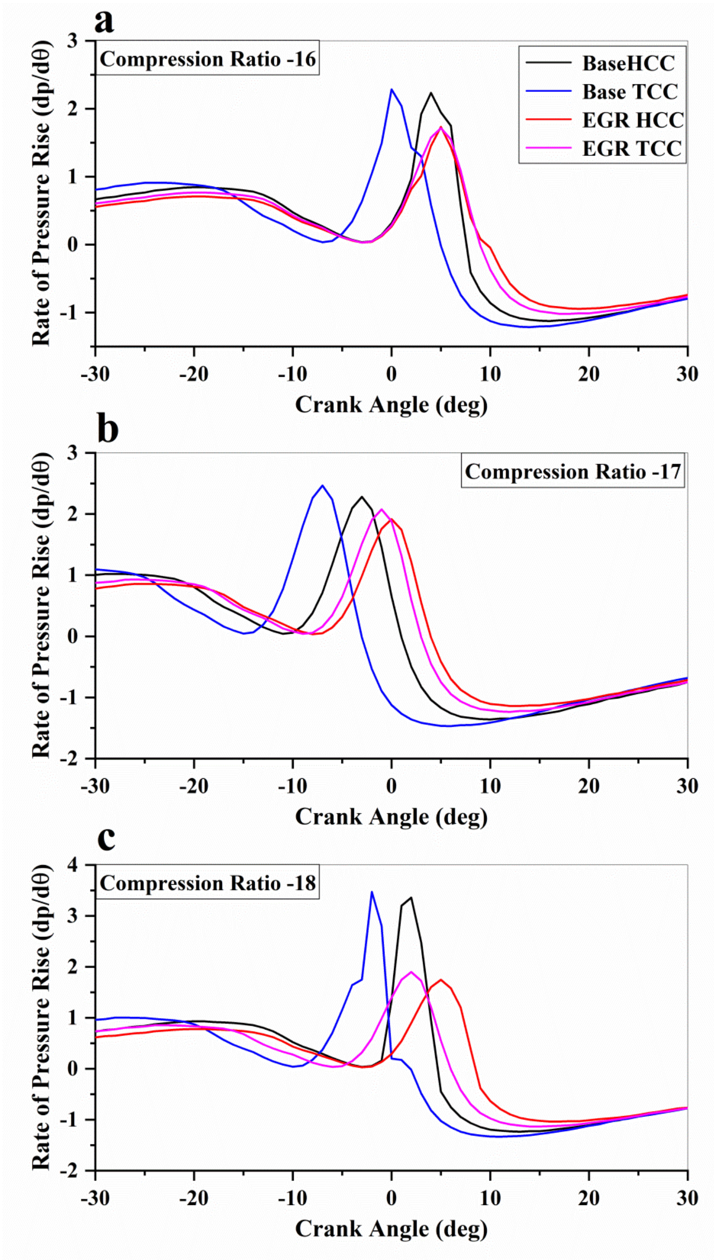


Figure 5.7: Rate of pressure rise at various compression ratios and 10% EGR at full engine load

5.1.3 Engine emissions reduction

5.1.3.1 NO_x Emissions

NO_x emissions are produced at high combustion temperature, and it depends on several engine parameters like compression ratio, piston bowl geometry, equivalence ratio, etc. (Heywood, 1988). It is noticed in Figure 5.8 that NO_x emissions were decreased with the use of EGR.

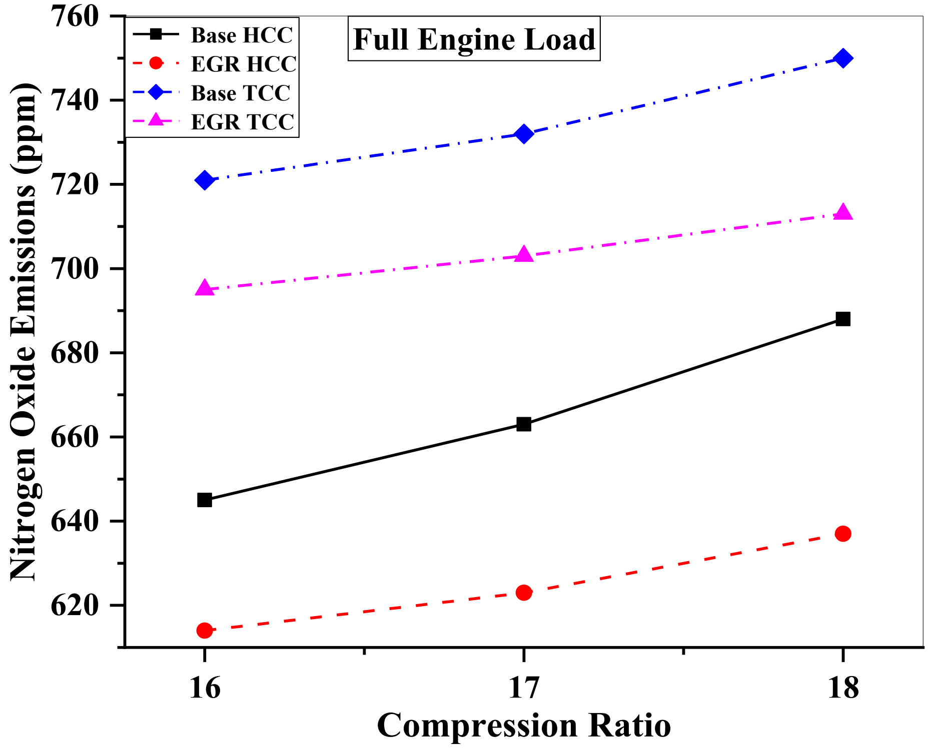


Figure 5.8: NO_x at various compression ratios and 10% EGR at full engine load

This trend was observed due to decrement in O_2 availability because of mixing a partial amount of the O_2 of fresh intake air with recirculated gas (D. Agarwal et al., 2011; M. Guo et al., 2015). This reduces the local flame temperature because of the flame's spatial broadening due to oxygen reduction (Hussain, Palaniradja, Alagumurthi, & Manimaran, 2012). In the end, because of endothermic chemical reactions like the dissociation of H_2O and CO_2, the combustion temperature was decreased (Maiboom, Tauzia, & Hétet, 2008). At full load and 18:1 CR, NO_x emissions were recorded as 745 ppm for base-TCC, which was maximum among all specified conditions. However, for EGR-TCC, it was decreased by 6.4% than base-TCC. EGR-TCC and base-HCC have produced the same amount of NO_x emissions at 18:1 CR. It was noticed that NO_x emissions steadily increased with increasing CR. The increment in NO_x emissions was due to reduced ignition delay and increased peak pressure, resulting in increased combustion temperature.

5.1.3.2 **Carbon monoxide emissions**

Carbon monoxide (CO) emissions are produced due to incomplete combustion, and it mainly depends on the A-F ratio (Park, Youn, & Lee, 2010).

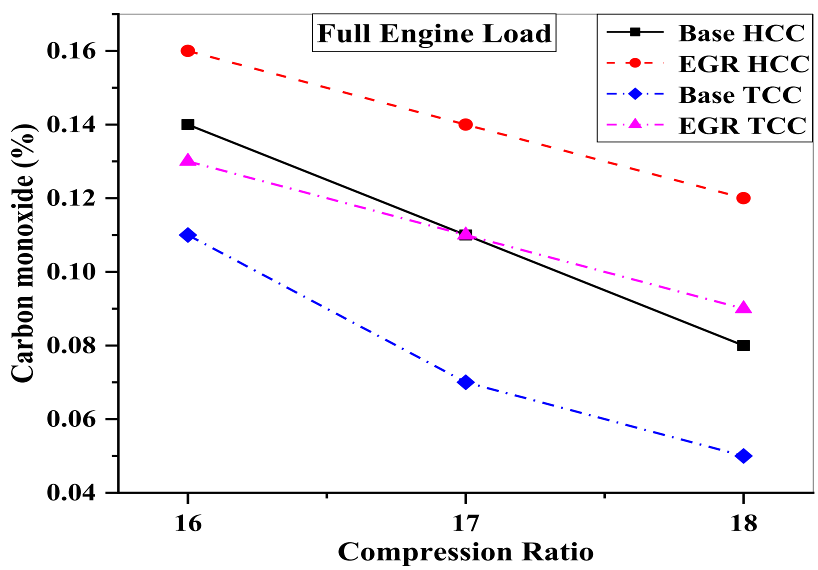


Figure 5.9: CO emissions at various compression ratios and 10% EGR at full engine load

The use of EGR decreases in-cylinder O_2 availability during combustion and slows down the A-F mixture's reaction rates, hence producing lower temperatures (D. Agarwal et al., 2011). At such a temperature, the flame front propagation could not be sustained with lean mixtures. Thus, the A-F mixture does not combust completely, causing CO emissions, as depicted in Figure 5.9. It was observed that CO emissions are decreasing with an increase in the compression ratio due to better combustion. Agreeing with the references (De Serio, de Oliveira, & Sodré, 2017; B. R. Kumar, Saravanan, Rana, Anish, & Nagendran, 2016), a peak in CO emissions were observed as EGR proportion increased in fresh A-F mixture. Minimum CO emissions (0.05%) were achieved at base-TCC with 18:1 CR. TCC piston geometries had shown lower CO emissions than HCC due to enhanced combustion (S Jaichandar & Annamalai, 2012; Karthickeyan, 2019). An interesting fact came to observation was that the CO emissions for EGR-TCC and base-HCC were almost the same at all the compression ratios.

5.1.3.3 **Hydrocarbon emissions**

Figure 5.10 shows the HC emissions at varying compression ratios and 10% EGR at full engine load. It is observed that the HC emissions steadily decrease with increasing compression ratio. This is because the increase in the intake air temperature at the end of compression stroke improves the combustion temperature and reduces the charge dilution, leading to better combustion and reduction in HC emissions. It was seen that with the induction of exhaust gases with

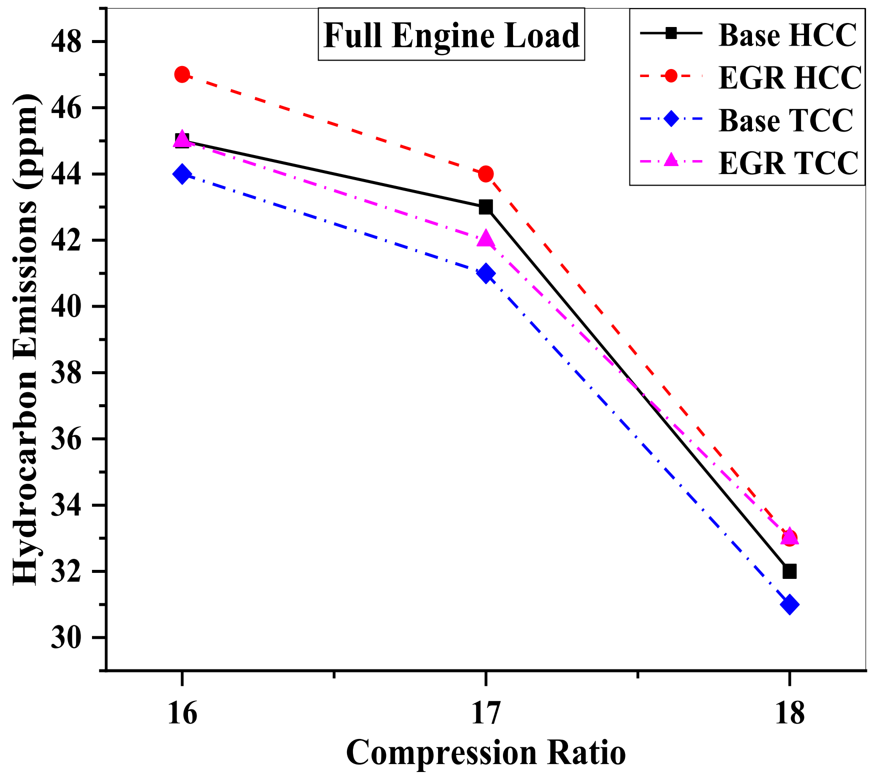


Figure 5.10: HC emissions at various compression ratios and 10% EGR at full engine load

fresh charge, HC emissions were increased. Fuel quantity being injected for any specific condition with or without EGR is remains the same. Recirculated exhaust gas decreases the O_2 availability for combustion, leading to an increase in HC emissions from an engine than the base condition (De Serio et al., 2017). The maximum value of HC emissions was observed to be 48 ppm at 16:1 CR with EGR-HCC. TCC has shown lower HC emissions at all the specified conditions than HCC; it is due to better mixing of A-F because of improved air swirl (Jyothi & Reddy, 2017). At 18:1 CR base-TCC has shown 8.82% of decrement in HC emissions than base-HCC. The summary of the results is shown in

Table 5.2, which represents the effect of piston geometries, CR, and EGR on engine behaviour.

Table 5.2 Summary of results

Parameter	TCC compared to HCC	Effect of CR	Effect of EGR
BTE	Increased with use of TCC	Increased with the increase of CR	Decreased with EGR
EGT	Increased with use of TCC	Decreased with an increase in CR	Decreased with EGR
Volumetric efficiency	Higher in TCC	Increased with an increase in CR	Decreased with EGR
In-cylinder combustion pressure	Higher in TCC	Increased with an increase in CR	Decreased with EGR
NHRR	Higher in TCC	Increased with an increase in CR	Decreased with EGR
CO emissions	Lower in TCC	Decreased with an increase in CR	Increased with EGR.
HC emissions	Lower in TCC	Decreased with an increase in CR	Increased with EGR.
NO_x emissions	Higher in TCC	Increased with an increase in CR	Decreased with EGR

The present study reveals that the performance, emissions, and combustion characteristics of the variable compression ratio test rig implemented with EGR can be improved using a suitable combustion chamber geometry and compression ratio. It also understood that the TCC geometry with EGR might also provide better results on the PCCI engine set up. The TCC geometry can help in overcoming the drawback of PCCI engines. It can reduce the penalty of HC and CO emissions in PCCI engines.

5.2 Comparative study of PCCI engine characteristics with HCC and TCC combustion chamber

Experiments were conducted to understand the combined effect of PCCI and varying piston geometry against the conventional DI engine. The engine was operated at 2 ml/min and 4 ml/min diesel vapor induction rate in PCCI mode at constant rpm. Each induction was accompanied by two different piston bowl geometry, HCC, and TCC. Observations were scrutinized based on performance, combustion, and emission characteristics of the engine.

5.2.1 Engine Performance

5.2.1.1 Brake thermal efficiency

The variation in brake thermal efficiency has been studied at all engine loads. The influence of piston bowl geometry and varying diesel vapor during combustion is clearly outlined in Figure 5.11a. The increase in brake thermal efficiency with engine load is observed in the whole engine operations. At full engine load, elevated demand for useful power prevents heat loss due to the loss of exhaust gas and cooling water. Studies have shown the fall in thermal efficiency using the HCCI engines adversely affecting brake thermal efficiency (Yao et al., 2009). Diesel vapour induction via HCCI has been a limiting factor for the brake thermal efficiency (D. Ganesh et al., 2008). As seen in Figure 5.11 a**,** 2 ml/min fuel vapour induction with TCC geometry has shown minimal penalty in BTE. However, with HCC geometry it was decreased by 4.63% at full engine load compared to the conventional diesel engine. At the low engine load, the penalty in BTE was lower compared to the higher load. At low engine load, vaporized diesel fuel initiates homogeneous mixing with the intake air. However, diesel vapour's high flow rate compared with the intake air leads to the formation of a rich mixture, causing excess fuel to be left unmixed, relatively decreasing the brake thermal efficiency.

Among all the PCCI mode combinations, the maximum BTE obtained was 28.45% with TCC geometry, 2 ml/min fuel vapour induction at full engine load. It has been observed that, the difference in BTE with both piston geometry is almost the same with 2 ml/min and 4 ml/min fuel vapour induction. However, the penalty in BTE with the 4 ml/min fuel vapour induction was higher than the 2 ml/min induction of fuel vapour. Enhanced piston bowl geometry creates the compensation factor for the loss of brake thermal efficiency due to diesel vapour induction. The toroidal chamber contributes to the effective combustion in several ways. However, increasing the mixture's turbulence is the most crucial TCC geometry property, leading to amplification in mixing the homogenous charge followed by proper charge distribution.

5.2.1.2 **Exhaust gas temperature**

Exhaust gas temperature is an essential factor in determining the heat loss from the exhaust gases and the actual combustion temperature. Varying trends of exhaust gas temperature at all engine loads is shown in Figure 5.11 b. The increase in engine load is accompanied by an increase in exhaust gas temperature at all engine operations. The increase in fuel quantity with engine load in the conventional engine and sufficient ratio of homogenized diesel vapour at full load in the PCCI engine creates a rich mixture that elevates the combustion, leading to increased temperature of exhaust gas. The HCCI engines shown the low-temperature of combustion (Stanglmaier & Roberts, 1999). A similar result was achieved while implementing PCCI. The diesel vapour induction effectively minimizes the exhaust gas temperature. The exhaust gas temperature was decreased at low engine load by 11°C and 52°C when the engine was induced with 2 ml diesel vapour with TCC and HCC piston geometry compared to a conventional engine. However, it was higher at full engine load with TCC geometry at 2 ml due to the higher temperature and in-cylinder pressure during combustion. The combustion temperature elevation is heavily affected by the high rate of pre-mixed combustion occurring in the conventional engine due to fuel accumulation during the initial injection stage. The external mixing of vaporized diesel avoids fuel accumulation with the intake air.

As a result, the charge entering the chamber is entirely homogenized, contributing towards controlled combustion. However, at full engine load, the effect of diesel vapour induction contradicts, especially for 2 ml induction with TCC piston geometry increasing the exhaust gas temperature by 2%. At full engine load, the PCCI setup is accompanied by a small amount of injected fuel. The atomized drops are devoid of air molecules forming a rich zone contributing to high temperature during combustion. The enhanced air swirl due to TCC piston bowl geometry provides optimum conditions for high combustion rate resulting in high exhaust gas temperature. At half engine loads, the exhaust gas temperature decreases by 7% for 2 ml and 21% for 4 ml diesel vapour induction with TCC piston bowl geometry. The decrease is primarily due to induced diesel vapour.

5.2.1.3 **Volumetric efficiency**

The intake air flow rate for a given engine load is a crucial factor for achieving ideal combustion. Figure 5.11c shows the volumetric efficiency variation at all engine loads for the conventional and PCCI mode along with varying combustion chamber geometry. The volumetric efficiency decreases with an increase in engine load due to increased fuel demand at high load. The piston bowl geometry has a considerable effect on volumetric efficiency.

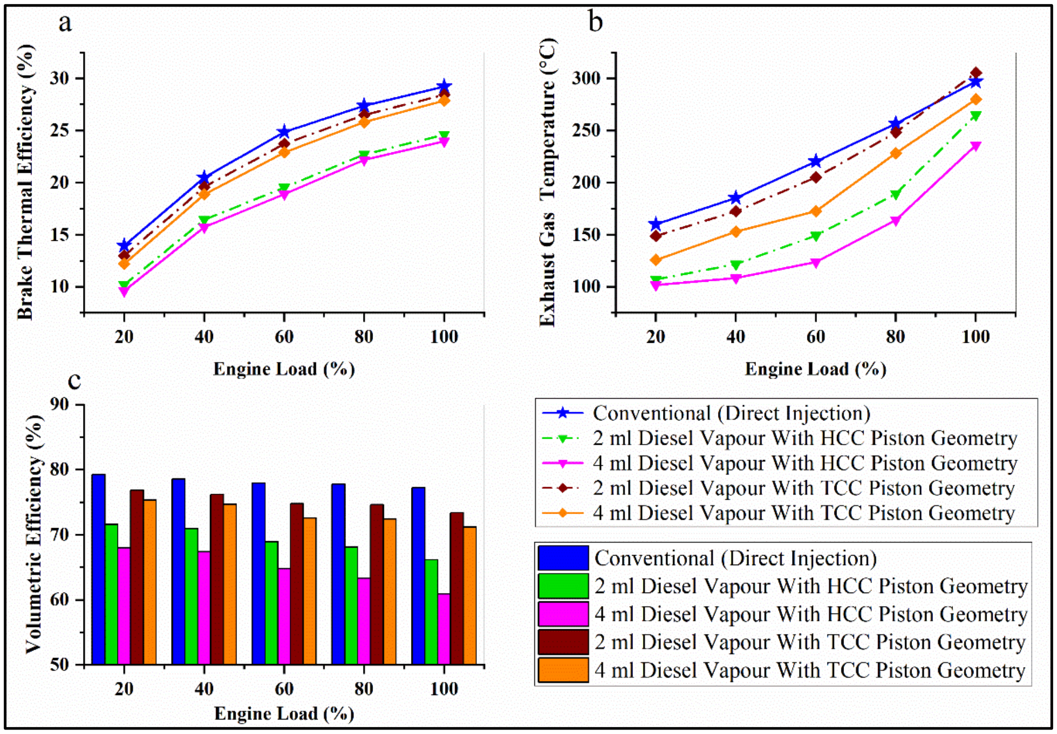


Figure 5.11: Engine performance characteristics a) Brake thermal efficiency Vs engine load b) Exhaust gas temperature Vs engine load c) volumetric efficiency Vs engine load

At full engine load, volumetric efficiency decreases was 3.83% and 11% for diesel vapour induction at 2 ml with TCC and HCC piston bowl geometry compared to a conventional engine. It was one of the causes of decrement in BTE with both the piston geometry. However, the loss was minimized using TCC piston bowl geometry. It was decreasing the penalty in volumetric efficiency in PCCI mode operation. The TCC geometry acts as both guideways and turbulence initiators for the air-fuel vapour mixture. It has drawn a sufficient amount of intake air hindered by the diesel vapour flow in the modified intake manifold. The decrease in volumetric efficiency with 2 ml/min and 4 ml/min fuel vapour induction in TCC geometry was 3.2%. It shows that the excess quantity of fuel vapour was creating a hindrance to the intake air.

5.2.2 Engine Combustion

5.2.2.1 Pressure vs. crank angle (degrees)

It is always necessary to control the in-cylinder pressure during combustion in the engine cylinder. During the combustion, high-pressure generation is one of the disadvantages of the conventional engine, which creates conditions leading to high NO_x emission. The NO_x emission is a function of in-cylinder pressure and temperature. Figure 5.12a outlines the engine operation's peak pressure in the conventional mode and PCCI mode with varying diesel vapour flow rate and piston bowl geometry at full engine load. The figure highlights the effect of diesel vapour induction optimizing the combustion timing towards the ideal requirements thereby, shifting the peak pressure closer to the top dead center by 3 °CA and 6 °CA for diesel vapour induction at 2 ml and 4 ml with HCC piston bowl geometry. Moreover, the peak pressure is reduced by 9.3% and 12% with this piston geometry.

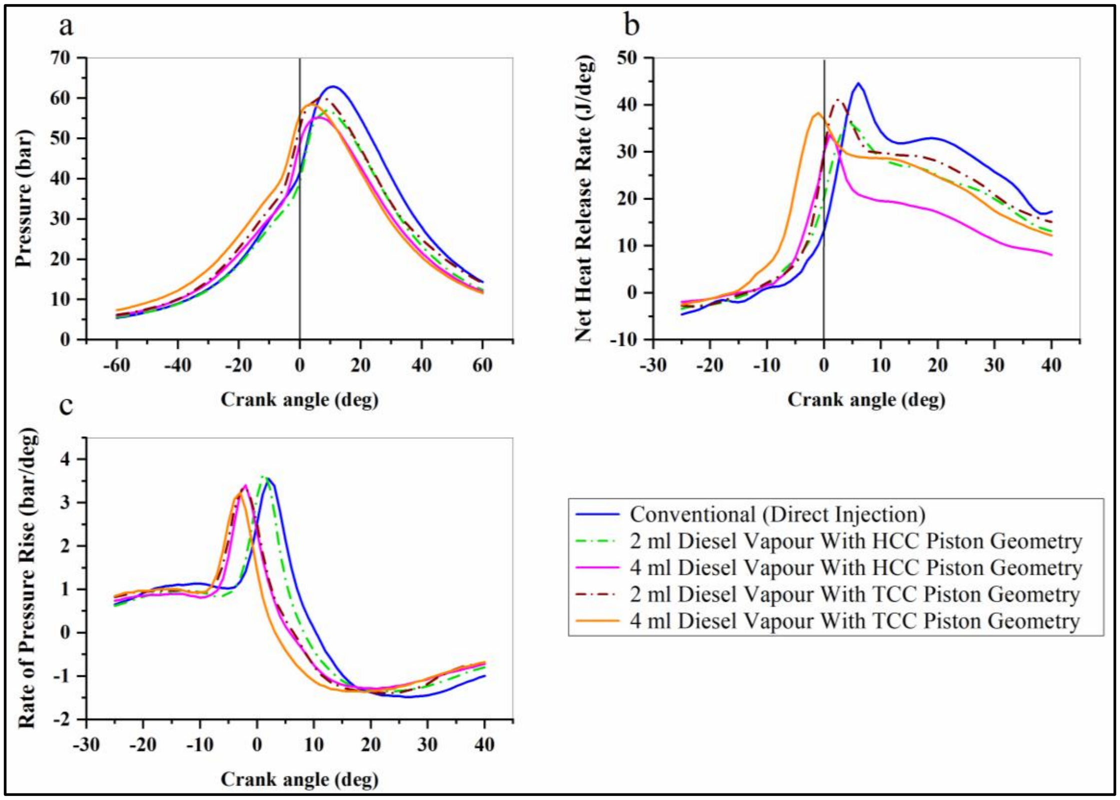


Figure 5.12: Engine combustion characteristics a) pressure Vs. Crank angle b) net heat release rate Vs. Crank angle c) Rate of pressure rise Vs. crank angle

Premixing of diesel vapour with intake air minimizes the diesel accumulation and delay in combustion occurring due to heterogeneity of liquid diesel with air, resulting in proper combustion initiation timing with reduced peak pressure. The use of TCC is effective for high combustion rate with proper combustion duration however; it amplifies the peak pressure compared to HCC piston geometry while the combined effect of TCC and diesel vapour induction 2 ml and 4 ml decreasing the peak in-cylinder pressure by 4.2% and 7.1% respectively compared to a conventional engine.

5.2.2.2 Net heat release rate vs. crank angle (degrees)

The HCCI technology has been limited by a high heat release rate resulting in ringing (Eng, 2002). No control over combustion is the primary reason for ringing. PCCI technology has a tendency to counter the issue. Figure 5.12 b shows the variation in the net heat release rate at full engine load. Variation in the peak value of heat release rate with respect to the top dead center has been excessively affected by diesel vapour induction, especially when the engine runs at a diesel induction rate of 4 ml, advancing the heat release by 6 °CA and 8 °CA for HCC and TCC piston bowl geometry. The TCC geometry has shown the peak in heat release rate at a 4 ml/min induction rate just before the top dead center, indicating early combustion. Due to improved mixture homogeneity by high vapour flow rate and enhanced charge distribution through the TCC geometry, the ignition conditions are achieved even at the initial stage of compression stroke due to pre-heated heated diesel vapors resulting in pre-ignition. It is affecting the brake power significantly due to early heat release. However, the net heat release rate is reduced by 13% for the same, which also helps in reducing NO_x emissions. The reduction primarily affects diesel vapour's high induction rate, hindering the required intake air flow rate. Insufficient air molecules in the combustion chamber restrict the effective utilization of diesel vapour limiting the combustion resulting in low net heat release rate. Optimum crank angle for net heat release is achieved when the engine runs on diesel vapour induction at 2 ml/min with TCC piston bowl geometry. The vapour induction rate is sufficient to allow the required quantity of air forming the proportioned mixture. However, in the partial PCCI technique the vaporized diesel is introduced in the combustion chamber during the suction stroke, the temperature difference between the intake air and diesel may lead to minor condensation of vapor that are unable to combine with air molecules due to natural heterogenic characteristic of liquid diesel making the condensed fuel inactive for combustion leading to reduction in peak value of net heat release rate by 8%.

5.2.2.3 **Rate of pressure rise vs. crank angle (degrees)**

Higher the irregularity in the rate of change pressure higher is the tendency of the engine to knock. Figure 5.12 c shows the RoPR at full engine load. Studies on HCCI using diesel fuel have resulted in premature ignition followed by knock when the engine operated on normal diesel compression ratio (Gray & Ryan, 1997). It is observed that the PCCI engine operation at 2 ml diesel vapour induction rate with HCC piston geometry increases RoPR by 3% while engine operation at 4 ml diesel vapour induction rate with the same piston bowl geometry decreases RoPR by 4.2 % indicating the ambiguous nature of PCCI. At full engine load, increased demand of fuel is compensated by direct injection along with 2 ml induction rate however, the unavailability of the required intake air for the injected fuel creates a rich mixture zone that attains high temperature prior to combustion due to heat transfer from diesel vapour. As all the conditions are favorable uncontrolled combustion may initiate at the junction formed between the atomized diesel and causing abrupt rise in the rate of pressure. Diesel vapour induced at 4 ml rate lessens the amount of directly injected fuel thereby, preventing the rich zone formation and hence, decrease in RoPR is observed. Though TCC piston bowl geometry promotes rise in peak pressure but the effective charge distribution in the combustion chamber provides smooth combustion even at low induction rate thereby the abruptness in pressure rate. RoPR decreases by 5% and 9% for diesel vapour induction at 2 ml and 4 ml with TCC piston bowl geometry.

5.2.3 Engine Emissions

5.2.3.1 HC and CO emission

The HC and CO emission categorize combustion quality. Proper combustion reduces both CO and HC while the disproportioned charge promotes incomplete combustion elevating HC and CO emissions in the exhaust gas. Figure 5.13 a and b show variation in HC and CO emissions at all engine loads in the PCCI and conventional mode. Lean charge formation due to low fuel flow rate deteriorates combustion quality leading to high CO and HC emission at low load (Dec, 2002). However, TCC piston bowl geometry plays a vital role in minimizing the HC and CO emission even at low load. The diesel vapour induction provides better homogeneity, the decrease in volumetric efficiency, especially in a 4 ml vapour induction rate at low engine load, hampers the stoichiometric requirements for optimum combustion. In PCCI mode, at 4 ml/min vapour rate, it increases HC and CO emissions by 0.7 g/kW-hr and 2.8 g/kW-hr for HCC piston bowl geometry compared to a conventional engine. With the TCC piston bowl geometry, an increase of 0.41 g/kW-hr in HC and 1.19 g/kW-hr in CO was observed. The Diesel vapour induction at 2 ml provides lesser restriction to the air aspiration. Its combination with TCC piston bowl geometry further minimizes the loss due to hindrance, especially at moderate engine load compared to a conventional engine. Favorable combustion using TCC geometry has shown the drastic reduction in the HC and CO emissions penalty compared to HCC geometry in the PCCI mode at both the fuel vapour induction rate. Compared to a conventional engine at full engine load, very minimal increment in HC and CO emissions were observed in PCCI mode compared to the difference of emissions at low engine load.

5.2.3.2 **NO_x emission and Smoke opacity**

Reduction in NO_x emission is one of the most remarkable characteristics of PCCI and HCCI technology. Figure 5.12 c and d highlights the variation in NO_x and smoke at all engine load Combustion in conventional DI engine is accompanied by high in-cylinder temperature leading to activation of nitrogen molecules that reacts with excess oxygen in air resulting in NOx formation through Zeldovich mechanism (Heywood, 1988). Low temperature combustion as well as low peak cylinder pressure due to the formation of homogeneous charge especially at low engine load decreases NO_x emission by 3% and 5% for 2 ml and 4 ml diesel vapour induction rate with HCC piston geometry. TCC piston bowl geometry encourage high combustion rate followed by increase in cylinder temperature and pressure. However, combustion in toroidal chamber in combination with diesel vapour induction compensates the rise in temperature and pressure thereby keeping NO_x emission lower than conventional DI by 2 g/kW-hr and 3 g/kW-hr at 2 ml and 4 ml/min induction rate.

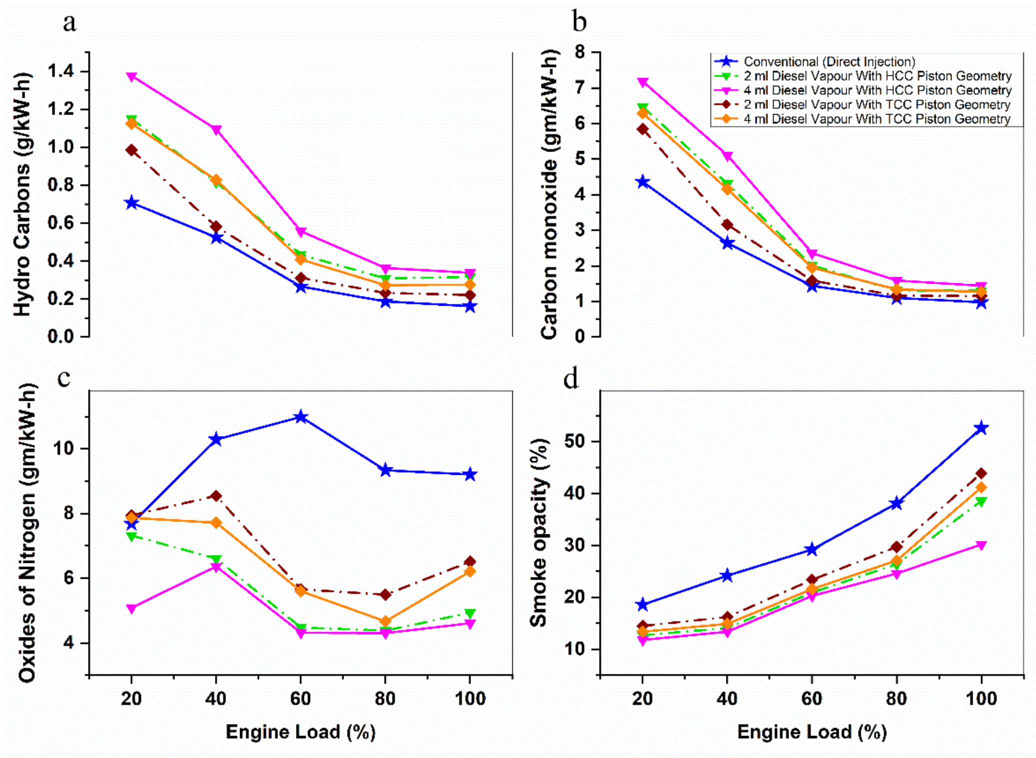


Figure 5.13: Engine emission characteristics a) Hydrocarbon emissions Vs engine load b) carbon monoxide emissions Vs engine load c) Oxides of nitrogen emissions Vs engine load d) Smoke emissions Vs engine load

5.3 Investigation of PCCI engine characteristics with TCC geometry and EGR

The engine was tested with four specified combinations, each defining PCCI operation. Initially, the conventional (DI) was made to run on PCCI mode by external addition of diesel vapor in the intake manifold. The engine was operated with 2 ml and 4 ml fuel vapor with (10%) and without EGR. All the combinations were tested with the toroidal (TCC) piston bowl geometry, and the data were analyzed based on performance, combustion, and emission characteristics and compared with the conventional engine (with TCC piston geometry).

5.3.1 Engine Performance

5.3.1.1 Brake thermal efficiency

The change in BTE with engine load, diesel vapor and EGR is depicted in Figure 5.14. At full engine load, the BTE was decreased by 2.7% and 4.6% with 2 ml and 4 ml diesel vapor without EGR as compared to a conventional engine.

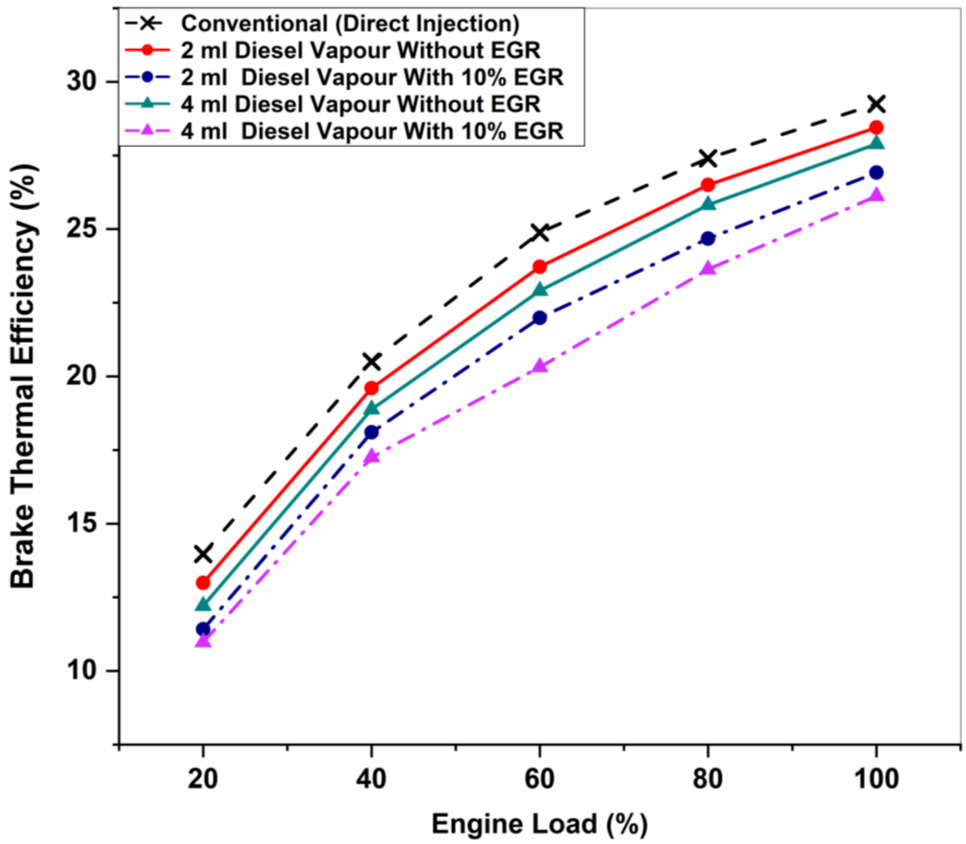


Figure 5.14: Deviation in brake thermal efficiency with engine load

The diesel vapor was externally mixed with the intake air in the intake manifold; the temperature difference between the two fluids (intake air and diesel vapor) may lead to condensation of some quantity of fuel vapors. Hence, a lesser amount of fuel contributes to charge formation that leads to a decrease in BTE with PCCI combustion (D. Ganesh et al., 2008). Higher diesel vapor increases the amount of fuel dilution, thereby decreasing the BTE. At full engine load, BTE was reduced by 8% and 10.7% with 2 ml and 4 ml diesel vapor with EGR. The EGR is a technique to minimize the NOx emission by diluting the oxygen in the charge (Thangaraja & Kannan, 2016). However, external mixing with diesel vapor adds a diluting element that mitigates the required combustion conditions inversely affecting the brake thermal efficiency. TCC geometry has provided better performance characteristics when compared with the conventional (HCC) piston bowl geometry, as seen in the research conducted by Jyothi et al. (Jyothi & Reddy, 2017). The swirl-squish flow of air and fuel is enhanced by the toroidal chamber providing better mixing conditions for improved combustions. Therefore, the use of TCC piston may reduce the percentage reduction in BTE in PCCI combustion.

5.3.1.2 **Exhaust gas temperature**

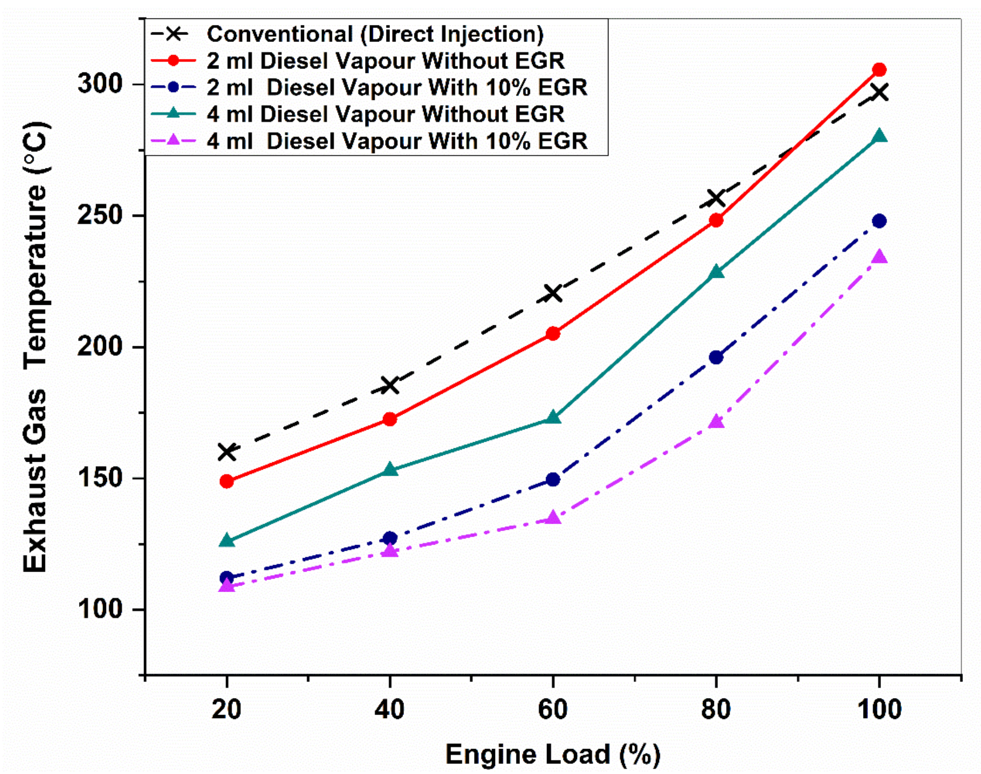


Figure 5.15: Deviation in exhaust gas temperature with engine load

The variation in exhaust gas temperature (EGT) at 20%, 40%, 60%, 80%, and 100% engine load with conventional as well as PCCI combustion as depicted in Figure

5.15 . An increase in load increases the combustion rate that further increases both temperature, as well as pressure and so EGT, elevates with increasing engine load. The maximum EGT of 305 °C was achieved at full engine load when the engine operated in PCCI mode with 2 ml diesel vapor without EGR. This may be due to the combined effect of cylinder temperature and TCC geometry. The toroidal combustion chamber promotes effective combustion that, in turn, favors the EGT (Khan, Panua, & Bose, 2018). At low engine load, when compared with conventional (DI), EGT decreases by 7% and 21% for 2 ml and 4 ml fuel vapor induction without EGR and by 4 ml and 32% for 2 ml and 4 ml diesel vapor respectively with 10% EGR. The primary factor of reduction in EGT in PCCI mode was lean A-F charge results in lower combustion temperature (D. Ganesh & Nagarajan, 2010). Several researchers have also obtained the same effects of decrement in EGT at all the engine loads (A. K. Agarwal et al., 2013) Though the diesel vapor enhances the homogeneity. However, the intake air temperature is responsible for its phase change that prevents contribution in combustion. EGR adds more to the dilution of charge, making it lean (D. Agarwal et al., 2011).

5.3.1.3 **Volumetric efficiency**

The breathing capacity of the engine is one of the characteristics that is inversely affected by the EGR and diesel vapor induction. The variation in volumetric efficiency at all the engine loads is represented in Figure 5.16   It can be seen from the trends that the volumetric efficiency is decreasing with EGR and fuel vapor quantity. In conventional (DI), suction stroke allows only air through the intake manifold while the charge formation occurs at the end of compression stroke during the injection of fuel in the combustion chamber. The entire space in the manifold is occupied by the intake air leading to high volumetric efficiency; however, a very short charge mixing span is available. Due to the shorter mixing span, the intake air is not entirely utilized during combustion, making a significant fraction of air useless. Whereas in the PCCI, the volume occupied by fuel vapor in the intake manifold and additional occupancy due to recirculated exhaust gas obstruct the intake air flow rate in the manifold results in decreasing the volumetric efficiency. However, the pre-mixing initiates with the start of the suction stroke and continues to the start of combustion, providing longer span for homogeneous charge formation, thereby completely utilizing the available air and eliminating excess air unlike conventional (DI) resulting in excess air contributing to high NOx emission. It can be observed that the volumetric efficiency is decreasing by 5% and 16% with 2 ml diesel vapor induction with and without EGR. Therefore, the impact of EGR on decrement in volumetric efficiency is more than the diesel vapor on an engine.

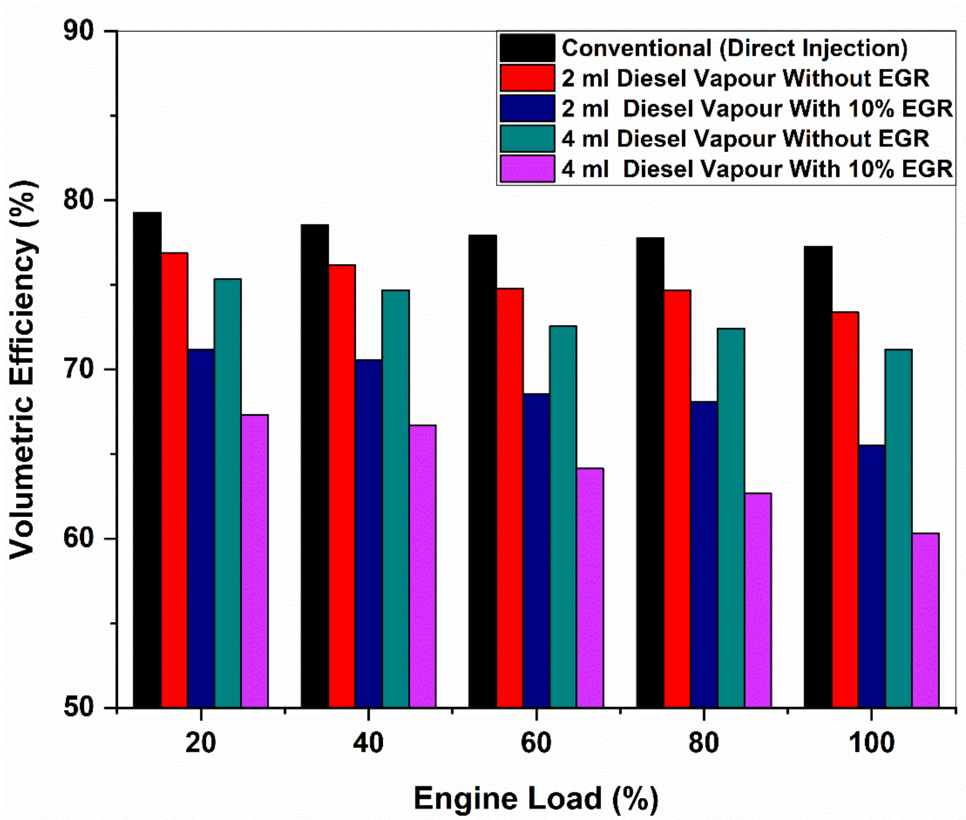


Figure 5.16: Deviation in volumetric efficiency with engine load

This could be another reason for using a low proportion (10%) EGR in this experimental study. The same results of decrement in volumetric efficiency were obtained by a few researchers with the induction of EGR (Ghazikhani, Feyz, & Joharchi, 2010; Nakano, Mandokoro, Kubo, & Yamazaki, 2000). The TCC piston bowl geometry creates turbulence during the intake stroke. The turbulence draws in a higher amount of air in the combustion chamber due to enhanced swirl. Hence, the use of TCC piston bowl geometry helps in minimizing the reduction in volumetric efficiency, especially at moderate engine load.

5.3.2 Engine Combustion characteristics

5.3.2.1 **Pressure vs crank angle (degrees)**

Pressure - crank angle (CA) diagram as shown in Figure 5.17 demonstrate the effect of EGR and diesel vapour induction on the cylinder pressure at full engine load. From the Figure 5.17, the use of 10% EGR has brought down the peak pressure rise compared to conventional DI engine, with the similar result obtained by Morsy et al. (Morsy, 2007). Due to the low proportion of EGR, the combined effect of EGR and diesel vapour has not shown the substantial decrement in in-cylinder pressure.

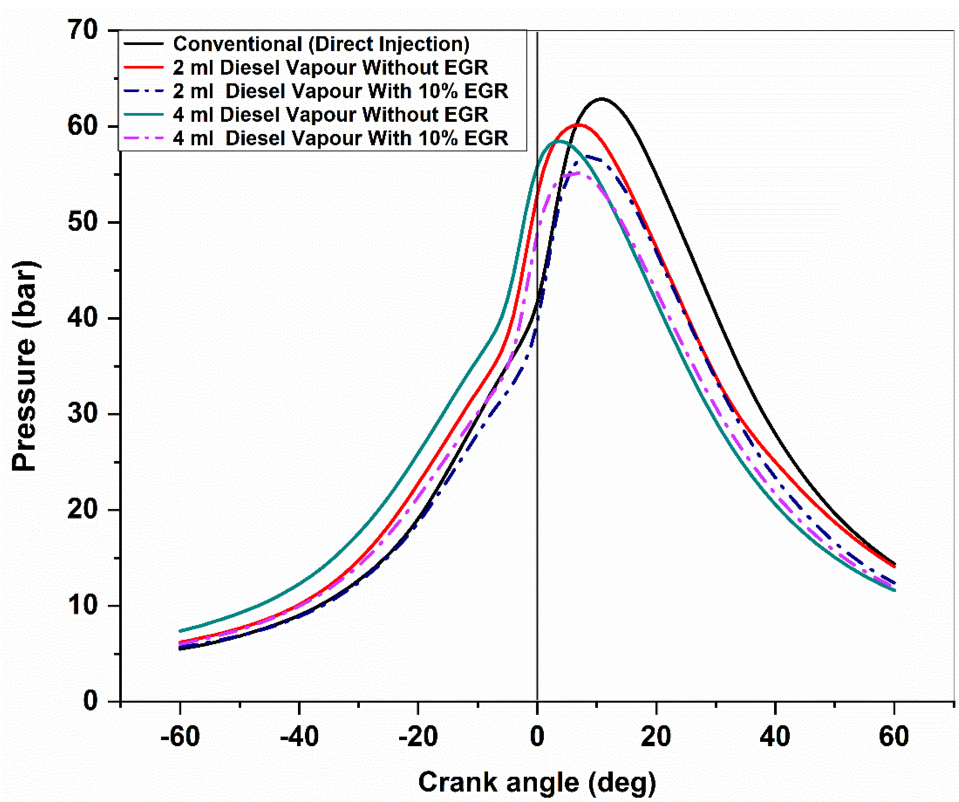


Figure 5.17: Deviation in combustion pressure at various crank angle at full engine load

However, it has shown the decrement of 12% and 10% with 2 ml and 4 ml diesel vapour with EGR as compared to conventional DI. Diesel vapour induction is another critical factor in peak pressure reduction. It has also observed that the increase in diesel vapour quantity, shifting the peak in-cylinder pressure closer to TDC. It shows the higher advancement in combustion with 4 ml diesel vapour quantity. Ganesh et.al had done the same with diesel vapour in HCCI engine and it is observed that the maximum in-cylinder pressure with diesel vapour was obtained exactly at (360°) TDC (D. Ganesh et al., 2008). It shows the better mixture formation in PCCI mode compared to conventional DI. The peak in-cylinder pressure obtained with 4 ml diesel vapor was 58 bar that was the lowest among all the combinations used in the study. The peak pressure decreases by 7% and 11% when 2 ml and 4 ml diesel vapor without EGR. Beyond 4 ml of diesel vapor combustion may tend to deviate towards the rich mixture and it may be resulting in pre-ignition of the A-F mixture which may results in an increase in peak in-cylinder pressure during combustion. Bhiogade et al. had performed the same experiment on a PCCI engine with external mixture formation of diesel vapour. It has found that full vapour induction not allowing the engine to operate on full engine load (G. Bhiogade & Suryawanshi, 2016).

5.3.2.2 **Net heat release rate**

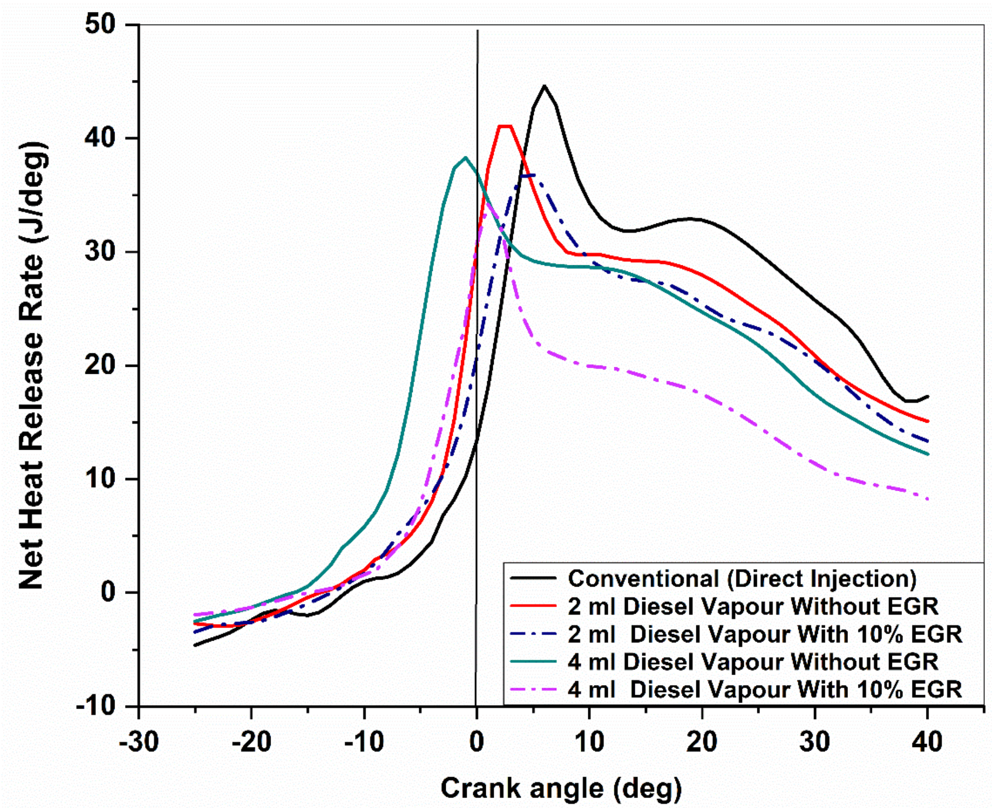


Figure 5.18: Deviation in NHRR at various crank angle at full engine load

Conventional (DI) engine is accompanied by a high heat release rate due to the injection followed by atomization, vaporization and mixing. The net heat release rate (NHRR) is characterized by two curves signifying low and high temperature region (Komninos & Rakopoulos, 2012; Lü et al., 2005). The heterogeneity of diesel causes its accumulation after flame generation followed by rich mixture formation near the injector resulting to high heat release rate at the first curve as represented in Figure 5.18. Diesel fuel induction favours ignition advancement by 5 °CA and 10 °CA for 2 ml and 4 ml diesel vapour without EGR as compared to the conventional (DI). Furthermore, at full engine load the NHRR decreases by 9% and 2 ml for the same. The gas-to-gas diffusion in fuel induction technique eliminates the delay due to heterogeneity while the limited entry of diesel fuel via injection prevents the rich mixture accumulation resulting in a reduction of NHRR.

Excess oxygen dilution due to 10% EGR at full engine load along with 4 ml diesel vapour induction brings down the NHRR by 24%, which is the maximum fall with the peak value almost before the TDC, enabling optimum combustion timing. The excessive ignition advance is limited by the entry of EGR with 4 ml diesel vapour flow unlike the 4 ml induction without EGR that contributing more towards pre-ignition.

5.3.2.3 **Rate of pressure rise**

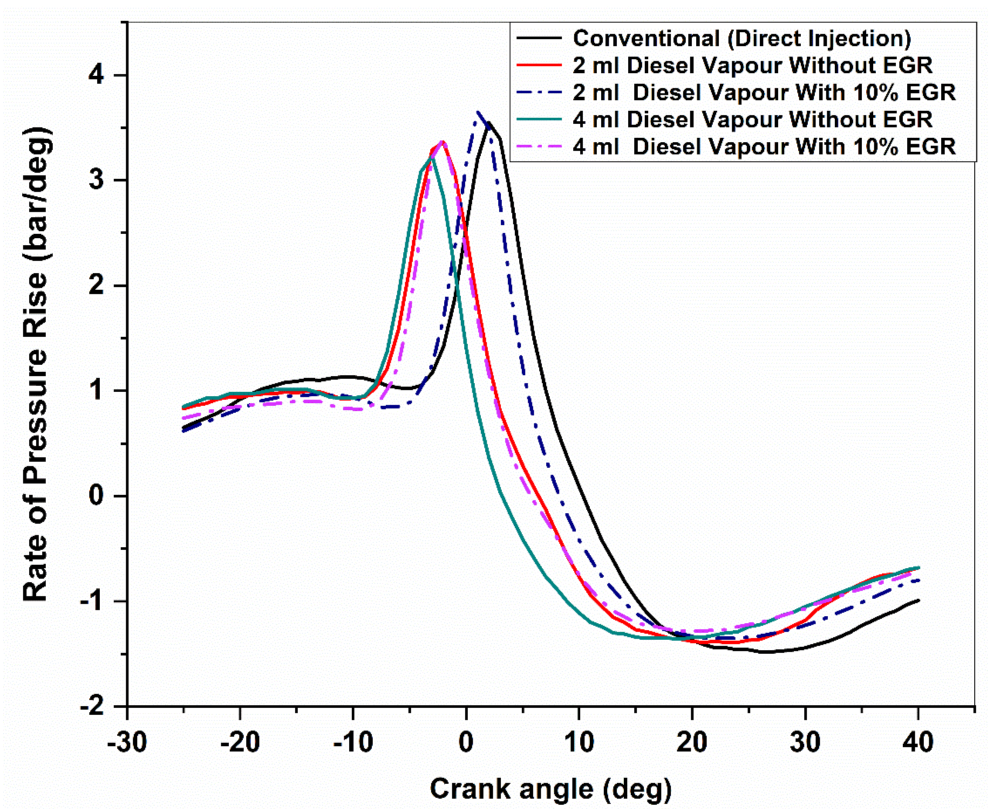


Figure 5.19: Deviation in rate of pressure rise at full engine load

Fluctuations in the rate of pressure rise (RoPR) determine the engine's tendency to knock. Hence, RoPR roughness is a critical factor in engine operation. Figure 5.19 shows RoPR at full engine load for all test runs. The peak value of RoPR for conventional (DI) and 2 ml diesel vapour with EGR is almost equal. The HC in the exhaust gas at full engine load may react with the oxygen in intake air at elevated temperature during compression along with the pre-mixed air-vapour mixture and minor amount diesel injection collectively forming a rich zone creating a condition of pre-ignition resulting in sudden energy release denoted by fluctuation in RoPR. However, diesel vapour induction without EGR eliminates rich mixture conditions as the air is pre-mixed with diesel vapour and excess is utilized by the injected diesel contributing to controlled combustion rate. The RoPR is decreased by 0.2 bar/ °CA and 0.4 bar/°CA for 2 ml and 4 ml fuel vapour without EGR, when compared with conventional DI.

5.3.3 Engine Emissions

5.3.3.1 Hydrocarbon and carbon monoxide emissions

The change in HC and CO emissions at all engine loads is shown in Figure 5.20 and Figure 5.21. Both emissions are bi-product of incomplete combustion and are highest at low engine load as seen in the trends. High CO and HC emission at low engine load is due to lean mixture formation in the conventional engine causing an early occurrence of lean flame blowout region (A. P. Singh & Agarwal, 2012). When compared with conventional (DI) at low engine load, 2 ml diesel vapour induction increases HC emission by 0.28 g/kW-hr and CO emission by 1.49 g/kW-hr while the 4 ml diesel vapour induction increases HC emission by 0.41 g/kW-hr and CO emission by 1.93 g/kW-hr.

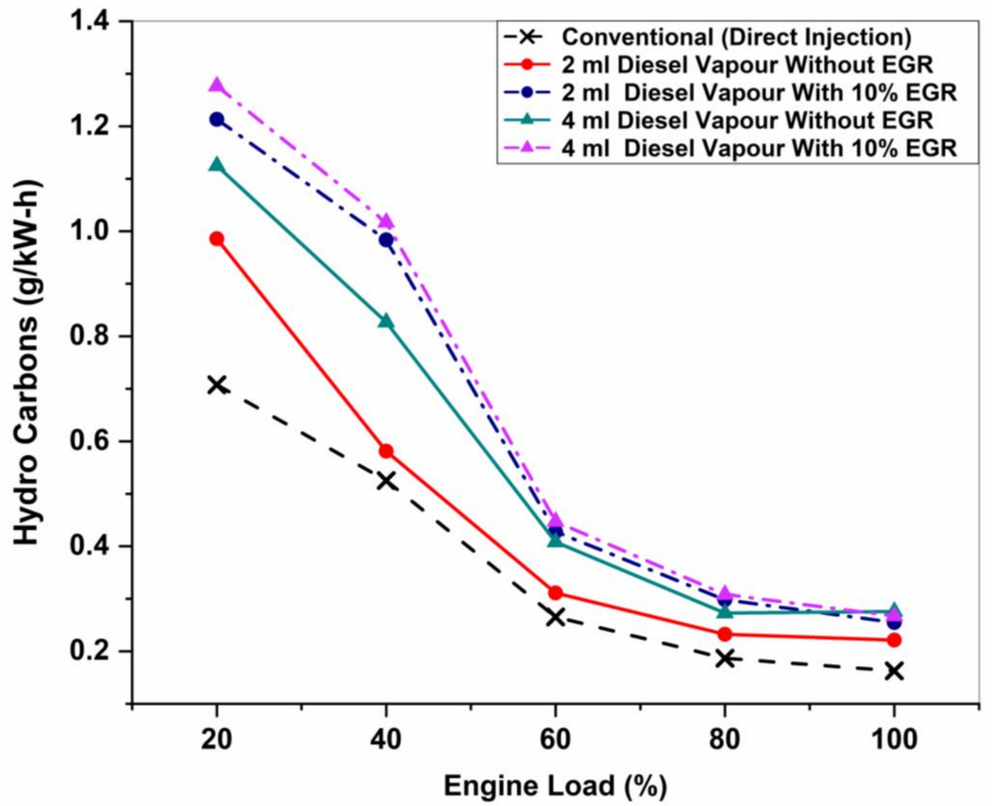


Figure 5.20: Deviation in hydrocarbon emissions with engine load

Diesel vapour may improve homogeneity; however, it limits the volumetric efficiency causing fluctuation of required air-vapour ratio leading to improper combustion. As the engine load increases the air-vapour, ratio matches the required ratio for the optimum combustion and hence, variation in emission tends to decrease at part load, mid load and is least at low engine load. The EGR is yet another factor that cuts off more than required air at low engine load resulting in high fuel vapour quantity against the intake air leading to inappropriate combustion conditions thereby increasing HC and CO emission by 0.5 g/kW-hr and 2.28 g/kW-hr for 2 ml vapour induction and 0.71 g/kW-hr, 2.45 g/kW-hr for 4 ml vapour induction.

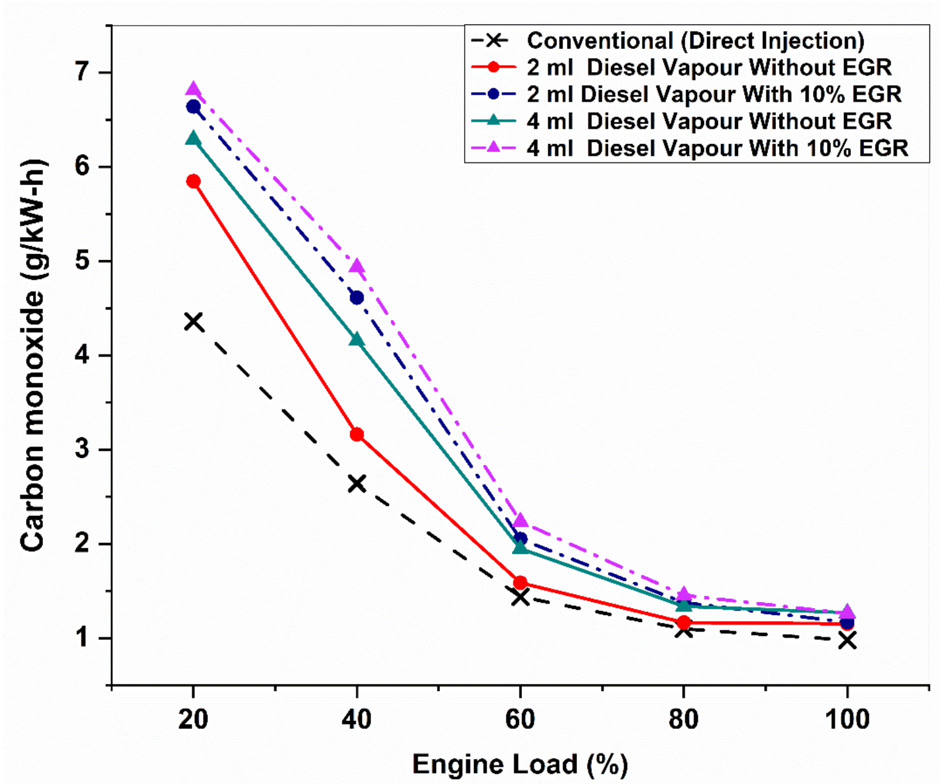


Figure 5.21: Deviation in carbon monoxide emissions with engine load

The TCC piston geometry plays crucial role in minimizing HC and CO emission (S Jaichandar & Annamalai, 2012; S. Jaichandar & Annamalai, 2013) due to its ability to optimize the air-vapour mixing and hence, the variation is brought down to its minimum possible limit especially when 2 ml diesel vapour is induced without EGR. At full engine load the 4 ml, diesel vapour induction attains the highest limit of HC and CO emission. Diesel vapour along with injected fuel (DI) may create the condition of choke that inversely affecting the combustion quality with increased HC and CO output.

5.3.3.2 **NOx emission**

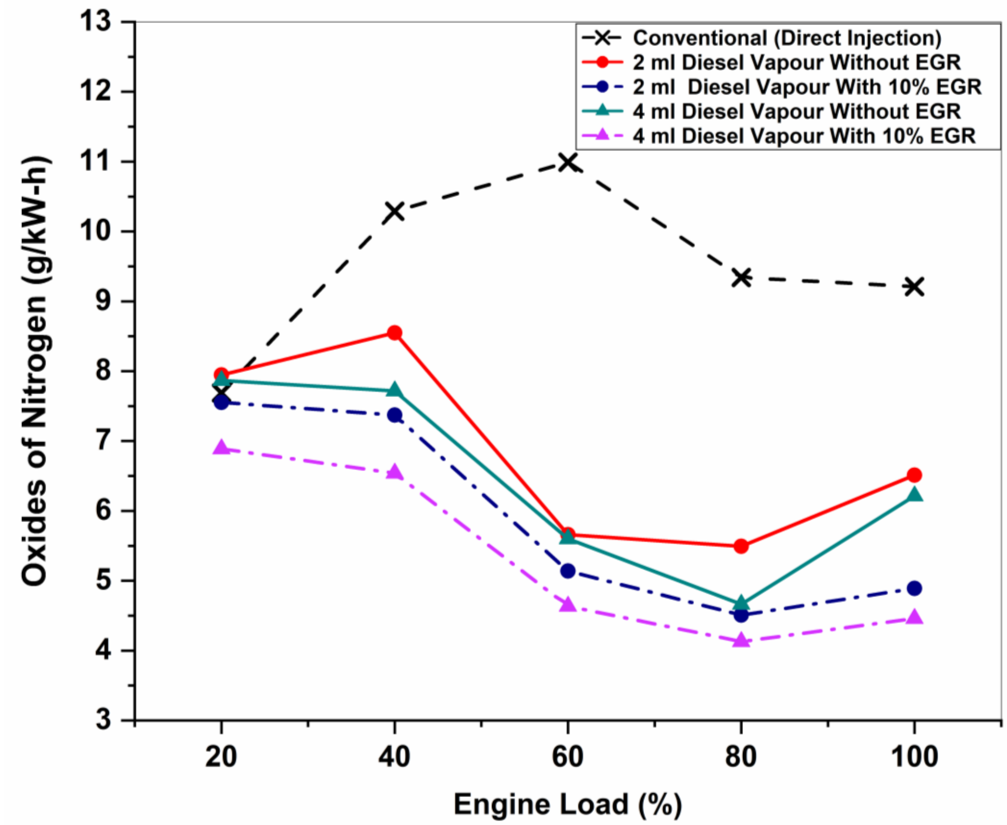


Figure 5.22: Deviation in oxides of nitrogen emissions with engine load

Due to the advantage of similar bulk and local temperature range in PCCI combustion technology, it is considered the best for NOx reduction (G. Singh et al., 2014). As shown in Figure 5.22 minimum NOx emission was achieved at moderate engine load while it is highest at part engine load for all test runs. At low engine load, fuel atomization tends to momentum loss of fuel in the conventional (DI) engine preventing charge formation due to inefficient mixing and hence, a distinct layer of diesel and air is formed generating a rich zone at the juncture of the two layers creating conditions for NO_x formation. However, diesel vapour induction at low load provides homogeneity to the charge unlike the distinct layer formation in the case of conventional (DI). The NO_x emission was reduced by 22% and 23% for

2 ml and 4 ml fuel vapour induction without EGR. EGR along with diesel vapour induction further brings down NO_x emission by 29% and 36% for 2 ml and 4 ml diesel vapour induction and the similar trends were obtained by Bhaskar et al. (Bhaskar & Sendilvelan, 2018). The premixing duration available for air and diesel vapour is larger than the time available in conventional DI, resulting in homogeneous charge formation in PCCI mode, moreover, excess air is cut off due to decrease in volumetric efficiency in the partial PCCI engine. Both conditions minimize the tendency of nitrogen to react due to the decreased number of free oxygen molecules. The excess oxygen in the combustion chamber can be minimized by the use of EGR. The unburned hydrocarbons and carbon monoxide in the exhaust gas react with the oxygen in the intake air making it unavailable for the nitrogen thereby reducing NO_x emission. However, the TCC is an encouraging factor for NO_x emissions (Jyothi & Reddy, 2017) due to the improved NHRR as compared to conventional (DI). Hence, its combination with fuel vapour induction and EGR compensates the increase in NO_x emissions.

5.3.3.3 Smoke opacity

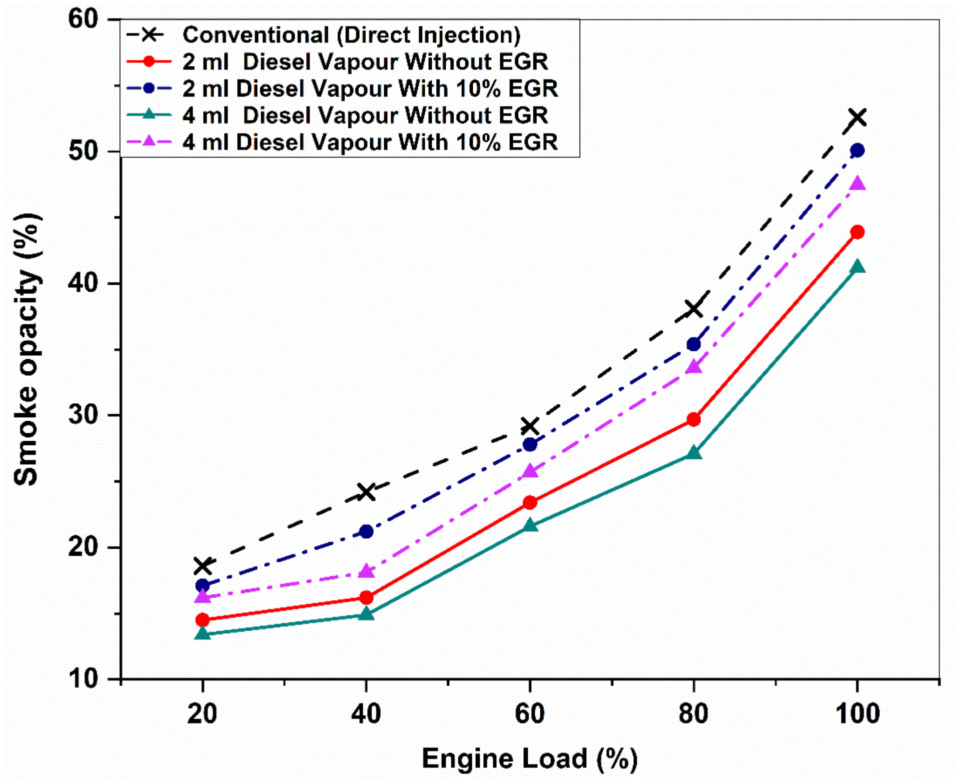


Figure 5.23: Deviation in smoke opacity with engine load

Rich mixture formation and EGR are responsible for an increase in smoke opacity in the exhaust gas (G. Singh et al., 2014). For the opacity to be minimum, the combustion needs to occur at the stoichiometric air-fuel ratio. Variation in smoke opacity at all engine loads is illustrated in Figure 5.23. At the higher engine loads smoke emissions increases due to richer mixture formation. Higher engine load leading accumulation of soot, unburned hydrocarbons and partially burned gases in the exhaust due to insufficient air for ideal combustion contributing to smoke opacity. The smoke opacity, at low engine load, is decreased by 33% and 38% when the engine was operated with 2 ml and 4 ml fuel vapour without EGR. The decrease in opacity, when compared with conventional (DI), is due to pre-mixing of homogeneous air-vapour mixture preventing the formation of rich mixture pockets as in case of conventional (DI) engine. The 10% EGR dilution is not advantageous at part load concerning smoke opacity as HC and CO that tends to make the mixture rich either by re-utilization of UHC or due to loss of required oxygen from the actual A-F mixture is utilizing the required amount of oxygen in the air. However, optimized combustion using TCC piston bowl geometry aids in bringing down the smoke opacity by 13% and 25% at part load for 2 ml and 4 ml diesel vapour induction without EGR.

CHAPTER 6 CONCLUSIONS AND FUTURE WORK

Conclusions

The present research work was conducted on a naturally aspirated single-cylinder, four-stroke water-cooled direct injection diesel engine modified to partially premixed charge compression ignition (PCCI) engine by mounting a fuel vapouriser at the air intake manifold. The combustion chamber was also replaced from hemispherical (HCC) to toroidal (TCC) to enhance the PCCI engine's performance. Experiments were conducted in the following three phases,

Experimental condition-I Assessment of compression ratio of the engine with TCC geometry. Experimental condition-II Investigation of PCCI engine characteristics with HCC and TCC piston geometry. Experimental condition-III Investigation of PCCI engine characteristics with TCC piston geometry and EGR

The following conclusions were made from the various experiments conducted on the conventional and PCCI engine set up.

In experimental condition-I, The effect of toroidal combustion chamber (TCC) geometry on performance, combustion, and emissions of a conventional compression ignition (CI) engine with variable compression ratio (CR) 16:1, 17:1, and 18:1 and exhaust gas recirculation (10% EGR) was investigated. The conclusions derived from the study are encapsulated as below:

- Improved air swirl in the TCC piston enriches the air-fuel mixing and enhances the brake thermal efficiency (BTE) than HCC. The maximum BTE achieved with base-TCC was 33.12%, which was higher by 5.67% than base-HCC.
- Transcend combustion because of enhanced mixture formation in TCC lowers CO and HC emissions compared to HCC piston geometry. However, NO_x emissions increased with an increase in the compression ratio and decreased

with EGR induction. NO_x emissions increased with base-TCC by 10.54% than base-HCC, whereas it decreased by 3.19% with EGR-TCC at 18:1 CR. The increment in NO_x emissions was observed due to an increased maximum rate of pressure rise caused by higher air swirl by the TCC piston bowl as compared to HCC.

- The study revealed that the performance, emissions, and combustion characteristics of the variable compression ratio test rig implemented with EGR could be improved using a suitable combustion chamber geometry and compression ratio. It also understood that the TCC geometry with EGR may also provide better results on the PCCI engine set up.
- **In Experimental condition-II,** The experiments were performed to compare the PCCI engine characteristics mounted with fuel vapouriser for external mixture formation with HCC (default) and TCC (modified) geometry at all the engine loads. The engine characteristics were studied, and the following observations were made
- Diesel PCCI engine has shown high potential for using it in CI engines. It provides the solution for the major concern of the diesel engine, like NO_x and smoke emissions. The low NO_x and smoke emissions were achieved with little compromise in BTE.
- Comparison of 2 ml/min fuel vapour induction with HCC and TCC geometry has shown that TCC has shown 8.7% and 9.5% of decrement in HC and CO emissions with a marginal increment in NO_x by 6.59%. BTE was higher by 5.31%.
- The PCCI mode at 4 ml/min fuel vapour induction compared to a conventional engine, an increase in HC and CO emissions by 0.7g/kW-hr, and 2.8g/kW-hr for HCC piston bowl geometry was observed. With the TCC piston bowl geometry, an increase of 0.41g/kW-hr in HC and 1.19g/kW-hr in CO was noticed. The only advantage of using the higher fuel vapour induction in PCCI mode was the decrement in NO_x emissions compared to the low quantity of fuel vapour.

- Overall, the PCCI mode operation with 2 ml/min fuel vapour induction with TCC geometry can be a replaceable option to HCC piston geometry in the PCCI engine to reduce the penalty in the BTE as compared to a conventional engine.
- **<u>In Experimental condition-III,</u>** The experiments performed to analyze the PCCI engine's engine characteristics with TCC geometry and EGR. The fuel vapour inducted was the same as in experiment condition-II. The conclusions made from the investigation are mentioned below:
- To control the PCCI combustion, cooled EGR was used up to 10%. Delayed combustion was occurred due to the use of EGR. The Combustion occurred near TDC rather than before TDC with low combustion temperature and pressure compared to a conventional engine.
- The PCCI mode with 2 ml/min fuel vapour and 10% EGR has shown the decrement in HC and CO emissions by 4.35 and 7.14%, respectively, with a minute increment in NO_x by 2.36% compared to 2 ml/min vapour induction with HCC geometry. The BTE was higher by 2.23% with TCC geometry.
- Compared to Conventional engines, 2 ml/min TCC without and with EGR has shown 19% and 13% of decrement in smoke opacity with a marginal penalty of 3.76% and 6.6% in BTE. The minimal increment in HC emissions was also observed by 0.4 g/kW-hr and 0.6 g/kW-hr. The decrement in NO_x emissions by 29% and 31%, respectively, was noticed.

Overall, the external mixture formation technique can successfully be implemented on a conventional DI engine with minimal cost and small modification on the existing intake manifold. TCC piston's use reduced the percentage reduction in BTE in PCCI combustion due to improve A-F mixing. Due to its improved geometry, the enhanced swirl motion in TCC helps maintain the BTE as close to the conventional engine.

Future Scope

The present study reveals that the PCCI engine emits very low NO_x and smoke emissions with a marginal penalty in HC and CO emissions, which can be reduced further by using after-treatment technologies like DPF and diesel oxidation catalysts (DOC).

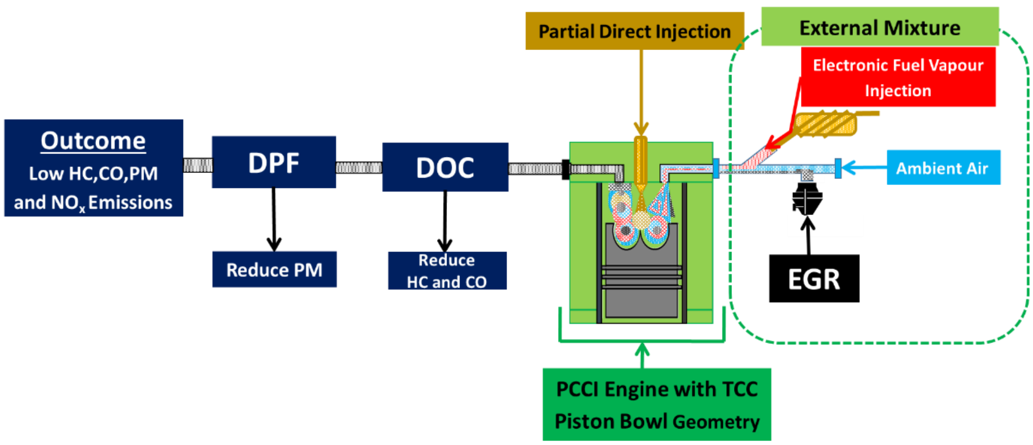


Figure 6.1: Future Scope of this Research to reduce the emissions of HC and CO

- The fuel vaporizer used in the study was mechanically controlled, in future there will be further scope to add electronically controlled fuel vaporizer which can be beneficial to supply desired amount of fuel vapor at any engine loading conditions.
- The Exhaust gas recirculation used in the study was also mechanically controlled, there will be further scope to minimize the NO_x emissions of an engine if EGR will be electronically controlled.
- Use of Exhaust gas to partially heat up the vaporizer so that the external power supply requirement can be reduce as shown in Figure 6.2.

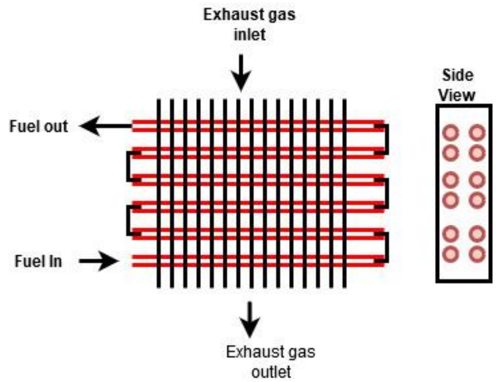


Figure 6.2: Cross Flow Heat Exchanger

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Publications and Patent from PhD

1 Bhurat, Swapnil Sureshchandra, et al. "Technical barriers and their solutions for deployment of HCCI engine technologies–a review." International Journal of Ambient Energy (2019): 1-14.

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5 **Patent:** A Fuel Vaporizer Unit Bhurat, Swapnil Sureshchandra; Pandey, Shyam; Venkat Chintala, Patent No. 201911044038 Published (2019)

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Sources included in the report

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