EXPERIMENTAL INVESTIGATION OF EFFECT OF HYDROGEN AND HIGHER ALCOHOL BLENDS ON ENGINE CHARACTERISTICS OF CRDI DIESEL ENGINE

Thesis

Submitted in partial fulfillment of the requirements for the Degree of

DOCTOR OF PHILOSOPHY

by

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D E C L A R A T I O N

I hereby *declare* that the Research Thesis entitled **"Experimental Investigation of Effect of Hydrogen and Higher Alcohol Blends on Engine Characteristics of CRDI Diesel Engine"** which is being submitted to the **National Institute of Technology Karnataka, Surathkal** in partial fulfilment of the requirements for the award of the Degree of **Doctor of Philosophy** in **Mechanical Engineering** is a *bonafide report of the research work carried out by me*. The material contained in this Research Thesis has not been submitted to any other Universities or Institutes for the award of any degree.

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C E R T I F I C A T E

This is to certify that the Research Thesis entitled **"Experimental Investigation of Effect of Hydrogen and Higher Alcohol Blends on Engine Characteristics of CRDI Diesel Engine"** submitted by **Mr. SANTHOSH K (Register Number: 177035)** as the record of the research work carried out by him, *is accepted as the Research Thesis submission* in partial fulfilment of the requirements for the award of the degree of **Doctor of Philosophy.**

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ABSTRACT

The depletion of fossil fuels had alarmed the researchers to search for new sustainable renewable fuels. Biofuels and hydrogen are the most promising sustainable and renewable alternatives to diesel fuel. In the present renewable energy world, alcohols are gaining prime importance due to their nature of production and fuel properties. The experiment was conducted in a CRDI CI engine test rig at a constant speed of 2000 rpm and the load is varied from 20% to 80% in step of 20%. The main aim of the present investigation is to study the effect of higher alcohol and hydrogen on combustion, performance, and emission characteristics of CRDI CI engine, and also the present investigation aims to reduce the NO_X emission by adopting the EGR and SCR of NO_X techniques.

Initially, a comparative analysis of 1-Pentanol and 1-Hexanol is done on engine characteristics of CRDI CI engine at standard engine conditions to find the best higher alcohol. From the results, it is noted that 1-Hexanol performs well compare to 1-Pentanol and from the results, the 30H70D blend is chosen as an optimized blend for further study. With the use of an optimized 1-Hexanol blend, further improvement in engine characteristics is done by varying the injection time to 9˚BTDC and 15˚BTDC. Compared to standard (12˚BTDC) and late injection (9˚BTDC), advanced injection time (15˚BTDC) showed better combustion, performance, and emission characteristics. Both BTE and BSEC showed only 1.77% and 1.8% reduction at 15˚BTDC compared to pure diesel operation. The optimization of alcohol blend and injection time showed a positive impact on performance and emission. The CO emission was almost equal to pure diesel operation and NO_X emission was still lower compared to diesel fuel operation.

The 1-Hexanol blends showed lagging in combustion and performance compared to pure diesel operation. For further improvement in combustion and performance, enrichment of hydrogen fuel through an inlet manifold is adopted. From the experimental investigation, it was noted that 10 lpm of hydrogen enrichment is optimum and it gives better compatibility with 1-Hexanol blends. The enrichment of hydrogen reduces the ignition delay and improves the poor combustion, performance, and emission characteristics of the 1-Hexanol blend in the engine. The hydrogen enriched 1-Hexanol blend showed an increment of 4.9% of BTE at 80% load compared to 1-Hexanol/diesel alone.

Exhaust gas recirculation is the in-cylinder strategy and SCR of NO_X is an after-treatment technique to reduce the NO_X emissions from the engine. These two methods are adopted in this investigation to reduce the NO_X emission from the engine with the optimized hydrogen-enriched 1-Hexanol blend. Here 10% and 20% EGR rate and Pt-Pd SCR catalyst

are used for the investigation. The 10% EGR rate and Pt-Pd SCR catalyst showed a positive effect on the reduction of NO_X.

The main intention of the present investigation was to study the compatibility of 1- Hexanol in a diesel engine without compromising in performance and emission characteristics of the CI engine. The present investigation was also intended to study the new combustion and emission control technologies, which can be co-developed with alternative fuels. Finding and securing a renewable alternative energy source for diesel fuel and providing a compatible technology to work with alternative fuels in the engine were also conscious efforts. The present investigation was also focused to create huge options in the production and supply of alternative fuels.

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NOMENCLATURE

BDC	Bottom dead centre	HC	Hydrocarbon
BTE	Brake thermal efficiency	H_2O	Water
BSEC	Brake specific energy consumption	HRR	Heat release rate
BSFC	Brake specific fuel consumption	ID	Ignition delay
CA	Crank angle	LHE	Latent heat of evaporation
CD	Combustion duration	LHV	Lower heating value
CI	Compression ignition	MGT	Mean gas temperature
CN	Cetane number	NHR	Net heat release rate
CO	Carbon monoxide	NO	Nitric oxide
CO_2	Carbon dioxide	NO_X	Nitrogen Oxides
CP	Cylinder pressure	O_2	Oxygen
CRDI	Common Rail Direct Injection	ppm	Parts per million
ECU	Electronic control unit	rpm	Revolutions per minute
EGR	Exhaust gas recirculation	TDC	Top dead centre
EGT	Exhaust gas temperature		

CHAPTER 1

1.1. Background

For energy solutions, the entire world depends upon unsustainable and non-renewable fossil fuels. Only a small amount of energy is drawn from a renewable energy source. With the world population explosion, a drastic increase in the consumption of energy for a day-to-day life is seen (EL-Seesy et al. 2019; Nanthagopal et al. 2020). The liquid fossil fuels like petroleum-based fuel reservations are limited and are mainly located in certain parts of the globe. This limitation creates a crisis in non-reserved countries. The use of automobiles in the last two decades for transportation and agriculture purposes is also increased at a rapid rate (Nanthagopal et al. 2020). The efficiency of the diesel engine is more compared to the spark ignition engine because of its better fuel economy and higher compression ratio, which makes the use of the diesel engine increases day by day. The growth of the automotive industry and an increase in the use of commercial vehicles around the world increases the demand for petroleum based fuel. In 2015 it was estimated that more than 335 million commercial vehicles were used around the globe (Mamat 2019). According to the World Energy Outlook (WEO), the demand for the diesel engine in the transportation sector will increases by about 40% by 2040 (Ghadikolaei et al. 2018). The cost of the energy market in the world is about 1.5 trillion dollars, a major portion of which is occupied by fossil fuels (Shafiee and Topal 2009). In the world's total energy demand more than 31% of energy share is occupied by oil and 23% by gas. Petroleum products will be the prime source of the energy mix by 2040 (James Griffin 2019). In total oil demand, the road transport section consumes about 43% of total demand by 2040. Most of the developed and developing countries made a policy to reduce the demand for petroleum oil in the transportation sector. This can be implemented by adopting biofuels, electrical vehicles, and the use of hydrogen. Nuclear and biomass are expected to see demand growth of approximately 7 mboe/d each by 2040 (James Griffin 2019).

According to the Ministry of Petroleum and Natural Gas (Govt. of India), it is estimated that the demand for diesel and petrol will rise from 69.4 MMT and 19.1 MMT (Million Metric Tons) respectively in the year 2014-15 to 110 MMT and 31.1 MMT by the year 2021-22 if the present situation prevails (Ministry of Petroleum and Natural Gas 2017). The transportation sector in India contributes about 3% of the GDP (Gross Domestic Product), about 210 million vehicles are registered in 2015 in India. India is the 3rd largest importer of crude oil in the world. Due to the large requirement for the energy and to reduce the dependency on crude oil India promotes the use of biodiesel in the transportation sector,

production of second-generation ethanol, and conversion of CO_2 (carbon dioxide) into biofuel (Ministry of Petroleum and Natural Gas 2019; Paladugula et al. 2018).

Table 1.1 Causes and consequences of automotive pollutants

Automotive pollutants	Reason	Adverse effect
Hydrocarbon	• Incomplete combustion • Rich fuel/air mixture • Fuel in contact with cold spots in the cylinder	Cause cancer or cell mutations, Difficulty in breathing, Irritates the eyes and throat and bronchi, Hemotoxic, neurotoxic, and cancer effect
Carbon monoxide	• Incomplete combustion • Lack of oxygen	Miscarriage, Coma, Heart failure & Headache
Nitrogen oxide (NO and NO_2)	• High cylinder temperature • Oxygen concentration • Residence time.	Genetic mutations, decrease infemale fertility, Swelling of the throat, Spasms, Rapid pulse & Dilated heart
Particulate matter	• Incomplete combustion • Rich fuel/air mixture	Nonfatal heart attacks, Aggravated asthma, coughing, or difficulty breathing & Premature death in people with heart or lung disease
Sulphur oxides (SO_2, SO_4, and H_2SO_4)	• Burring of Sulphur in diesel fuel and lubricating oil. • Some part of SO_2 in the exhaust reacts with oxygen and forms SO_3.	Lung infections, Respiratory tract infections

The exhaust emission from the burning of fossil fuels in automobiles causes air pollution, global warming, acid rain, smog, and odours. The increase in toxic pollution in the environment causes adverse effects on human health and the environment. NO_X in the presence of UV radiation produces photochemical smog and reduces visibility. NO_X also reacts with moisture and raindrops to form nitric acid (HNO_3) aerosols that lead to acid rain and it also acts as an indirect greenhouse gas. For a human being, exposure to NO_X leads to respiratory disease lowers the resistance to influenza and bronchitis and prolonged exposure causes lung cancer (Kampa and Castanas 2008). Around 10% of the aggregate national greenhouse gas emission is contributed by India's transportation segment and 87% of the aggregate emissions in the sector (Paladugula et al. 2018). The reason for the

formation of automotive pollution and its adverse effect is shown in Table 1.1. Due to the depletion of petroleum based fuel and adverse effects on the environment from emissions, the lawmakers made strict rules to mitigate these issues. The policymakers and automotive companies are continuously trying to reduce the harmful emissions from the combustion of fossil fuels in the engines (Mamat 2019). All these issues with the use of petroleum based fuels had made researchers think about sustainable alternative fuels.

1.2 Alternative fuels to diesel

The above statistics and data confirm that there is a huge demand for petroleum products in the transportation sector and also the demand will be doubled in the future leads to rise in the price of crude oil. This all needs a promising alternative to fossil fuel to meet the demand. Biofuels are the most promising alternative renewable source of energy. The major biofuels are biodiesel and bio-alcohols, the scientist tremendously explored the use of biodiesel in the compression ignition engine, which gave them a big success; many countries permitted to use of up to 20% of biodiesel with diesel (Ministry of Petroleum and Natural Gas 2017).

For the production of biofuels, the only organic carbon available on the earth is biomass, which effectively substitutes petroleum based fuels and chemicals (Serrano-ruiz et al. 2012). The biofuels are mainly produced from various biomasses like agriculture products, agriculture waste, animal manure, tall oil pitch, crude glycerine, municipal and industrial wastes. It was noted that with the use of biofuels more than 60% of the carbon emission was reduced (Kumar et al. 2016). The use of biofuels effectively reduces the demand for petroleum based fuel, greenhouse gas, and the cost of energy (Çelebi 2019).

The main feedstock for the production of biodiesel is agricultural products, the biodiesel market in the future may get expanded, and most of the biodiesel production depends on agricultural products. The use of biodiesel is not a permanent or better solution to the existing problem. The feedstock required to produce biodiesel are may compete with food crops. the increase in dependency on biodiesel and biodiesel production affects the world agriculture commodity market and it will be a major threat to food security (Koizumi 2015). The production of biodiesel has both risks and opportunities. Hence, most of the researchers divert their concentration to explore the better fuel properties of alcohols. Alcohols are produced from bio-waste, which will not alter the food chain supply. The main ingredient of alcohol is lignocellulosic material which can be obtained from agricultural waste (Sarathy et al. 2014).

The alcohols are of two types, lower alcohols, and higher alcohols. If the number of carbon

atoms is less than four are considered as a lower carbon chain alcohols and carbon atoms above four are considered as a higher carbon chain alcohols (B. Ashok et al. 2019; De Poures et al. 2017). An extensive study had already been done with the use of lower- carbon chain alcohols and concluded that they are not suitable to use in CI engines due to lower CN, lower LHV, higher LHE, and also the presence of higher oxygen content and water molecules had an adverse effect on combustion (Campos-Fernández et al. 2012; Suhaimi, Adam, Anes G. Mrwan, et al. 2018; Yilmaz 2012). However, an increase in the number of carbon molecules in the structure improves the fuel properties which makes them suitable to use in the CI engine (B Ashok et al. 2019; Atmanli and Yilmaz 2020). Compared to lower alcohol, higher alcohols had higher CN, LHV, lower LHE, and less hygroscopic nature, and were less corrosive (Rajesh Kumar, Saravanan, Rana, and Nagendran 2016). Due to these improved fuel properties, many researchers and biotechnology companies are interested to produce higher alcohols on large scale ("BP/DuPont launches Butamax biobutanol" 2009, "Gevo resumes commercial production of isobutanol at its Luverne plant" 2013; Lp and Sm 2012).

1.3 Higher alcohols as an alternative fuel

The term 'higher alcohol' usually refers to the series of straight chain alcohols containing four or more carbon atoms, viz. butanol (C_4), pentanol (C_5), hexanol (C_6), octanol (C_8), dodecanol (C_{12}), phytol (C_{20}), etc. In recent years higher carbon chain alcohols are more popular due to their better blend stability, fuel properties, and hydroxyl molecules in the structure help in complete burring compared to lower carbon chain alcohols and less hygroscopic nature (B. Ashok et al. 2019; De Poures et al. 2017; Rajesh Kumar and Saravanan 2016). With an increase in the number of carbon atoms in the chain, its CN, LHV, density, and viscosity get increases, while its solubility, and oxygen (O_2) content gets decreases (De Poures et al. 2017).

The excellent blending stability of higher alcohol with diesel enables it to inject directly into the combustion chamber of a diesel engine (Li et al. 2017). The spray characteristics of the diesel fuel could be improved with the addition of higher alcohols which increases the spray cone angles, increasing the puffing intensity, shortening spray tip penetration and spray area increased. Further, the local turbulence and ambient gas entrainment are enhanced. The R-OH group in the pentanol is more efficient in the reduction of soot formation (Chen et al. 2019a; Li et al. 2017).

Table 1.2 Comparative properties of alcohols (Rajesh Kumar and Saravanan 2016)

Properties	Diesel	Methanol	Ethanol	Propanol	Butanol	Pentanol	Hexanol	Octanol	Decanol
Molecular formula	C_xH_y	CH_3-OH	C_2H_5-OH	C_3H_7-OH	C_4H_9-OH	C_5H_{11}-OH	C_6H_{13}-OH	C_8H_{17}-OH	C_9H_{19}-OH
Molecular weight (Kg/kmlo)	190-211.7	32.04	46.07	60.09	74.12	88.15	102.18	130.23	158.28
C (%wt)	86.13	37.48	52.14	59.96	64.82	68.13	70.52	73.72	68.23
H (%wt)	13.87	12.48	13.02	13.31	13.49	13.61	13.70	13.82	12.64
O (%wt)	0	49.93	34.73	26.62	21.59	18.15	15.70	12.29	10.11
Solubility (g/L)	Immiscible	Miscible	Miscible	Miscible	77	22	7.9	4.6	3.7
Cetane number	52	5	8	12	17	18.2-20	23	39	50
Self-ignition temperature(°C)	254-300	463	420	350	345	300	285	270	255
Density (kg/m^3) at15°C	835	791.3	789.4	803.7	809.7	814.8	821.8	827	830
Viscosity at 40°C (mm/s^2)	2.72	0.58	1.13	1.74	2.22	2.89	5.32	-	6.5
Lower heating value (MJ/Kg)	42.49	19.58	26.83	30.63	33.09	34.65	39.10	52.94	-
Latent heat of evaporation (kJ/kg)	270-375	1162.64	918.42	727.88	581.4	308.05	486	-	-
Vapor pressure (mmHg)	0.4	127	55	20	7	6	1	0.08	<0.1
CFPP b (°C)	180-360	64.7	78.3	97.1	117.5	137.9	157	195	233
Boiling point(°C)	180-360	64.7	78.3	97.1	117.5	137.9	157	195	233
Flash point (°C)	> 55	11-12	17	11.7	35-37	49	59	81	108

The comparative properties of (physical and chemical) some higher and lower alcohols with diesel fuel are shown in Table 1.2. Compared to lower alcohol, higher alcohol has greater potential to replace diesel fuel fully or partially. Less hygroscopic nature, high carbon content, and low polarity of higher alcohol make them to blend with diesel fuel without phase separation. Higher alcohols do not require any co-solvent or emulsifying agent to maintain the blend stability. The lower water content of higher alcohols is less corrosive and an increase in length of carbon chain and absence of branched chain improves the fuel properties of higher alcohol and less tendency to knock. The lower oxygen content of higher alcohol improves the calorific value. The storing and handling of higher alcohols are easy with the existing distribution infrastructure due to higher flashpoint. Compare to lower carbon chain alcohol, higher carbon chain alcohols consume less energy during their production since the breaking down of larger molecules can stop earlier (Rajesh Kumar and Saravanan 2016).

Earlier the use of the higher alcohols was limited to only for the food industry due to its high cost and the production of higher alcohols only from the petroleum source. Annually the production of 1-pentanol, 1-hexanol, 1-heptanol, and 1-octanol are reached one million pounds by chemical synthesis (Marcheschi et al. 2012). Because these higher alcohols are not accessible through the biosynthesis pathway. Later several researchers expose the fuel properties of the higher alcohol and conclude that it may be a better alternative to petroleum fuels. With the effort of many researchers finally, the production of higher alcohols from non-petroleum resources like various non-food biomasses, including agricultural residuals, forest wood, and marine algae was found (Pan et al. 2018a). Recently in the last 10 years, many research organizations and biotechnology companies are very interested in the use of higher alcohol as a sustainable vehicle fuel and they are trying to increase the yield of higher alcohol from cellulose. Higher alcohols are artificially produced from genetically engineered micro-organisms like Clostridium species, Escherichiacoli micro-organisms, Cyanobacteria, and Saccharomyces cerevisiae are used to produce the butanol, pentanol, hexanol, etc., from cellulose (Desai et al. 2015; Formighieri 2015; Higashide et al. 2011; Ofuonye et al. 2013).

The carbohydrates and lipids are the major substrates for the production of alcohols, whereas Huo et al. use protein biomass to produce alcohols and they succeed in the production of higher alcohols (C_4-C_5) by using genetically engineered E. coli bacteria to generate protein hydrolysates, which enables the cells to convert proteins into butanol and pentanol at 56% of the theoretical yield (Huo et al. 2011). The syngas produced from coal or biomass is also used as a substrate to produce alcohols. The synthesis gas contains about

25-35% of H_2 (Hydrogen), 40-65% of CO (carbon monoxide), 1-20% of CO_2, 0-7% of CH_4 (methane), and a small trace of sulphur gas. Only CO, CO_2, and H_2 in the syngas are used to produce alcohols, the anaerobic bacteria called bacterium C. ljungdahlii was engineered to produce alcohols (Phillips et al. 1993).

Dekishima et al. conducted research to produce 1-Hexanol from 1-Butanol by extending the 1-butanol production pathway. Initially, 1-Butanol was produced from glucose by various chemical reactions; in later steps, the Escherichia coli strain was engineered to synthesize 1-Hexanol. After 48h of a chemical reaction, a yield of 47 mg/l is found (Dekishima et al. 2011). Marcheschi et al. adopted a +1 pathway for the carbon chain of 2-ketoacids to produce higher alcohols from glucose by using a combination of quantum mechanical modeling, protein−substrate modeling, protein and metabolic engineering. For enzyme mutation H97A, S139G, G462D, a yield of 302 mg/l of 1-Hexanol is produced (Marcheschi et al. 2012). Higher alcohols are produced not only form biomass but also from syngas and carbon dioxide can be used to produce. Phillips et al. develop a Clostridium carboxidivorans organism to produce higher alcohol from syngas (CO:H_2:CO_2 [70:20:10]) at a low production cost. They develop butanol, hexanol, and ethanol. The yield of both butanol and hexanol is 1 g/l and ethanol yield is 3 g/l (Phillips et al. 2015). With natural metabolic reactions, it is not possible to produce non-natural alcohols like pentanol, hexanol, decanol, etc. Zhang et al. conducted an experiment to produce non-natural alcohols by expanding the metabolism. Escherichia coli organism was engineered to develop higher alcohol by using glucose as a substrate, which gives a yield of 18.5 mg/l of 1-Hexanol. The production of higher alcohols through the biosynthesis method and its yield are shown in Table 1.3.

Table 1.3 Production of 1-Hexanol and 1-Pentanol with their yield

Higher alcohol	Engineered Host	Source	Yield	Reference
1-Hexanol	*Escherichia coli*	Glucose	47 mg/l	(Dekishima et al. 2011)
1-Hexanol	*Escherichia coli*	Glucose	302 mg/l	(Marcheschi et al. 2012)
1-Hexanol	*Clostridium carboxidivorans*	Syngas	1 g/l	(Phillips et al. 2015)
1-Hexanol	*Escherichia coli*	Glucose	18.5 mg/l	(Zhang et al. 2008)
1-Pentanol	*Escherichia coli*	Glucose	204.7 mg/l	(Zhang et al. 2008)
1-Pentanol	*Escherichia coli*	Glucose	2.22 g/l	(Dekishima et al. 2011)

1.4 Hydrogen as an alternative fuel

The clean burning and better performance of hydrogen make as one of the most promising alternative fuel. Solar energy can be indirectly converted into hydrogen by using biomass-based renewable sources. Also, hydrogen can be produced by naturally available sources like uranium, coal, and oil shale. Gasification of coal and electrolysis of water is another commercial way of producing hydrogen. Solar photo-electrolysis and thermochemical decomposition of water are the other methods used to produce hydrogen. Whereas, these methods are still are under research and currently not available commercially. Present hydrogen fuelled vehicle technology is not more sophisticated to compete with gasoline or higher alcohol fuelled vehicles (Saravanan and Nagarajan 2008). In SI engines, hydrogen can be directly used as a fuel and the ignition of H_2 can be done by using a spark plug. Whereas it is not possible to use H_2 as the sole fuel in a CI engine due to its high self-ignition temperature of about 858 K and absence of a spark plug. However, the hydrogen can be used in dual fuel mode in a CI engine (Saravanan et al. 2008; Saravanan and Nagarajan 2008).

In a dual fuel mode, the intake of hydrogen can be done in three ways: direct injection of hydrogen into the combustion chamber, continuous injection of hydrogen into the intake manifold, and timing injection of hydrogen into the intake manifold. All these three injection methods have their own advantages and disadvantages. The hydrogen injection into the intake manifold leads to a reduction in intake air due to a higher density of hydrogen thereby reducing the volumetric efficiency of the engine and also reducing the power output due to poor combustion. Compared to timed injection of hydrogen, the continuous injection causes backfiring, and the engine operating range is limited. In the case of direct hydrogen injection, a dedicated hydrogen injector is required this can withstand high in-cylinder pressure and temperature (Karagöz et al. 2016). Hydrogen is free from carbon and sulphur which do not produce particulate matter, carbon dioxide, hydrocarbon, or carbon monoxide (Saravanan et al. 2008; Talibi et al. 2014). Hydrogen in a dual fuel mode produces higher combustion efficiency and performance compared to diesel only operation. This can be attributed that the higher flame propagation of hydrogen which shortens the combustion duration and reduces the loss of heat to the cylinder wall (Talibi et al. 2014). Due to higher in-cylinder temperature, the hydrogen- diesel fuel operated engines experience higher NO_X emissions. The presence of nitrogen in air and fuel reacts with oxygen at high in-cylinder temperatures and produces NO_X emission (Saravanan et al. 2008). The high diffusive nature of hydrogen helps in the perfect mix with the air in the intake manifold and inducted into the engine cylinder. This reduces the heterogeneous mixture with diesel fuel spray in the

cylinder, making a more uniform fuel-air mixture (J. H. Zhou et al. 2016). At higher loading conditions the addition of more than 20% hydrogen energy share leads to the release of the maximum amount of heat before TDC, which leads to abnormal combustion and misfire. This would result in unstable combustion and the engine may get shut down automatically. To avoid all these adverse effects on engine performance some technical measures have to be taken. This can be done by diluting the intake charge by adding the inert (CO_2, N_2) gas or EGR and water injection, or to reduce the combustion chamber temperature hydrogen stream will be mixed with the water vapours (J. H. Zhou et al. 2016).

The adoption of a dual fuel mode of operation is easy for the existing engine which requires a minor modification in the fuel supply system. Here hydrogen is supplied to the intake manifold where it mixes with the intake air which acts as secondary fuel. Diesel is the main fuel that is directly injected into the combustion chamber during the compression stroke BTDC, diesel fuel acts as an ignition source for hydrogen fuel.

1.5 NO$_X$ controlling techniques

The most common emissions from C.I. engines are hydrocarbon, particulate matter, CO, and NO$_X$. When NO$_X$ chemically reacts with HC under sunlight, ground-level ozone is formed. The ground level ozone may cause several health issues like asthma, reduction in lung function, and difficulty in respiration (Heck and Farrauto 2001). Around 10% of the aggregate national greenhouse gas emission is contributed by India's transportation segment (Paladugula et al. 2018). The most unfavourable emission from the CI engine is NO$_X$ and particulate matter, which limit the application of the CI engine due to more strict emission norms. To control these pollutants from the CI engine more stringent rules are implemented by the government.
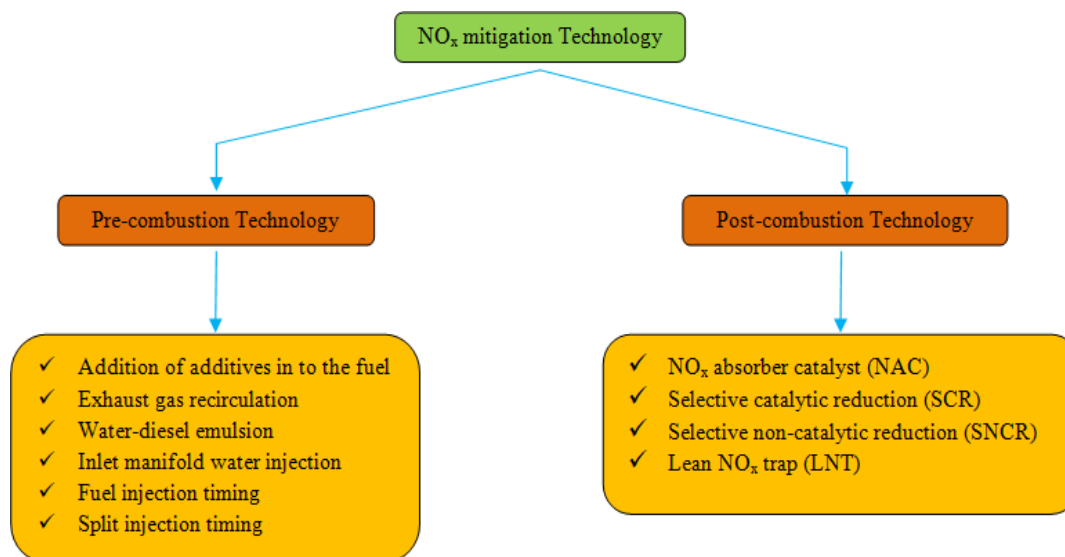
Fig.1.1 NO$_X$ mitigation technology

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Emission from the engine can be controlled by various combustion techniques and/or by exhaust treatment. The pictorial presentation of NO_X controlling techniques is shown in Fig. 1.1.

Many researchers are continuously working to mitigate the NO_X from the lean combustion engine in the exhaust to N_2. The simultaneous controlling of both NO_X and PM is a challenge, NO_X forms due to higher cylinder temperature and PM forms due to poor combustion. They developed several in-cylinder methods such as water–diesel emulsion, exhaust gas recirculation (EGR), inlet manifold water injection, fuel injection timing, and split injection techniques have been studied (Qi 2017; Subramanian 2011; Taghavifar et al. 2017; Thangaraja and Kannan 2016). Akihama et al. are the first research group to address the PM- NO_X dilemma in CI engines. In their research work, they recommended using exhaust gas recirculation to reduce both NO_X and PM (Akihama et al. 2001).

The use of only these in-cylinder strategies cannot meet the required NO_X reduction efficiency without compromising the engine performance, hence the several researchers working on the various after-treatment methods. Several after-treatment technologies like SCR, selective non-catalytic reduction (SNCR), lean NO_X trap (LNT), and diesel oxidation catalyst (DOC) have been developed. Compared to all these after-treatment technologies, SCR of NO_X is the promising De NO_X technology (Praveena and Martin 2017).

1.5.1 Selective catalytic reduction of NO_X

A catalytic converter contains a catalyst, which lowers the energy required for the reaction called activation energy and increases the reaction rate, whereas, the catalyst is not consumed during the reaction. The NO_X can be decomposed at a temperature below 900°C in the presence of a catalyst; the catalyst lowers the activation energy required for the reaction (Guan et al. 2014). During the catalytic reaction process, the catalyst reduces the NO_X and oxidizes the CO and HC.

$$2CO + O_2 \rightarrow CO_2 \qquad\qquad 1.1$$

$$C_xH_y + O_2 \rightarrow CO_2 + H_2O \qquad\qquad 1.2$$

$$2NO \rightarrow N_2 + O_2 \qquad\qquad 1.3$$

Equations 1.1 and 1.2 are referred to as oxidation reactions and equation 1.3 is a reduction reaction. In both equations 1.1 and 1.2, the carbon and hydrocarbon are being oxidized and the molecular oxygen is being reduced. With the introduction of reducing agents like ammonia, hydrocarbon, or H_2, the reaction rates can be increased (Liu and Ihl Woo 2006). The NO_X can be decomposed at a temperature below 900 °C in the presence of a catalyst; the catalyst lowers the activation energy required for the reaction (Liu and Ihl Woo 2006)

and also with the introduction of the reducing agents like ammonia (NH_3), hydrocarbon or hydrogen, the reaction rates can be increased (Dasireddy and Likozar 2017). The type of active catalytic material used in the SCR plays an important role in the NO_X reduction efficiency like platinum (Pt), palladium (Pd) (Goula et al. 2016), Rhodium (Rh) (Ayodhya, Lamani, Thirumoorthy, et al. 2018), Gold, silver (More et al. 2016), however, limited literature is documented on bimetallic Platinum/palladium coated on aluminium oxide (Al_2O_3).

1.5.2 Exhaust gas recirculation (EGR)

Diesel fuel vehicles are the major source of NO_X and particulate emissions, which destroy the quality of ambient air. Many countries like China, Denmark, Paris, and India made a new policy to regulate the NO_X emission from diesel vehicles. Recently in India, the government banned the registration of diesel passenger cars of capacity 2000 CC and above in Delhi. The simultaneous reduction of NO_X and PM in a diesel engine through in- cylinder technology is a challenge. The exhaust gas recirculation strategy is an effective technology to control NO_X from a diesel engine (Thangaraja and Kannan 2016).

The pictorial representation of the effect of EGR on combustion and emission of CI engine is shown in Fig. 1.2. The exhaust gas recirculation is used to reduce the in-cylinder temperature by introducing high specific heat inert gases. Researchers found that the use of up to 20% EGR is optimum for low NO_X emission with minimum possible smoke, CO, and HC (Damodharan, A. P. Sathiyagnanam, et al. 2018; Mani et al. 2010). In EGR the desired percentage of exhaust gas is re-circulated back into the cylinder.

The replacement of fresh air with burnt gas reduces the oxygen supplies, which deteriorates the combustion inside the cylinder. This leads to a reduction in in-cylinder temperature and eliminates the suitable environment to form the NO_X emission. The reduction in NO_X emission is directly proportional to the percentage of EGR applied. Exhaust gas recirculation had three different effects on combustion in diesel engine i.e., Thermal effect, chemical effect, and dilution effect. In thermal effect, the EGR increases the specific heat capacity of the intake charge. The presence of CO_2 and H_2O increases the temperature of the charge. In chemical effect, the presence of CO_2 and H_2O in exhaust gas dissociates during combustion and absorbs the heat from the cylinder. In case of the dilution effect, the EGR replaces the considerable amount of O_2 in the intake charge makes the unavailability of oxygen during combustion (Damodharan, A. P. Sathiyagnanam, et al. 2018; Hountalas et al. 2008). The use of EGR affects the oxygen concentration in the cylinder which changes the structure of the flame, which leads to increases in combustion duration, and also the

addition of EGR increases the ignition delay. The reduction in flame temperature is an important factor in reducing the NO emission (Agrawal et al. 2004).
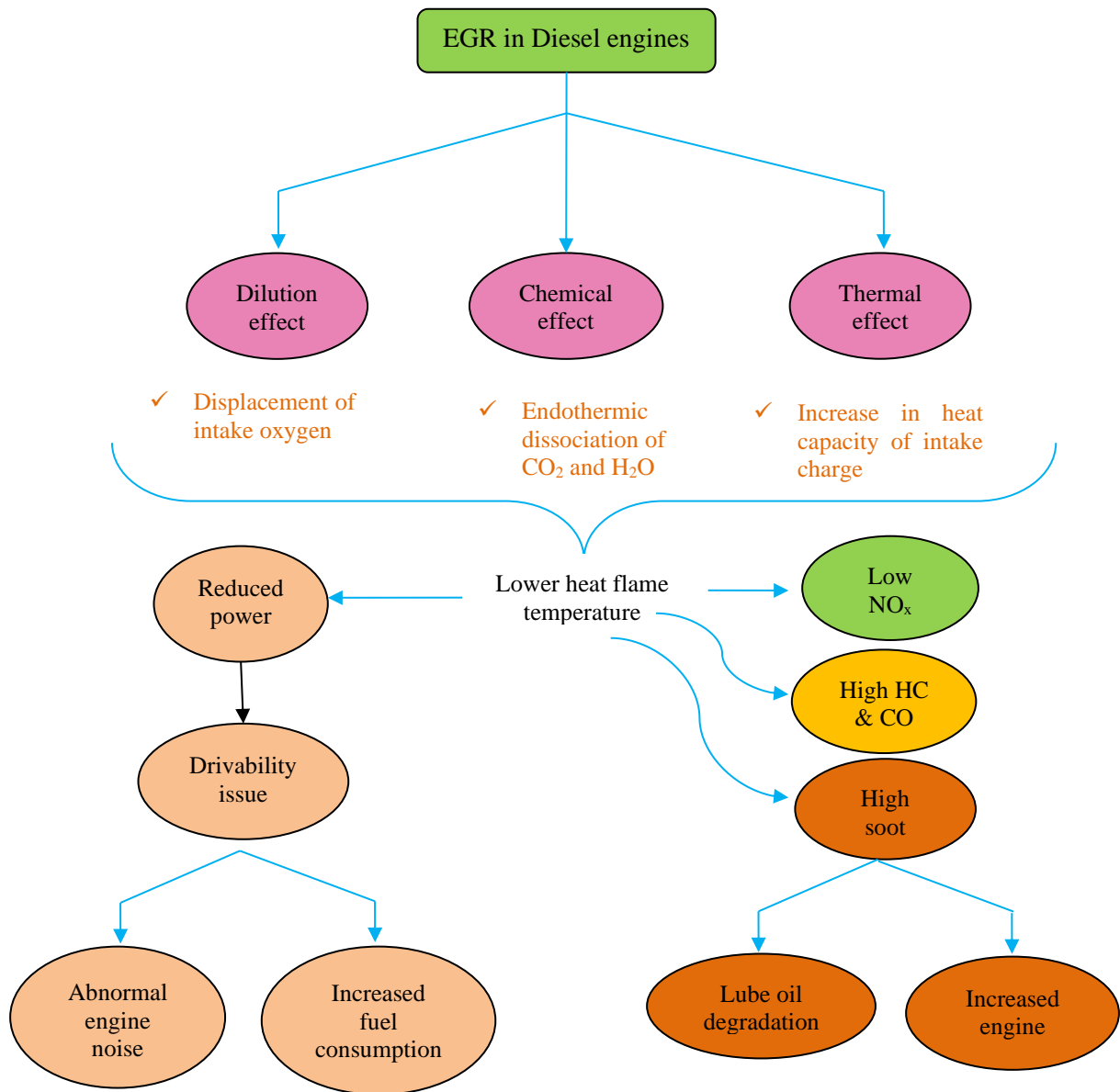
Fig. 1.2 The effect of EGR on combustion (Thangaraja and Kannan 2016)

1.6 Thesis outline

The main aim of the present investigation is to find the effect of hydrogen and higher alcohol blends on combustion performance and emission characteristics of the CRDI CI diesel engine. The thesis has been organized into five chapters.

The first chapter is about the introduction to the present investigation. In this chapter background of the present investigation and the available alternative to diesel fuel is discussed. Also, the feasibility of hydrogen and higher alcohols as an alternative to diesel fuel is discussed. Along with alternatives to diesel fuel, the different NO_X controlling technologies for diesel emissions are also discussed.

The second chapter is about a summary of the literature survey related to the present investigation. This chapter consists of the use of higher alcohol in diesel engine as an additive and alternative to diesel fuel. The use of hydrogen in CI engine and its effect on engine characteristics and finally the NO_X controlling technologies like EGR and SCR of NO_X are presented. Based on the literature research gap is found and the objectives of the present investigation are drawn.

In the third chapter development of the experimental setup and methodology to conduct the experiment are discussed. It includes the development of hydrogen setup, the development of SCR setup, and the instruments used in the experiment are discussed. Here also discussed the preparation of the blends and properties of the fuels used in the experiment.

The results of the present investigation are discussed in the fourth chapter. In this chapter, the effect of higher alcohol like 1-Pentanol and 1-Hexanolon on engine characteristics and the effect of hydrogen in a diesel engine are discussed. Finally, the effect of EGR and SCR of NO_X on emission is discussed in this chapter.

The conclusions are drawn from the present investigation and the scope of the future work is discussed in the fifth chapter. The references, list of publications, bio-data and Appendix are provided after Chapter 5.

CHAPTER 2

LITERATURE SURVEY

In this chapter, a broad literature is done to show the compatibility of higher alcohol- diesel blends and hydrogen in a dual fuel mode in a compressed ignition engine. Proper techniques to improve the combustion, performance, and emission characteristics of CI engine are discussed and it includes the following key points

- The use of higher alcohol-diesel blends in CI engine
- The use of higher alcohol-diesel blends at various injection time
- The use of exhaust gas recirculation in CI engine
- The use of hydrogen in dual fuel mode in CI engine
- The use of selective catalytic reduction of NO_X in CI engine

2.1 Use of higher alcohol-diesel blends in CI engine

2.1.1 1-Hexanol-diesel blends in CI engine

Limited studies are available on the use 1-Hexanol/diesel blends in CI engine and some other studies used 1-Hexanol as an additive or surfactant.

Sundar Raj et al. made the first attempt to use 1-Hexanol as a fuel in diesel engine and they studied the effect of 1-Hexanol on engine characteristics. They found that the use of 1-Hexanol in a diesel engine increases the ID period and NO_X emission was observed, while smoke density was reduced. The higher concentration of 1-Hexanol in the blends reduces the BTE and increases the BSEC (Sundar Raj and Saravanan 2011).

De Poures et al. investigated the effect of 1-Hexanol/diesel blend (10%, 20%, and 30% v/v) on engine characteristics of CI engine under the influence of EGR and injection timing. The presence of 1-Hexanol in the blend increases the ID, CP, and HRR and it increases with a higher fraction of 1-Hexanol in the blend. This is due to lower CN which increases the ignition delay. The addition of EGR further reduces the in-cylinder temperature, HRR and prolongs the ID. At higher load conditions, higher NO_X was noted which increases with an increase in the Hexanol content. The retardation of injection timing and application of EGR gradually reduces the NO_X emission. Whereas, HC emission increases with an increase in the fraction of 1-Hexanol at all engine loads. Retardation of the injection angle affects negatively on combustion characteristics and positively on NO_X emission (De Poures et al. 2017).

2.1.2 1-Hexanol as an additive/co-solvent in CI engine

In most of the cases, the researchers use 1-Hexanol in lower concentration as an additive/surfactant with ethanol/diesel blend, and biodiesel/diesel blends to improve the phase stability of the blend.

Babu et al. investigated to find the effect of n-Pentanol and n-Hexanol as an additive to biodiesel/diesel blend on engine characteristics of diesel engine. The concentration of both the alcohols varied from 5 to 10% v/v. Compared to other ternary blends the addition of a small amount (5%) of n-Hexanol enhances the BTE and lowers the BSFC. An increase in the concentration of n-Hexanol in the blend to 10% v/v leads to a higher heat release rate due to a prolonged delay period allowing more fuel to accumulate in the chamber. Longer combustion duration was noted for both 5% v/v n-Pentanol and n- Hexanol blend. Lower CO and HC emissions were noted for the 10% n-Pentanol blend compared to other blends in the test due to higher oxygen content. Compared to n- Pentanol, n-Hexanol showed the highest NO_X emission at full load due to higher heating value and CN, which is lower compared to diesel fuel operation (Babu and Anand 2017). Pandian et al. examined the effect of neat cashew nut shell oil on engine characteristics by doping 1-Hexanol (10% and 20% by volume) as an additive. They found that the addition of 1-Hexanol reduces HC, CO, smoke, and BSFC and marginally increases the BTE with increases in 1-Hexanol concentration in the blend. The doping of 1-Hexanol to cashew nut shell oil enhances the oxygen content, reduces viscosity improves the atomization (Pandian et al. 2018). Damodharan et al. et al. use n-Pentanol, n-Hexanol, and n-Octanol as an additive to the waste plastic oil to find its effect on engine characteristics of CI engine. The engine was run with 70% waste plastic oil and 30% higher alcohol. The effect of injection time (21˚, 23˚ & 25˚CA bTDC) and EGR rate (10%, 20% & 30%) were also tested. Lower NO_X emission was noted for late injection with a 30% EGR rate due to lower cylinder temperature. Higher NO_X, higher cylinder pressure, and HRR rate were noted for n-Pentanol followed by n-Hexanol and n-Octanol due to lower CN and longer ignition delay. The lower EGR rate, advanced injection time, and addition of alcohol lower the formation of smoke. The lowest smoke production was noted for n-Pentanol due to higher oxygen concentration followed by n-Hexanol and n-Octanol. Higher BTE and lower BSFC were noted for lower EGR and advanced injection time. Higher BTE was found for n-Pentanol addition due to lower viscosity followed by n-Hexanol and n-octanol. Overall, they concluded that 70% of waste plastic oil-30% of n-Pentanol injected at 21˚CA bTDC with 10% EGR have optimum emission and performance (Damodharan, A.P. Sathiyagnanam, et al. 2018).

Abdullah et al. investigated the effect of Hexanol/biodiesel/diesel blend on combustion and emission characteristics of CI engine. The concentration of 1-Hexanol was varied as 5%, 10%, and 15% by volume. At all engine loads, lower CP was noted for all 1-Hexanol blends due to prolonged combustion. At 50% and 100% load, HRR was increased with an increase in the concentration of Hexanol in the blend compared to diesel fuel. Higher CO and lower NOx emissions were noted at all engine loads due to lower in-cylinder temperature and higher LHE. However, 15% of 1-Hexanol produces lower CO emissions compared to diesel fuel (Abdullah et al. 2019).

Swaminathan et al. used 1-Hexanol and 1-Octanol as secondary oil for neem and wintergreen oil blends to know the effect on engine characteristics of diesel engine. During the experiment, the alcohols were injected into the combustion through inlet air in the 10%, 20%, and 30% ratio by mass. The addition of alcohol reduces the HC, CO, and smoke emissions and increases the NO_X emission due to reduced viscosity, better atomization, and higher oxygen concentration of higher alcohols compared to neat neem and neem-wintergreen oil. The lower fraction of alcohol improves the performance characteristics of the engine. The addition of alcohol improves the combustion parameters like CP and HRR and also reduces the combustion duration (Swaminathan et al. 2019).

Ashok et al. studied the effect of 1-Hexanol and Decanol as an additive for Calophyllum Inophyllum biodiesel on engine characteristics of diesel engine. The concentration of 1-Hexanol and Decanol was varied from 30% to 40% v/v, biodiesel was varied from 10% to 20% v/v and the concentration of diesel was fixed, it was 50% v/v. The addition of higher alcohol performs well at all engine loads, which increases the CP, HRR, BTE and reduces the smoke, and NO_X emission compared to pure biodiesel and diesel/biodiesel blend. Compared to the 1-Hexanol blend Decanol performs well and had higher BTE and lower BSEC. Lower CO, HC, NO_X, and smoke emissions were noted for 40% Hexanol share, compared to other ternary blends (B. Ashok et al. 2019).

Ramesh et al. conducted an experimental investigation to find the effect of Calophyllum Inophyllum methyl ester/Hexanol/diesel blends on engine characteristics. The peak CP and HRR increase for a higher fraction of 1-Hexanol in the blend and its peak pressure shifts towards the right due to lower LHV and CN. Higher BTE and lower BSFC were noted for the 40% of 1-Hexanol blend compared to diesel/biodiesel blend. The addition of 1-Hexanol reduces the CO, smoke, and HC emission at all loads. Overall, the addition of 1-Hexanol (40%) had a positive effect on HRR, CP, BTE, BSFC, CO, and HC compared to biodiesel/diesel blends. However, the NO_X emission for a higher concentration of 1-Hexanol was higher compared to biodiesel and nearly equal to diesel fuel (Ramesh et al.

2019).

2.1.3 1-Pentanol-diesel blends in CI engine

Wei et al. conducted an experimental investigation to know the effect of 1-Pentanol/diesel blends on engine characteristics of diesel engine. The fraction of 1-Pentanol varied from 10% to 30%. The addition of 1-Pentanol increases the delay period which adversely affects the engine characteristics. An increase in CP, HRR, BSFC, and decrease in BTE was noted. The poor combustion characteristics of 1-Pentanol increase HC and CO. The prolonged delay allows more fuel to accumulate in the combustion chamber resulting in a higher heat release rate which may be the main cause for the increment in the NO_X emission. (Wei et al. 2014).

Kumar et al. experimented to know the effect of 1-Pentanol and EGR (10%, 20%, and 30%) on the performance and emission characteristics of the diesel engine. The concentration of Pentanol in diesel was varied as 10% 20% 30% and 45% v/v. The higher concentration of both 1-Pentanol and EGR adversely affects on performance and emission characteristics of the engine. Lower BTE, NO_X, and higher BSEC, HC, and CO were noted with 1-Pentanol compared to diesel fuel. The smoke opacity increases for all the blends beyond 20% EGR rate and also the increase in HC and CO emission was noted with an increase in the EGR rate (Rajesh Kumar and Saravanan 2015).

Yilmaz et al. studied the effect of Pentanol/diesel blends (up to 35% v/v) on the performance and exhaust emission of a diesel engine. The addition of Pentanol to diesel blends deteriorates the combustion due to low CN and higher LHE. At all engine loads, the value of BSFC increased with the addition of Pentanol and the BTE also lower BTE was noted for all Pentanol blends compared to diesel fuel. Poor performance characteristics was noted for all Pentanol fraction compared to diesel. All the Pentanol blends exhibit higher CO and HC emissions compared to diesel fuel. However, only 5% and 35% of Pentanol blends show a reduction in NO_X emission compared to diesel fuel (Yilmaz and Atmanli 2017).

Chen et al. experimentally analyzed the effect of Pentanol on combustion and emission characteristics of CI engine. The fuel used in the test was aviation kerosene rocket propellant 3 (RP-3), which was blended with Pentanol. The Pentanol shares 20% and 40% by volume, reaming was RP-3. The 20% of Pentanol blend showed nearly equal CP and HRR at all loading conditions compared to RP-3. The higher concentration of Pentanol deteriorates the combustion due to the low CN. The ID of 1-Pentanol was reduced at advanced injection time. Higher NOx and HC emission and lower soot emission was noted

with the addition of Pentanol compared to RP-3 (Chen et al. 2017).

Pan et al. investigated the effect of Pentanol/diesel blends on spray and evaporation characteristics of blended fuel and engine characteristics of DI diesel engine. Pentanol concentration of 25% and 50% was used in the test. The spray liquid column for both the Pentanol blends was short and the front end of the column was less misty compared to diesel fuel. The spray penetration of both Pentanol blends was short compared to diesel due to lower energy density, which decreases the kinetic energy of the oil column and the low viscosity of Pentanol improves the fuel droplet breakage. The spray cone angle of the injected fuel increase with the addition of Pentanol compared to diesel fuel, which further increases with an increase in the concentration of Pentanol in the blend. At low and medium loads the CP and HRR rate for both the Pentanol blends were higher compared to diesel fuel. Lower BTE and higher BSFC values were noted for Pentanol blends compared to diesel fuel. Lower Smoke, lower CO, higher HC, and higher NO_X emission was noted with the addition of Pentanol compared to diesel fuel (Pan, Huang, Liao, Jia, Zhou, Huang, et al. 2019).

Chen et al. examined the effect of diesel/n-Pentanol/methanol blends on engine characteristics of a CRDI diesel engine. The addition of alcohol increases the ignition delay and heat release rate and shorter combustion duration compared to neat diesel. Higher ignition delay and HRR were noted for methanol blends, it increased with a higher concentration of methanol in the blend. The formation of NO_X for all the alcohol blends was higher compared to neat diesel, among the blends higher NO_X was noted for the methanol blend. Advancing the injection angle increases the formation of NO_X whereas, retardation reduces the NO_X formation. Also, these blends exhibit lower soot (Chen et al. 2019b).

2.1.4 1-Pentanol as an additive/co-solvent in CI engine

Jianxin et al. used 1-Pentanol as an additive in a diesel/biodiesel blend to examine the engine characteristics of the CI engine. The addition of 1-Pentanol in the diesel/biodiesel blend improves the spray and combustion characteristics. Lower NO_X, soot, HC, and higher CO emission was noted for 1-Pentanol blends (Li, Jianxin, et al. 2015).

Zhu et al. added Pentanol as an additive to waste cooking oil biodiesel to find its effects on combustion and emission in a diesel engine. Higher cylinder pressure and heat release rate were noted for Pentanol blends compared to biodiesel and it was increased with a higher fraction of Pentanol in the blend. Both BTE and BSFC of Pentanol blends are higher than neat biodiesel and diesel fuel. Lower NO_X emission was noted for 10% and 20% Pentanol

fraction and for 30% Pentanol fraction an increase in the NO_X emission was noted compared to diesel and biodiesel fuel. Both HC and CO emissions were higher for Pentanol blends compared to diesel and biodiesel fuel (Zhu et al. 2016).

Atmanli et al. added Propanol, n-butanol, or 1-Pentanol to the diesel/waste cooking oil biodiesel blend, to know the effect on engine characteristics of CI engine. The fraction of each alcohol in the blend was 20% by volume separately. Butanol and Pentanol blends had slightly lower BSFC and the propanol blend showed higher BSFC compared to the diesel/biodiesel blend. Similarly, the Butanol and Pentanol blend showed improved BTE and the propanol blend showed lower BTE compared to the diesel/biodiesel blend. The addition of butanol and Pentanol reduces the HC emission, whereas, propanol addition increases the HC emission compared to the diesel/biodiesel blend. All alcohol blends in the test showed higher CO and lower NO_X as compared to diesel/biodiesel blends (Atmanli 2016).

Yilmaz et al. added 1-Pentanol as an additive to waste oil methyl ester to check its effect on the performance and emission characteristics of the CI engine. 1-Pentanol was added to both diesel and biodiesel separately by adding 10% and 20% Pentanol to create D90P10, D80P20, B90P10, and B80P20 blends. The addition of Pentanol increases the fuel consumption and reduces the thermal efficiency, the poor performance of 1-Pentanol was due to lower heating value, and higher latent heat of evaporation. It was noted that compared to pure biodiesel operation, the addition of Pentanol showed little enhancement in BTE. The addition of 1-Pentanol adversely affects the HC and CO emission which increases with a higher fraction of Pentanol in the blends. Also, higher blends showed higher NO_X emissions (Yilmaz et al. 2017).

Yilmaz et al. investigated the effect of waste oil biodiesel, diesel-waste oil biodiesel (D/B), and diesel-waste oil-soybean oil (D-B-VO) on engine characteristics in a CI engine. Also, they used 10% of propanol and Pentanol as an additive with the D-B-VO blend. The higher BSFC and lower BTE were noted for all the blends compared to diesel fuel. Compared to the propanol blend, the addition of Pentanol showed reduced fuel consumption and improved BTE. The lower CN and heating value of propanol increases the fuel consumption and ignition delay compared to Pentanol. The addition of alcohol increases the exhaust gas temperature, propanol blends showed higher EGT than Pentanol blends. Higher CO and HC emissions were noted with the addition of alcohol, which was higher than the diesel/biodiesel blend. However, lower NO_X was noted for both the alcohol bend compared to diesel fuel (Yilmaz et al. 2018).

Ashok et al. added n-Pentanol as an additive to Calophyllum Inophyllum biodiesel to know

the effect on engine characteristics. The volume of n-Pentanol in the blend varied from 10 to 50% in step of 10%. The addition of alcohol to biodiesel improves the cylinder pressure and heat release rate compared to pure diesel operation. Alcohol blends showed a positive effect till 30% of Pentanol concentration in the blend. Longer delay period was experienced with alcohol blends compared to pure biodiesel and diesel blends. A longer delay period was proportional to the concentration of 1-Pentanol in the blend. Compared to pure biodiesel operation, the addition of Pentanol showed improvement in both BTE and BSFC. Among all the alcohol blends, 10% Pentanol concentration showed higher BTE and lower BSFC. Higher HC and CO emission were noted for Pentanol blends compared to neat biodiesel operation, whereas, lower NO_X emission was noted with the addition of alcohol compared to pure biodiesel (B Ashok et al. 2019).

2.1.5 Comparative analysis of 1-Hexanol and 1-Pentanol with other alcohols in CI engine

Fernández et al. investigated the effect of 1-butanol/diesel and 1-Pentanol/diesel blends on performance of DI diesel engine. The concentration of butanol and Pentanol in the blend was varied as 10%, to 25% v/v. All butanol blends except 25% and 30% butanol showed lower consumption of fuel at higher brake mean effective pressure compared to diesel fuel. Pentanol also shows similar consumption of fuel compared to diesel fuel. They noted that 15% of butanol and Pentanol shows comparatively better results compared to diesel. Both Pentanol and butanol blends showed higher BTE compared to diesel fuel operation (Campos-Fernández et al. 2012).

Kumar et al. conducted an experimental investigation to know the effect of iso-butanol, n-Pentanol, n-Hexanol, and n-octanol on combustion and emission characteristics of diesel engine. Each higher alcohol shares 30% by volume with diesel fuel. Compared to diesel and other higher alcohols, iso-butanol showed a longer ID period, higher HRR, and higher cylinder pressure due to lower CN. The shortest combustion duration was noted for iso-butanol compared to other fuels in the test. A higher fraction of smoke and CO emission was noted for diesel followed by octanol, Hexanol Pentanol, and iso-butanol. A decrease in the concentration of oxygen in the fuel structure increases the smoke and CO formation. Higher HC emission was noted for iso-butanol followed by Pentanol, Hexanol, and octanol. Lower NO_X emission was noted for all alcohol blends due (Rajesh Kumar, Saravanan, Rana, and Nagendran 2016).

Yi et al. examine the effect of n-Pentanol or n-butanol/diesel blends on spray characteristics in a constant volume chamber. The concentration of n-Pentanol and n- butanol was varied

from 10% to 40% v/v in diesel fuel. The fuel was injected with a single hole fuel injector by varying the injection pressure as 60 MPa, 80 MPa, and 100 MPa, and also the ambient pressure was varied as 1 MPa and 2 MPa. Compared to other blends in the test, diesel fuel had longer spray tip penetration the higher density of diesel allows to have larger initial axial momentum, which helps to overcome the resistance of ambient air compared to blends. Higher injection pressure increases the spray tip penetration compared to lower injection pressure. The increase in ambient pressure from 1 MPa to 2 MPa which offers more resistance thereby reducing the axial momentum of the fuel leads to decreases in spray tip penetration. The increase in the ambient pressure increases the spray cone angle of the blended fuels. At all injection pressure, n-butanol blends showed a larger spray cone angle compared to n-Pentanol blends due to lower kinematic viscosity of n-butanol compared to n-Pentanol. An increase in the injection pressure reduces the gap between the spray cone angles caused by different ambient pressure. The spray cone angle of blended fuels was higher than diesel fuel. The spray area of the blended fuels was higher than diesel fuel, also it was noted the spray area of Pentanol blended fuels was slightly higher than butanol blended fuels. The peak tip velocity increases with a higher fraction of alcohols in the blend and it increases with injection pressure and decreases with ambient pressure. The peak tip velocity for the Pentanol blend was higher than the butanol blend (Li et al. 2017).

Atmanli et al. conducted a comparative analysis of n-butanol and 1-Pentanol blends on the performance and emission characteristics of CI engine. The variation of percentage of both alcohols in bends was varied as 5%, 25%, and 35% by volume. Alcohol blends had lower BTE and higher BSFC compared to neat diesel fuel operation at all engine loads. Compared to n-Pentanol, n-butanol blends had higher BTE and lower BSFC. The exhaust gas temperature of both the alcohol blends are higher compared to diesel, the 1-Pentanol blends had higher EGT compared to n-butanol blends. Lower NO_X, higher HC, and CO emission was noted for all alcohol blends. Compared to n-butanol blends, n-Pentanol blends experience higher NO_X emissions and HC emissions (Atmanli and Yilmaz 2018a). De Poures et al. (2019) conducted a comparative analysis of n-Pentanol and n-Hexanol blends on engine characteristics of CI engine. The ratio of both n-Pentanol and n-Hexanol varied to 30% in diesel. The effect injection time (21˚CA, 23˚CA, and 25˚CA) and EGR (10%, 20% and30%) were studied. n-Pentanol/diesel blends showed a higher CP and HRR rate compared to the n-Hexanol/diesel blend. The longer ignition delay was also experienced for n-Pentanol blends compared n-Hexanol and neat diesel. The delayed injection time and EGR reduces the peak pressure and HRR for all the fuel blends. The BSFC of both the alcohol blends was higher compared to diesel fuel operation too, the addition of EGR

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further increases the fuel consumption. Fuel consumption of n-Pentanol was higher compared to the n-Hexanol blend. However, it was noticed that the BTE of n- Pentanol was higher compared to the n-Hexanol blend. Irrespective of the EGR rate n- Pentanol showed higher BTE compared to n-Hexanol at all injection times. The advanced injection time improves the combustion and thereby increases the BTE and reduces the BSFC of all alcohol blends. Lower NO_X emission was noted for both alcohol blends, whereas, higher NO_X formation was noted for n-Pentanol blends compared to n-Hexanol blends. The delaying the injection time and application of EGR further reduces the NO_X emission compared to neat diesel. HC emission of all the alcohol blends was higher than diesel fuel. n-Pentanol blends showed higher HC emissions compared to n-Hexanol and diesel. The increase in EGR further increases the HC emission at all injection times. Lower CO and smoke emission was noted for all alcohol blends compared to diesel fuel. The advancing injection time further reduces the CO emission. The CO emission for Pentanol was lesser than Hexanol followed by diesel (De Poures et al. 2019)

2.2 Use of hydrogen in CI engine

The use of hydrogen in CI engine in dual fuel mode is not a new concept and varieties of research were already done to know the combustion, performance, and emission characteristics of CI engine in dual fuel mode. However, still, the use of hydrogen in CI is under investigation due to contradictory results from researchers. Some of the recent and important research is discussed here.

Saravanan et al. (2008) investigated the effect of hydrogen in a dual fuel mode on engine characteristics of diesel engine. The hydrogen flow rate was varied from 10% to 90% by volume (l/min). Higher BTE was obtained at 30% of H_2 enrichment and lower SEC was obtained for 90% of H_2. Performance of the engine increases with higher energy share of the hydrogen, whereas it was limited to lower hydrogen energy share due to the knocking effect. NO_X emission from the engine increases up to 30% hydrogen enrichment, further increase in the hydrogen energy share reduces the NO_X emission. This was due to lean-burn operation (equivalence ratio decreases) which leads to a reduction in the peak combustion temperature. The lowest smoke, particulate matter, and HC were noted at 90% hydrogen energy share due to the absence of hydrocarbon on hydrogen. Higher peak pressure and heat release rate were noted for 30% hydrogen energy share (Saravanan and Nagarajan 2008).

Bose et al. (2009) investigated the effect of hydrogen enrichment (0.15 kg/h) and EGR (10% and 20%) on the engine characteristics of the diesel engine. At 80% load, an increase

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in BTE and reduction in BSEC was noted with hydrogen enrichment compared to diesel only operation. Whereas, the addition of EGR deteriorates the combustion, which reduces the BTE and increases the BSEC of the engine. The volumetric efficiency of the engine was reduced with the addition of hydrogen due to the high density of hydrogen. Lower CO_2, CO, smoke, and HC emissions are observed for hydrogen enrichment compared to diesel only operation. The addition of EGR negatively affects on exhaust emission, which increases the CO and HC emissions. NO_X emission for hydrogen enrichment was higher at 80% load compared to diesel only operation (Bose and Maji 2009).

Deb et al. (2015) investigated the effect of hydrogen on engine characteristics in dual fuel mode operation. In their research work timed injection system was used for hydrogen injection into the inlet manifold. The injector opening time and duration of injector opening were controlled by ECU. The hydrogen is injected into the manifold after the scavenging period. The addition of hydrogen increases the BTE and reduces the energy consumption compared to neat diesel fuel; this improvement was increased with an increase in the hydrogen enrichment. A very small increment in cylinder pressure is seen for 11% and 42% hydrogen energy share and all other HES show lower cylinder pressure compared to pure diesel operation. The lower heat release rate was seen for all HES except 42% HES which is a little higher compared to no hydrogen content. Lower HC, CO_2, CO, and smoke emissions were noted for all HES which were due to an increase in H/C ratio and complete combustion due to the high flame speed of H_2. NO_X emission for hydrogen was higher compared to diesel fuel and it was increased with an increase in the hydrogen energy share (Deb et al. 2015).

Kargoz et al. (2015) investigated the effect of hydrogen enrichment on the engine characteristics of CI engine. The 30% hydrogen energy share was used as a secondary fuel which was injected in to the inlet manifold, through an LPG-CNG injector, the flow rate and injection duration were controlled by ECU. The obtained results in the experiment indicate that lower thermal efficiency and an increase in fuel consumption were noted compared to diesel only operation. Lower CO, CO_2, and smoke emissions were noted with hydrogen enrichment compared to neat diesel. Lower NO_X emission was obtained at low and medium load conditions, whereas at higher load condition NO_X emission increases compared to diesel only operation. It was noted that both cylinder pressure and heat release rate at all engine load was increased with the addition of hydrogen compared to diesel only operation. The higher flame speed and explosive type combustion of hydrogen increase the heat release rate (Karagöz et al. 2015). The same group continued the investigation with varying hydrogen energy shares of 22% and 53% with diesel fuel. The same injection

method for hydrogen enrichment was also used in this experiment. The same kind of results was also observed in this experiment. (Karagöz et al. 2016).

Parthasarathy et al. (2015) studied the effect of hydrogen enrichment with ethanol–biodiesel blend on engine characteristics of a DI diesel engine. The flow rate of hydrogen was 4 lpm, which was constant thorough the experiment. In the biodiesel-ethanol blend, the ethanol concentration was varied as 10%, 20% and 30%. Higher BTE and lower energy consumption were noted for all the blends in the test compared to neat diesel operation. The hydrogen took part in the combustion and improves the performance of the engine. The hydrogen assisted combustion of biodiesel-ethanol blends shows lower HC, CO, and NO$_X$ emissions compared to diesel fuel (Parthasarathy et al. 2016).

Sharma et al. (2018) experimented to know the effect of hydrogen on engine characteristics in dual fuel mode. The hydrogen was supplied to the intake manifold and the HES in the test were varied as 5, 10 and 20%. The hydrogen assisted combustion reduces the maximum pressure, which reduces the BTE and increases the BSEC of the engine at all engine loads due to a reduction in volumetric efficiency of the engine. The lower HES reduces the NO$_X$ emission at low and mid load, whereas at higher load the NO$_X$ emission was increased for all HES. Higher HC formation was noted for 5% HES, and with an increase in HES the reduction in HC was noted at all loads, whereas, lower CO was noted for all HES and at all engine (Sharma and Dhar 2018).

Yilmaz et al. (2018) examined the effect of hydrogen on engine characteristics of CI engine; hydrogen volume was varied as 20 and 40 lpm. It was noted that higher CP and lower HRR were noted with hydrogen compared to diesel fuel. Higher BTE and lower BSEC were noted with 20 lpm hydrogen at 50 and 75 Nm load, whereas, 40 lpm of hydrogen showed lower BTE and higher BSEC at 100 Nm load, which was due to unavailability of oxygen for complete combustion of inducted fuel. Lower NO$_X$ emission was obtained for 20 lpm at all loading conditions whereas, an increment in NO$_X$ was noted for 40 lpm at 50 and 75 Nm load. Lower CO$_2$ and HC emission was noted at all loading condition due to no carbon content in hydrogen (Yilmaz and Gumus 2018).

Juknelevičius et al. (2019) conducted an experiment to study the effect of hydrogen on engine characteristics in a dual fuel mode. The hydrogen volume was varied as 10, 20 and 30 lpm which was supplied to the intake manifold of the engine. The in-cylinder pressure was reduced at all engine loads with the addition of hydrogen compared to diesel fuel due to reduced ignition delay. Thermal efficiency of the engine decreases with a higher fraction of hydrogen energy share, whereas, at low load and medium load the consumption of diesel was reduced with an increase in hydrogen flow rate. Lower NO$_X$ was obtained with

hydrogen, the addition of hydrogen reduces the ignition delay of diesel which reduces the in-cylinder pressure, and the heat release rate leads reduction in NO_X. Due to lower volumetric efficiency, incomplete combustion of diesel fuel occurs which leads to an increase in HC emission, whereas, the lower H/C ratio reduces the CO emission with hydrogen (Juknelevičius et al. 2019).

2.3 The use of selective catalytic reduction (SCR) of NO_X in CI engine

Macleod et al. (2002) investigated the effect of $Pt/\gamma-Al_2O_3$, $Pd/\gamma-Al_2O_3$, and Pd/SiO_2, Pt and Pd loading was 0.5 wt% respectively. Hydrogen was used as a reductant in the experiment. The light-off temperature was discussed in the study, light-off temperature was defined as the temperature required to obtain 50% conversion of the reductant. The exhaust gas contains 4000 ppm H_2, 500 ppm NO, and 5% of O_2. For $Pt/\gamma-Al_2O_3$ and $Pd/\gamma-Al_2O_3$ catalysts, the light-off temperature was obtained at 100°C and 110°C respectively. Maximum NO conversion for $Pt/\gamma-Al_2O_3$ and $Pd/\gamma-Al_2O_3$ was 80% (145°C.) and 10% respectively. It was also, noted that the Pd catalyst produced less NO_2 compared to the Pt catalyst. With an increase in the CO content in the exhaust, increases in the light-off temperature of H_2. The addition of 500 ppm of CO reduces the NO conversion efficiency of Pt catalyst, maximum conversion was only 43% at 165°C. Whereas the NO conversion was increased for Pd catalyst with CO addition which was 25% with 500 ppm CO, NO conversion increases with an increase in CO. SiO_2 showed low conversion efficiency compared to $\gamma-Al_2O_3$, support material plays important role in the conversion of NO. Under lean-burn conditions with both H_2 and CO, Pt catalyst showed higher NO conversion efficiency, whereas, Pd catalyst showed low performance under lean-burn condition (Macleod and Lambert 2002).

Kaneeda et al. (2009) examined the effect of Pt/Al_2O3 and $Pt-Pd/Al_2O_3$ catalysts on NO oxidation. It was noted that Pt (0.77) showed 70% NO conversion efficiency at 300°C, with severe heat treatment the conversion efficiency was reduced to 40%. The addition of a Pd catalyst improves the performance of the catalyst compared to a monometallic Pt catalyst. After the severe heat treatment, the NO conversion was 58% for the Pt-Pd catalyst (Kaneeda et al. 2009).

Herreros et al. (2014) investigated the effect of monometallic Pt/Al_2O_3 and bimetallic $Pt-Pd/Al_2O_3$ catalysts on NO reduction in a diesel engine. Both Pt and Pd loading was 1 wt% on the ceramic honeycomb structure, which has a cell density of 4000 cpsi, and the surface area of the washcoat was 100 m^2/g. Hydrogen was used as a reductant, the flow rate of the hydrogen was varied as 500, 1000, 2500 and 8000 ppm. The NO_X conversion efficiency

increases with a higher flow rate of H_2, maximum NO_X reduction was noted for 8000 ppm of H_2 and Pt-Pd catalyst compared to monometallic Pt catalyst. Along with NO_X reduction, HC and CO emissions were also reduced, the addition of hydrogen reduced the light-off temperature of both HC and CO (Herreros et al. 2014).

Caravaggio et al. (2016) investigated the effect of Pd catalyst and various support (Al_2O_3, CBV-2314 & TiO_2) on NO_X conversion in the stream of exhaust. Initially, the experiment was conducted to find the right composition of the supporting material, and later the experiment was continued to find the right loading of Pd. In the first part of the experiment, they noted that Pd with 0.2% loading and supporting material compositions like $Al_2O_3(60\%)$+CBV-2314(20%) and $TiO_2(20\%)$ showed the best results compared to other support compositions in the experiment. This was due to the highest formation of NH_3 in the situ sites in the presence of exhaust stream $NO/H_2/CO/O_2$, NH_3 was consumed in NO_X conversion. The catalyst with zeolite was highly active in the conversion of NO_X. In the formation of NH_3, CO was involved, hence in the experiment NO conversion was increased with an increase in the CO conversion, in the absence of CO, NO_X conversion was dropped for the catalyst with zeolite content. With the continuation of this experiment, the author varies the loading of Pd to 0.05, 0.2, and 0.5 wt%. The lowest wt% (0.05) of Pd showed the highest $DeNO_X$ activity, however, the NO_X conversation temperature shifted to the higher side. This was due to decreasing the loading of Pd and increasing the dispersion of metal on the support material. Also, the 0.5 wt% loading of Pd showed the best NO_X conversion efficiency at very temperature (115°C) compared to other Pd loadings (Caravaggio et al. 2016).

Abu-Jrai et al. (2018) et al. investigated the effect of hydrogen and methane on the engine characteristics of CI engine. To reduce the NO_X emission in the exhaust 1wt% Pt/Al_2O_3 SCR catalyst was used, here unburnt HC was used as the main reductant. The 20% of intake air was replaced with a mixture of H_2+CH_4 gas and manually the supply of diesel fuel was reduced to maintain the constant speed and load. The combination of H_2+CH_4 was fed into the intake manifold with different ratios as 75:25, 50:50, and 25:75 ratios. The overall efficiency of the engine at a higher load for a 50:50 ratio was high compared to diesel fuel. The unburnt H_2 and HC were high for all ratios of hydrogen and methane mixture. For gaseous fuel mixture at low and medium load, the lower NO_X emission was noted, whereas at high load NO_X emission was more compared to diesel fuel operation. At lower exhaust gas temperature (<150°C) SCR of NO_X was low which was less than 20%. The maximum NO_X reduction at low, medium and high loads was 62%, 59%, and 56% respectively at 230°C. The further increase in the temperature leads to a decrement in NO_X reduction was

noted due to oxidation of HC. The use of H_2-CH_4 (25:75 & 50:50) increases the SCR activity at a lower temperature. This was due to oxidation of HC which increases the temperature of the catalyst bed. The performance of the SCR was reduced with an increase in the load due to the reduction of unburnt HC in the exhaust (Abu-Jrai et al. 2017).

Kim et al. (2018) investigated the combined effect of α-Al_2O_3 and γ-Al_2O_3 with 1 wt% of Pt catalyst; CH_4 was used as a reductant. The activity test was conducted in a reactor and the reactor temperature was maintained at 120°C. The 4000 PPM of reductant (CH_4) and 400 PPM of NO were passed to the reactor for SCR reaction. The highest specific surface area was shown by γ-Al_2O_3 compared to α-Al_2O_3. The highest CH_4 absorption was shown by α-Al_2O_3 and the highest NO absorption was shown by γ-Al_2O_3. In presence of O_2, NO oxidized to NO_2, and the highest absorption of NO_2 was found for Pt/γ-Al_2O_3 catalyst. The highest NOx conversation efficiency was shown by Pt/γ-Al_2O_3 compared to Pt/α-Al_2O_3 (Kim et al. 2012).

Ayodhya et al. (2018) studied the effect of Rh/Al_2O_3 catalyst with NH_3 as a reductant on NO_X emission of the CI engine fuelled with blends of waste plastic oil and diesel. The diesel engine was run with a mixture of 30% plastic oil and 70% diesel fuel. At 80% load, an increase in BTE was noted with the use of plastic oil compared to diesel fuel. The various flow rate of NH_3 was used in the experiment to know the optimum flow rate of the reductant. The maximum NO_X reduction was noted at 60% load for 0.5 kg.hr flow rate of NH_3, in between the temperature range 250-300°C (Ayodhya, Lamani, Thirumoorthy, et al. 2018).

2.4 Summary of the literature review

Some of the most notable features from the literature are as follows:

- The fast depletion of fossil fuel reserves has alarmed to find the alternative renewable source which could be an effective replacement for the conventional fuels.

- Diesel fuelled engines have been proven to be an efficient solution to the hunt for alternative fuels.

- A limited amount of literature is available on 1-Pentanol as an alternative fuel for diesel and also had no agreement on exhaust emission.

- In most of the cases, the researchers use 1-Hexanol in lower concentration as an additive/surfactant with ethanol/diesel blend, and biodiesel/diesel blends to improve the phase stability of the blend. Till now only three studies are available that use 1-Hexanol/diesel blends in CI engine and some other studies used 1-

Hexanol as an additive or surfactant.

- Higher alcohols virtually decrease all regulated emissions and an urgent need is to develop advanced technologies and concepts in this area to optimize the various engine parameters that will result in a clean environment and more efficient diesel engine.

- A good amount of research has been done on hydrogen in the dual fuel mode of operation in a diesel engine. However, still, the use of hydrogen in CI engine is under investigation due to contradictory results from researchers.

- The use of a higher concentration of hydrogen in dual fuel mode causes knocking in the engine.

- The combination of higher alcohol and hydrogen in a diesel engine is not available in the literature.

- A decent amount of work has been done on the SCR of NOx by using different ratios of Pt and Pd catalysts.

2.5 Research gap

- The following gaps were found in the literature survey

- Most of the researchers used higher alcohol like 1-Hexanol and 1-Penatnol as additives or co-solvent. Among all the higher alcohol 1-Hexanol is the least explored higher alcohol.

- The least explored combination is the EGR as emission controlling technique with 1-Hexanol and 1-Pentanol as fuel in a CRDI CI engine.

- Many researchers had shown that hydrogen enrichment has a positive effect on CO_2 and smoke emissions of diesel engines in the literature. However, researchers have not shown agreement on cylinder pressures, ROHR, BTE, BSEC, HC, and NO_X emissions from diesel engines, and these parameters need to be investigated to make up for this deficiency in the literature.

- None of the researchers had explored the combination of Hydrogen and higher alcohol in a dual fuel mode in the CRDI CI engine.

- None of the researchers had explored the combination of EGR and SCR of NO_X as an emission controlling technology in a CRDI CI engine fuelled with higher alcohol and hydrogen in a dual fuel mode.

2.6 Objective of the present investigation

Research objective:

The main objective of the present investigation is to study the effect of higher alcohol and hydrogen in a dual fuel mode on combustion, performance, and emission characteristics of CRDI diesel engine and the reduction of NO_X emission with the application of EGR and SCR of NO_X.

Specific Objectives of the Research Work:

- To study the effect of higher alcohol/diesel blends on performance, combustion, and emission characteristics of CRDI diesel engine.
- To study the effect of hydrogen/diesel blends on performance, combustion, and emission characteristics of CRDI diesel engine.
- To study the combined effect of hydrogen and higher alcohol/diesel blends on performance, combustion, and emission characteristics of CRDI diesel engine.
- To study the effect of EGR and SCR of NO_X on the emission characteristics of CRDI diesel engine fuelled with optimized higher alcohol/diesel blends and optimized hydrogen/higher alcohol blend.

CHAPTER 3

Experimental Setup and Methodology

3.1 Development of experimental setup

In this research work four-stroke, twin-cylinder, naturally aspirated common rail direct injection diesel engine was used. The engine used in this experiment is a Mahindra make and Maxximo model engine, which is used in minitrucks. The minitruck engines have now become more popular on Indian roads; it boosts the automobile industry. The E- commerce sector plays an important role in the small commercial vehicle segment. Small businesses, traders, and captive customers depend on this mini truck and more than 50% of the sale is shared by these category users. More than 12 million vehicles are there on Indian roads (Thakkar 2017). The fuel consumption of these vehicles is also more; hence in this study mini truck engine is selected. The technical details of the engine are shown in Table 3.1. The engine is coupled with an eddy current dynamometer for loading purpose and connected with an open ECU to control the various parameters of the engine. The detailed schematic diagram of the engine used in this study is given in Fig. 3.1.
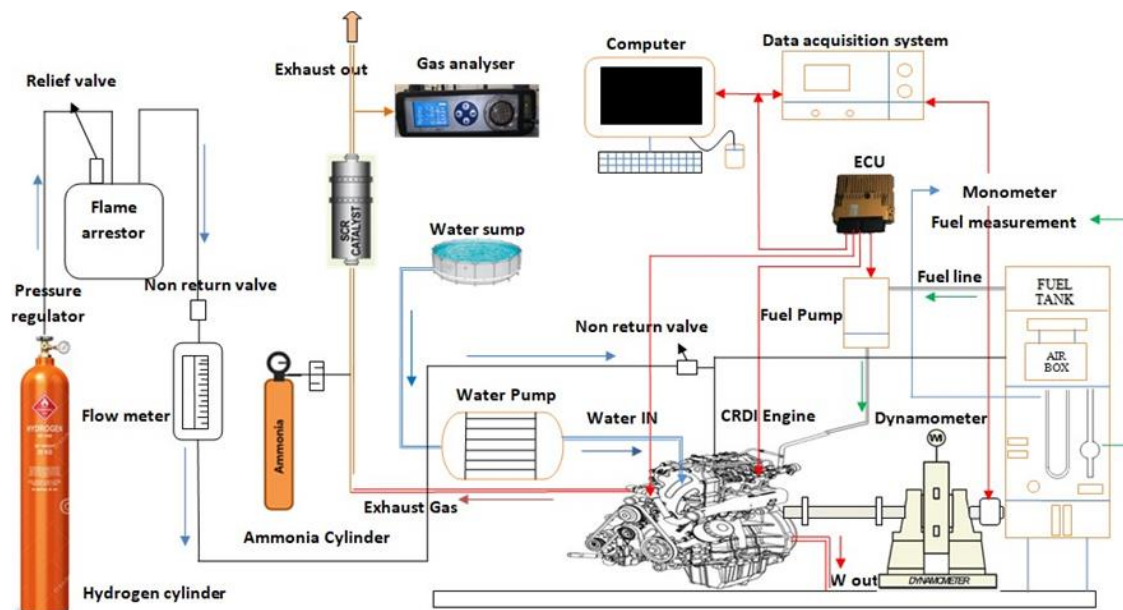
Fig. 3.1 Schematic diagram of the experimental setup

The fuel is delivered from the fuel tank to the electric feed pump through the fuel filter and enters into the high pressure pump. The high pressure pump is used to supply the pressurized fuel through the high pressure pipe to the common rail injection. High pressure pump maintains the required pressure in the common rail injection system under all operating conditions. The solenoid valve controls the amount of required fuel supplied to the injector.

Table 3.1 Details of the Test Rig.

Specification	Details
Make	Mahindra
Model	Maxximo (900 cc)
No. of cylinder	2 Cylinder
Stroke × bore (mm)	83×84
Horsepower (rated)	24. 67 hp (18.4kW) at 3600rpm
Compression ratio	18.5:1
Injection timing	12° BTDC
Injection pressure	1000 bar
Injection type	Common ail (CRDI)
Dynamometer	Eddy current dynamometer

3.2 Development of hydrogen line setup

The schematic diagram of the hydrogen fuel line is included in the test rig (Fig. 3.1). In this study, the hydrogen is continuously supplied into the intake manifold during the suction stroke which replaces the calculated amount of air. The hydrogen in the intake manifold mixed well with air and forms a homogeneous mixture. Prefilled hydrogen cylinder was procured from Lakshmi cryogenics Mangalore, with a purity of 99.99%. The high pressured hydrogen in the cylinder was reduced to 2 bar using a pressure regulator. A flame arrester is installed before hydrogen is inducted into the intake manifold to avoid backfire and explosion. The flame arrestor is equipped with a pressure relief valve to avoid overpressure. Two non-return valves are used; one non-return valve is set before entering into the flow meter and the other one before entering into the inlet manifold. The hydrogen flow meter is used to control the flow rate of the hydrogen. The photographic view of the hydrogen setup is shown in Fig. 3.2.

The hydrogen flow rate of 10 lpm, 20 lpm, 30 lpm, 40 lpm, and 50 lpm is inducted into the intake manifold. The flow rate of the hydrogen is converted into hydrogen energy share by using the following Eq.3.1.

$$HES = \frac{\dot{m}_{hydrogen} \times LHV_{hydrogen}}{\left(\dot{m}_{hydrogen} \times LHV_{hydrogen}\right) + \left(\dot{m}_{diesel} \times LHV_{diesel}\right)} \qquad 3.1$$

Where,

$\dot{m}_{hydrogen}$ = Mass of hydrogen (g/s)

$\dot{m}_{diesel}$ = Mass of diesel (g/s)

| $LHV_{hydrogen}$ | = | Lower heating value of hydrogen (kJ/kg) |
| LHV_{diesel} | = | Lower heating value of diesel (kJ/kg) |

Energy share value for 10, 20, 30, 50 and 50 litter per minute of hydrogen are 5.88%, 11.76%, 17.63%, 23.51% and 29.39% respectively.
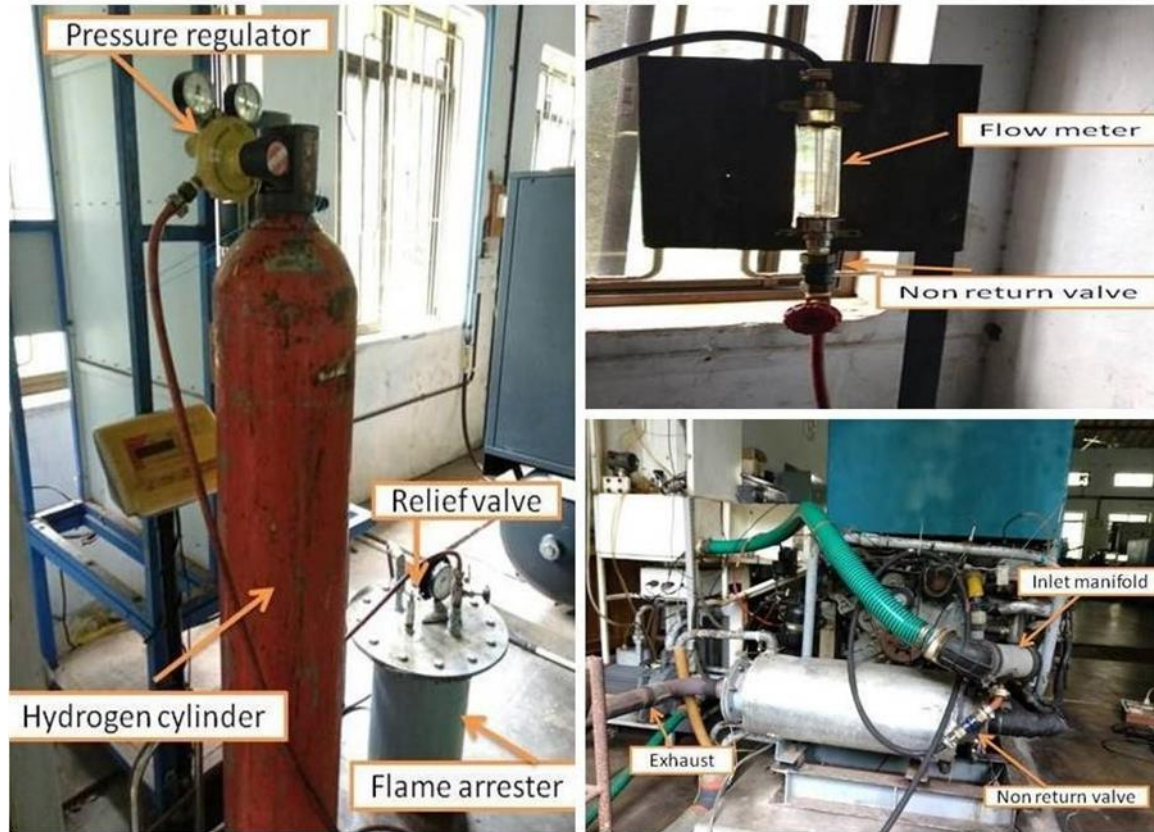
Fig. 3.2 Photographic view of Hydrogen fuel line

3.3 Selective catalytical reduction (SCR) of NOₓ catalyst setup

SCR catalyst used in this research work is to reduce the NO_X emission in the engine exhaust. Platinum (Pt) and Palladium (Pd) bimetallic catalyst is used in the work. Both Pt and Pd share 1wt% each, Al_2O_3 is used as a base material for the catalyst. 1wt% Pt-Pd/Al_2O_3 is coated on a ceramic honeycomb structure. The bimetallic Pt-Pd/Al_2O_3 catalyst caned with a stainless steel alloy sheet was supplied from Techinstro, India and the supplied catalyst was directly coupled to the engine exhaust pipe without further modification. The ceramic honeycomb and stainless steel casing with Pt-Pd/Al_2O_3 catalyst is shown in Fig.3.3. The selective catalytic reduction unit comprises a catalyst, ammonia cylinder for reductant, rotameter, and pressure regulating globe valve to control the flow rate of the ammonia. The pictorial representation of the SCR setup is shown in Fig.3.4.
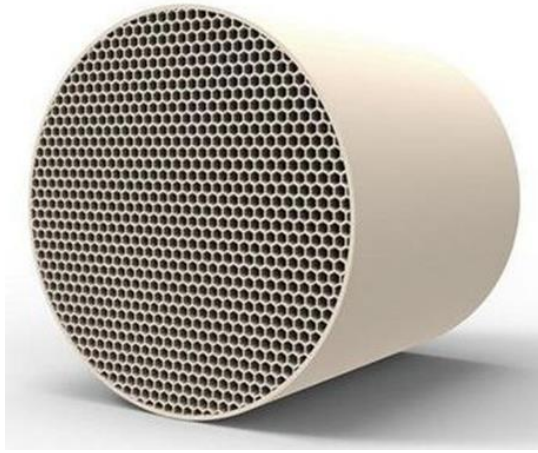
| Ceramic honeycomb structure | SS casing with Pt-Pd/Al_2O_3 catalyst |

Fig. 3.3 Selective catalytical reduction (SCR) of NO_X catalyst

$$8NH_3 + 6NO_2 = 7N_2 + 12H_2O$$
$$4NH_3 + 4NO + O_2 = 4N_2 + 6H_2O$$
$$2NH_3 + NO + NO_2 = 2N_2 + 3H_2O$$

Monolith Ceramic Honeycomb Coated with Pt & Pd Catalyst

$$CO(NH_2)_2 + H_2O = 2NH_2 + CO_2$$
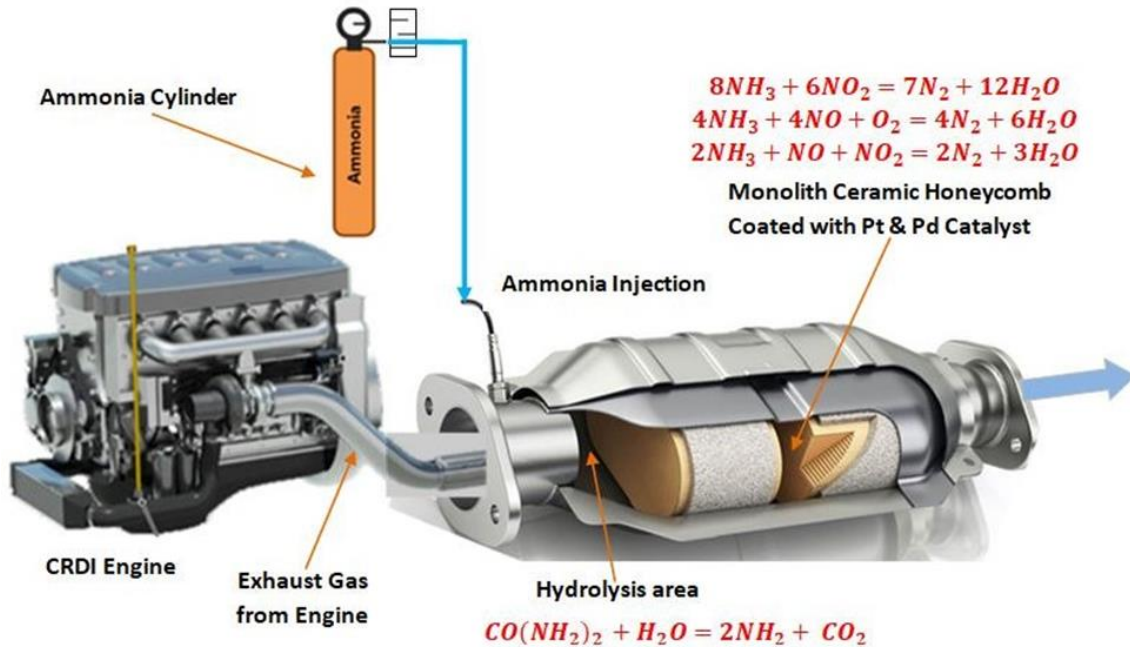
Fig. 3.4 Pictorial representation of the SCR setup

Ammonia gas was inducted into the tailpipe well before the SCR catalyst through a non-return valve, which avoids the flow of the exhaust gas to the ammonia cylinder. The ammonia flow rate is varied from 0.25kg/hr to 0.75 kg/hr. The technical detail of the catalyst used in this experiment is shown in Table 3.2.

Even though it is easy and safe to use urea as a reducing agent, it had associated with some serious problems like the emission of iso-cyanic acid at very low and very high temperatures, these iso-cyanates lead to the hyper-reactive respiratory diseases. The freezing point of urea is only -11⁰C which leads to a technical problem in winter use (Koebel et al. 2000). Hence, in this research work, ammonia gas was directly injected into the tailpipe, so an efficient reaction will occur.

Table 3.2 Technical Details of the Catalyst.

Parameters	Details
Catalyst coating	Platinum and Palladium
Substrate type	Ceramic Honeycomb
Catalyst type	Cylindrical honeycomb structure
Diameter	150 mm
Height	80 mm
Cell density	400 CPSI

3.4 EGR control

Exhaust gas recirculation (EGR) is an effective in-cylinder strategy to control engine emissions, in which a fraction of exhaust gas is re-circulated to the intake manifold where it mixes with fresh air and enters the combustion chamber replacing the same volumetric amount of fresh air. The EGR unit is controlled by ECU, vacuum pump, and a solenoid valve. As per the input given by the user, the ECU sends a signal to the vacuum pump which operates the solenoid valve. The solenoid valve regulates the circulation of the exhaust gases to be re-circulated. EGR rates of 0%, 10%, and 20% were employed to reduce NO_X emissions. The exhaust gas flow rate is will be compared with the air-intake flow rate and calculated as per EGR rate Eq.3.2 i.e.

$$EGR = \left\{ \frac{Mass\ of\ air\ without\ EGR - Mass\ of\ air\ with\ EGR}{ass\ of\ air\ without\ EGR} \right\} \qquad 3.2$$

3.5 Instrumentation

The instruments that are used for sensing and measuring the various characteristics like performance, combustion and exhaust emissions of the engine are installed at the right place in the test rig. With the use of these instruments, the following basic quantities are measured during the experiment.

- Engine speed
- Engine load
- Fuel consumption
- Cylinder pressure
- Crank angle encoding
- Temperatures of the air inlet, water inlet, and exhaust gas.
- Exhaust emissions- HC, CO, NO, CO_2 and O_2

The details of these instruments with corresponding parameters are discussed in detail in

the following sections.

3.5.1 Load measurement system

The experiment is conducted at a constant speed and the load is varied. To measure the load an accurate measuring system is necessary. In this experimental research work, the load system consists of an eddy current dynamometer, strain gauge type load cell, and loading unit. An electric current is supplied to the dynamometer through the loading unit to apply the load. The load cell measures the load applied to the engine.

The eddy current dynamometer is directly coupled to the output shaft of the engine and the load is varied as 20%, 40%, 60%, and 80% of the rated load which correspond to the brake power of 2 kW, 4 kW, 6 kW, and 8 kW. The speed of the engine is measured through the crank angle sensor connected to the engine shaft. The components of the eddy current dynamometer are the rotor, shaft, bearings, casing, and bedplate. The stator in the eddy current dynamometer consists of several electromagnets and the rotor rotates within the electromagnetic field. The rotor of the eddy current dynamometer is connected to the output shaft of the engine as shown in Fig.3.5.
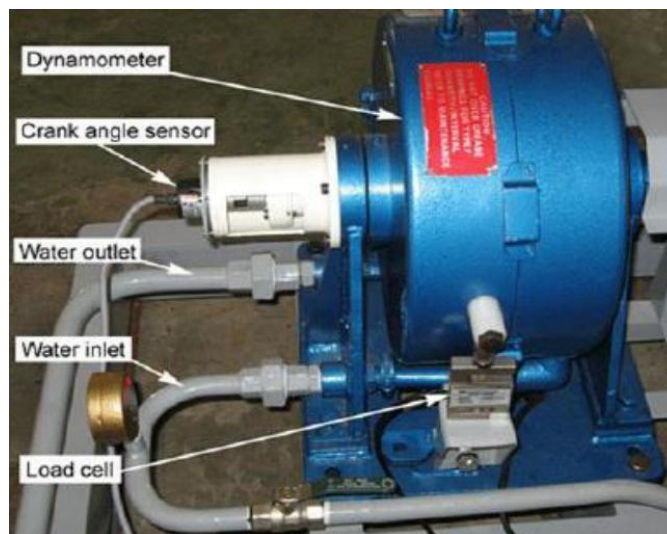
Fig. 3.5 Various components of the eddy current dynamometer

When the direct electric current is passed to the electromagnetic coils in the stator it produces an eddy current due to magnetic flux. When the rotor starts to rotate the produced eddy current field creates a break effect on the shaft thus loading on the engine. The produced rotational torque in the rotor is measured by the strain gauge load cell, which is placed between the horizontal shaft connected to the rotor and the dynamometer bed. The produced eddy current has degenerated as heat, which is absorbed by the cooling water supplied to the dynamometer. The produced current in the electromagnet coil is regulated to control the load on the engine. Dynamometer load measurement is from a strain gauge

load cell and speed measurement is from a shaft mounted sixty tooth wheel and magnetic pulse pick up. AG80 series, Saj Test Plant Pvt. Ltd. make is used in this experiment. The technical specification of the eddy current dynamometer is shown in Table 3.3.

Table 3.3 Specification of eddy current dynamometer

Make	Saj Test Plant Pvt. Ltd
Model	AG80 series
Torque	11.5
Hot coil voltage	75
Continuous current amps	5.0
Speed max.	9000rpm
Load	40kg

The load cell is a device that is used to convert the mechanical force obtained from the eddy current dynamometer to electrical signals. The force applied on the load cell deforms the strain gauge, the electrical arrangements measure the strain and convert it into electrical signals. The strain gauge arrangements in the load cell are in the form of a Wheatstone bridge configuration. The obtained signals from the load cell are to be applied before it uses. The load cell is shown in Fig 3.6 and the configuration of the load cell is shown in Table 3.4.

Table 3.4 Specification of load cell

Make	Sensotronics
Model	60001
Type	S – Beam Universal
Capacity	0 – 50 kg
Non-linearity (FSO)	<+/-0.025%
Hysteresis (FSO)	<+/-0.020%
Non-repeatability	<+/-0.010%
Bridge resistance	350 Ohms (Nominal)

Fig. 3.6 Strain gauge load cell

A dynamometer unit consists of a Dimmer-stat and an ON/OFF switch. The Dimmer-stat is used to control the electric supply to the eddy current dynamometer and the ON/OFF switch is used to control the working of the dynamometer. The dynamometer loading unit used in the experiment is shown in Fig.3.7.

Fig. 3.7 Dynamometer loading unit

3.5.2 Cylinder pressure transducer

The in-cylinder pressure of the engine is measured by using a piezoelectric type pressure transducer with a quartz element is used. A pressure transducer is installed in the cylinder head when the cylinder gas produces pressure on the quartz element. The stress produced by the quartz element is converted into an electric charge which is equal to cylinder pressure. For experimental recording purposes, continuous change in the electric charge due to continuous change in cylinder pressure is converted into electric voltage by using a charge amplifier. The obtained signals from the charge amplifier are sent to the data acquisition card. The specification of the piezoelectric type pressure transducer is given in Table 3.5.

Table 3.5 Specification of pressure transducer

Make	PCB Piezotronics, INC.
Type	ICP, Model SM 111 A22
Sensitivity	1 mV/psi
Resolution	0.1 psi
Cylinder pressure transducer range	10000 psi

3.5.3 Crank angle encoder

The combustion characteristics of the engine are the important parameters to study the combustion behaviour of particular fuel in a combustion chamber. The pressure produced

due to the burning of fuel in a cylinder is drawn against time (crank angle). The knowledge of crank angle corresponding to the variation of cylinder pressure is important. To obtain the signals of crank angle position and top dead centre, a crank angle encoder is used. It produces one pulse for one crank angle degree i.e. it produces 360 pulses for one rotation. The crank angle encoder is fitted to the shaft of the dynamometer as shown in Fig.3.8. The crank angle sensor of Kubler-Germany make and model 8.3700.1321.0360 is used in this research work. The obtained signals from the crank angle sensor are fed to the data acquisition card. The specification of the crank angle encoder is shown in Table 3.6.
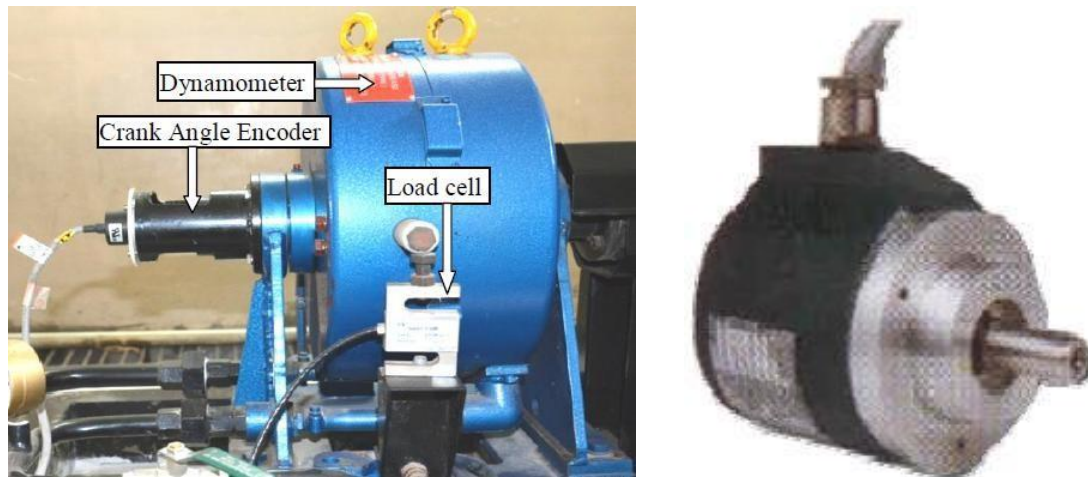
Fig. 3.8 Crank angle encoder

Table 3.6 Specification of the crank angle encoder

Make	Kubler-Germany
Model	8.3700.1321.0360
Supply voltage	5-30V DC
Output	Push Pull (AA,BB,OO)
Output pulses	360 PPR
Diameter	37 mm

3.5.4 Air consumption measurement

The pulsating flow of air is observed in the CI engine due to its reciprocating nature. The measurement of the flow of air is important because it shows the breathing capacity of the engine. To measure the flow rate of the air an airbox with an orifice diameter of 35mm and orifice co-efficient discharge of 0.6 is used. The U-shaper manometer is connected to the orifice of the airbox to measure the differential pressure across the orifice. The mass flow rate of the air is calculated by using Eq 3.3.

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$$\dot{m}_a = C_d \times A \sqrt{\left(\frac{2g \times h_w \times \rho_w}{\rho_a}\right) \times \rho_a \times 3600} \qquad 3.3$$

Where,

C_d	=	coefficient of discharge of orifice (0.6)
A	=	cross section area of the orifice (m^2)
g	=	gravity acceleration (9.8 m/s^2)
h_w	=	differential head across the orifice (m of water)
ρ_w	=	density of water (kg/m^3)
ρ_a	=	density of air (kg/m^3)

3.5.5 Fuel consumption measurement

The volumetric measurement method is used to measure the flow rate of the fuel. A glass burette and stopwatch are used to measure the flow rate of the fuel. The burette has one entry and one exit, the entry pass of the burette is connected to the fuel tank and the exit is connected to the fuel line of the engine. The graduation in the glass burette is in ml, during the measurement, the fuel flow knob is open and the fuel consumption of the engine for 60 sec is noted in ml using the stopwatch. The fuel measurement system is shown in Fig.3.9.

Fig. 3.9 Fuel measurement system

The mass flow rate of fuel in terms of kg/hr is calculated by the following Eq.3.3.

$$\dot{m}_f = \frac{v}{t} \times \rho_f \times 3600 \qquad 3.4$$

Where,

$\dot{m}_f$	=	mass flow rate of fuel consumption (kg/hr)
v	=	fuel volume consumption (ml)

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$$v \qquad = \qquad \text{time of fuel consumption (s)}$$

$$\rho_f \qquad = \qquad \text{density of fuel (kg/m}^3\text{)}$$

3.5.6 Rotameters

The cooling of the engine is necessary during the operation, here the water cooling system is employed to cool the engine. The flow rate of the water to the engine and calorimeter is controlled by using rotameters. The two rotameters fitted to the panel box are shown in Fig 3.10. one is used to control the flow rate of the water to the engine and the second one is to control the flow rate of the water to the calorimeter. The working principle of the rotameter is a variable area, the float in the transparent measuring glass tube moves up and down. Upward flow causes the float to take up a position in which the buoyancy forces and the weight are balanced. The vertical position of the float as indicated by scale is a measurement of the instantaneous flow rate. Eureka makes rotameters used in the system, one ranging from 40-400 lph is used for the engine and 25-250 lph is used for the calorimeter to measure the flow rate.

Fig. 3.10 Rotameter

3.5.7 Exhaust gas measurement

The exhaust gas from the engine like CO, CO_2, HC, NO, and O_2 are measured to know the quality of combustion. NO_X is a combination of NO and NO_2, together it is called NO_X. The toxicity of NO_2 is much higher than NO, and NO_2 has unique characteristics, NO is uncoloured and odourless gas whereas, NO_2 is brown in colour and had a pungent smell. The maximum part of the NO_X is constituted with NO. For exhaust gas measurement AVL Digas 444 exhaust gas analyser is used as shown in Fig.3.11, it works on the principle of Non Dispersive Infra-Red (NDIR) by selective absorption of infrared energy. The exhaust gas from the engine is supplied to the gas analyser through the probe. The probe contains filters to filter the moisture and carbon deposit. The NO and O_2 in the exhaust are measured

through electrochemical sensors and CO, CO_2, and HC are measured through the spectrum analysis principle in AVL DI 444 gas analyser. This measures the NO_X and HC in parts per million and CO, CO_2, and O_2 in percentage volume. The specification of the AVL Digas 444 exhaust gas analyser is shown in Table 3.7.

Fig. 3.11 AVL Digas 444 exhaust gas analyzer

Table 3.7 Specification of AVL Digas 444 exhausts gas analyzer

Model	AVL DiGas 444
Measuring range	HC: 0-20000 ppm volume
	CO: 0-10 % by volume
	CO_2: 0-20 % by volume
	O_2: 0-22 % by volume
	NO: 0-5000 ppm by volume
	Engine speed: 400-6000 rpm
	Lambda: 0-9.999
Oil temperature	-30 to 125°C
Connector Gas In	Approx. 180 lph, max. Overpressure 450 hPa.
Warmup time	Approx. 7 min.
Power supply	11-22 V DC
Relative humidity	≤ 95 %, non-condensing

3.5.8 Data Acquisition System (DAQ) and electronic controlling unit (ECU)

The DAQ is used to analyse the signal from the various measuring units like pressure transducer, loadcell, dynamometer, crank angle encoder, air transmitter and thermocouple, etc., used in the engine setup. DAQ is used in the engine setup is made from National instruments and model USB-6210, 16-bit, 250kS/s is shown in Fig.3.12. "Enginesoft" software package is used for online measurement of performance. This enables to study of

the engine performance and combustion analysis including cylinder pressure, heat release rate, mass fraction burnt, and cumulative heat release rate.

The open ECU is used to control the various operation of the engine like injection timing, injection pressure, pilot injection timing, pilot injection quantity, and exhaust gas recirculation. In this engine setup, NIRA i7r ECU is used with NIRA rk software which enables to control of the various parameters according to the convenience. The NIRA i7r ECU is shown in Fig. 3.13.

Fig. 3.12 Data Acquisition System   **Fig. 3.13** NIRA i7r ECU

3.6 Test fuel

In this research work, 1-Pentanol (CAS No: 71-41-0) and 1-Hexanol (CAS No: 111-27-3) were procured from Loba Chemie PVT. LTD (India) and Diesel was purchased from the local vendor. The test fuels were prepared by mixing 10%, 20%, 30%, 40%, and 50% of 1-Pentanol and 1-Hexanol separately with pure diesel by volume. To prepare the blends, the flash bending method was followed. After flash blending, blended fuels were stirred for 15 min in a magnetic stirrer to form a homogeneous mixture. The list of blends and its designation is given in Table 3.8.

The prepared blends were undergone a phase stability test and it is observed for 24 hours and no separation was found. Properties of pure diesel, 1-Pentanol, 1-Hexanol, and its blend are listed in Table 3.9 and Table 3.10.

Prefilled hydrogen cylinder was procured from Lakshmi cryogenics Mangalore, with a purity of 99.99%. For the convenience of the work, hydrogen is added in terms of volume flow rate. The hydrogen flow rate of 10 lpm, 20 lpm, 30 lpm, 40 lpm, and 50 lpm is inducted into the intake manifold. The properties of hydrogen are shown in Table 3.11.

Table 3.8 List of blends and their designation

1-Pentanol + Diesel blends by volume	
Blend Designation	Fraction of Diesel and 1-Pentanol
10P90D	1-Pentanol (10%) + Diesel (90%)
20P80D	1-Pentanol (20%) + Diesel (80%)
30P70D	1-Pentanol (30%) + Diesel (70%)
40P60D	1-Pentanol (40%) + Diesel (60%)
50P50D	1-Pentanol (50%) + Diesel (50%)
1-Hexanol + Diesel blends by volume	
Blend Designation	Fraction of Diesel and 1-Pentanol
10H90D	1-Hexanol (10%) + Diesel (90%)
20H80D	1-Hexanol (20%) + Diesel (80%)
30H70D	1-Hexanol (30%) + Diesel (70%)
40H60D	1-Hexanol (40%) + Diesel (60%)
50H50D	1-Hexanol (50%) + Diesel (50%)

Table 3.9 Properties of 1-Pentanol fuel

Properties	100D	100P	10P90D	20P80D	30P70D	40P60D	50P50D
Cetane index	50	20	47	44	41	38	35
Viscosity at 40°C (mm^2/s)	2.76	2.88	2.772	2.784	2.796	2.808	2.82
Lower heating value (MJ/kg)	44.99	34.65	43.956	42.992	41.888	40.854	39.82
Density (kg/m^3)	832	815	830.3	828.6	826.9	825.2	823.5

Table 3.10 Properties of 1-Hexanol fuel

Properties	100D	100H	10H90D	20H80D	30H70D	40H60D	50H50D
Cetane index	50	23	47.3	44.6	41.9	39.2	36.5
Viscosity at 40°C (mm^2/s)	2.76	4.64	2.948	3.136	3.324	3.512	3.7
Lower heating value (MJ/kg)	44.99	39.10	44.401	43.812	43.223	42.634	42.045
Density (kg/m3)	832	821.8	830.98	829.96	828.94	827.92	826.9

Table 3.11 Properties of hydrogen in comparison with diesel (Deb et al. 2015; Yilmaz and Gumus 2018)

Properties	Diesel	Hydrogen
Molecular formula	C_XH_Y	H_2
Autoignition temperature (°C)	254–300	585
Stoichiometric air-fuel ratio on a mass basis	14.5	34.3
Molecular weight (g/mole)	190–211.7	2
Density (kg/m^3) at 15°C	835	0.0838
Lower heating value (MJ/kg)	42.49	119.93
Flame velocity (cm/s)	30	265-325
Octane number	30	130
Cetane number	52	-
Specific gravity	0.83	0.091

3.7 Experimental procedure

The experiment is conducted in various modes are explained in this section. The engine was allowed to run for 15min to get stable before the collection of actual data. The gas analyser was also switched on to stabilize before starting the measurement. All the instruments used in the experiment are calibrated. The injection pressure of 1000 bar was kept constant throughout the experiment. The pressure crank angle data of 100 cycles were measured and data of 100 cycles were averaged to obtain cylinder pressure was averaged to diminish the influence of cycle by cycle variation. Throughout the experiment, the speed of 2000 rpm was kept constant and the load is varied as 20%, 40%, 60%, and 80% of the rated load which correspond to brake power of 2 kW, 4 kW, 6 kW, and 8 kW. At each load, the data of fuel measurement, air measurement, exhaust gas temperature, and exhaust gas emission were noted. The flow chart of the experimental procedure is shown in Fig. 3.14. In the experiment, exhaust gas recirculation was employed to know its effect on combustion, performance, and emission characteristics of the engine. 1 wt% Pt-Pd/Al$_2$O$_3$ SCR catalyst is employed to know the effect on NO$_X$ emission.

Phase I: Initially experiment was conducted to compare 1-Pentanol/diesel and 1-Hexanol/diesel blends on engine characteristics of CRDI CI engine at standard injection time 12°BTDC and injection pressure 1000 bar. For the best alcohol/diesel blend the effect of retard (9°BTDC) and advanced (15°BTDC) injection time was studied to know its effect on combustion, performance, and emission characteristics of CRDI CI engine.
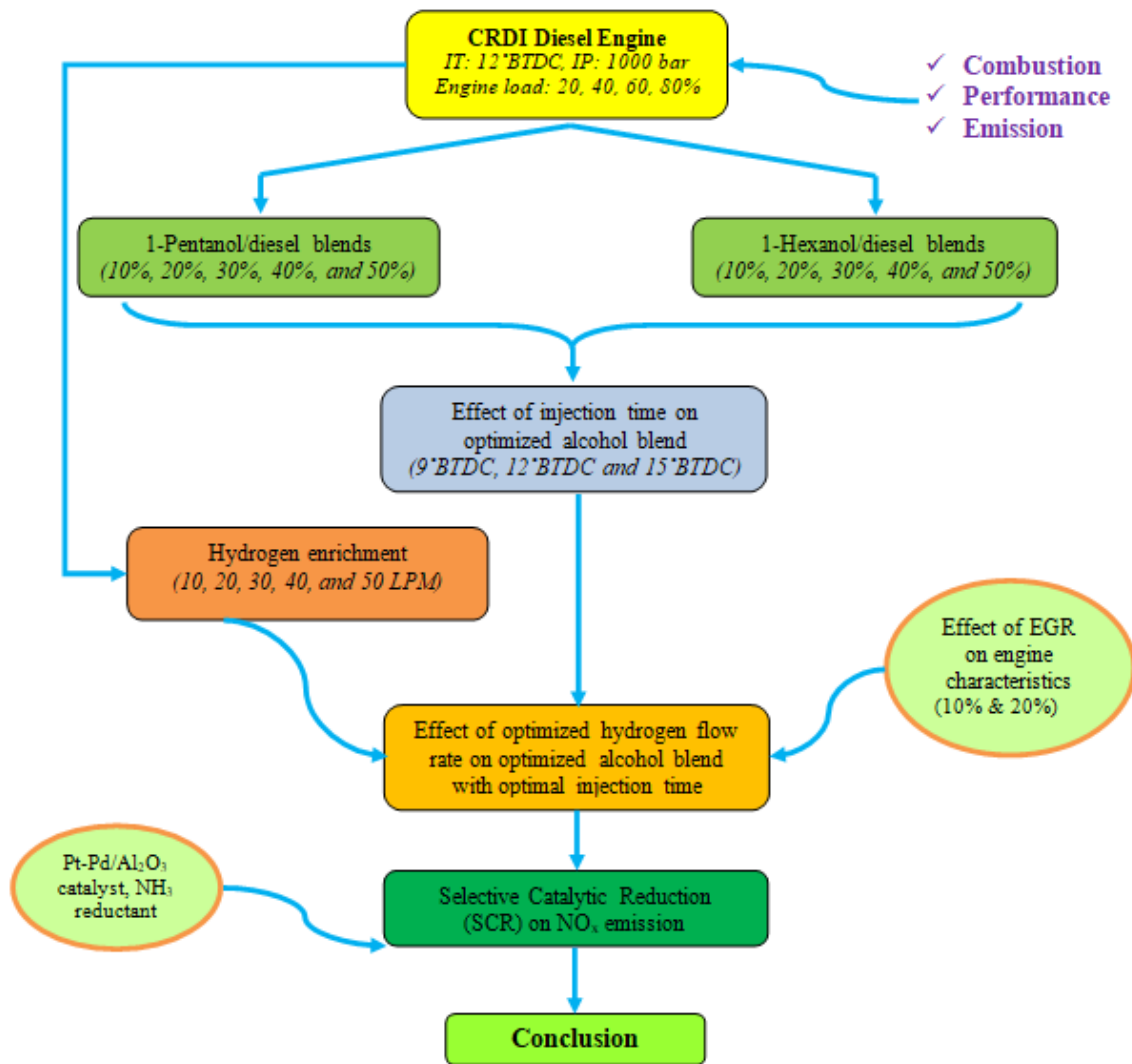
Fig. 3.14 Experimental procedure

Phase II: In this phase, the effect of hydrogen on combustion, performance, and emission characteristics of the CRDI CI engine in dual fuel mode is analysed. Here the hydrogen is inducted into the intake manifold as a secondary fuel and diesel is injected into the combustion chamber conventionally as a primary fuel. The hydrogen flow rate was varied as 10 lpm, 20 lpm, 30 lpm, 40 lpm, and 50 lpm. The optimized flow rate of the hydrogen was used with optimized alcohol/diesel blends.

Phase III: In this phase, the effect of an optimized flow rate of hydrogen, optimized alcohol/diesel blend, and optimized injection time on combustion, performance, and emission characteristics of the CRDI CI engine was studied.

Phase IV: In this phase, the effect of EGR and 1wt% Pt-Pd/Al$_2$O$_3$ SCR catalyst on combustion, performance, and emission characteristics of the CRDI CI engine is studied for the optimized injection time with hydrogen enriched optimized alcohol/diesel blends.

CHAPTER 4

RESULTS AND DISCUSSION

The experiment was conducted at a constant speed of 2000 rpm and the engine load was varied as 20%, 40%, 60%, and 80% load. The fuel injection pressure of 1000 bar was kept constant throughout the experiment. Initially, the experiment was conducted at a standard injection time of 12° BTDC for pure diesel, 1-Pentanol/diesel, and 1-Hexanol blends. Later the experiment was continued with an optimized 1-Hexanol/diesel blend to find the optimized injection time by varying the injection time as 9° BTDC and 15° BTDC. To find the optimized hydrogen ratio, 10 lpm to 50 lpm in the step of 10 lpm was enriched in dual fuel mode. With optimized 1-Hexanol/diesel blend, injection time, and hydrogen ratio the effect of EGR and 1wt% Pt-Pd/Al$_2$O$_3$ catalyst was examined to reduce the NO$_X$ emission.

4.1 Effect of 1-Pentanol/diesel blends on combustion, performance, and emission characteristics of CRDI CI engine.

In this section, the effect of 1-Pentanol/diesel blends on engine characteristics are discussed and compared with base fuel. These are the important parameters that help to understand the behaviour of the fuel in the diesel engine.

4.1.1 Combustion Characteristic

4.1.1.1 Cylinder Pressure

The variation of cylinder pressure under the influence of 1-Pentanol at 80% load is shown in Fig. 4.1. The peak cylinder pressure increases with an increase in the fraction of 1-Pentanol in the blend. The maximum peak cylinder pressure at 80% load for 100D, 10P90D, 20P80D, 30P70D, 40P60D, and 50P50D is 80.71 bar, 80.77 bar, 81.7 bar, 82.8 bar, 81.1 bar, and 80.85 bar respectively. Maximum cylinder pressure is noted for 30P70D, which is 1.47% higher compared to neat diesel fuel operation. The increment in the cylinder pressure is seen only at 80% load, whereas at lower load, the peak cylinder pressure for 1-Pentanol blends is lower than diesel fuel. The addition of 1-Pentanol retard the ignition and the ID period increases with a higher concentration of 1-Pentanol in the blend (Wei et al. 2014). The combustion characteristics depend on both the physical and chemical properties of the fuel. The higher concentration of 1-Pentanol in the blend leads to a reduction in the cylinder pressure. This is due to the lower CN, LHV, and higher LHE of 1-Pentanol. The lower CN of 1-Pentanol prolongs the ID and the higher LHE causes a cooling effect in the

combustion chamber and absorbs more amount of heat for evaporation all these factors reduce the peak cylinder pressure. The prolonged delay period extends the combustion to the expansion stroke, which causes an increase in the combustion chamber volume due to the movement of the piston from TDC to BDC, this results in lower CP (Pan et al. 2018b). Whereas, at higher load, CP of 1-Pentanol blends is higher compared to neat diesel fuel operation due to higher in-cylinder temperature. At a higher load, more amount of fuel is drawn and higher cylinder temperature reduces the effect of LHE of 1-Pentanol, all these factors positively affect combustion. However, lower CN prolongs the ID and CD, and the pressure peak shifts away from TDC this reduces thermal efficiency as most of the heat is lost to the surroundings due to longer CD.
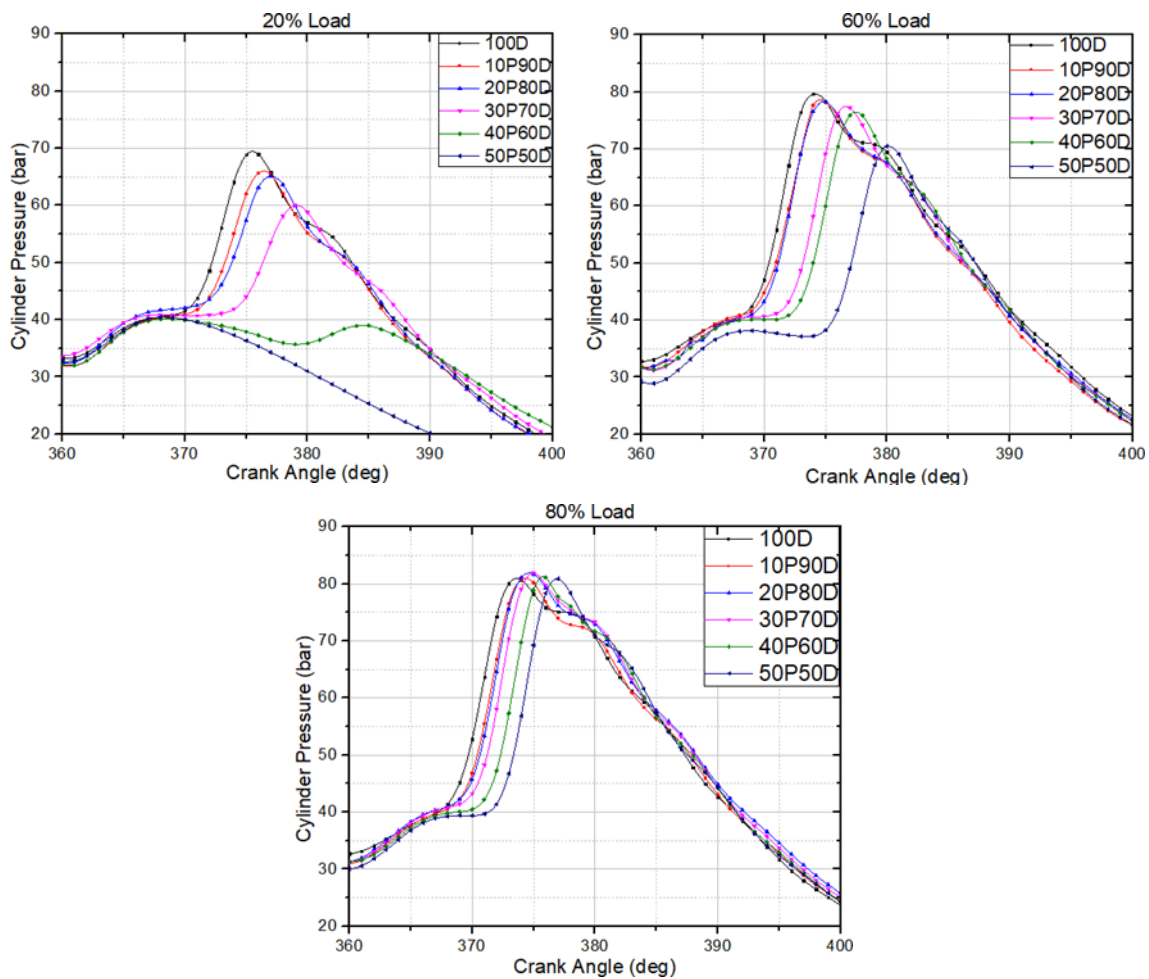
Fig. 4.1 Cylinder Pressure v/s crank angle for different test fuels

4.1.1.2 Mean Gas Temperature

The effect of 1-Pentanol blends on MGT with reference to the crank angle is shown in Fig. 4.2. An increase in the MGT is observed for all the test fuel with an increase in the engine load. The MGT at 80% load for 100D, 10P90D, 20P80D, 30P70D, 40P60D and 50P50D is 1871.7˚C, 1878.42˚C, 1862.49˚C, 1861.99˚C, 1872.5˚C, and 1884.28˚C respectively. The

maximum MGT at 80% load is obtained for 10P90D which is 0.35% higher and 50P50D which is 0.66% higher, the increment in the MGT for 1-Pentanol blends is negligible compared to pure diesel fuel operation. The lowest MGT has been noted at low load conditions, it is due to the reason that, low LHV and higher LHE. Higher LHE quenches the cylinder wall which reduces the MGT, this reduction increases with an increase in the fraction of 1-Pentanol in the blend (Suhaimi, Adam, Anes G Mrwan, et al. 2018). At mid and higher load, the increment in the MGT is due to lower CN of 1-Pentanol, which increases the delay period thereby increasing the proportion of premixed combustion which accelerates the combustion. On the other hand, the O_2 content of the pentanol helps in the complete combustion of the fuel resulting in higher MGT (Pan et al. 2018b; Wei et al. 2014).
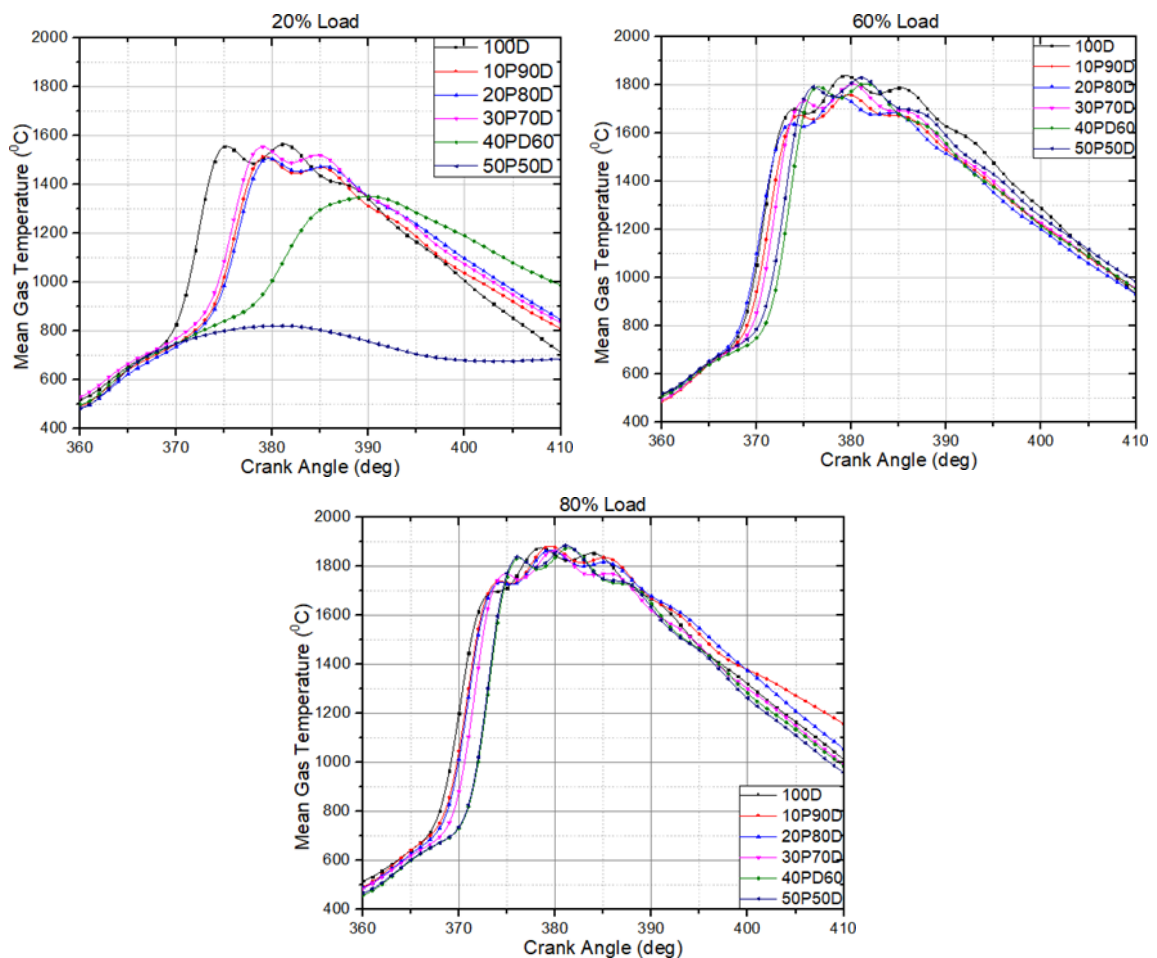
Fig. 4.2 Mean Gas Temperature v/s crank angle for different test fuels

4.1.1.3 Net Heat Release Rate

The effect of 1-Pentanol blends on NHR with reference to crank angle is shown in Fig.4.3. The NHR increases with an increase in the load and also it increases with a higher proportion of 1-Pentanol in the blend at 80% load. The NHR rate at 80% load for 100D,

10P90D, 20P80D, 30P70D, 40P60D and 50P50D is 96.27 J/deg, 98.92 J/deg, 101.4 J/deg, 104.59 J/deg, 110.25 J/deg and 105.53J/deg respectively. A combustion characteristic of any fuel mainly depends on fuel properties, poor fuel properties deteriorate the combustion, which is responsible for lower CP, NHR, and MGT. At 20% load condition the NHR for all 1-Pentanol blends is lower compared to neat diesel fuel operation. This is due to low cylinder temperature; the higher LHE of 1-Pentanol requires more heat for evaporation. The lower in-cylinder temperature and low CN increase the ignition delay (Li, Wang, et al. 2015). More fuel is accumulated in the combustion chamber due to a longer delay period, and only a small quantity of fuel will get burnt in the premixed combustion zone and diffusion combustion zone. Most of the unburnt hydrocarbon comes in the exhaust. Whereas, at 80% engine load an increment in NHR is observed from the graph for all 1-Pentanol blends compared to neat diesel fuel.
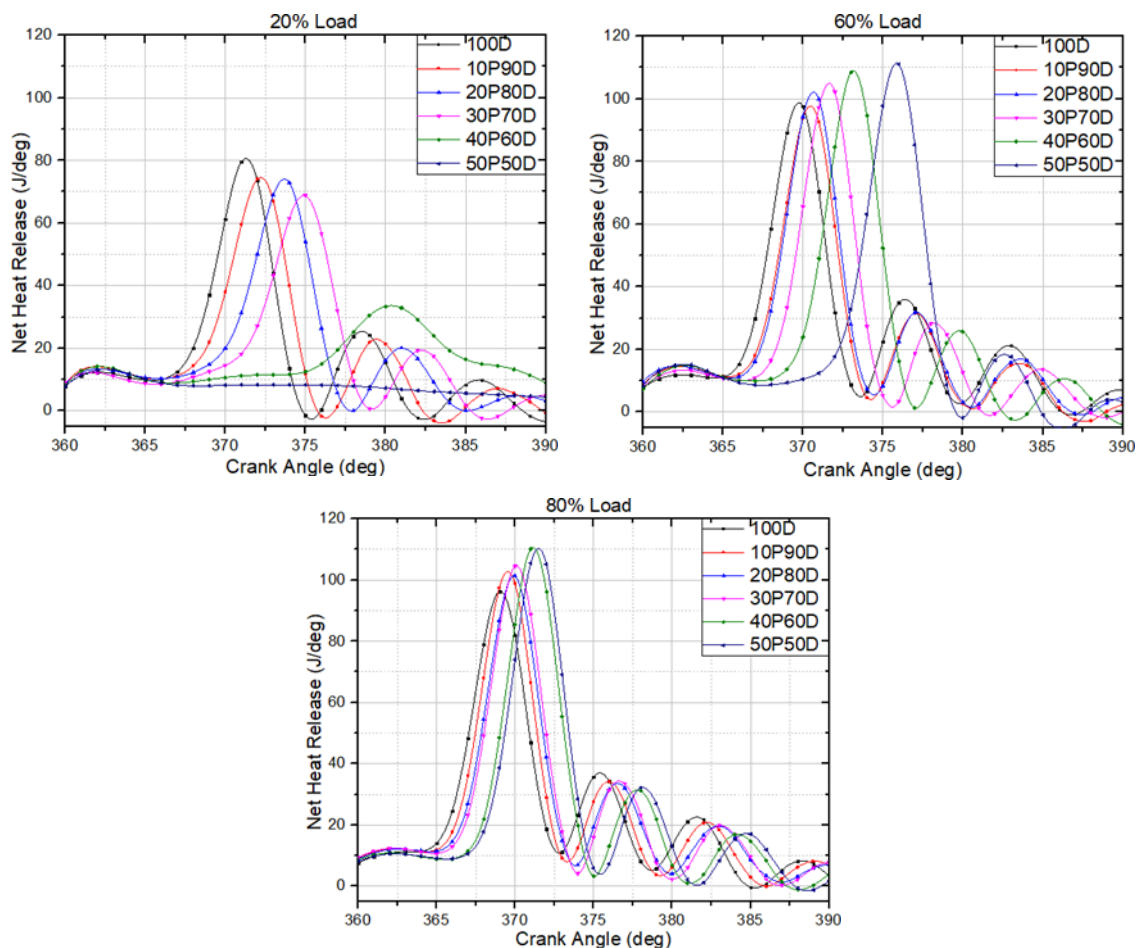
Fig. 4.3 Net Heat Release rate v/s Crank angle for different test fuels

The increase in NHR for 1-Pentanol blends is 3.40% at 10P90D, 8.01% at 20P80D, 12.09% at 30P70D, 9.76% at 40P60D and 9.71% at 50P50D. The longer ID allows to the accumulation of more quantity of fuel in the combustion chamber and a higher cylinder temperature at 80% load helps in rapid evaporation of accumulated 1-Pentanol. Also, the higher O_2 content of 1-Pentanol helps in the complete combustion of accumulated fuel. The

prolonged ID due to lower CN of 1-Pentanol increases the combustion duration, which can be seen in the graph, the peak shifts away from the TDC.

4.1.2 Performance Characteristics

4.1.2.1 Brake Thermal Efficiency
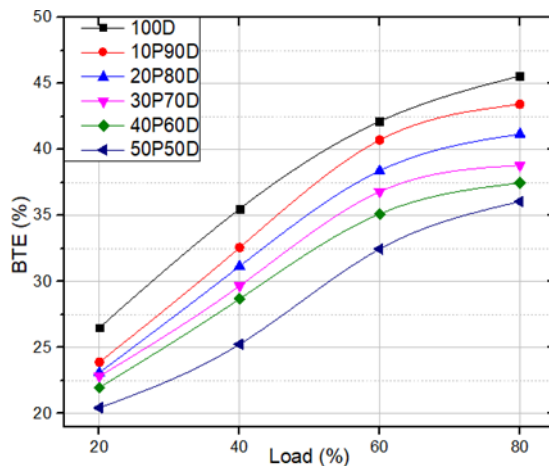
Fig. 4.4 BTE v/s engine load for different test fuels

Fig. 4.4 shows the change in BTE with engine load for different fuel blends. It is noted from the graph that BTE increases with an increase in the load for all the fuel blends. The results indicate that there is a reduction in BTE with the use of 1-Pentanol blends and an increment in the reduction is marked for higher proportion of 1-Pentanol in the blends. The similar reduction of BTE with the use of 1-Pentanol in the blends was noted by Yilmaz et al. (Yilmaz and Atmanli 2017) and Atmanli et al. (Atmanli and Yilmaz 2018b). The reduction in BTE at 80% load for 10P90D, 20P80D, 30P70D, 40P60D, and 50P50D is 4.70%, 9.65%, 14.86%, 17.78%, and 26.14% lower respectively when compared with baseline readings with neat diesel as a fuel. The lower LHV and higher LHE of 1-Pentanol are the possible reason for the reduction of BTE (Atmanli and Yilmaz 2018b; Rajesh Kumar and Saravanan 2015; Yilmaz and Atmanli 2017). The higher LHE of the 1-Pentanol absorbs more amount of heat from the combustion chamber, and also the lower LHV of the 1-Pentanol results in a low heat release rate, both these parameters deteriorate the combustion which harmed the BTE of the engine. On the other hand, the lower CN of 1-Pentanol prolongs the ID, which expands the combustion duration and it continues till the piston reaches the BDC. Due to this the amount of heat transferred to the engine parts increases during the combustion process (Atmanli and Yilmaz 2018b). This enhanced loss of heat to the engine parts reduces the BTE.

4.1.2.2 Brake Specific Energy Consumption
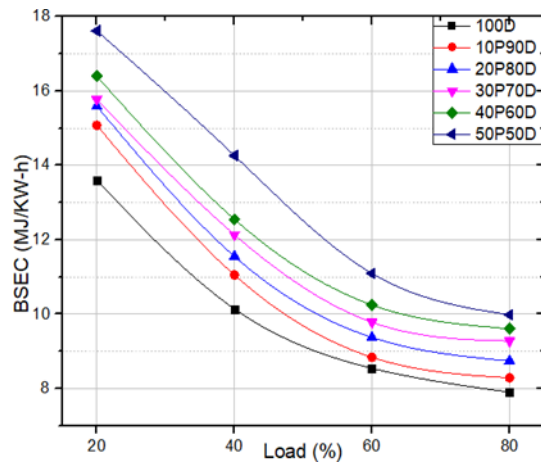
Fig. 4.5 BSEC v/s engine load for different test fuels

The change in BSEC with respect to load for 1-Pentanol blends is shown in Fig. 4.5. The BSEC decreased with an increase in the engine load. At 80% load, increase in BSEC is noted for 10P90D, 20P80D, 30P70D, 40P60D, and 50P50D which is 4.70%, 9.652%, 14.86%, 17.78% and 26.14% higher respectively compared to pure diesel fuel operation. The increasing trend of BSEC is observed for a higher percentage of 1-Pentanol in the blend. The lower LHV of 1-Pentanol causes a reduction in the heat release rate resulting in a higher BSEC. Further, 1-Pentanol consumes higher latent heat from the combustion chamber to evaporate due to its higher LHE, which deteriorates the combustion by causing a cooling effect in the cylinder resulting in higher BSEC (Yilmaz and Atmanli 2017).

4.1.3 Emission Characteristics

4.1.3.1 Hydrocarbon Emission
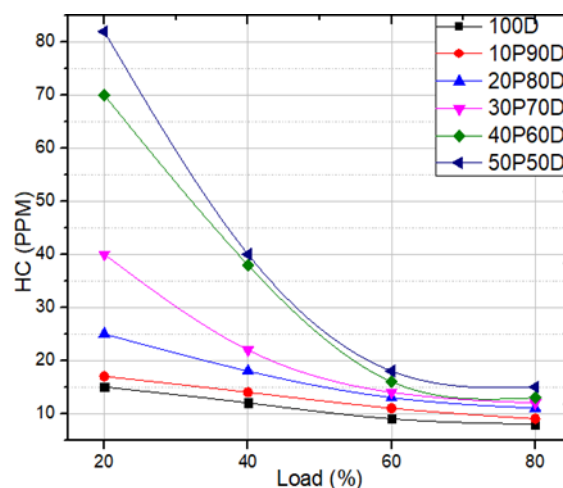
Fig. 4.6 Change in HC emission with load for different test fuels

The formation of HC emission with load for different test fuels is shown in Fig. 4.6. The HC emission for 10P90D, 20P80D, 30P70D, 40P60D and 50P50D at 80% load is 11.11%,

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27.27%, 33.33% 38.46% and 46.66% higher respectively than neat diesel fuel operation. Among all the five alcohol blends, higher HC emission is observed for 50P50D at all engine loads. The HC emission from the engine increases as the percentage of 1-Pentanol increases in the blends at all loading conditions. The low CN and high LHE of higher alcohol are the probable cause of high HC discharge (Wei et al. 2014). The longer combustion duration due to low CN, presence of oxygen molecules, and water content in the fuel weakens the self-ignition properties of the fuel and causes a quenching effect in the engine cylinder (Nanthagopal et al. 2018). Due to longer ID, the fuel would take more time to evaporate in the cylinder which leads to a broader and leaner outer flame zone, resulting in a higher HC emission (Nadir Yilmaz 2016; Pan, Huang, Liao, Jia, Zhou, Huang, et al. 2019). The formation of the too lean mixture due to the presence of oxygen content in 1-Pentanol also results in a higher HC emission. Also, the higher LHE of 1-Pentanol creates the quenched zone in the engine cylinder by absorbing the higher amount of heat to evaporate, the flame front reaches the quenched zone faster due to the lower viscosity of 1-Pentanol and loss of heat to the cold region (Atmanli and Yilmaz 2018b).

4.1.3.2 Carbon Monoxide Emission
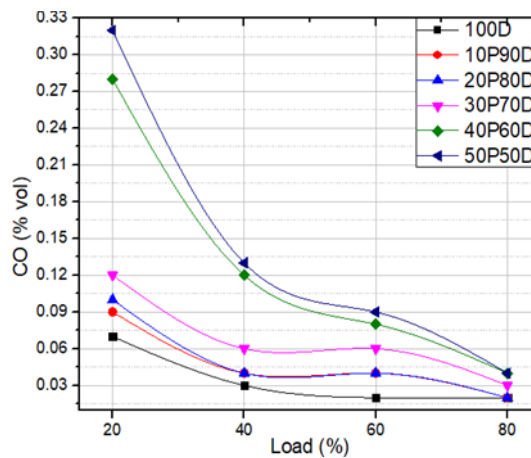
Fig. 4.7 Change in CO emission with load for different test fuels

The change in CO emission for all the test fuels with engine load is shown in Fig. 4.7. At a lower load, the temperature of the engine cylinder is low which causes higher CO emissions compared to a higher loading condition. At 80% load, 10P90D and 20P80D showed similar CO compared to neat diesel fuel. It is a positive sign of good combustion, the higher oxygen content of 1-Pentanol in their molecular structure and higher cylinder temperature helps in the combustion process which causes a lower CO emission. The CO emission from the engine is comparatively higher for alcohol blends than diesel fuel operation and its concentration in the exhaust increased with a higher percentage of higher alcohol in the

blends. Higher CO emission is noted for 50P50D, which is 78.12%, 76.92.33%, 77.77% and 50% higher at 20%, 40%, 60% and 80% engine load compared to diesel fuel. The increase in CO emission for 1-Pentanol blends is due to increased ID and higher LHE. The longer ID is due to low CN of higher alcohol which evaporates most of the fuel before the start of ignition. The higher LHE quenches the cylinder results in lean combustion. Due to the quenching effect, the temperature of combustion is reduced which decreases the post oxidation of fuel. All these factors are in favour to produce CO emission (Atmanli and Yilmaz 2018b). The presence of oxygen molecules in the structure of higher alcohol directly participates in the reaction to produce CO, the bond between the CO remains strong during the combustion hence more CO in the exhaust is observed (Devarajan et al. 2017; Westbrook et al. 2006a). Also, the oxygen molecule in alcohol lowers the calorific value of fuel and CN (Nadir Yilmaz 2016). This directly affects the delay period and combustion duration.

4.1.3.3 Nitrogen Oxide Emission
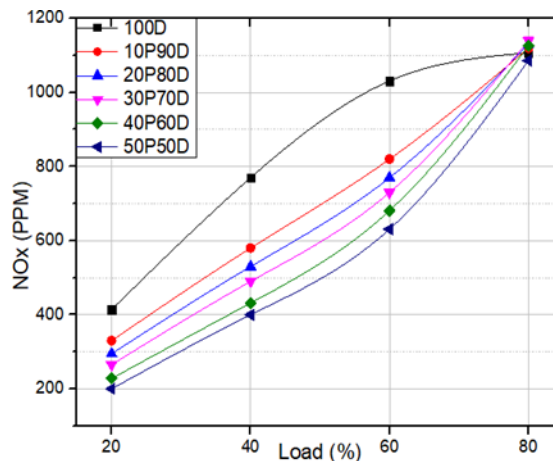
Fig. 4.8 Change in NO_X emission with load for different test fuels

The formation of NO_X emission for all the test fuels with load is shown in Fig. 4.8. The NO_X emission is increased with engine load however, 1-Pentanol had a positive effect on the NO_X formation. Lower NO_X formation is noted with the addition of 1- Pentanol at all loading conditions except 80% load. At 80% load NO_X emission is slightly higher than diesel fuel, which is 1%, 2.03% 2.89%, and 1.68% higher for 10P90D, 20P80D, 30P70D, and 40P60D respectively compared to diesel fuel operation. The main reason for the increase in NO_X at 80% load is higher cylinder temperature and pressure which improves the evaporation of 1-Pentanol blends and forms a homogenous mixture. Due to the longer ID period and CD of 1-Pentanol allows more fuel to accumulate and O_2 of 1-Pentanol helps in the complete combustion of accumulated fuel, resulting in a higher temperature.

Temperature is a key factor in forming the NO_X emission. Whereas, at lower load NO_X emission drastically reduces for all 1-Pentanol blends. At 60% load the reduction in NO_X is noted for 10P90D, 20P80D, 30P70D, 40P60D and 50P50D which is 20.45%, 25.31% 29.19% 33.94% and 38.79% lower respectively compared to neat diesel fuel operation. This is due to lower cetane number, calorific value, and higher latent heat of vaporization of higher alcohol. The higher LHE and lower LHV reduce the combustion temperature, which causes a cooling effect in the engine cylinder. Also, the low CN of alcohol increases the ID, this results in prolonged combustion duration (Devarajan et al. 2017). Thus, the combustion happens when the piston approaches (TDC) and it will go on till the bottom dead centre (BDC). Because of the expanded ignition span more measure of warmth is exchanged to the engine parts resulting in a decline in burning temperature and NOx discharge will be watched (Atmanli and Yilmaz 2018b). Overall, the higher LHE, lower CN, lower LHV, oxygen, and water content of higher alcohols are the reason for the reduction of NO_X.

4.1.4 Summary

- Poor combustion characteristics are observed with the use of 1-Pentanol in the blends. Only the lower fraction of 1-Pentanol had the combustion characteristics are equal to pure diesel operation.

- For all the 1-Pentanol blends the BTE got reduced and BSEC got increased compared to pure diesel operation.

- The fraction of unburnt hydrocarbon emission got increased with the use of 1-Pentanol with diesel. However, the CO emission is similar to that of diesel fuel operation and a negligible amount of NO_X emission is increased.

4.1.5 Conclusion

The poor fuel characteristics of 1-Pentanol showed poor combustion and emission characteristics. Only at 80% load 1-Pentanol/diesel blends showed some better performance. This is due to higher cylinder temperature which helps for the better combustion of the fuel. Due to incomplete combustion, the HC emission got increased in the exhaust. It is concluded that a lower fraction of 1-Pentanol at high in-cylinder temperature can perform well in the diesel engine.

4.2 Effect of 1-Hexanol/diesel blends on combustion, performance, and emission characteristics of CRDI CI engine.

In this section, the effect of 1-Hexanol/diesel blends on engine characteristics are discussed and compared with base fuel. These are the important parameters that help to understand the behaviour of 1-Hexanol in the diesel engine.

4.2.1 Combustion Characteristics

4.2.1.1 Cylinder Pressure
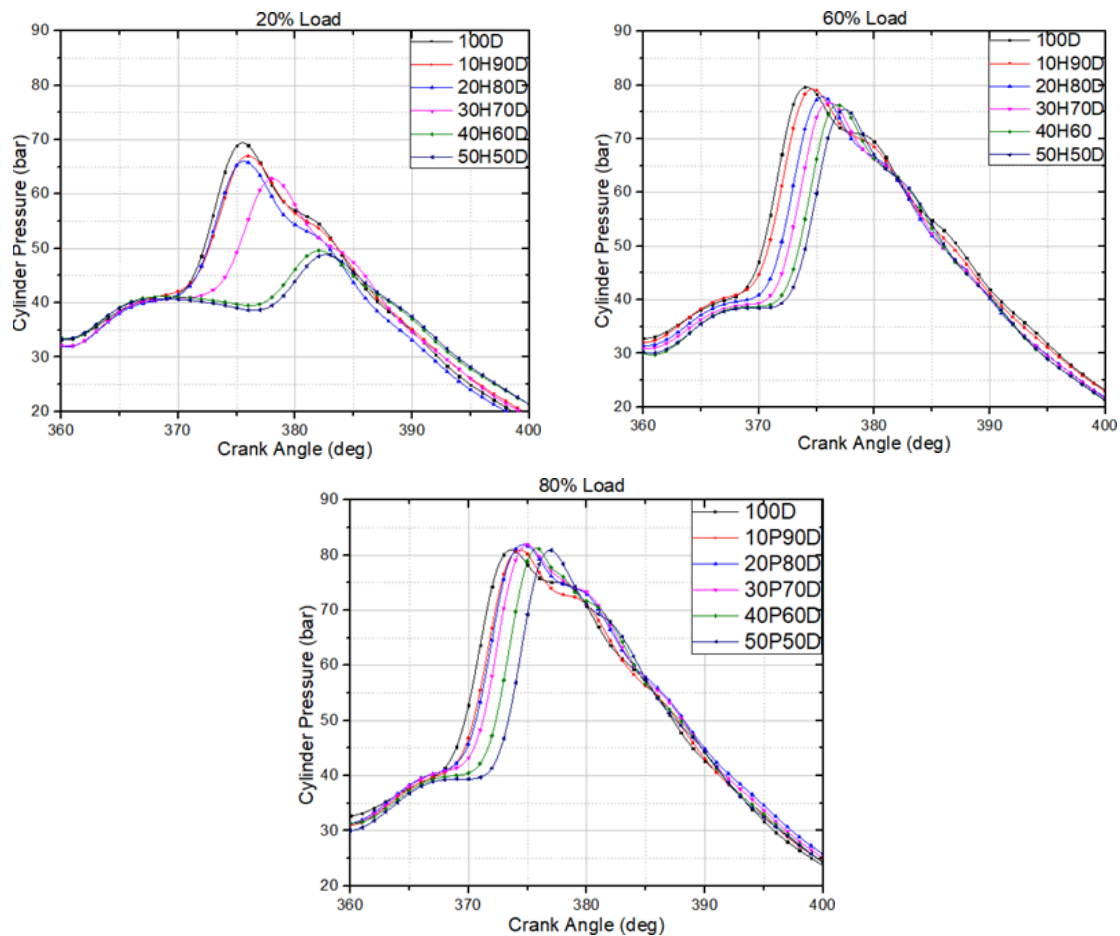
Fig. 4.9 Cylinder Pressure v/s crank angle with various EGR rates

The variation of CP with a crank angle with the addition of 1-Hexanol in the blend is shown in Fig. 4.9. An increment in CP is noted for all 1-Hexanol blends at 80% load compared to pure diesel fuel operation. The maximum CP at 80% load for 100D, 10H90D, 20H80D, 30H70D, 40H60D and 50H50D is 80.71 bar, 81.7 bar, 82.7 bar, 82.2 bar, 82.8 bar, and 81.5 bar respectively. The maximum increment in CP is noted for 40H60D at 80% load which is 2.5% higher compared to diesel fuel. Expect at 80% load, all 1-Hexanol blends experience lower cylinder pressure. As previously explained the combustion characteristics depend on both the physical and chemical properties of the fuel. 1-Hexanol had higher LHE, higher viscosity, lower LHV, low CN, and lower density (De Poures et al. 2017;

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Rajesh Kumar, Saravanan, Rana, and Nagendran 2016). All these properties of 1-Hexanol directly affect the combustion process. At lower engine load the high LHE dominates the cylinder temperature and consumes a higher quantity of heat to evaporate, lower CN prolongs the ID period and allows more fuel to accumulate in the cylinder. Along with these drawbacks, the higher kinematic viscosity of 1-Hexanol causes poor atomization of fuel and injects bigger fuel droplets into the combustion chamber. Whereas, at a higher engine load the higher cylinder temperature reduces the effect of high LHE and also increases the atomization of viscous fuel. The longer ID allows more fuel to accumulate in the combustion chamber and the higher O_2 content of 1-Hexanol helps in the complete combustion of accumulated fuel. Hence at higher engine load 1-Hexanol blends shows a higher peak compared to diesel fuel operation.

4.2.1.2 Mean Gas Temperature
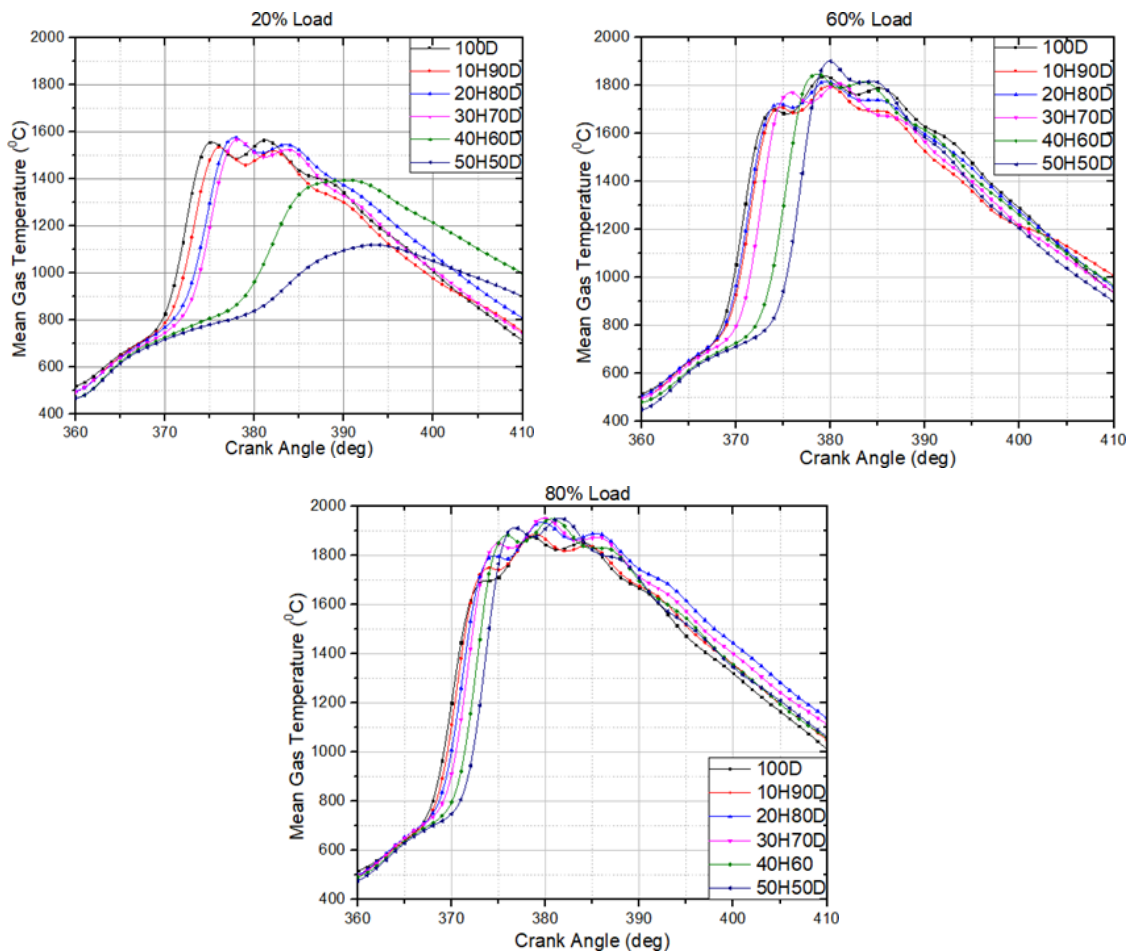
Fig. 4.10 Mean Gas Temperature v/s Crank angle for different test fuels

The variation of MGT with load for all the fuels in the test is shown in Fig. 4.10. The MGT at 80% load for 100D, 10H90D, 20H80D, 30H70D, 40H60D and 50H50D is 1871.71˚C, 1887.17˚C, 1934.03˚C, 1951.9˚C, 1947.1˚C and 1947.11˚C respectively. The maximum

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increment in the MGT is noted for 30H70D which is 4.28% higher compared to pure diesel fuel operation. In comparison with 1-Pentanol, 1-Hexanol had higher MGT, which is due to the improved heating value and cetane number of 1-Hexanol. Ignition delay is associated with physical delay and chemical delay, higher in-cylinder temperature at 80% load reduces the chemical delay by improving chemical reaction. However, there is no change in the physical delay due to higher viscosity and lower CN of 1-Hexanol which allows more fuel to accumulate in the combustion chamber. Due to improved combustion reaction and O_2 in 1-Hexanol improves the combustion and increases the MGT at 80% load. Whereas, at lower and mid-load conditions, the lower cylinder temperature and higher viscosity of 1-Hexanol deteriorate the combustion process.

4.2.1.3 Net Heat Release Rate
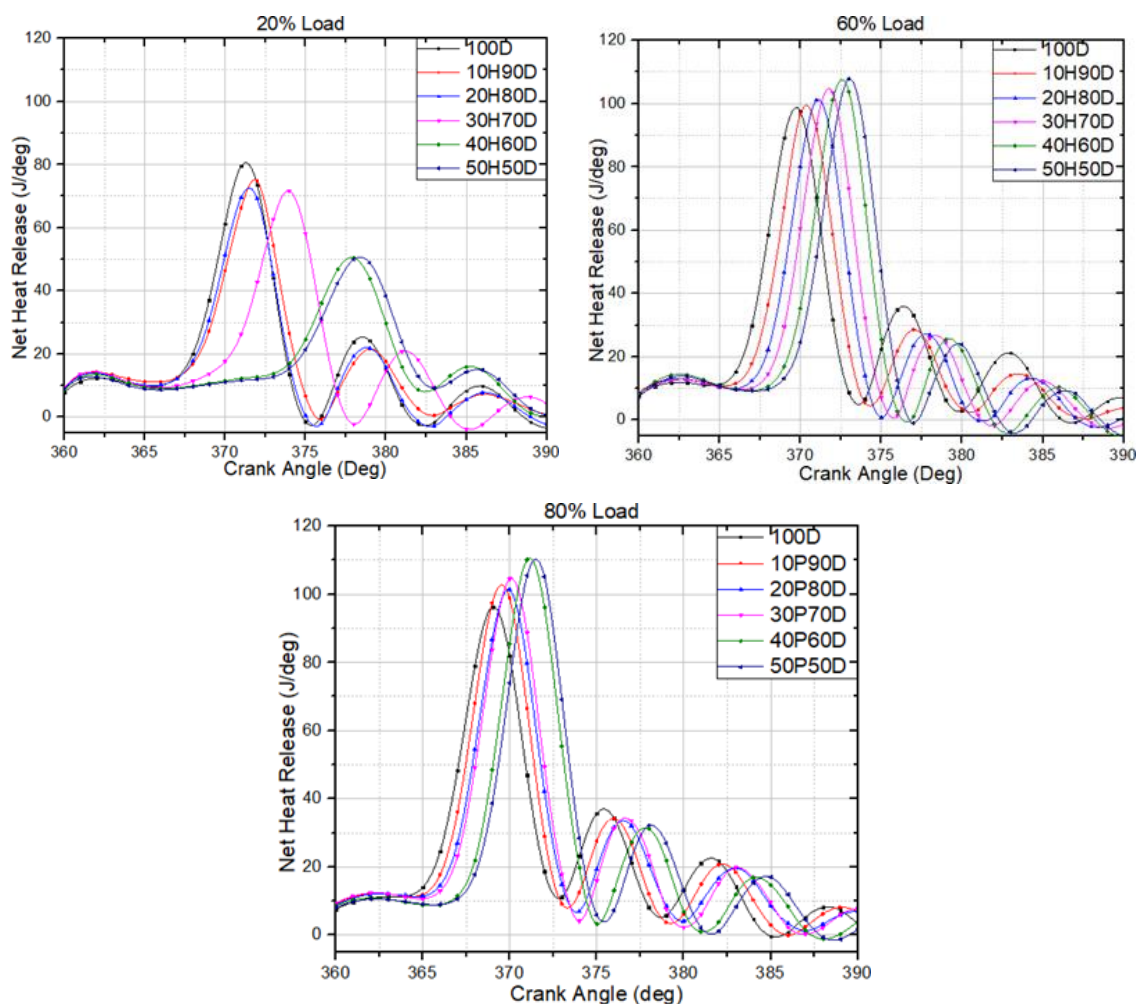
Fig. 4.11 Net Heat Release rate v/s Crank angle for different test fuels

The variation of NHR with respect to crank angle with the addition of 1-Hexanol at 80% load for all fuels in the test is shown in Fig. 4.11. It is noted from the graph that an improvement in NHR is observed with the addition of 1-Hexanol, increment in NHR increases with an increase in the fraction of 1-Hexanol in the blend. Compared to lower

load the increment in NHR is improved at higher engine load. The OH radicals in the alcohol are not so active at low temperature which reduce the rate of reaction in the premixed combustion zone (Chen et al. 2019b; Mathan Raj et al. 2018). The NHR at 80% load for 100D, 10H90D, 20H80D, 30H70D, 40H60D and 50H50D is 96.27 J/deg, 99.66 J/deg, 104.66 J/deg, 109.52 J/deg, 113.52 J/deg and 115.7 J/deg respectively. The ID mainly depends on the ignition quality of the fuel which includes CN, oxygen content, and kinematic viscosity of the fuel. The lower CN and higher oxygenated nature of 1-Hexanol blends increase the delay period. Due to increased delay, more fuel will be accumulated and higher oxygen content helps for improved combustion in the premixed combustion zone which leads to an increase in the NHR rate at a higher load (Rajesh Kumar, Saravanan, Rana, Anish, et al. 2016). At a higher load, compared to diesel fuel the consumption of 1-Hexanol is more due to lower CN, hence more fuel is accumulated and at a higher engine load, a suitable environment is there to burn the accumulated fuel which increases the NHR at higher engine load. However, at a lower engine load, higher viscosity reduces the formation of the suitable combustible mixture due to poor atomization (Nour et al. 2019a), hence less fuel burnt in the premixed combustion zone.

4.2.2 Performance Characteristics

4.2.2.1 Brake Thermal Efficiency
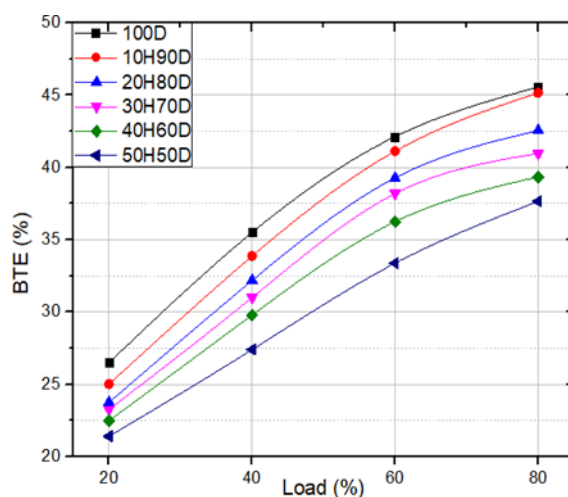
Fig. 4.12 BTE v/s engine load for different test fuels

The variation of BTE as a function of load is shown in Fig. 4.12. The 1-Hexanol blends showed lower BTE, the highest percentage of decrement in the BTE is seen for 50H50D, which is 22.86%, at 40% engine load compared to pure diesel fuel operation. The BTE decreases with a higher fraction of 1-Hexanol in the blend. At 80% engine load the decrement in BTE for 10H90D, 20H80D, 30H70D, 40H60D and 50H50D is 0.90%, 6.63%, 10.10%, 13.68% and 17.37% lower respectively compared to pure diesel operation. The

main reason for the decrement in the BTE for 1-Hexanol blends is low LHV compared to diesel fuel. Along with low LHV, higher LHE, low CN, and high viscosity of 1-Hexanol are equally responsible for poor BTE. 1-Hexanol is an oxygenated fuel, the heating value of any oxygenated fuel will decrease with an increase in the oxygen atom in the fuel (Atmanli 2016). Poor premixed combustion is experienced with 1-Hexanol blends, which is due to the higher viscosity of the blend, large fuel droplets will inject into the combustion chamber which will take more time for evaporation (Verma et al. 2019). On the other hand, the water content of 1-Hexanol decreases the fuel vaporization rate and consumes more heat to vaporize. This endothermic reaction creates cold spots in the cylinder and decreases the propagation of flame speed. Hence the small quantity of charge will be burnt in the premixed combustion zone and the whole combustion shifts to diffusion combustion. In diffusion combustion, the time for complete combustion of charge is much less due to longer premixed combustion. Also, the diffusion combustion extends till the completion of the expansion stroke, due to longer CD further loss of heat to the various engine parts; this decreases the thermal efficiency of the engine. Sundar Raj et al. shows that BTE increases with the addition of 1-Hexanol in their studies (Sundar Raj and Saravanan 2011).

4.2.2.2 Brake Specific Energy Consumption

The effect of 1-Hexanol on BSEC at all loading conditions for various fuel blends in the test is shown in Fig. 4.13. Higher fuel consumption is noticed for 50H50D and lower fuel consumption is noticed for 10H90D compared to other fuel blends in the test. At 80% load the increment in the fuel consumption for 10H90D, 20H80D, 30H70D, 40H60D and 50H50D is 1.003%, 6.73%, 10.13%, 13.77% and 17.37% higher respectively compared to pure diesel fuel operation. The fuel consumption of the engine is based on the fuel properties, like heating value, CN, and LHE. 1-Hexanol has a lower heating value, low CN, and higher LHE, which affects the combustion negatively and reduces the amount of energy liberated during the combustion process. As previously explained the viscosity of the fuel also causes a negative effect on vaporization due to larger fuel droplets. This causes improper air/fuel mixture, which affects negatively on premixed combustion (Nour et al. 2019a). Due to the large CD, most of the heat from combustion is lost to the engine parts. Also, the water content of alcohol affects the combustion process by decreasing the evaporation rate of the fuel, hence most of the fuel comes out in the exhaust without proper burning. All these causes are responsible for poor BSEC of 1-Hexanol blends.
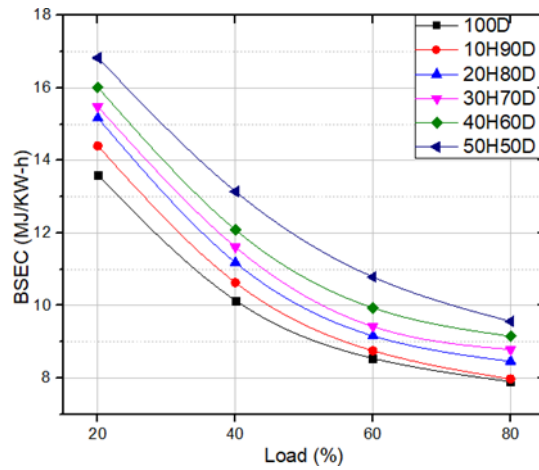
Fig. 4.13 BSEC v/s engine load for different test fuels

4.2.3 Emission Characteristics

4.2.3.1 Hydrocarbon Emission
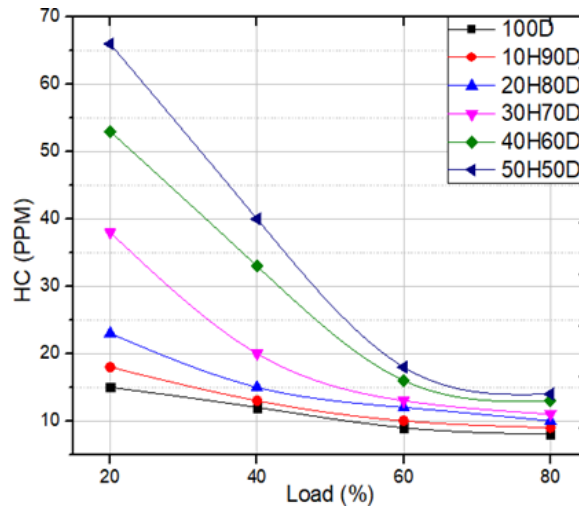
Fig. 4.14 HC emission v/s load for different test fuels

Fig. 4.14 shows the variation of HC concentration in the exhaust for all the fuels in the test. It is seen from the graph that higher HC emission is observed for the 1-Hexanol blend at low load and its concentration increases with a higher fraction of 1-Hexanol in the blend. However, the HC concentration in the exhaust reduces at higher engine load for all the fuel blends in the test. Increase in HC emission for 10H90D, 20H80D, 30H70D 40H60D and 50H50D at 80% load is 11.11%, 20%, 27.27%, 38.46% and 42.85% respectively compared to pure diesel fuel operation. This can be attributed that, 1- Hexanol is a high viscous fuel, during the injection process larger fuel droplets and more fuel is being injected into the combustion chamber. The larger fuel droplets are not completely vaporized and a higher concentration of fuel is accumulated in the cylinder (Catalu and Silva 2012). On the other hand, 1-Hexanol had higher LHE, and it contains some quantity of water with it (Huang et al. 2018; Pan, Huang, Liao, Jia, Zhou, and Huang 2019). All these factors hinder the

complete vaporization of the fuel droplets and quench the cylinder wall. This results in a poor combustion characteristic both in the premixed combustion phase and the controlled combustion phase and reduces the speed of the flame. Even though 1-Hexanol is an oxygen-rich fuel, its HC emission in the exhaust gas is high, accumulation of more fuel in the premixed combustion due to longer ID, poor combustion due to higher viscosity, and loss of heat to the surrounding due to longer CD, all these factors negatively affect the combustion which hinders the effect of oxygen on combustion. Also, the lower CN of 1-Hexanol prolongs the CD which weakens the self-ignition characteristic of 1-Hexanol and quenches the cylinder wall resulting in a higher HC emission (De Poures et al. 2017). Poures et al. show similar results for 1-Hexanol/diesel blends in their experiment (De Poures et al. 2017).

4.2.3.2 Carbon Monoxide Emission
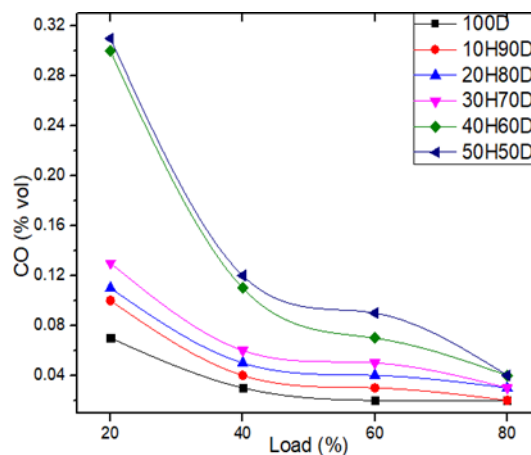
Fig. 4.15 CO emission v/s load for different test fuels

The variation of CO emission with engine load for different fuel blends is shown in Fig. 4.15. At 80% load the increment in CO emission for 10H90D, 20H80D, 30H70D, 40H60D and 50H50D is 0%, 33.33%, 33.33%, 50% and 50% higher respectively compared to baseline readings. The highest CO emission is noted for 50H50D at 20% load which is 77.77% higher compared to diesel fuel operation. The oxygenated nature of 1-Hexanol reduces the CN and calorific value. The lower CN of alcohol increases the delay period, due to the longer delay period more fuel will be accumulated in the cylinder. The higher viscosity reduces the fuel atomization and higher LHE reduces the fuel vaporization rate. Hence most of the fuel in the premixed combustion zone is unburnt and forms CO. Also, the O_2 in the 1-Hexanol fuel structure participated in the formation of CO emission. According to Westbrook et al. chemical kinetics modelling for oxygenated fuels, during the combustion process, the oxygen atoms in the fuel make a strong bond with the C atom

to form CO, and this CO atom comes out in the exhaust without oxidation to CO_2 (Westbrook et al. 2006b). Lower in-cylinder temperature for 1- Hexanol blends is also a reason for the formation of higher CO emission.

4.2.3.3 Nitrogen Oxide Emission
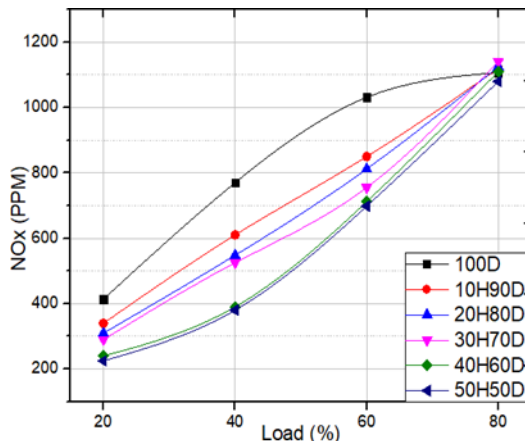
Fig. 4.16 NO_X emission v/s load for different test fuels

The change in NO_X emission with engine load for all the fuel blends in the test is shown in Fig. 4.16. At 60% load, the reduction in NO_X for 10H90D, 20H80D, 30H70D, 40H60D, and 50H50D is 17.55%. 21.24% 26.77%, 30.94% and 32.29% lower respectively compared to diesel fuel operation. Diesel fuel has higher CN compared to 1-Hexanol, which reduces the ID and helps in the early start of the combustion, which increases the combustion temperature, which leads to higher NO_X emission. The fuel properties of 1-Hexanol are suitable to produce low NO_X emission, NO_X is a function of combustion temperature. The lower CN prolongs the ID period, which allows the accumulation of more fuel in the combustion chamber, higher LHE traps more heat for evaporation. Also, the higher viscosity leads to poor atomization in a cylinder, low-temperature premixed combustion happens which results in a low NO_X emission (Ickes et al. 2009) and water content of higher alcohol consumes heat for disassociation of water molecule. All these poor fuel properties of 1-Hexanol deteriorate the combustion. The lower CN prolongs the ID which extends the combustion duration and the higher viscosity allows more quantity of fuel to be entered into the combustion chamber with a bigger fuel droplet size. Due to this effect the combustion duration extends till the piston reaches BTDC, due to longer CD most of the heat is lost to the surrounding, this reduces the in-cylinder temperature. The overall drawback of 1-Hexanol combustion characteristics results in a lower NO_X emission. Whereas at 80% load increase in NO_X emission is noted for all 1-Hexanol blends, this is due to high cylinder temperature and long residence time. At 80% load more fuel is drawn and higher cylinder temperature reduces the negative effect of low LHE and O_2 of 1-

Hexanol supporting the combustion positively. All these elements lead to an increase in NOx emission at 80% load. Increment in NOx at 80% load for 10H90D, 20H80D, 30H70D, and 40H60D is 1.1%, 1.6%, 2.8%, and 0.27% higher respectively compared to pure diesel operation.

4.2.4 Summary

- 1-Hexanol exhibits poor combustion characteristics. At higher load conditions there was an improvement in the combustion characteristics for a lower fraction of 1-Hexanol in the blend.
- The performance characteristics of the engine decrease with an increase in the fraction of 1-Hexanol in the blend.
- The emission characteristics are increased with an increase in the fraction of 1-Hexanol in the blend.

4.2.5 Conclusion

From the investigation it is noted that, 1-Hexanol performs well up to 30% of 1-Hexanol fraction and performance becomes worsen at higher fraction of 1-Hexanol in the blend. Due to higher ignition delay more amount of fuel could accumulate that may increase the combustion chamber temperature leads to higher NO_X emissions. It is concluded that lower fraction of 1-Hexanol at higher load gives comparable results with diesel fuel operation.

4.3 Comparative study of 1-Pentanol and 1-Hexanol blends on combustion, performance, and emission characteristics of CRDI CI engine.

In this section comparative study of 1-Pentanol and 1-Hexanol is shown. From the previous results, it was observed that better combustion, performance, and emission are noted up to 30% of higher alcohol with diesel. Further increment in the percentage of alcohol, deteriorates the combustion at lower and medium load conditions. Hence for the comparative purpose, only 30P70D and 30H70D blends are used.

4.3.1 Combustion Characteristics

4.3.1.1 Cylinder Pressure
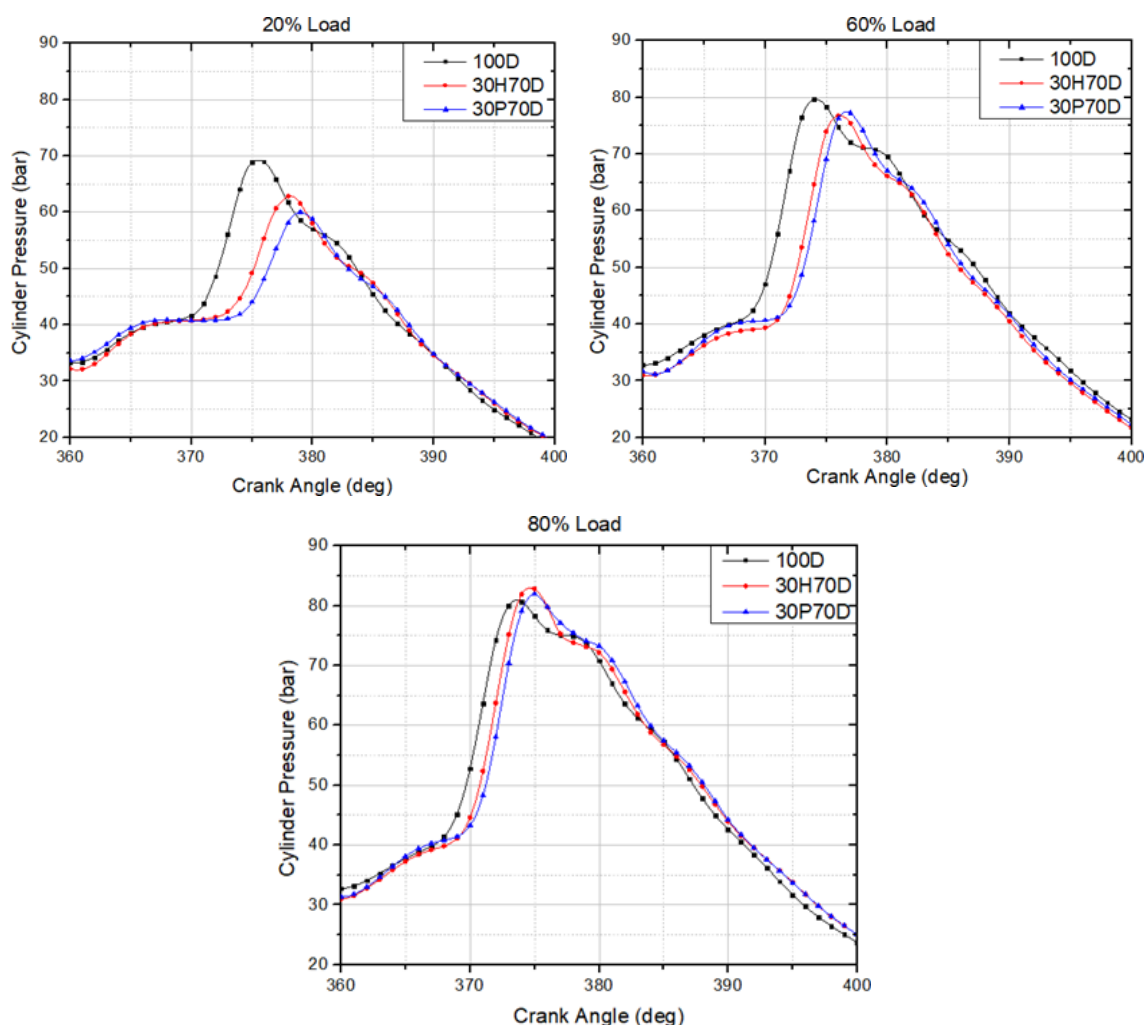
Fig. 4.17 Cylinder Pressure v/s Crank angle for different test fuels

The comparative effect of 1-Pentanol and 1-Hexanol on cylinder pressure is shown in Fig. 4.17. The trend of cylinder pressure for both the alcohols is similar to that of diesel fuel. Higher peak pressure is noted for 30H70D compared to neat diesel and 30P70D. At 80% load 1-Hexanol shows a 1.07% increment in CP compared to 30P70D. The fuel properties of 1-Hexanol are higher compared to 1-Pentanol. Higher LHV, CN, and lower LHE

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improve the combustion compared to 1-Pentanol. The only drawback for 1-Hexanol is higher viscosity compared to 1-Pentanol, which may prolong the ID and allow more fuel to accumulate in the combustion chamber.

4.3.1.2 Mean Gas Temperature
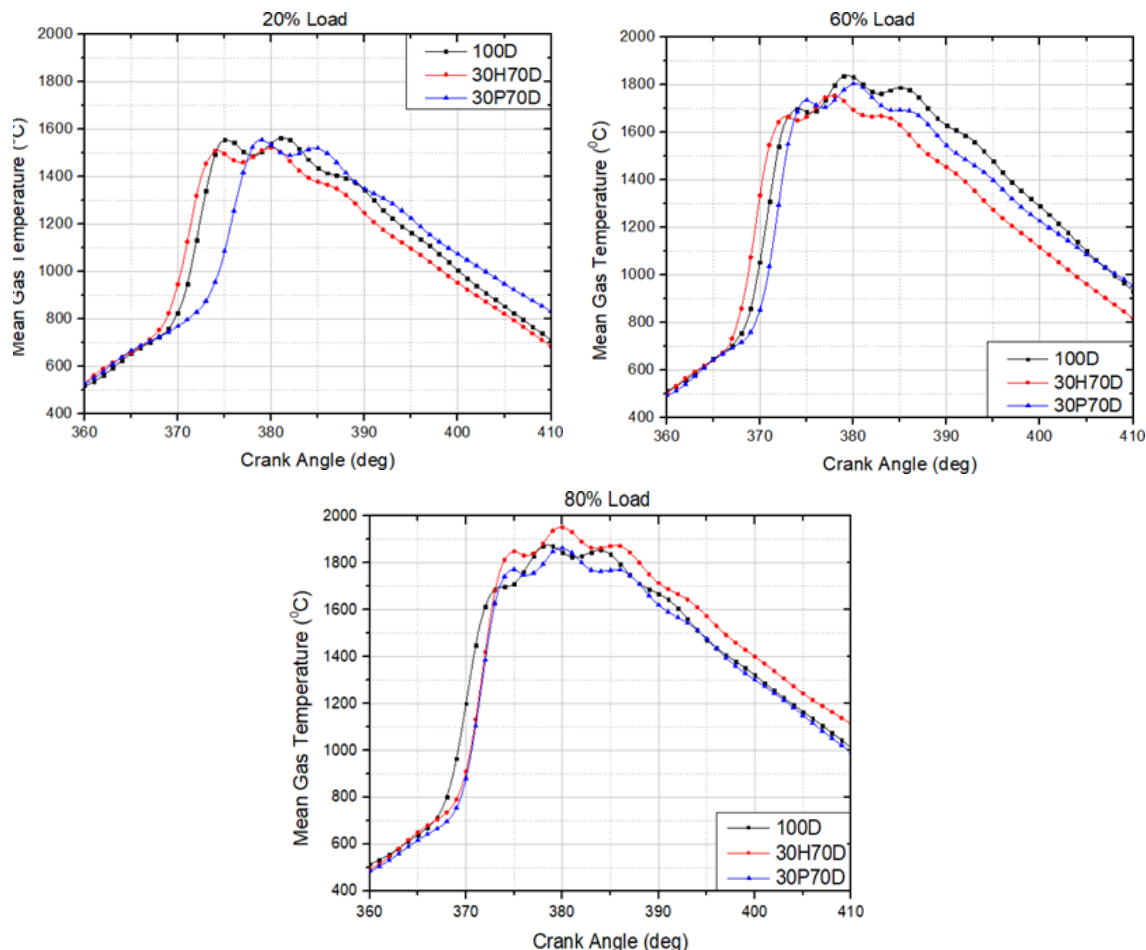
Fig. 4.18 Mean Gas Temperature v/s Crank angle for different test fuels

The comparative effect of 1-Pentanol and 1-Hexanol on mean gas temperature is shown in Fig. 4.18. It is noticed from the graph that both the alcohol blends have similar trends of MGT. The peak MGT of 1-Hexanol is near to TDC whereas, the peak for 1-Pentanol is moved away from the TDC. It clears the impact of the delay period and combustion duration on MGT. Higher peak pressure is noted for 30H70D compared to neat diesel and 30P70D. At 80% load 1-Hexanol shows a 4.06% increment MGT compared to 30P70D.

4.3.1.3 Net Heat Release Rate

The comparative effect of 1-Pentanol and 1-Hexanol on cylinder pressure is shown in Fig. 4.19. From the graph, it is noted that both the fuel blends have similar NHR patterns. At 20% load, both 1-Pentanol and 1-Hexanol showed a lower NHR rate compared to diesel

fuel. At 60% load, the peak NHR rate of 1-Pentanol is equal to 1-Hexanol. However, at 80% load, higher peak pressure is noted for 30H70D compared to neat diesel and 30P70D. At 80% load 1-Hexanol shows a 4.5% increment in net heat release rate compared to 30P70D.
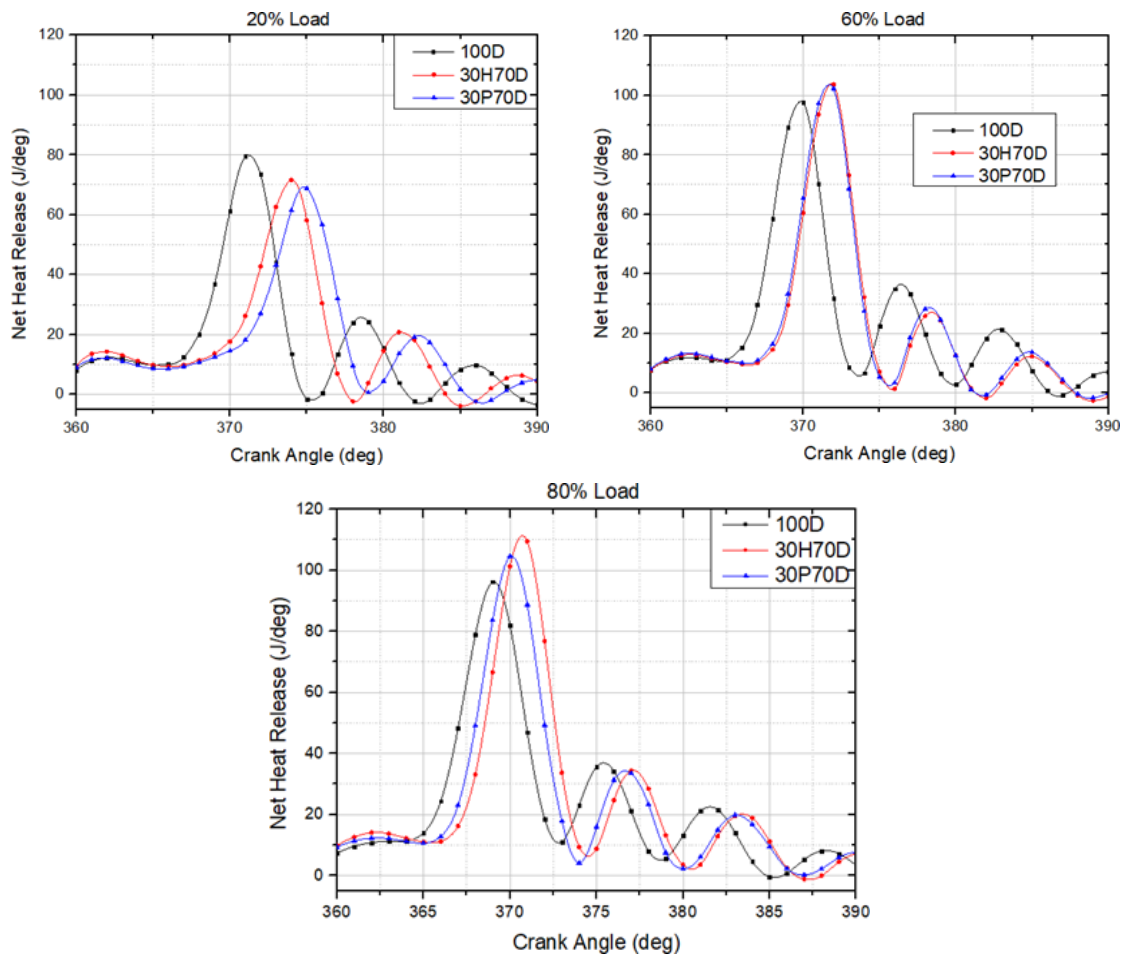
Fig. 4.19 Net Heat Release rate v/s Crank angle for different test fuels

4.3.2 Performance characteristics
4.3.2.1 Brake Thermal Efficiency
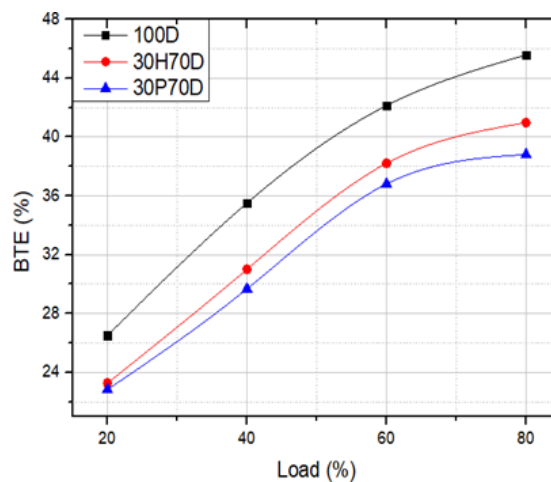
Fig. 4.20 BTE v/s engine load for different test fuels

The variation of BTE as a function of load for 100D, 30P70D, and 30H70D is shown in Fig. 4.20. The BTE increases with an increase in load. BTE trend for 1-Pentanol and 1-Hexanol is similar to diesel fuel. 1-Hexanol blend had higher BTE compared to 1-Pentanol blends at all loading conditions. At 80% load, the BTE of 30H70D is 5.29% higher compared to the BTE of 30P70D. Both the alcohol has similar patterns of BTE. 1-Hexanol blends showed higher BTE compared to 1-Pentanol blends due to higher heating value and cetane number. The higher CN reduces the ignition delay and the higher heating value increases the thermal efficiency of the engine.

4.3.2.2 Brake Specific Energy Consumption
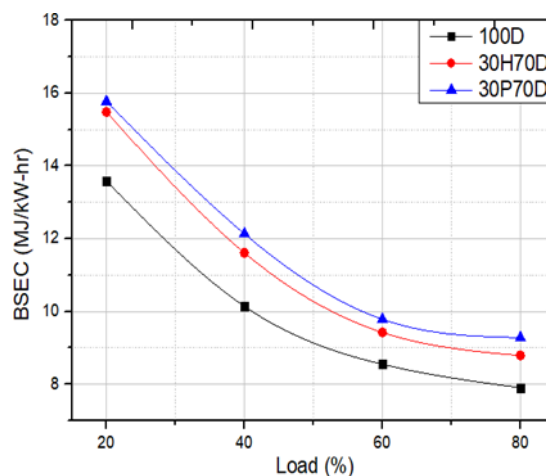
Fig. 4.21 BSEC v/s engine load for different test fuels

The variation of BSEC as a function of load for 100D, 30P70D, and 30H70D is shown in Fig. 4.21. The BSEC decreases with an increase in load. BSEC trend for 1-Pentanol and 1-Hexanol is similar to diesel fuel. 1-Hexanol blend had lower BSEC compared to 1- Pentanol blends at all loading conditions. At 80% load, the reduction in BSEC is noted for 30H70D which is 5.3% lower compared to 30P70D.

4.3.3 Emission Characteristics

4.3.3.1 Hydrocarbon Emission

Fig. 4.22 shows the variation of HC concentration in the exhaust for 100D, 30H70D and 30P70D. Both 30H70D and 30P70D show similar HC emission trends. It is seen from the graph that higher HC emission is observed for the 1-Pentanol blend at all loading condition compared to 1-Hexanol. At 80% load, higher HC emission is noted for 30P70D which is 8.33% higher compared to 30H70D. The poor fuel properties of 1-Pentanol compared to 1-Hexanol is the reason for higher HC emission.
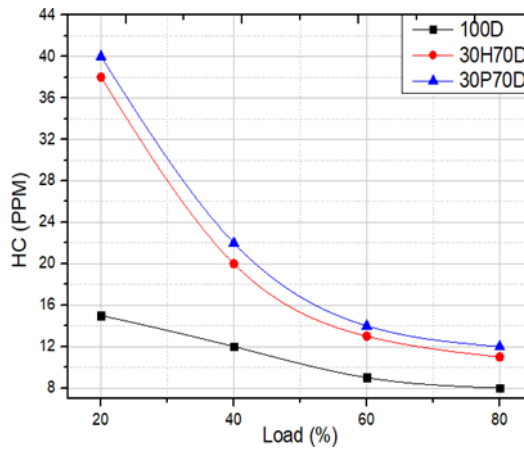
Fig. 4.22 HC Emission v/s engine load for different test fuels

4.3.3.2 Carbon Monoxide Emission
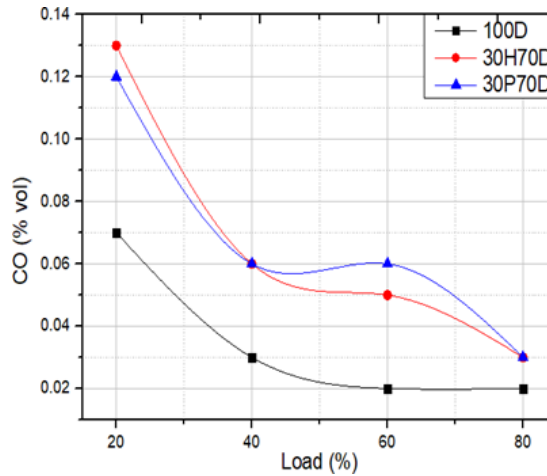
Fig. 4.23 CO Emission v/s engine load for different test fuels

The variation of CO emission with engine load for 100D, 30P70D, and 30H70D is shown in Fig. 4.23. At 40% and 80% load no changes are found in CO emission for both the alcohols blend. The main reason for CO emission is incomplete combustion due to the unavailability of oxygen. Both the alcohols have oxygenated in nature and have a similar trend in CO emission.

4.3.3.3 Nitrogen Oxide Emission

Variation of NO_X with load for 100D, 30P70D, and 30H70D is shown in Fig. 4.24. The NO_X increases with an increase in load. The NO_X trend for both the alcohols is similar. At 80% load, it is seen that alcohol blends have higher NO_X emissions compared to diesel fuel. It is attributed that a longer ignition delay, which allows the accumulation of more quantity of fuel in the combustion chamber. At 80% load, the cylinder temperature is high enough to burn all the accumulated fuel hence its NO_X got increased. There is no notable difference in NOx noted for both 30P70D and 30H70D at 80% load.
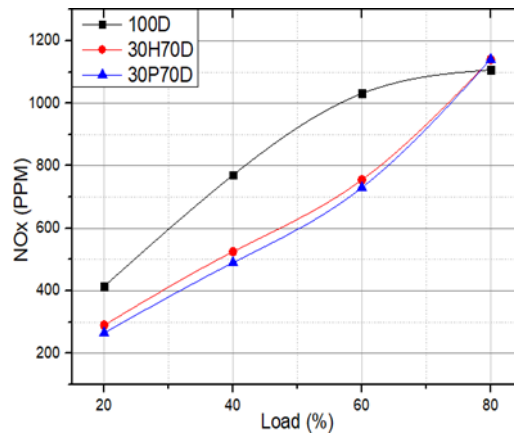
Fig. 4.24 NOx Emission v/s engine load for different test fuels

4.3.4 Summary

- At 80% load 1-Hexanol shows a 1.07% increment in-cylinder pressure compared to 30P70D.
- The peak MGT of 1-Hexanol is near to TDC whereas; the peak for 1-Pentanol is moved away from the TDC. It clears the impact of the delay period and combustion duration on MGT.
- At 80% load, the BTE of 30H70D is 5.29% higher compared to the BTE of 30P70D.
- At 80% load, higher HC emission is noted for 30P70D which is 8.33% higher compared to
- 30H70D. Both the alcohol blends showed a similar trend for CO and NO_X emissions.

4.3.5 Conclusion

From the comparative study, it was noted that 1-Hexanol performs well in the combustion, emission, and performance characteristics of the engine compared ton1-Pentanol blend. The ignition delay of 1-Hexanol is low compared to 1-Pentanol, hence better combustion happens. Both the alcohol performs well at a higher load, where sufficient temperature to vaporize the accumulated fuel. It is concluded that compare to 30P70D, 30H70D performs well and for the future study, 30% of the 1-Hexanol fraction is considered.

4.4 Effect of injection time on combustion, performance, and emission characteristics of CRDI CI engine for the optimized alcohol blend (30H70D)

In this section, the effect of advanced and retarded injection time on engine characteristics is discussed and compared with the standard injection time. These are the important parameters that help to understand the behaviour of the fuel at various injection times.

4.4.1 Combustion Characteristic

4.4.1.1 Cylinder Pressure
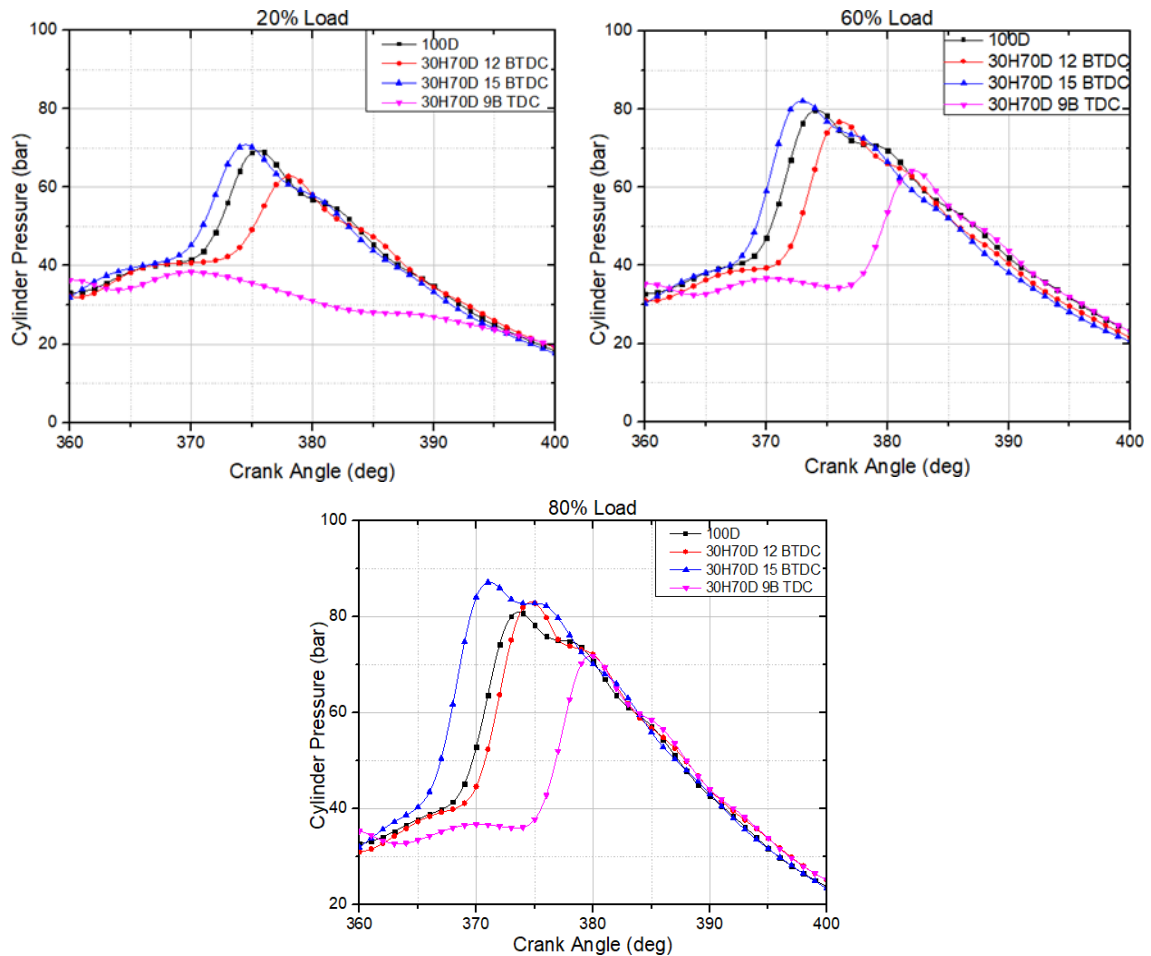
Fig. 4.25 Cylinder pressure v/s Crank angle for different injection angle

The effect of change in fuel injection timing on cylinder pressure with crank angle is shown in Fig. 4.25. The maximum peak cylinder pressure at all loading conditions is noted for advanced injection time (15˚BTDC) compared to standard (12˚BTDC) and late injection (9˚BTDC). The peak cylinder pressure of 30H70D at 80% load for 12˚BTDC, 15˚BTDC, and 9˚BTDC is 82.8 bar, 87.17 bar, and 71.79 bar respectively. Compared to the standard injection angle (12˚BTDC), early injection gives the highest peak pressure. The lower CN and higher viscosity of 1-Hexanol increase the delay period, this allows the accumulation of more quantity of fuel in the combustion chamber (Nour et al. 2019a). Due to an increase

in the delay period the overall combustion duration gets increases. When the fuel is injected at standard injection time (12ºBTDC), due to a prolonged delay period only a small amount of fuel is burnt in the premixed combustion phase and the maximum amount of charge is moved to the diffusion combustion phase. In diffusion combustion due to a large amount of unburnt hydrocarbon incomplete combustion occur and the fuel/air charge is moved to late combustion. The prolonged delay period allows the maximum amount of fuel to get burned in the expansion stroke, it is evidenced that the peak cylinder pressure at both 12˚BTDC and 9˚BTDC shifts away from the TDC. Whereas, at advanced injection time (15˚BTDC) the peak pressure shifts towards TDC. When the fuel is injected before the completion of the compression stroke (15˚BTDC), it had more time to evaporate and form a homogeneous mixture with air. When the mixture attains the self-ignition temperature it starts to burn and the peak pressure shifts toward TDC.

4.4.1.2 Mean Gas Temperature

Fig. 4.26 shows the effect of change in injection timing on MGT at 20%, 60%, and 80% load. At 60% load the MGT for 30H70D at 12˚BTDC, 15˚BTDC and 9˚BTDC is 1807.87˚C, 1753.44˚C and 1851.51˚C respectively. It is noted that at 60% load there is a reduction in MGT at advanced injection time compared to both standard and late injection. As previously explained 1-Hexanol blend had a higher ignition delay compared to diesel fuel, which makes the accumulation of a high volume of fuel into the combustion chamber. Due to an increase in the delay period, the premixed combustion phase also extends and only a small quantity of fuel is burnt and combustion is extended to the diffusion combustion phase. In the diffusion combustion phase, incomplete combustion occurs due to the high volume of unburnt hydrocarbon. Hence most of the fuel mixture gets burnt in expansion stroke which is not accounted as useful energy. Also, when the injection happens nearer to TDC (9˚BTDC), high temperature and pressure are experienced near TDC (Mochida et al. 1991), which makes the rapid evaporation of accumulated fuel and start burning instantaneously. When the injection is advanced to 15˚BTDC, an early start of combustion avoids the loss of fuel burning in the expansion stroke and also reduces combustion duration and delay period. Due to the reduced delay period, a lower amount of fuel gets accumulated and low LHV of 1- Hexanol produces lower heat in the cylinder. On the other hand, at 80% load, an increment in MGT is noted for the early start of injection. At 80% load the MGT for 30H70D at 12˚BTDC, 15˚BTDC and 9˚BTDC is 1951.9˚C, 1957.7˚C and 1879.92˚C respectively. This is attributed that at 80% load higher in-cylinder

temperature increases the fuel vaporization rate and makes a homogeneous combustible mixture and consumption of a higher amount of fuel are responsible for higher MGT.
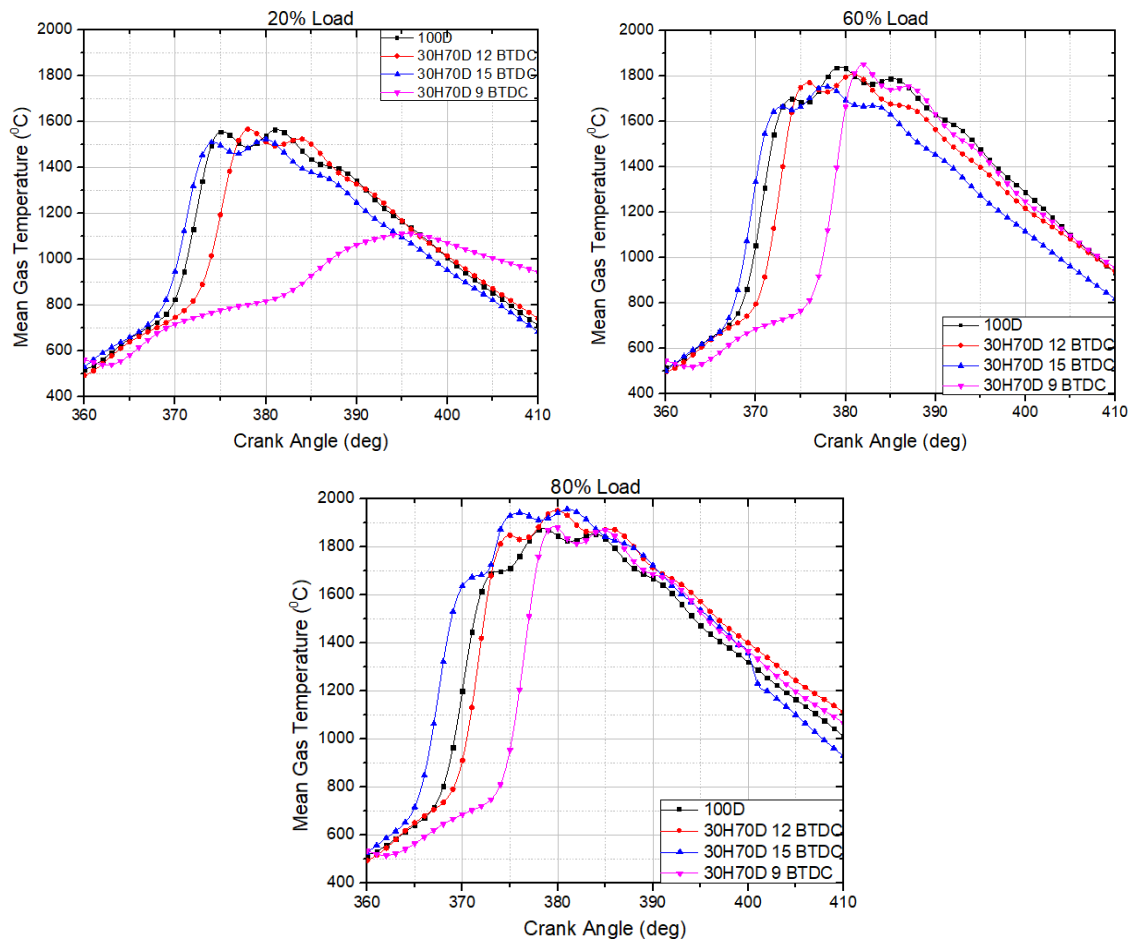
Fig. 4.26 Mean Gas Temperature v/s Crank angle for different injection angle

4.4.1.3 Net Heat Release Rate

The change in NHR rate at 20%, 60%, and 80% load for different injection timing is shown in Fig. 4.27. At 60% and 80% load higher NHR rate is noted for late injection timing (9˚BTDC) compared to standard and advanced injection. At the end of compression or near TDC the in-cylinder pressure and temperature are higher (Mochida et al. 1991), when the injection is done at this point, fuel will rapidly evaporate and form an ignitable mixture. Hence, at the late injection, the NHR rate is higher compared to the other two injection timing. However, it is noted that during the late injection timing the peak heat release rate moves away from TDC. It is attributed that, higher viscosity and lower CN prolongs the delay period, which leads increase in the combustion duration, combustion mixture burns during expansion stroke (Pan et al. 2018b). Hence the NHR peak shifts away from TDC. Even though, the higher peak was obtained for late injection timing which is not useful in terms of brake thermal efficiency. At 80% load the NHR rate for 30H70D at 12˚BTDC, 15˚BTDC and 9˚BTDC is 109.52 J/deg, 95.21 J/deg and 110.59 J/deg respectively. The

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NHR at 80% load obtained at 15˚BTDC is nearer to diesel fuel operation. A similar kind of result for higher alcohol at standard injection is obtained for Kumar et al. (Rajesh Kumar, Saravanan, Rana, and Nagendran 2016).
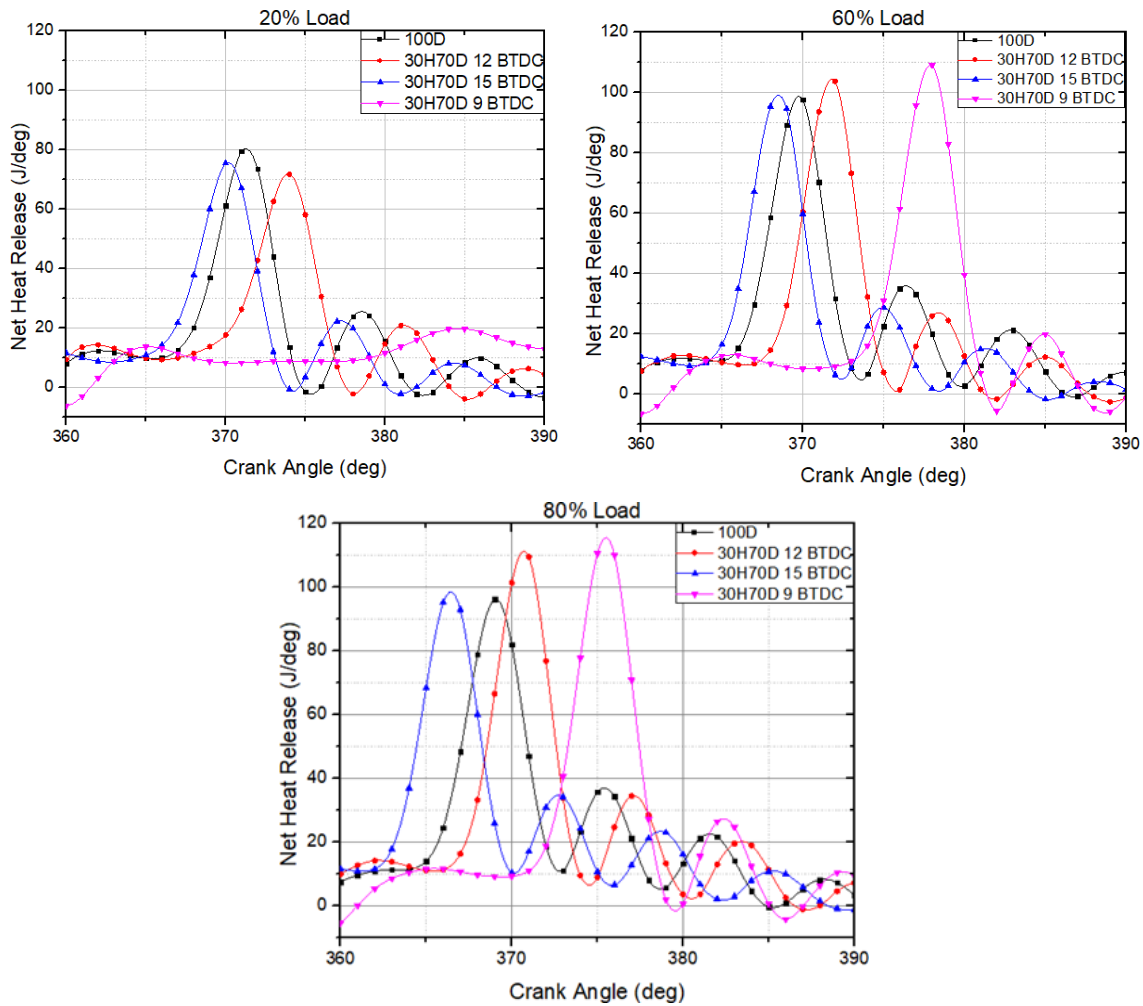
Fig. 4.27 Net Heat Release Rate v/s Crank angle for different injection angle

4.4.2 Performance Characteristics
4.4.2.1 Brake Thermal Efficiency
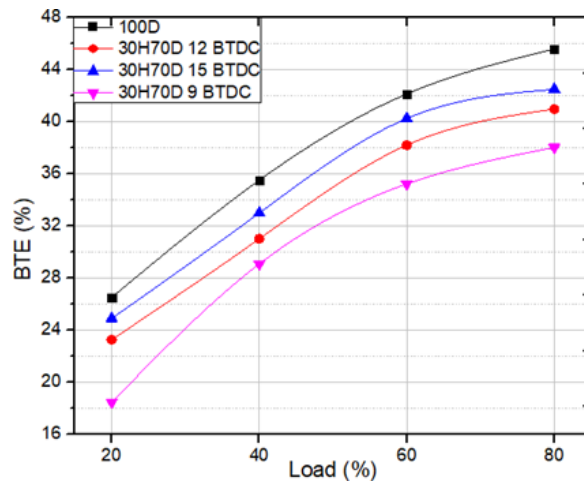
Fig. 4.28 Change in BTE with load for different injection angle

The variation in BTE with load for 30H70D at different injection angles is shown in Fig. 4.28. The higher BTE is noted for 15°BTDC compared to standard and late injection. At 80% load BTE for 30H70D at 12°BTDC, 15°BTDC and 9°BTDC is 12.23%, 1.77% and 24.75% lower compared to diesel fuel operation at standard injection time. The ID period for alcohols is higher compared to diesel fuel due to lower CN, higher viscosity, higher oxygen content, and presence of water molecules (Damodharan, A.P. Sathiyagnanam, et al. 2018). Due to a longer delay period, the fuel preparation phase takes much more time than usual which leads to an increase in combustion duration. When the fuel is injected at 9°BTDC, due to prolonged combustion duration the fuel combustion extends to expansion stroke and there is a loss of heat energy. Whereas, at 15°BTDC due to the early start of the injection fuel preparation phase have sufficient time and the combustion peak is nearer to TDC. Hence maximum thermal efficiency can be obtained at 15°BTDC.

4.4.2.2 Brake Specific Energy Consumption
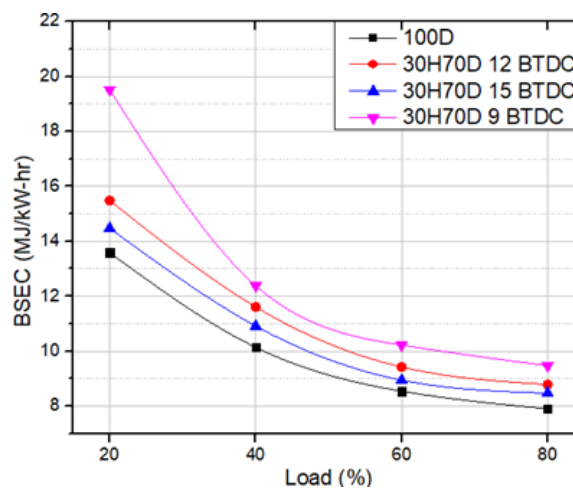
Fig. 4.29 Change in BSEC with load for different injection angle

The variation in BSEC with load at all injection times for 100D and 30H70D is shown in Fig. 4.29. The consumption of fuel decreases with an increase in the engine load. At 80% load BSEC for 30H70D at 12°BTDC, 15°BTDC and 9°BTDC is 12.27%, 1.8% and 24.8% higher compared to diesel fuel operation at standard injection time. The accumulation of more quantity of fuel due to prolonged delay and improper combustion due to higher LHE reduces the thermal efficiency which results in a higher BSEC at all engine loads. The fuel consumption mainly increases due to the lower heating value of 1-Hexanol which consumes more fuel to produce the same amount of heat (Yilmaz and Atmanli 2017). It is noted that lower BSEC is noted for 15°BTDC compared to 12°BTDC and 9°BTDC. When the fuel is injected at 9°BTDC due to a prolonged delay period, most of the fuel goes out through the exhaust without burning which may be the possible reason for higher BSEC.

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4.4.3 Emission Characteristics

4.4.3.1 Hydrocarbon Emission
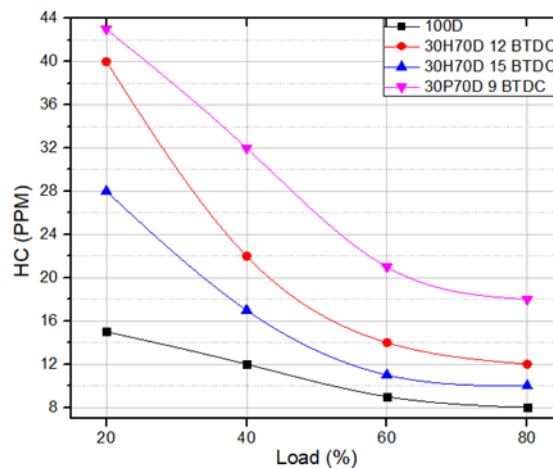
Fig. 4.30 Effect of injection timing on HC

The variation in HC emission with load at all injection times for 100D and 30H70D is shown in Fig. 4.30. At 80% load HC emission for 30H70D at 12˚BTDC, 15˚BTDC and 9˚BTDC is 33.33%, 20% and 55.55% higher compared to neat diesel fuel operation. Compared to standard and late injection time, the advanced injection time has lower HC emission, which is due to the complete combustion of accumulated fuel. At standard and late injection time, the fuel preparation phase got delayed due to higher LHE and lower CN, which increases the premixed combustion phase where only a small amount of fuel is burned and whole combustion shifts to diffusion combustion there also incomplete combustion occur due to higher amount of unburnt hydrocarbon deposit. This leads to incomplete combustion most of the unburnt fuel comes out through the exhaust. Also, the quenching of cylinder walls due to loss of heat to the surrounding due to prolonged combustion duration and high LHE takes more heat to evaporate are responsible for higher HC emissions (Nanthagopal et al. 2018; Yilmaz et al. 2018).

4.4.3.2 Carbon Monoxide Emission

The variation of CO emission at all loading conditions for all the fuels in the test is shown in Fig. 4.31. At 80% load the CO emission of 30H70D at 12˚BTDC, 9˚BTDC and 15˚BTDC is 33.33%, 60% and 0% respectively higher compared to neat diesel fuel operation. It was noted that late injection time increases the concentration of CO emission in the exhaust, whereas, the CO emission at 15˚BTDC is equal to diesel fuel operation. The formation of CO emission depends on oxygen concentration, reaction time, and cylinder temperature (Santhosh et al. 2020). At standard and late injection time, the combustion starts near TDC and it will continue till the end of BDC, which will increase the combustion duration. Due

to prolonged duration the produced heat by burning of fuel will be lost to the cylinder wall, along with this reason, the higher LHE and viscosity also reduce the combustion chamber temperature. These negative effects are responsible for the formation of CO at standard and late injection time (Atmanli and Yilmaz 2018a). However, at advanced injection time, the early start of the combustion reduces the combustion duration and ignition delay, which avoids the loss of the heat to the cylinder wall, and also the oxygenated nature of alcohol assists in the oxidation reaction of CO to CO_2 (Rajesh Kumar, Saravanan, Rana, and Nagendran 2016).
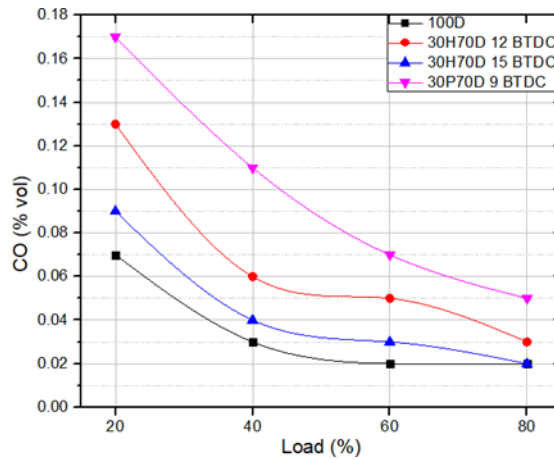
Fig. 4.31 Effect of injection timing on CO

4.4.3.3 Nitrogen Oxide Emission
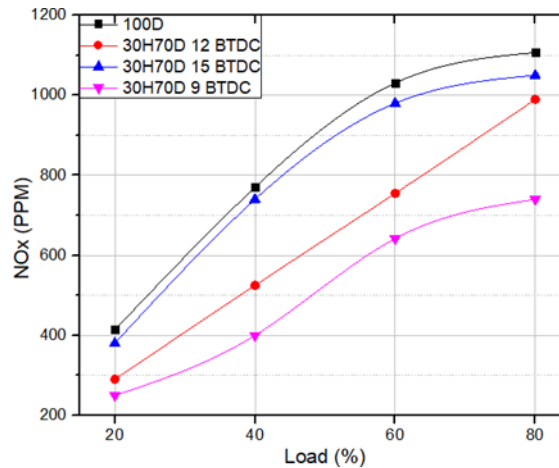
Fig. 4.32 Effect of injection timing on NOx

The effect of injection timing on NO_X emission for 30H30D is shown in Fig. 4.32. The NO_X emission depends on in-cylinder temperature, the higher the temperature of the cylinder higher the NO_X emission. All the injection times for 30H70D showed lower NO_X emission compared to diesel fuel operation. The advanced injection time showed a higher NOx emission compared to the standard and retard injection time of 30H70D. This can be attributed that, at 15°BTDC, the early start of combustion increases the cylinder

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temperature and pressure which participate in the formation of NO_X emission (Nour et al. 2019b). At 60% load NOx emission for 30H30D at 12˚BTDC, 15˚BTDC and 9˚BTDC is 26.77%, 4.94% and 37.73% lower compared to diesel fuel operation at standard injection time. During the early injection time, most of the fuel injected into the combustion chamber burns in the premixed combustion phase, which rapidly increases the combustion temperature leading to an increase in the concentration of NO_X in the exhaust. At 9˚BTDC, the injection happens near TDC, a considerable quantity of the fuel-air mixture is lost through exhaust gas and combustion of reaming fuel-air mixture happens in the expansion stroke. The heat release in the expansion stroke is not accounted as useful energy and is lost to the surrounding which is not participating in the formation of NO_X emission. These are the main reasons for lower NO_X at standard and late injection time.

4.4.4 Summary

- Advanced injection time showed improved cylinder pressure, mean gas temperature, and net heat release rate compared to standard and retarded injection time.

- The BTE of the engine at advanced injection time is almost equal to neat diesel fuel operation whereas, lower BTE is observed for standard and retarded injection time. Even fuel consumption at advanced injection time is improved.

- Higher NO_X emission was noted for advanced injection time compared to standard and late injection.

- Compared to standard and late injection time, the advanced injection time has lower HC and CO emissions, which is due to the complete combustion of accumulated fuel.

4.4.5 Conclusion

Compared to standard (12˚BTDC) and late injection (9˚BTDC), advanced injection time (15˚BTDC) showed better combustion, performance, and emission characteristics. The early start of injection had better space for an ignitable mixture and complete combustion improves the overall performance of the engine. It is concluded that 30% of 1-Hexanol with 70% diesel showed better performance at advanced injection time.

4.5 Effect of Hydrogen enrichment on combustion, performance, and emission characteristics of CRDI CI engine.

In this section, the effect of hydrogen enrichment on engine characteristics are discussed and compared with baseline readings. The hydrogen is supplied to the CRDI CI engine through the inlet manifold and runs the engine in dual fuel mode. These are the important parameters that help to understand the behaviour of the hydrogen in dual fuel mode.

4.5.1 Combustion Characteristics

4.5.1.1 Cylinder Pressure
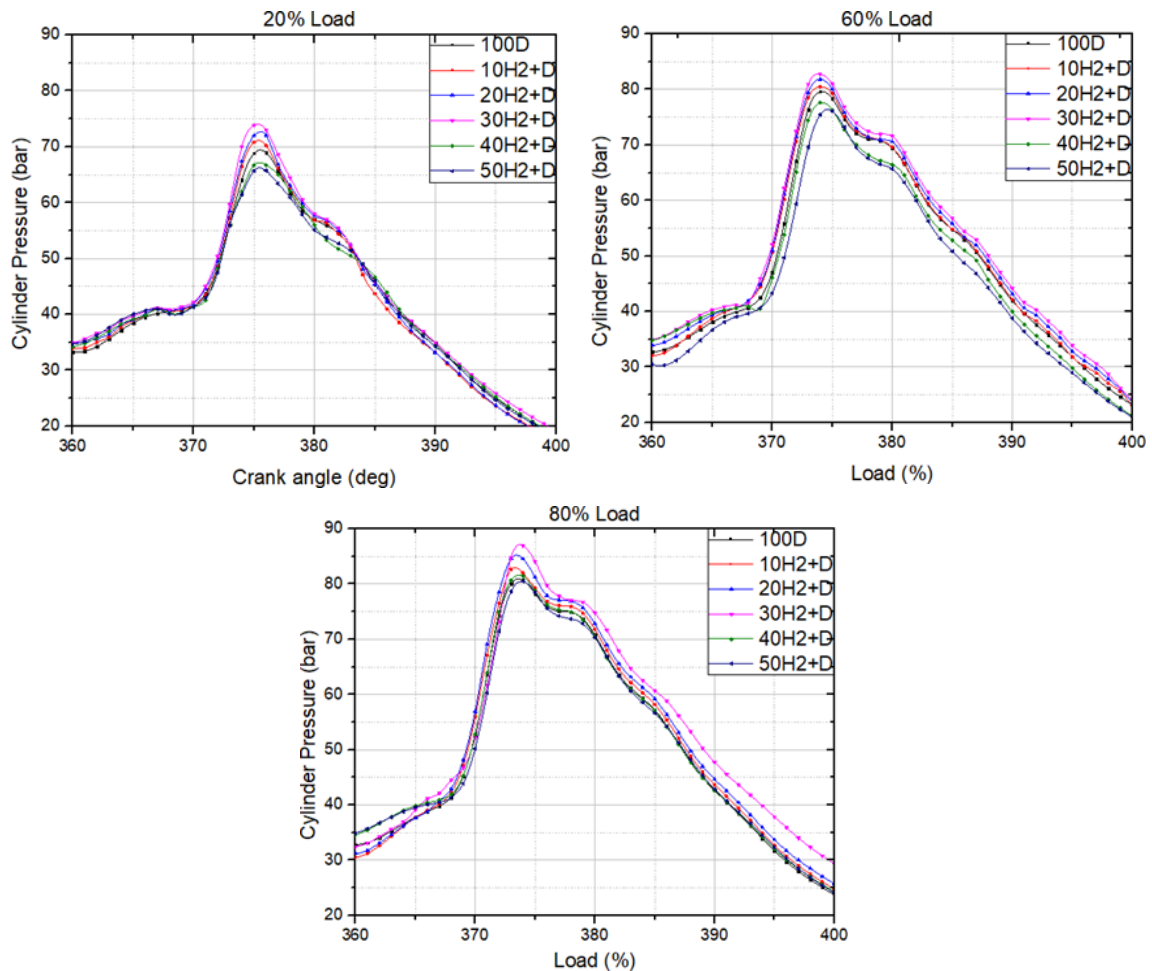
Fig. 4.33 Cylinder Pressure v/s crank angle for different H_2 flow rates

The change in CP with a crank angle for all hydrogen fractions is shown in Fig. 4.33. It is noted that the CP increases with an increase in load and also increases with an increase in hydrogen fraction. The peak cylinder pressure at 80% load for 100D, 10H_2+D, 20H_2+D, 30H_2+D, 40H_2+D, 50H_2+D is 80.71 bar, 82.64 bar, 84.64 bar, 86.96 bar, 81.42 bar and 80.5 bar respectively. Higher peak pressure is noted for 30H_2+D, which is 7.18% higher compared to diesel only operation. The inducted hydrogen-air mixture makes a homogeneous combustible mixture, which readily ignites once the diesel fuel is injected

into the combustion chamber, most of the hydrogen burnt in the premixed combustion phase. This increases the temperature and pressure in the premixed combustion phase, which reduces the ignition delay. Also, the injection of a higher quantity of fuel at a higher load makes difference in peak cylinder pressure. The H_2-air mixture motion in the cylinder is influenced by the in-cylinder temperature and pressure. At a higher engine load, higher in-cylinder temperature increases the turbulence of intake charge which is beneficial for the H_2-air mixture. At a high engine load, the fuel-air mixture is closer to stoichiometric condition; the higher in-cylinder gas temperature and pressure are favourable for hydrogen combustion. Zhou et al. conducted an investigation of diesel-hydrogen dual-fuel combustion at different loads and speeds. They also represent similar kind of results at both lower and higher loads (J H Zhou et al. 2016). Talibi et al. also obtained similar results at both lower and higher load conditions (Talibi et al. 2014). Barrios et al. also noted a reduction in cylinder pressure with the addition of 25% of fuel energy by hydrogen (Barrios et al. 2017).

4.5.1.2 Mean Gas Temperature
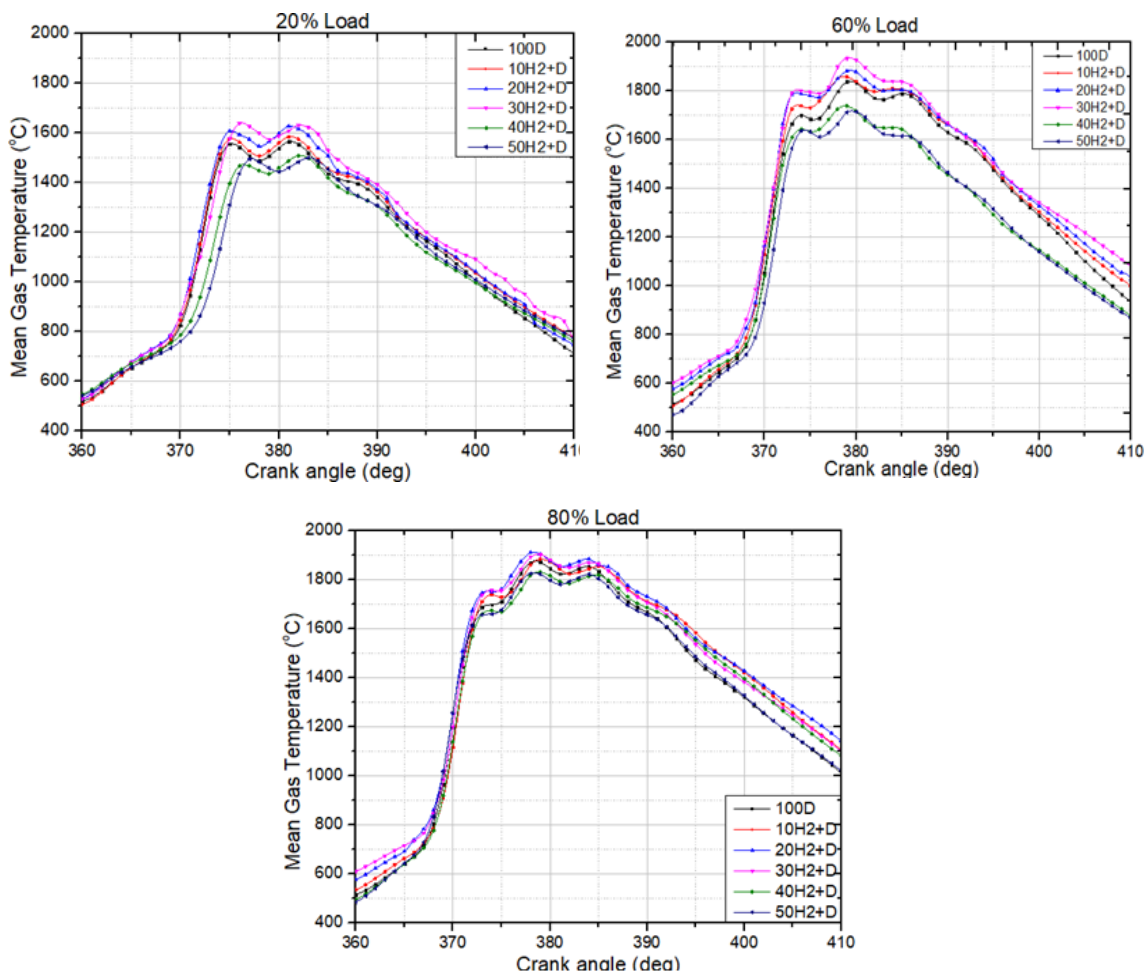
Fig. 4.34 Mean Gas Temperature v/s crank angle for different H_2 flow rates

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The variation of MGT with crank angle is shown in Fig. 4.34 for all hydrogen fractions in the test. It is noted that MGT increases with an increase in the hydrogen fraction up to $30H_2+D$ with further increase in hydrogen fraction decrement in MGT is noted at 80% load. MGT at 80% load for 100D, $10H_2+D$, $20H_2+D$, $30H_2+D$, $40H_2+D$, $50H_2+D$ is 1871.71ºC, 1883.53ºC, 1910.56ºC, 1902.34ºC, 1831.24ºC and 1825.86ºC respectively. At a higher load, the H_2-air mixture is relatively close to stoichiometric ($\varphi = 0.40$), due to the higher flame speed and high diffusive nature of hydrogen burnt in premixed combustion. This results in higher in-cylinder temperature and reduces the ID period. Higher fraction of hydrogen (40 and 50 lpm) exhibits poor combustion which is attributed to that, the high diffusive nature of hydrogen replacing the substantial amount of air. Also, a higher fraction of hydrogen increases the in-cylinder temperature and pressure which reduces the ignition delay. Due to shorter ID lower quantity of diesel is burnt in the premixed combustion zone and combustion shifts to the diffusion combustion phase. However, in diffusion combustion, less time to burn a huge quantity of combustible mixture and lower oxygen availability makes incomplete combustion and reduction MGT. Obtained results are in line with Talibi et al. and Lilik et al. at higher loading conditions (Lilik et al. 2010; Talibi et al. 2014).

4.5.1.3 Net Heat Release Rate

The variation of NHR with a crank angle for all hydrogen fractions is shown in Fig. 4.35. The NHR increases with an increase in load and also increases with an increase in hydrogen fraction up to 30 lpm, further increment leads to a reduction in NHR. The NHR rate for 100D, $10H_2+D$, $20H_2+D$, $30H_2+D$, $40H_2+D$, and $50H_2+D$ at 80% load is 96.27 J/deg, 98.44 J/deg, 99.31 J/deg, 100.13 J/deg, 93.57 J/deg, and 92.86 J/deg respectively. The higher burning velocity, calorific value, and diffusive nature of hydrogen burn rapidly and release a higher heat release rate nearer to TDC. Most of the inducted hydrogen burns in the premixed combustion zone, and the diesel combustion happens in diffusion combustion. If sufficient air is available in the cylinder the ID period reduces due to an increase in the in-cylinder temperature and the H_2-air mixture becomes stoichiometric level, which speeds up the chemical reaction, the laminar burning velocity of the H_2 develops into turbulent. Hence higher NHR rate is noted at high load (Talibi et al. 2014). As previously explained lower heat release rate is noted at 40 lpm and 50 lpm. For higher hydrogen fractions the availability of air is lower, which makes a rich fuel-air mixture, this reduces the complete combustion of liquid fuel in the diffusion combustion phase, which leads to a reduction in NHR at higher hydrogen fraction. The combustion process has three phases. Diesel-air

premixed combustion is first, second is the hydrogen-air premixed combustion, and third is diffused hydrogen-diesel-air controlled combustion. When the first stage of combustion happens (1.e. diesel-air premixed) which is followed by hydrogen premixed combustion phase, there is no significant boundary between the first and second premixed combustion. The CP and HRR are more rapid in the premixed combustion due to the burning of hydrogen than that of diesel.
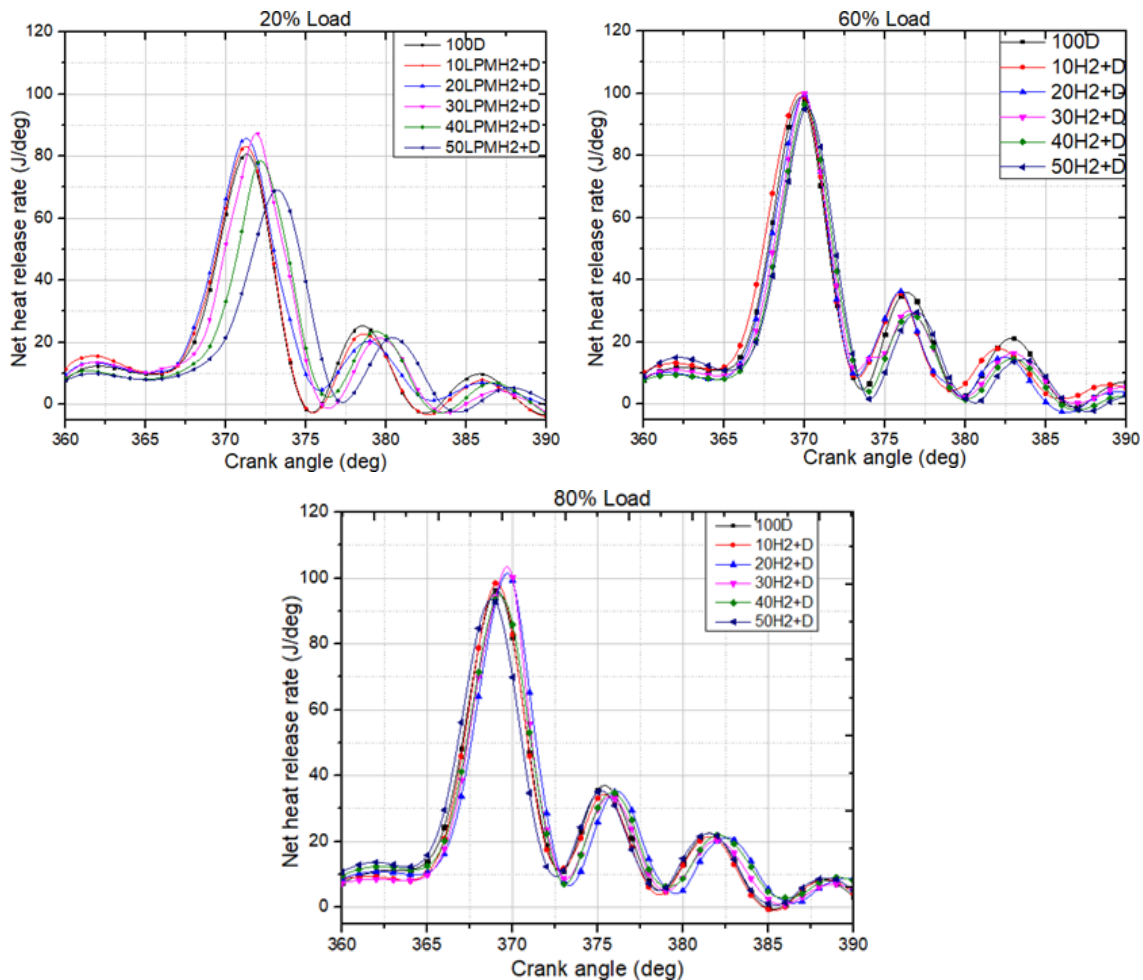
Fig. 4.35 Net Heat Release rate v/s crank angle for different H_2 flow rates

4.5.2 Performance Characteristics

4.5.2.1 Brake Thermal Efficiency

The effect of hydrogen co-combustion on BTE is shown for all hydrogen fractions in Fig.4.36. It is noted that the BTE decreases with a higher fraction of H_2 intake. For $30H_2+D$ the increment in BTE at 20%, 40% 60% and 80% load is 5.66%, 6.94%, 8.62% and 10.41% higher compared to neat diesel fuel operation. The increase in brake thermal efficiency is due to hydrogen's better mixing with air in addition to its faster burning characteristics. Saravanan et al. showed an improvement of 18.4% BTE in their study (Saravanan and Nagarajan 2008) and Bose et al. also shows an improvement in BTE (Bose and Maji 2009).

Whereas at a higher fraction of hydrogen there is a decrement in BTE which is due to low volumetric efficiency, the density of hydrogen is much less compared to air, which displaces an appreciable quantity of air and occupies the cylinder. During the induction process hydrogen mix with air and become a homogenous mixture when the diesel fuel is injected, the hydrogen starts to burn in the premixed combustion and consumes the maximum amount of air, there is a lack of oxygen for complete combustion of diesel fuel mixture in the diffusion combustion zone. This is the main reason for the reduction of BTE with a higher fraction of hydrogen.
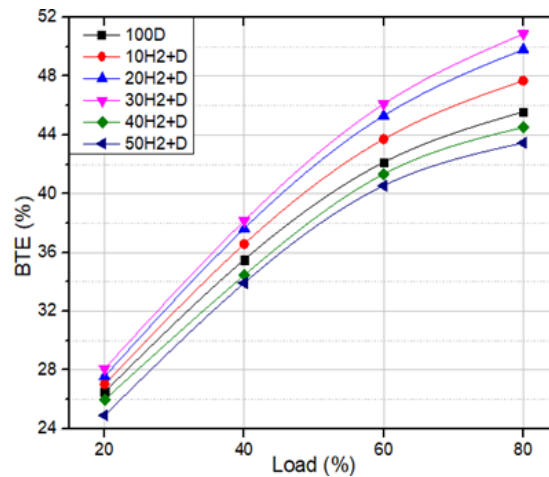
Fig. 4.36 BTE v/s engine load for different H_2 flow rates

4.5.2.2 Brake Specific Energy Consumption
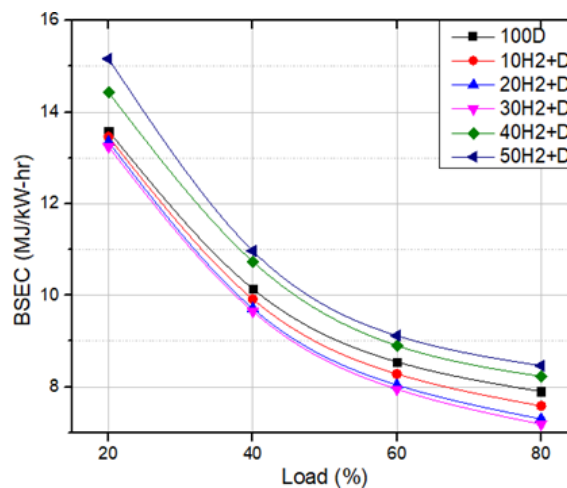
Fig. 4.37 BSEC v/s engine load for different H2 flow rates

The variation in BSEC with the addition of H_2 into the intake manifold at all loading conditions is shown in Fig. 4.37. BSEC decreases with an increase in the fraction of H_2 in the intake manifold at all loading conditions. At 80% load the decrement in BSEC for $10H_2+D$, $20H_2+D$, and $30H_2+D$ is 3.99%, 7.61% and 9.06%. Whereas an increment in BSEC is noted for $40H_2+D$ and $50H_2+D$ which is 4.1% and 6.6% higher compared to diesel

only operation. The higher LHV and higher flame velocity of hydrogen increase the rate of heat release rate compared to diesel fuel which reduces the BSEC. A similar trend is noted by Karagoz et al. they investigated 22% and 53% HES, and they noted a decrease in thermal efficiency and an increase in BSEC (Karagöz et al. 2016). The rapid burning rate of hydrogen consumes the fuel-air mixture rapidly without allowing more heat to the surrounding which increases the thermal efficiency. Hydrogen's quenching gap is smaller than diesels. Thus, the hydrogen flame can expand closer to cylinder walls, resulting in complete fuel combustion (Yilmaz and Gumus 2018).

4.5.3 Emission Characteristics
4.5.3.1 Hydrocarbon Emission
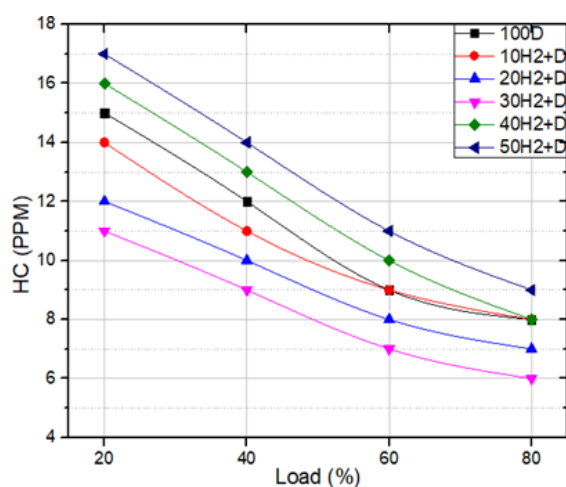
Fig. 4.38 HC emission v/s load for different H_2 flow rates

The variation of HC emission with the addition of hydrogen at all loading conditions is shown in Fig. 4.38. The drastic reduction in HC emission can be observed with the application of hydrogen up to $30H_2+D$, further increment in hydrogen fraction results in higher HC due to improper combustion. The highest decrement is shown for $30H_2+D$ which is 25% lower at 80% load and the highest increment is seen for $50H_2+D$ which is 18.18% higher at 60% load compared to diesel only operation. The formation of HC is low with the hydrogen due to no carbon content in the hydrogen and also higher cylinder temperature helps in rapid and complete combustion of the diesel-air mixture. However, the increase in hydrogen fraction leads to an increment in HC due to incomplete combustion of diesel fuel mixture both in premixed and diffusion combustion phase. The higher diffusive nature of hydrogen replaces a large amount of air during induction of H_2 which makes too lean mixture and the unavailability of oxygen makes poor combustion resulting in higher HC emission.

4.5.3.2 Carbon Monoxide Emission
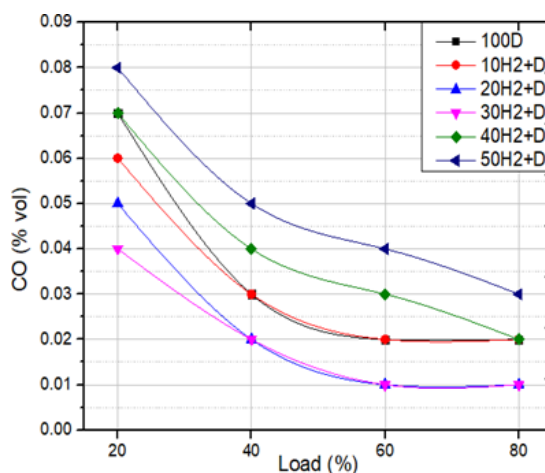
Fig. 4.39 CO emission v/s load for different H_2 flow rate

The effect of hydrogen induction on CO emission at all loading conditions is shown in Fig. 4.39. The key factors for the production of CO emission are oxygen and cylinder temperature. If sufficient oxygen is available at a higher cylinder temperature CO oxidizes to CO_2. If sufficient oxygen is available in the cylinder which results in complete combustion and generates the hydroxyl (OH), the hydroxyl reacts with CO at a higher temperature and forms CO_2 ($CO + OH \rightarrow CO_2 + H$). The diesel engine runs with a lean mixture which requires excess oxygen to oxidize the CO emission. Replacement of hydrogen above the 30 lpm reduces the availability of oxygen and leads to incomplete combustion which increases the production of CO due to the reduction of OH. Due to the addition of H_2 combustion will happen near the injector which reduces the diffusion of diesel with air. Due to the shortage of time for diesel-air mixture, incomplete combustion will lead to the production of CO and other pollutants. A similar trend in CO emission is noted by Hosseini et al. (Hosseini and Ahmadi 2017) and Nag et al. (Nag et al. 2019). Lower CO emission is noted for $30H_2+D$ which is 50% lower at both 60% and 80% load. An increase of 50% and 33.33% of CO emission is noted for $50H_2+D$ at 60% and 80% load respectively.

4.5.3.3 Nitrogen Oxide Emission

Variation of NO_X with load under the influence of hydrogen fraction and diesel is shown in Fig. 4.40. The higher in-cylinder temperature and pressure are the key factors for producing NO_X emission. The NO_X emission for $10H_2+D$, $20H_2+D$, and $30H_2+D$ at 60% load is 1.05%, 2.45%, 3.82%, higher compared to pure diesel operation and at 60% load for $40H_2+D$ and $50H_2+D$ is 2.90% and 4.75% lower compared to diesel. For hydrogen fraction up to 30 lpm, an increase in NO is observed due to higher cylinder temperature,

whereas further increment in H_2 fraction reduces the in-cylinder temperature and pressure due to poor combustion so a reduction in NO_X is noted which is lower than diesel only operation. The addition of a small amount of hydrogen makes the availability of oxygen and hydrogen have higher thermal efficiency due to a higher heat release rate. Most of the hydrogen burns rapidly in the premixed combustion zone and creates a comparatively higher temperature in the cylinder than diesel only operation. These parameters support the formation of NO_X and increase up to $30H_2+D$. The further increment in the hydrogen leads to a reduction of NO_X due to incomplete combustion. The hydrogen replaces the maximum amount of air during induction and makes too rich mixture, when the diesel fuel is injected hydrogen starts to burn within no time and consumes most of the available oxygen. Due to higher cylinder temperature ignition delay got reduced which reduces the residence time of the gas mixture and oxygen availability was also reduced. All these conditions negatively affect the formation of NO_X.
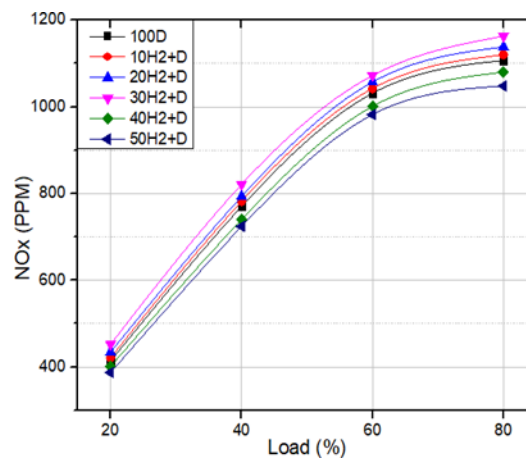
Fig. 4.40 NO_X emission v/s load for different H_2 flow rates

4.5.4 Summary

- Improved combustion characteristics are seen up to 30 lpm of Hydrogen. Further increment in hydrogen fraction deteriorates the combustion.
- For $30H_2+D$ the increment in BTE at 80% is 10.41% higher compared to neat diesel fuel operation. Both BTE and BSEC showed improved performance up to 30 lpm of hydrogen fraction.
- Lower CO and HC emissions and higher NO_X were noted up to 30 lpm of hydrogen fraction. A higher fraction of hydrogen enrichment showed poor emission characteristics.

4.5.5 Conclusion

The enrichment of hydrogen positively responds with diesel and improves combustion emission and performance characteristics of the engine. The hydrogen fractions up to 30 lpm gives a positive response further increment in H_2 fraction deteriorate the combustion due to unavailability of oxygen for complete combustion. It is concluded that up to 30 lpm of hydrogen fraction performs well in the diesel engine. However, the 30 lpm of hydrogen fraction increases the combustion chamber temperature which leads to an increase in the NO_X emission. NO_X is the most harmful emission which should be always kept low. Hence for the future study, only 10 lpm of hydrogen fraction is considered.

4.6 Effect of hydrogen enrichment on combustion, performance, and emission characteristics of CRDI CI engine with optimized alcohol blends at optimized injection time.

In this section, the effect of 10 lpm of hydrogen enrichment is done on 30% of 1-Hexanol/diesel blends to know its effect on engine characteristics of CI engine. The engine is operating at 15˚BTDC, 2000 rpm.

4.6.1 Combustion Characteristics

4.6.1.1 Cylinder Pressure
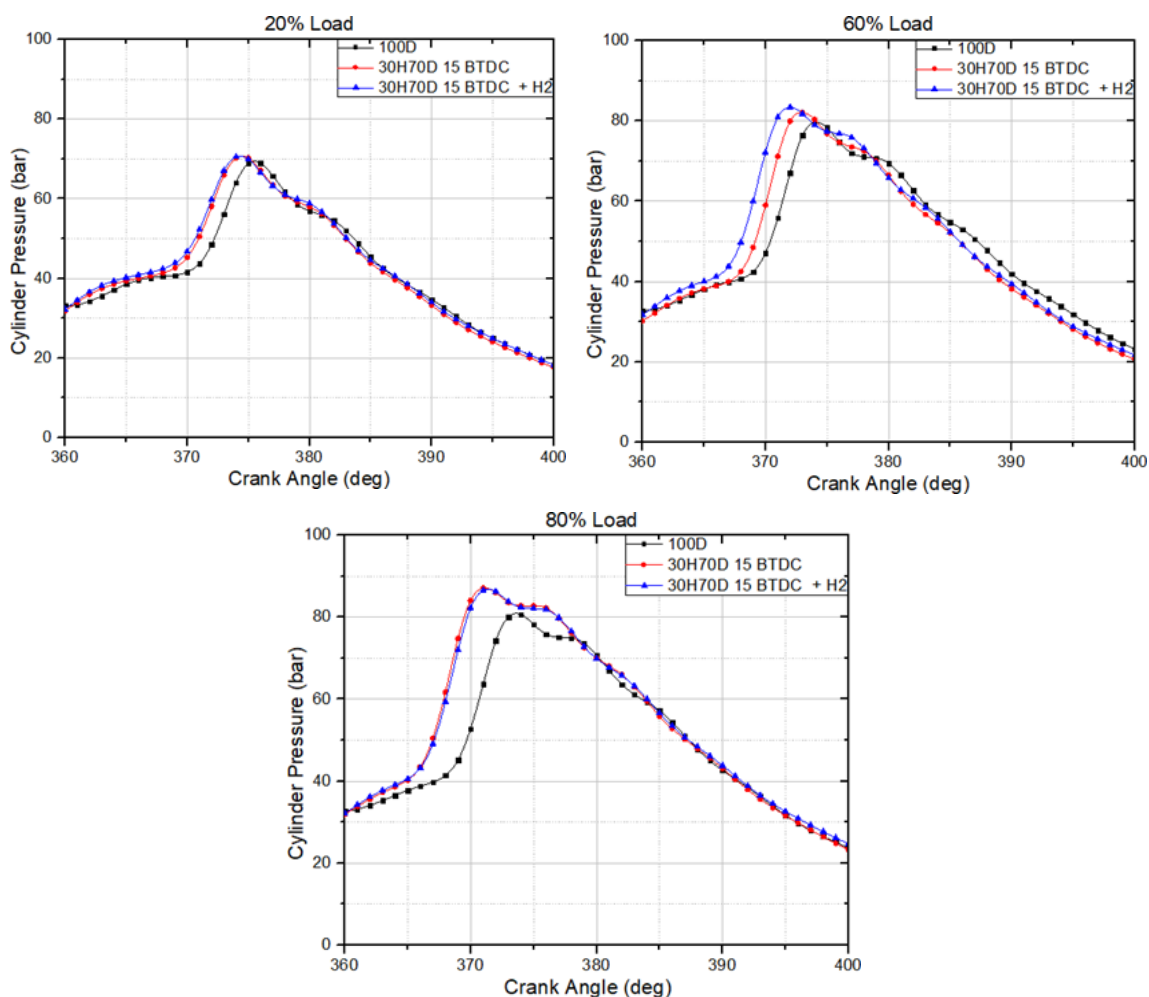
Fig. 4.41 Variation in Cylinder pressure with Crank angle

The effect of hydrogen addition with 1-Hexanol blend on cylinder pressure at 20%, 60%, and 80% load is shown in Fig. 4.41. The cylinder pressure for both hydrogen-enriched 1-Hexanol blend and 1-Hexanol blend alone is higher at all loading compared to neat diesel fuel operation. It is noted that the addition of hydrogen did not show any noticeable difference in CP at 20% and 80% load compared to the 1-Hexanol blend alone. However, at 60% load hydrogen enriched 1-Hexanol blends showed higher cylinder pressure compared to 30H70D alone. At 60% load the cylinder pressure for 100D, 30H70D and

30H70D+H_2 is 79.63%, 82.1% and 83.47% respectively. The higher heating value and higher flame speed of hydrogen reduce the ID (Juknelevičius et al. 2019). The lower quantity of fuel would accumulate due to reduced ID. At 60% load, the addition of hydrogen shows a 1.64% enhancement in cylinder pressure. At 80% load the cylinder pressure for 100D, 30H70D and 30H70D+H_2 is 80.71%, 87.17% and 86.58% respectively.

4.6.1.2 Mean Gas Temperature
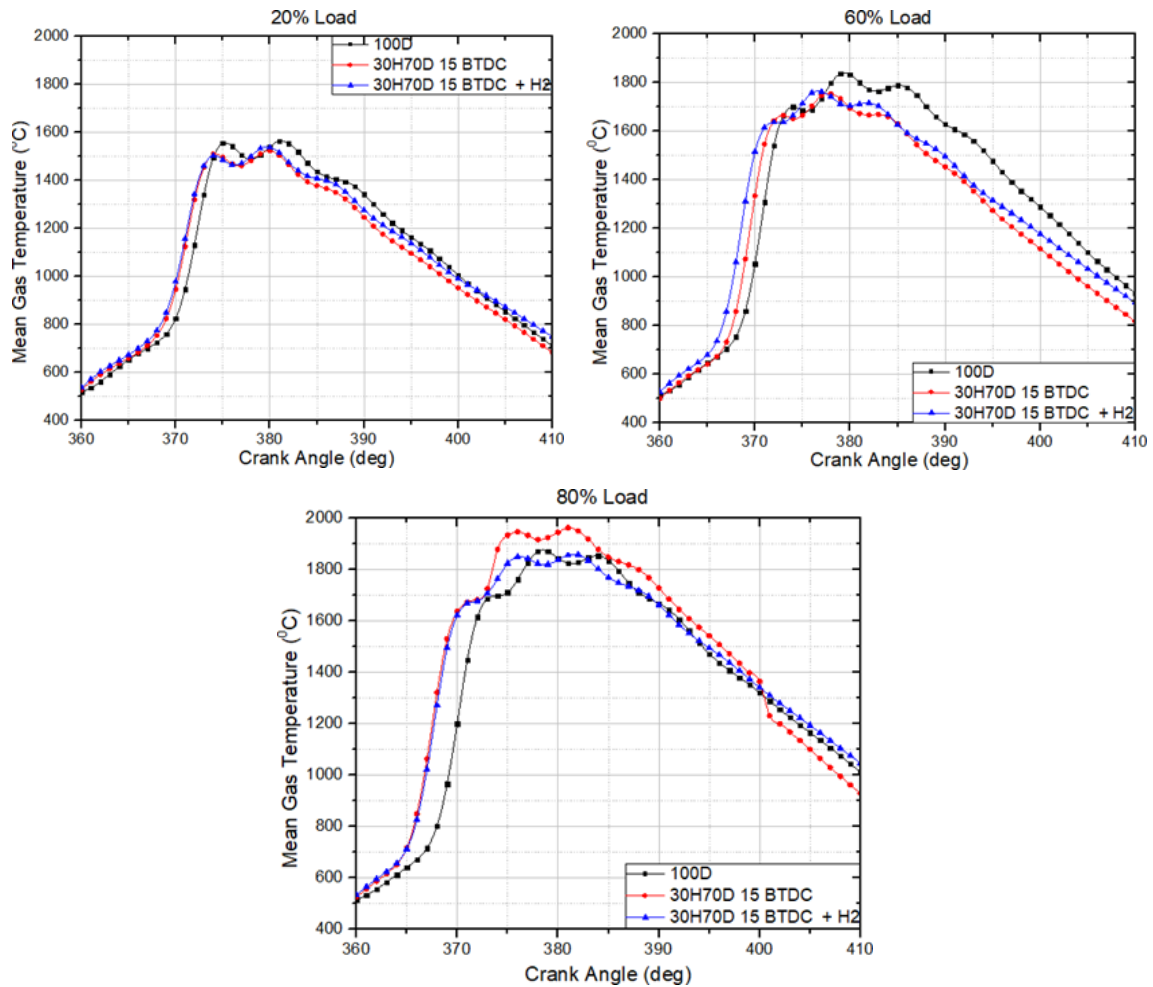
Fig. 4.42 Variation in Mean Gas Temperature with Crank angle

The effect of hydrogen addition on the mean gas temperature at 20%, 60%, and 80% load is shown in Fig. 4.42. The mean gas temperature for both 30H70D and 30H70D+H_2 is lower at 20% and 60% load compared to diesel fuel operation. At 60% load, MGT for 100D, 30H70D, and 30H70D+H_2 is 1836.95°C, 1753.44°C, and 1762.49°C respectively. The addition of 1-Hexanol increases the delay period due to the lower cetane number which allows the accumulation of a higher quantity of fuel into the combustion chamber. Whereas, hydrogen enrichment reduces the delay period due to a higher heat release rate and allows the fuel mixture to burn rapidly. 1-Hexanol has a lower heating value and higher viscosity

which may be the possible reason for lower MGT. At 80% load, MGT for 100D, 30H70D, and 30H70D+H_2 is 1871.95°C, 1947.64°C, and 1849.89°C respectively.

4.6.1.3 Net Heat Release Rate
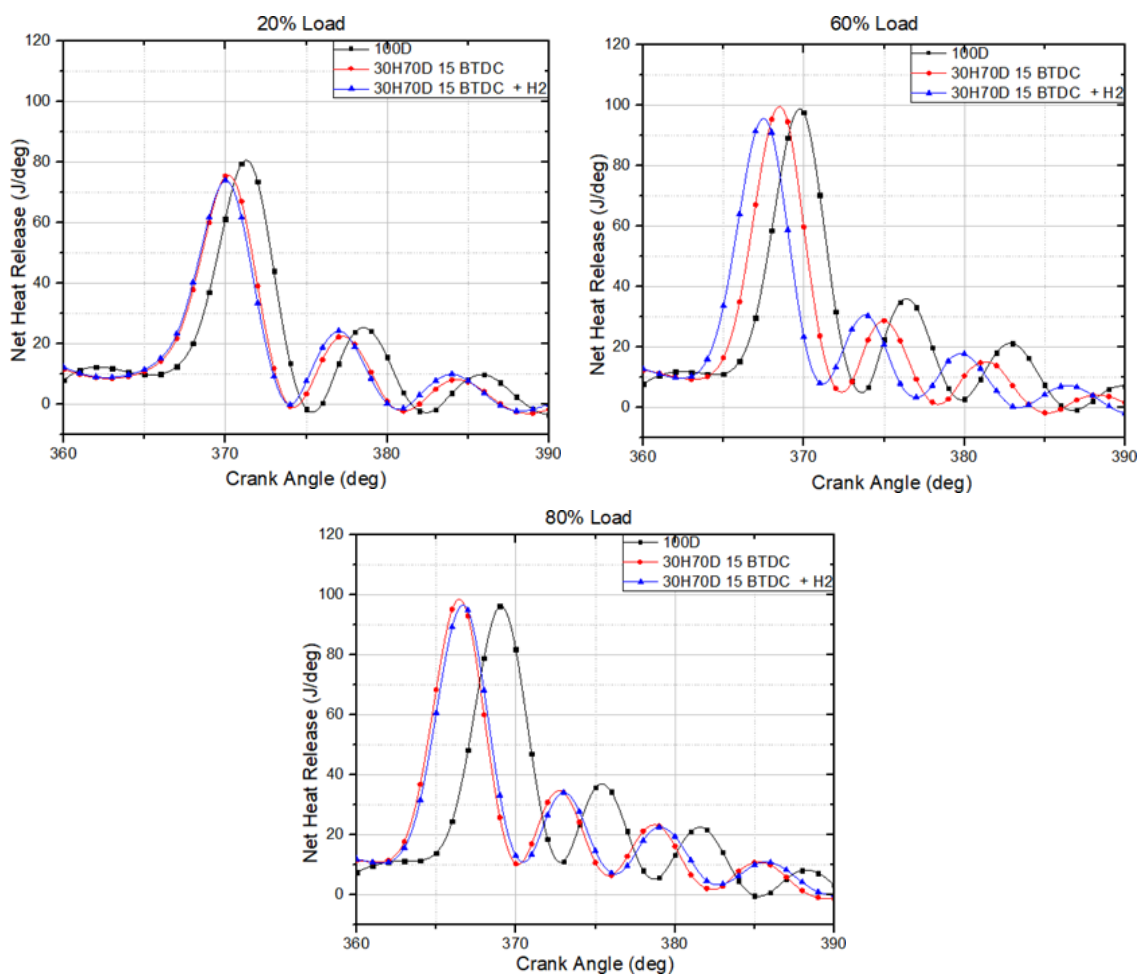
Fig. 4.43 Variation in Net Heat Release with Crank angle

The effect of hydrogen addition on net heat release rate at 20%, 60%, and 80% load is shown in Fig. 4.43. The addition of hydrogen reduces the net heat release rate at all engine load conditions compared to 30H70D alone. This is due to a reduction in ignition delay. Hydrogen had a higher heating value and the rapid flame speed of hydrogen makes the rapid burning of accumulated fuel mixture, which increases the temperature of the cylinder. The higher cylinder temperature reduces the ignition delay by increasing the fuel evaporation rate. At 80% load NHR rate for 100D, 30H70D and 30H70D+H_2 is 96.27 J/deg, 95.21 J/deg and 94.82 J/deg. It is noted that the net heat release rate of all the blends has the same trend and there is no notable difference is observed. Whereas, at 60% load the NHR rate for 30H70D+H_2 is 6.39% and 4.12% lower compared to 100D and 30H70D respectively.

4.6.2 Performance characteristics

4.6.2.1 Brake Thermal Efficiency
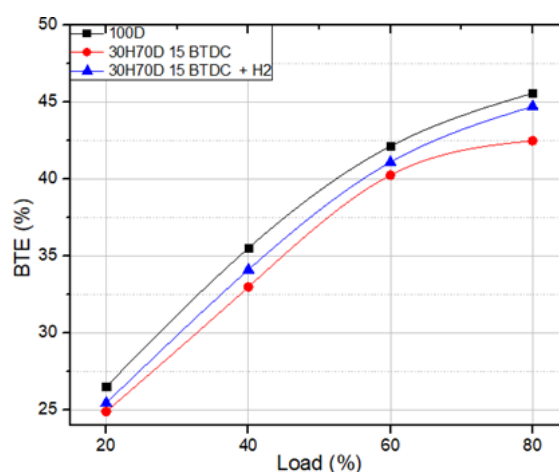
Fig. 4.44 Change in BTE with load for all the test fuel

The effect 1-Hexanol and hydrogen on brake thermal efficiency at all loading conditions is shown in Fig. 4.44. It is noted that the BTE of 30H70D+H_2 is higher compared to 30H70D. At 20%, 40%, 60% and 80% load the increment in BTE for 30H70D+H_2 is 2.19%, 3.27%, 2.04% and 4.99% higher compared to neat 30H70D. The main reason for the increment in BTE is the higher heating value of hydrogen. As previously discussed, with advanced injection time the 1-Hexanol/diesel blends show an improvement in BTE, further improvement in BTE can be seen with the addition of hydrogen. A small amount of hydrogen enrichment enhances the combustion capacity of the fuel/air mixture due to higher flame speed and rapid combustion.

4.6.2.2 Brake Specific Energy Consumption

The energy consumption of all the fuels in the test is shown in Fig. 4.45. Diesel fuel operation consumes lower energy to produce the same output compared to 1-Hexanol blend, due to higher heat content. The enrichment of hydrogen shows improvement in fuel consumption. Due to the higher heating value of hydrogen, it consumes lower energy to produce the same power output. At 20%, 40%, 60% and 80% load the decrement in BSEC for 30H70D+H_2 is 2.19%, 3.27%, 2.04% and 4.99% lower compared to 30H70 blend. The lower heating value and CN of 1-Hexanol consume more fuel to produce the same amount of heat energy. The lower CN of 1-Hexanol allows accumulation of more amount of fuel in the combustion chamber and due to longer ignition delay and longer combustion duration, there is a loss of fuel due to incomplete combustion and loss of heat due to longer combustion duration. However, the hydrogen enrichment improves the combustion and avoids the loss of heat to the cylinder wall by rapid combustion.
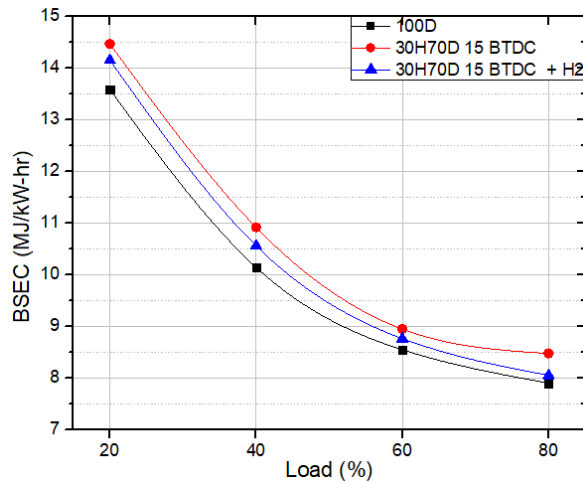
Fig. 4.45 Change in BSEC with load for all the test fuel

4.6.3 Emission characteristics

4.6.3.1 Hydrocarbon Emission
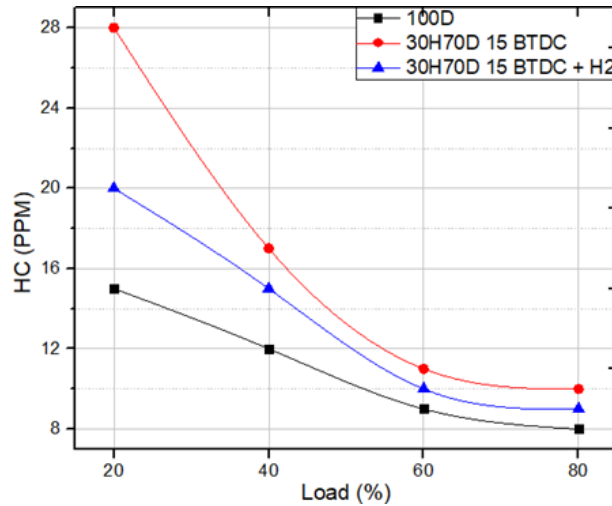
Fig. 4.46 Change in HC with load for all the test fuel

The variation of HC emission with the addition of hydrogen at all loading conditions is shown in Fig. 4.46. It is noted that the enrichment of hydrogen reduces the formation of unburnt hydrocarbon in the exhaust. At 20%, 40%, 60% and 80% load the decrement in HC for 30H70D+H_2 is 28.57%%, 11.76%, 9.09% and 10% lower compared to 30H70D alone. At 80% load, the HC emission of 30H70D+H_2 is only 11.11% higher compared to diesel only operation. The formation of HC is due to the incomplete combustion of accumulated fuel. The higher LHE and water molecule in 1-Hexanol lowers the cylinder temperature which causes incomplete combustion of hydrocarbon molecules. Whereas, the higher heating content and rapid flame speed of the hydrogen increase the combustion chamber temperature and eliminate the drawback of 1-Hexanol. Also, the absence of HC molecules in the hydrogen lowers the formation of HC. Hence the hydrogen enrichment shows lower HC emission.

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4.6.3.2 Carbon Monoxide Emission
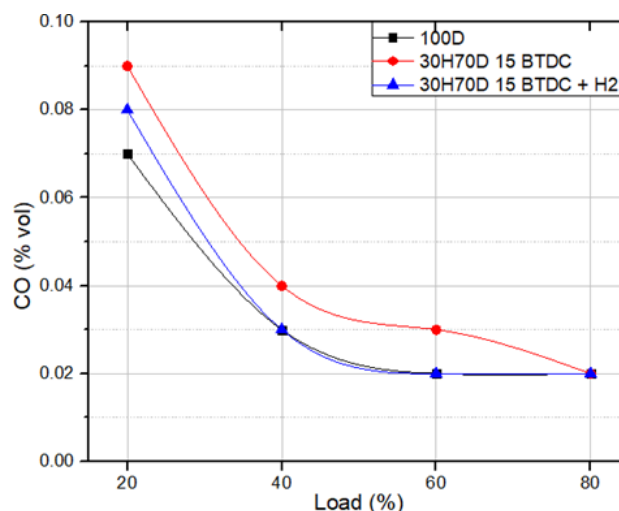
Fig. 4.47 Change in CO with load for all the test fuel

The effect of hydrogen enrichment on CO emission at all engine loads is shown in Fig.4.47. The enrichment of hydrogen to 30H70D reduces the concentration of CO in the exhaust emission. Only at 20% engine load 30H70D+H_2 showed 12.5% higher CO emission compared to diesel fuel operation, whereas at 40%, 60%, and 80% load the CO emission was almost similar to diesel fuel operation. As previously explained the hydrogen enrichment cause complete combustion of the accumulated fuel, due to higher burning velocity and flame velocity. The complete combustion generates OH radicals and also the presence of OH radicals in the alcohol reacts with CO at higher temperatures and forms CO_2 ($CO + OH \rightarrow CO_2+H$). Another main reason for the reduction in CO emission with the addition of hydrogen is the absence of carbon molecules in the hydrogen.

4.6.3.3 Nitrogen Oxide Emission

The effect of hydrogen enrichment on NO_X emission is shown in Fig. 4.48. It is noted that an almost similar kind of NO_X trend is observed for all the tested fuels. It is noted from the graph that the hydrogen enrichment reduces the formation of NO_X emission compared to 30H70D at advanced injection time at all engine loads. The formation of NO_X for 30H70D+H_2 at 80% load is 11.47% lower compared to neat diesel fuel operation. This is 6.66% lower compared to 30H70D only operation. Due to the addition of hydrogen both ignition delay and combustion duration of 30H70D were reduced. This helps in the accumulation of a lower amount of fuel in the combustion chamber and also the residence time of combustion products in the cylinder is reduced which acts negatively on the formation of NO_X.
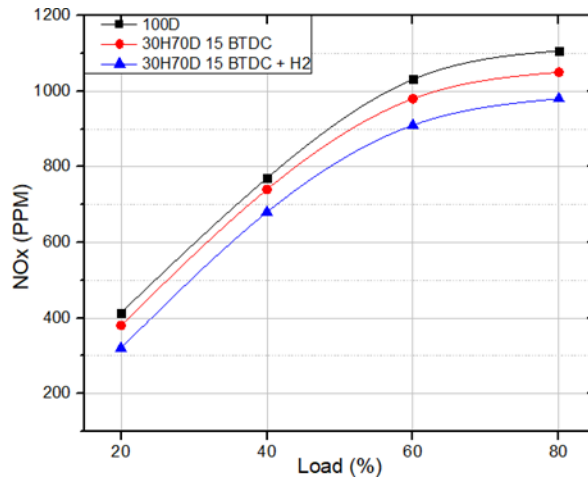
Fig. 4.48 Change in NO$_X$ with load for all the test fuel

4.6.4 Summary

- The hydrogen enriched 1-Hexanol blend showed improved combustion characteristics compared to the 1-Hexanol blend alone. The higher heating value and higher flame speed of hydrogen reduce the ID, which causes a positive effect on combustion.

- There is an improvement in thermal efficiency and a reduction in fuel consumption is noted for the hydrogen enriched 1-Hexanol blend.

- The absence of HC molecules in the hydrogen reduces the CO and HC emissions. The HC and CO emission of hydrogen enriched 1-Hexanol blend is almost equal to diesel fuel operation.

- Lower NO$_X$ emission is noted for the hydrogen enriched 1-Hexanol blend due to reduction of ignition delay with hydrogen enrichment.

4.6.5 Conclusion

The higher heating value and higher flame propagation speed of the hydrogen reduce the ignition delay period. This lowers the accumulation of more quantity of fuel in the combustion chamber. The reduced delay period and short combustion duration reduce the loss of heat to the surrounding. All these factors had a positive effect on the combustion, performance, and emission characteristics. It is concluded that 10 lpm of hydrogen enrichment to the 30H70D improves the overall performance of the engine.

4.7 Effect of EGR and selective catalytic reduction (SCR) of NOx on combustion, performance, and emission characteristics of CRDI CI engine for the optimized alcohol/diesel blends and optimized hydrogen enrichment with optimized injection time

In this section, the effect of EGR and SCR of NOX on engine characteristics are discussed. As EGR is an in-cylinder strategy, its effect on combustion, performance, and emission characteristics of the engine is discussed however, SCR of NOX is after treatment technique that did not affect combustion and performance. Hence, the effect of SCR of NOX is discussed only for the emission characteristics of the engine. The effect of 10% and 20% EGR and SCR of NOX on engine characteristics are examined for hydrogen- enriched 1-Hexanol/diesel blends at advanced injection time (15°BTDC).

4.7.1 Combustion Characteristics

4.7.1.1 Cylinder Pressure
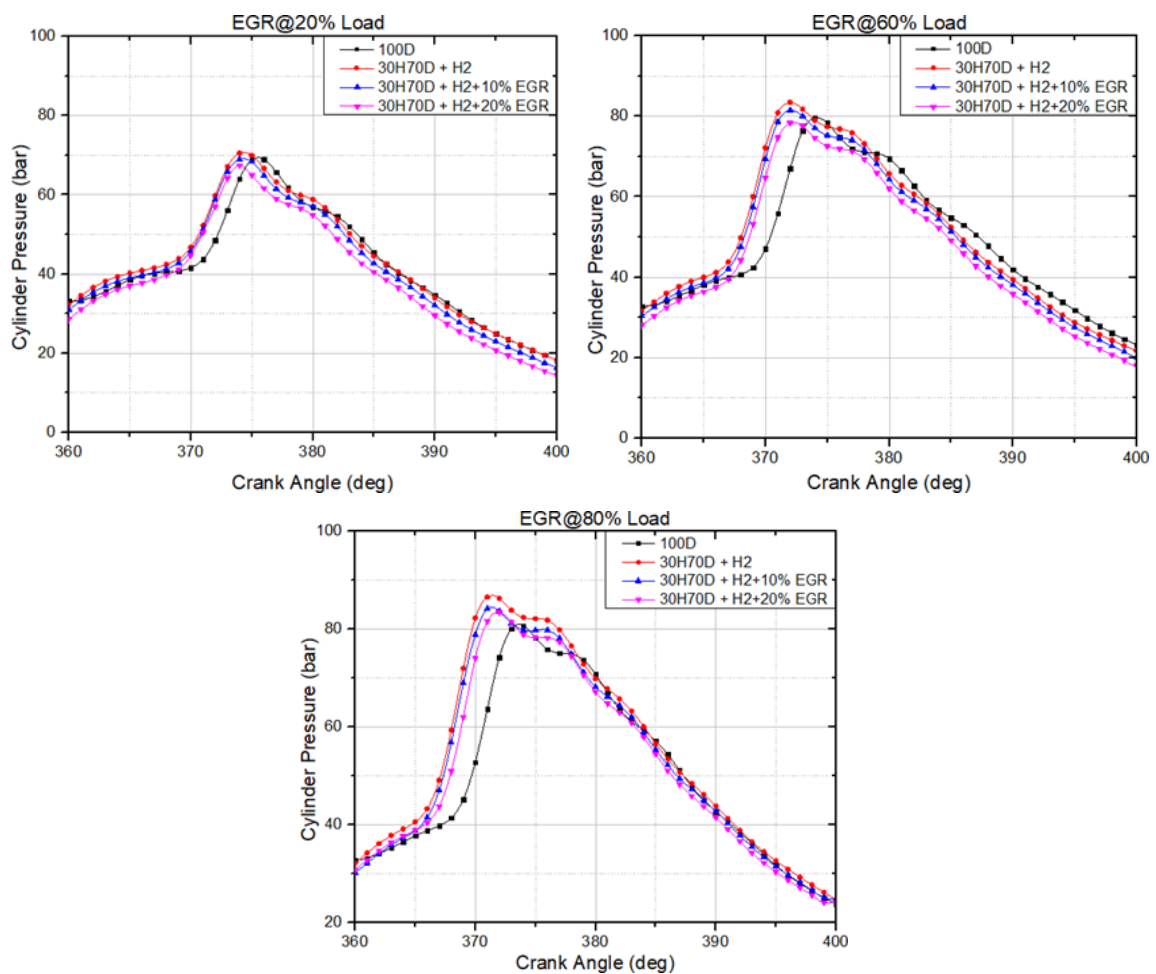
Fig. 4.49 Effect of EGR on cylinder pressure for hydrogen enriched 1-Hexanol blend

The effect of EGR on cylinder pressure at 20%, 60%, and 80% load is shown in Fig. 4.49. It is noted that in the presence of EGR there is a reduction in the cylinder pressure at all

loading conditions. At 10% EGR for 30H70D+H$_2$ at 20%, 60% and 80% load the reduction in CP is 2.155%, 2.34% and 2.73% respectively compared to no EGR condition. At 20% EGR for 30H70D+H$_2$ at 20%, 60% and 80% load the reduction in CP is 4.35%, 6.09% and 3.6% respectively compared to no EGR condition. It is noted that the cylinder pressure reduces with an increase in the EGR rate. The main reason is the reduction in the supply of oxygen, the application of EGR replaces the considerable amount of fresh air inducted into the engine. Already, the enrichment of hydrogen reduces some part of the air inducted into the engine, which is unavailable for combustion. When the EGR is employed, it further replaces the part of the fresh air in the fuel-air mixture with exhaust gas. Which deaccelerates the availability of the oxygen for combustion thereby deteriorating the combustion. Also, the exhaust gas contains CO_2 and water, which act as a heat sink and absorbs the heat from the engine cylinder (Ayodhya, Lamani, Bedar, et al. 2018; Hountalas et al. 2008; Nour et al. 2017). The availability of oxygen molecules in 1-Hexanol reduces the effect of EGR, when the EGR rate is increased to 20%, the cylinder pressure is affected and decreased.

4.7.1.2 Mean Gas Temperature
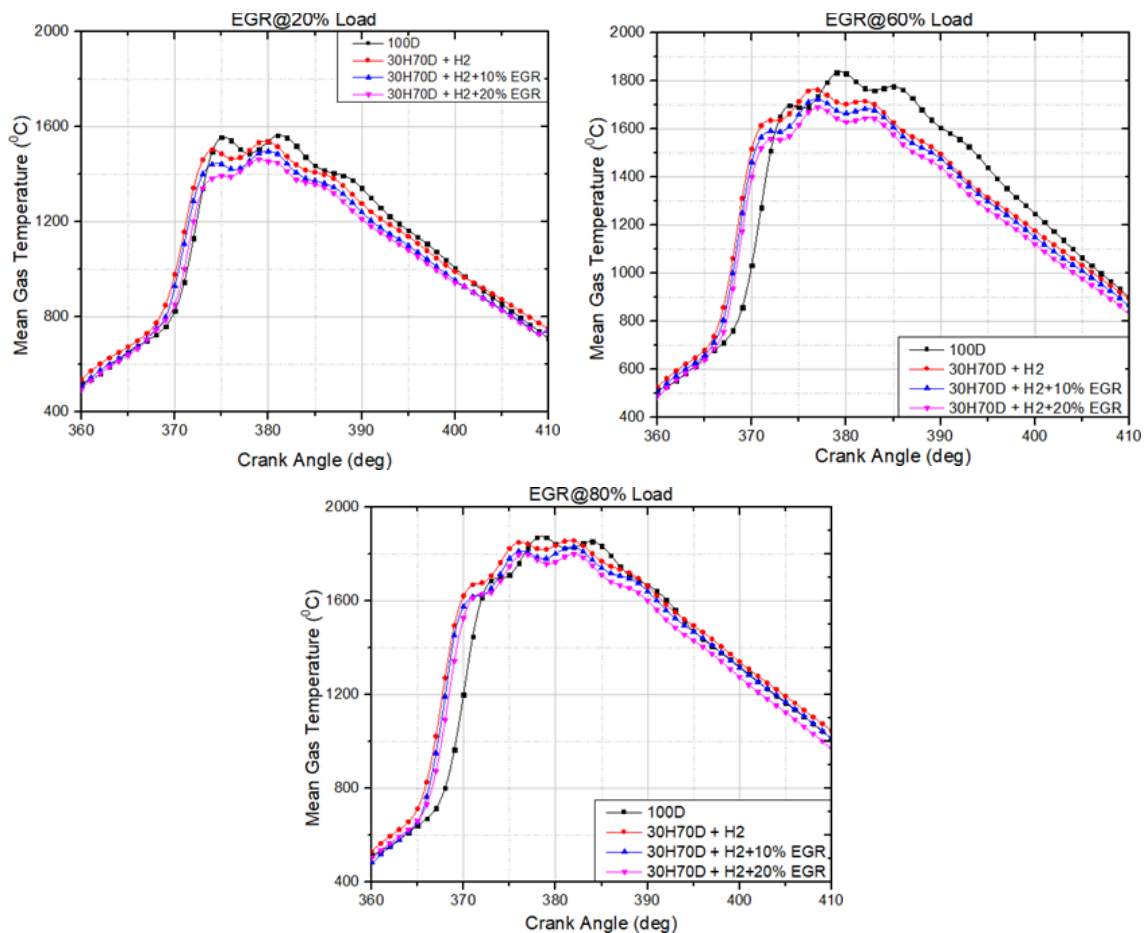
Fig. 4.50 Effect of EGR on MGT for hydrogen enriched 1-Hexanol blend

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The effect of EGR on mean gas temperature at 20%, 60%, and 80% load is shown in Fig. 4.50. The addition of EGR negatively affects MGT at all loading conditions. At 10% EGR for 30H70D+H_2 at 20%, 60% and 80% load the reduction in MGT is 2.72%, 2.20% and 1.44% respectively compared to no EGR condition. At 20% EGR for 30H70D+H_2 at 20%, 60% and 80% load the reduction in MGT is 4.87%, 4.13% and 3.07% respectively compared to no EGR. The main reason for the reduction in the MGT is the unavailability of oxygen for complete combustion. The inert gas in the EGR dilutes the combustible mixture and reduces the combustion rate and also, the specific heat capacity of the fresh charge increases due to recirculation of the CO_2, which reduces the combustion temperature (Hountalas et al. 2008). At 80% load, for both EGR rates the reduction in MGT is low, this is due to, at higher load more amount of fuel is injected into the combustion chamber, the fuel mixture contains 1-Hexanol, which is enriched with oxygen molecules which reduces the effect of EGR at higher load.

4.7.1.3 Net Heat Release Rate
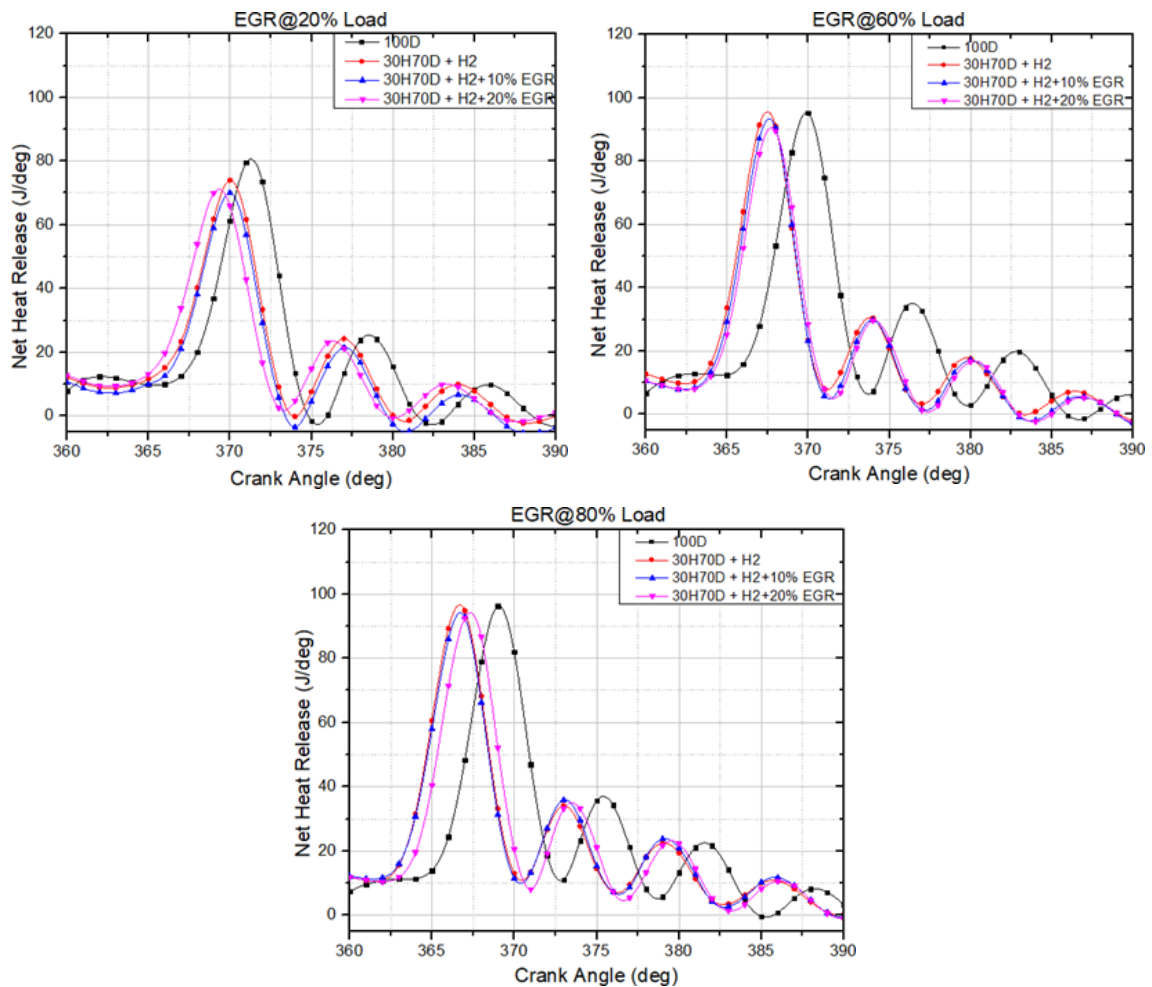
Fig. 4.51 Effect of EGR on Net heat release rate for hydrogen enriched 1-Hexanol blend

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The effect of EGR on net heat release rate at 20%, 60%, and 80% load is shown in Fig. 4.51. At 10% EGR for 30H70D+H_2 at 20%, 60% and 80% load the reduction in NHR is 5.32%, 0.98% and 2.11% respectively compared to no EGR condition. At 20% EGR for 30H70D+H_2 at 20%, 60% and 80% load the reduction in NGR is 5.50%, 2.18% and 2.35% respectively compared to no EGR. The application of EGR causes a dilution effect, chemical effect, and thermal effect on NHR. When the EGR is supplied through the inlet manifold it replaces the part of fresh oxygen in the charge (dilution effect) with CO_2 and H_2O, the dissociation of CO_2 and H_2O is an endothermic reaction (chemical effect) which tarp some heat for the dissociation process. The EGR increases the specific heat capacity of the intake charge (thermal effect) resulting in a lower heat release rate (Thangaraja and Kannan 2016). It is noticed that compared to other loading conditions the effect of EGR on NHR is minimum at 60% load. Maybe at 60% load, the combustion chamber temperature is suitable for complete evaporation of 1-Hexanol/diesel fuel, and the oxygen content of 1-Hexanol reduces the dilution effect of EGR. All these parameters may support the complete combustion of accumulated fuel which may reduce the effect of EGR on combustion.

4.7.2 Performance characteristics

4.7.2.1 Brake Thermal Efficiency
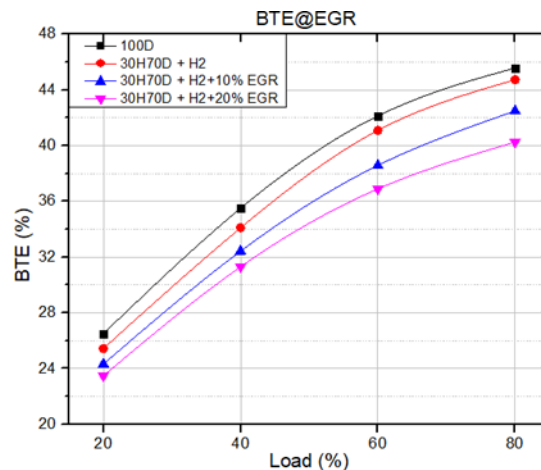
Fig. 4.52 Effect of EGR on BTE for hydrogen enriched 1-Hexanol blend

The effect of EGR on brake thermal efficiency for optimized 1-Hexanol/diesel blend with optimized hydrogen enrichment is shown in Fig. 4.52. It is noticed that the BTE of the engine decreased with the application of EGR at all loading conditions for all the test fuel. At 10% EGR rate the reduction in BTE for 30H70D+H_2 is 4.4%, 4.9%, 6.13% and 5% at 20%, 40%, 60% and 80% load respectively compared to no EGR condition. At 20% EGR rate reduction in BTE for 30H70D+H_2 is 7.66%, 8.19%, 10.22% and 9.99% at 20%, 40%, 60% and 80% load respectively compared to no EGR condition. It is noted that at both 10%

and 20% EGR rates the reduction in BTE increases with load. The maximum reduction in BET is seen at 60% load for both the EGR condition. The BTE of the engine is based on complete combustion of the fuel and utilization of produced heat due to combustion. The main cause for the reduction in BTE with EGR is the dilution effect. The enrichment of hydrogen through the inlet port replaces the part of inlet air and the addition of EGR also replaces the availability of fresh oxygen for combustion in the chamber. The addition of EGR increases the ID period, and it acts as a heat sink which reduces the combustion temperature and also reduces the propagation of flame speed (Xie et al. 2016). All these negative effects of EGR reduce the energy conversion of the engine thereby reducing the BTE.

4.7.2.2 Brake Specific Energy Consumption
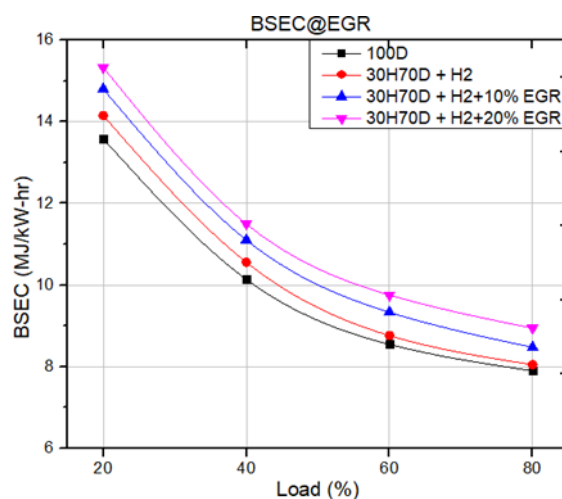
Fig. 4.53 Effect of EGR on BSEC for hydrogen enriched 1-Hexanol blend

The effect of EGR on brake specific energy consumption for optimized 1-Hexanol/diesel blend with optimized hydrogen enrichment is shown in Fig. 4.53. At 10% EGR, the increment in BSEC for 30H70D+H_2 at 20%, 40%, 60% and 80% load is 4.4%, 4.91%, 6.11% and 5.07% respectively compared to no EGR condition. At 20% EGR, the increment in BSEC for 30H70D+H_2 at 20%, 40%, 60% and 80% load is 7.69%, 8.2%, 10.19% and 10.06% respectively compared to no EGR condition. The trend of both BTE and BSEC is the same for the EGR application. The addition of EGR causes a negative effect on energy consumption. This is due to, the use of EGR reducing the in-cylinder temperature, which may result in a decrement in the combustion efficiency (Rajesh Kumar and Saravanan 2015). In addition to the lower CN of 1-Hexanol, the EGR also prolongs the delay period which will be a negative effect on combustion and a considerable amount of heat will be lost, which results in higher energy consumption.

4.7.3 Emission characteristics

4.7.3.1 Estimation of optimum ammonia flow rate
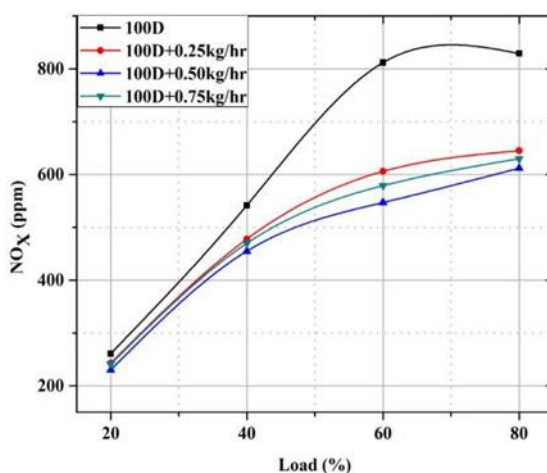
Fig. 4.54 Reduction of NO_X with load at a various ammonia flow rates for neat diesel

Fig. 4.54 shows the SCR of NO_X at various ammonia flow rates. Maximum of 25.36%, 32.63% and 28.69% of SCR of NO_X was attained for 0.25kg/hr, 0.5kg/hr and 0.75kg/hr of ammonia flow rate at 60% load respectively. It is seen from the graph that the highest SCR of NO_X has achieved at 60% load and a further increase in the load leads to a reduction in SCR of NOx due to an increase in the temperature at a higher load. The Pt and Pd catalysts are highly active at medium temperatures (250-300 ⁰C) than at higher temperatures (Goula et al. 2016). At higher temperatures, ammonia is desorbed from the walls of the monolith, higher hydrocarbon emission causes a blockage of pores through carbon particles and little residence time for the reactants within the catalyst for standard SCR reaction (Kumar et al. 2015). The flow rate of 0.25kg/hr of ammonia results in a lower reduction of NO_X emission due to the insufficiency of NH_3 for the SCR of NO_X, whereas 0.75kg/hr of ammonia flow rate results in a deposition and pore-blocking of SCR catalyst (Kumar et al. 2015). Hence 0.5kg/hr of ammonia flow rate has been considered an optimum flow rate for maximum NO_X reduction.

4.7.3.2 Hydrocarbon Emission

The effect of EGR and SCR of NO_X on HC emission at all loading conditions is shown in Fig 4.55. Higher HC emission is noted with the addition of EGR and it increases with a higher concentration of EGR for all the fuels in the test. At 60% load, the increment in HC emission for 30H70D+H_2 at 10% EGR is 9.09% and at 20% EGR is 23.07% compared to no EGR condition. There is a drastic increment in HC emission is seen with the addition of EGR. Even though a small amount of oxygen was present in the molecular structure of 1-Hexanol, it was suppressed by the addition of EGR. EGR replaces the considerable amount

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of fresh air in the inlet and also, the addition of Hydrogen through the inlet manifold may cause a further decrement in the availability of oxygen in the combustion chamber. When the exhaust gas is re-circulated it contains unburnt HC from the previous cycle, if enough air is available in the combustion chamber it will be burnt and increase the combustion temperature. However, EGR replaces part of fresh air with diluents, and not enough air is available to burn unburnt HC in the exhaust (Verma et al. 2019). Also, at a 20% EGR rate, higher HC emission is noticed for a higher fraction of the 1-Hexanol blend. This can be attributed to that, the higher concentration of EGR may cause a misfire, which requires higher fuel to sustain the combustion, along with this lower LHE and water content of 1-Hexanol lowers the evaporation rate of the fuel blends. This causes local cold spots and the accumulation of rich air-fuel mixture in the cylinder results in poor combustion. All these factors influence the formation of higher HC emission.

It is noted from the graph that with the presence of SCR the HC emission got reduced. At some point, even the HC emission was lower than pure diesel operation. At 60% load in the presence of SCR, the HC emission for 30H70D+H_2 is 27.77% lower compared to the 10% EGR rate and 30.76% lower compared to the 20% EGR rate. The presence of SCR catalyst in the tailpipe reduces the HC emission. HC emission in the exhaust helps to reduce the NO_X (increased HC:NO ratio), which acts as a reductant in the SCR process (Abu-Jrai and Tsolakis 2007). Oxidation of hydrocarbon is a function of exhaust gas temperature (Pérez-Hernández et al. 2005), hence in the presence of SCR with an increase in the load, HC oxidation also increases this leads to a reduction of HC emission at a higher load. The HC emission is oxidized by the oxygen present in NO as follows (Piumetti et al. 2015).

$$NO + hydrocarbon + O_2 \rightarrow N_2 + CO_2 + H_2O \qquad 4.1$$

$$2NO + 2C_3H_6 + 8O_2 \rightarrow N_2 + 6H_2O + 6CO_2 \qquad 4.2$$
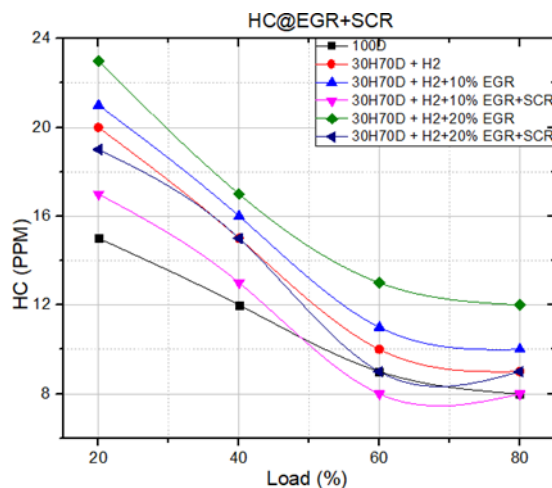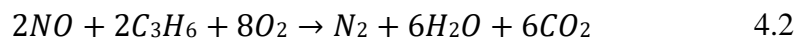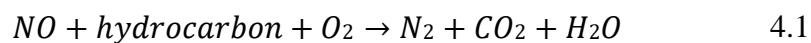
Fig. 4.55 Change in HC with load for all the test fuel with EGR and SCR

4.7.3.3 Carbon Monoxide Emission
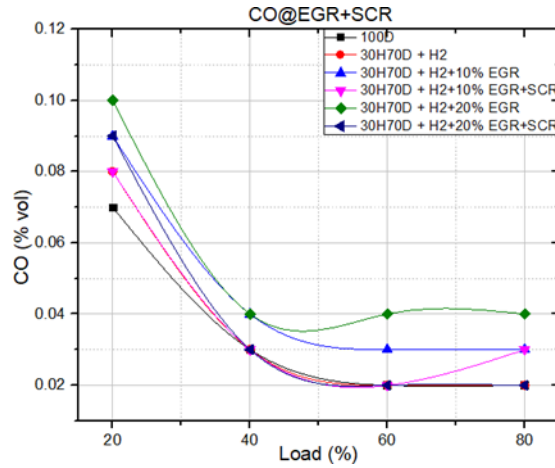
Fig. 4.56 Change in CO with load for all the test fuel with EGR and SCR

The effect of EGR and SCR of NO_X on CO emission at all loading conditions is shown in Fig 4.56. At 60% load, the increment in CO emission for 30H70D+H_2 10% EGR rate is 33.33% and at 20% EGR rate 50% higher compared to no EGR condition. The main cause for the formation of CO emission with EGR is the unavailability of enough oxygen in the combustion chamber. At the 10% EGR rate the increment in the CO emission is low compared to the 20% EGR rate, which is due to that, when the exhaust gas is re-circulated it re-burns the CO with it, and also the availability of free radicals in the exhaust helps for complete combustion and reduces the CO emission. However, with an increase in the rate of EGR, the inert nature of EGR dominates this effect and increases the CO in the emission (Verma et al. 2019). Also, the inert nature of exhaust gas further reduces the in-cylinder temperature and make a suitable environment to form CO emission.

It is noted that in the presence of Pt-Pd catalyst the CO emission was reduced and it is almost equal to the CO content of diesel at both 10% and 20% EGR rates. The highest CO conversion has happened at ~140°C. Hence the lowest CO was observed at 40% and 60% load for both 10% and 20% EGR rates. At 60% load in the presence of SCR, the CO emission for 30H70D+H_2 is 33.33% lower compared to 10% EGR rate condition and 50% lower compared to 20% EGR rate. HC in the exhaust acts as a reducing agent on the surface Pt:Pd catalyst and oxidation of CO happens with the presence of HC in the exhaust (Kang et al. 2018).

$$CO + NO \rightleftharpoons 0.5N_2 + CO_2 \qquad\qquad 4.3$$
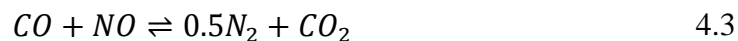
The reaction in Eq. 4.3 shows that CO reacts with NO and forms N_2

$$CO + H_2O \rightleftharpoons CO_2 + H_2 \qquad\qquad 4.4$$

The reaction in Eq. 4.4 is a water-gas shift reaction.

$$C_mH_n + 2mH_{2O} \rightarrow mCO_2 + (2m + 0.5n)H_2 \qquad\qquad 4.5$$
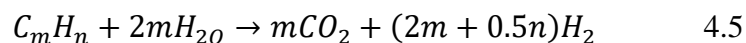
The reaction in Eq. 4.5 is a steam reforming reaction. Both water-gas shift reaction and steam reforming reaction remove the carbon monoxide and hydrocarbon from the exhaust.

4.7.3.4 Nitrogen Oxide Emission

The effect of EGR and SCR on NO_X emission at all loading conditions is shown in Fig 4.57. The NO_X emission decreases with the addition of EGR and its decrement increases with a higher concentration of EGR. At 60% load, the decrement in NO_X emission for 30H70D+H_2 10% EGR rate is 3.29% and at 20% EGR rate 5.16% lower compared to no EGR condition. The EGR dilution increases the ID period and delays the start of burning and heat release. The exhaust gas contains CO_2 and H_2O, which acts as a heat sink when the exhaust gas enters the combustion chamber and absorbs the heat to disassociate CO_2 and H_2O molecule, which is an endothermic reaction. The exhaust gas increases the specific heat capacity of the combustible mixture, which makes a lower combustion temperature. On the other hand, EGR makes the unavailability of oxygen for combustion. The total negative effect on combustion reduces the in-cylinder temperature, NO_X is a function of in-cylinder temperature hence lower NO_X is observed (Verma et al. 2019).
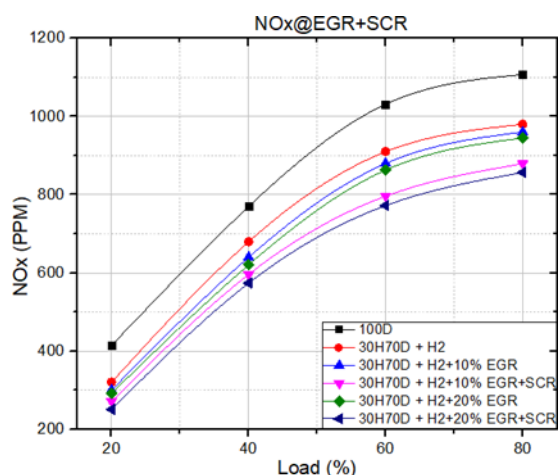
Fig. 4.57 Change in NO_X with load for all the test fuel with EGR and SCR

It is noted from the graph that, Pt-Pd catalyst further reduces the NO_X emission in the presence of NH3 as a reducing agent. At 60% load in the presence of SCR, the NO_X emission for 30H70D+H_2 is 9.54% lower compared to 10% EGR condition and 10.54% lower compared to the 20% EGR condition. When the exhaust gas is recirculated, it contains HC and CO emissions from the previous cycle. The HC and CO emission in the exhaust acts as a reducing agent and participant in the reaction to reduce NO_X emission. It can be seen from the graph.

The NO_2 oxidized into NO in the presence of O_2.

$$NO + \frac{1}{2}O_2 \rightarrow NO_2 \qquad\qquad 4.6$$

In the presence of NH_3, the reduction of NO is shown in Eq. 4.7 and 4.8.

$$4NH_3 + 4NO + O_2 \rightarrow 4N_2 + 6H_2O \qquad 4.7$$

$$2NO_2 + 4NH_3 + O_2 \rightarrow 3N_2 + 6H_2O \qquad 4.8$$
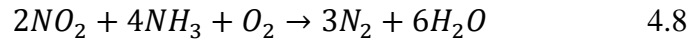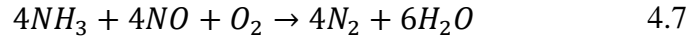
In this equation, the NO is reduced to N_2 in the presence of NH_3. The O_2 consumes H_2 in the NH_3 to form H_2O.

$$4NH_3 + 2NO + 2NO_2 \rightarrow 4N_2 + 6H_2O \qquad 4.9$$
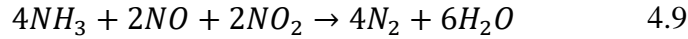
In this reaction, there is no participation of O_2. In the presence of a reducing agent NO is reduced to N_2.

$$6NO + 4NH_3 \rightarrow 5N_2 + 6H_2O \qquad 4.10$$

$$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O \qquad 4.11$$
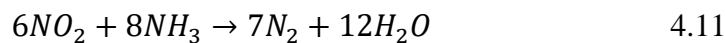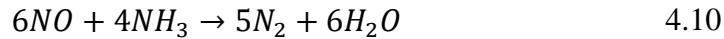
Eq. 4.7 is a standard SCR reaction that contains stoichiometric with identical amounts of NH_3 and NO. Eq. 4.9 is a fast SCR reaction, its reaction rate is 10 times faster than the reaction Eq.8 above 200 °C.

4.7.4 Summary

- The application of EGR negatively affects the combustion process. The addition of EGR replaces the considerable amount of fresh air inducted into the engine. The combustion characteristics of the engine are less affected by the 10% EGR rate, however, 20% EGR rate the reduction of CP, MGT, and NHR is high.

- Both thermal efficiency and fuel consumption of the engine are affected by the application of EGR. The highest decrement in BTE is 10.20% which is seen at 60% load for 20% EGR rate.

- The SCR of NO_X is an after-treatment technique that does not participate in the combustion reaction in the chamber. Hence SCR did not affect combustion and performance characteristics.

- The addition of EGR had a negative effect on both HC and CO emissions. However, the NO_X emission decreases with an increase in the rate of EGR.

- SCR of NO_X had a positive effect on all three emissions. Both HC and CO emission in the exhaust acts as reducing agents in the reduction reaction of NO_X.

4.7.5 Conclusion

Both combustion and emission characteristics of the engine deteriorate with the application of EGR. The 20% EGR rate had worsened effect on the engine characteristics of the diesel engine. SCR of NO_X is an after-treatment technique that had no effect on combustion and

performance. The application of SCR of NO_X effectively reduces HC, CO, and NO_X emissions in the exhaust. The loss that happened with EGR in exhaust emission was fulfilled by the SCR of NOx. It is concluded that 20% of EGR had a negative effect on engine characteristics except for NO_X emission. The application of SCR of NO_X effectively reduces HC, CO, and NO_X emissions.

CHAPTER 5

CONCLUSION

The main focus of the present investigation is to find a suitable higher alcohol blend for the diesel fuel without compromising the performance and emission characteristics of the CRDI CI engine. In this investigation, the effect of different alcohol blends and injection times were studied to know the compatibility of higher alcohols in the CI engine. Also, hydrogen enrichment was done to know its behaviour with higher alcohol blends in the CI engine. To mitigate the NO_X emission from the CI engine the in-cylinder strategy like EGR and the after-treatment technique like SCR of NO_X was employed in this investigation. Based on the detailed investigations the following conclusions were drawn.

- For all 1-Pentanol blends, at low load conditions poor combustion, performance and emission characteristics were seen. An improvement in engine characteristics was noted at 80% load. At higher load conditions cylinder temperature was high which suppress the poor fuel properties of 1-Pentanol and makes the complete fuel atomization. Due to incomplete combustion of 1-Pentanol blends showed higher HC, CO and lower NO_X emissions.

- The behaviour of 1-Hexanol blends was similar to 1-Pentnaol blends in the engine. 1-Hexanol showed poor combustion characteristics at low load conditions whereas, at higher load conditions the value of CP, MGT, and NHR was higher compared to diesel fuel.

- The brake thermal efficiency of the engine decrease with increment in the fraction of 1-Hexanol in the blend. Both HC and CO emissions of 1-Hexanol blend in the engine increase at low load conditions whereas, at higher engine load the emissions were nearly equal to diesel fuel operation.

- Better combustion, performance, and emission were noted up to 30% of higher alcohol (1-Hexanol and 1-Pentanol) with diesel. Further increment in the fraction of alcohol, deteriorates the combustion at lower and medium load conditions.

- Compared to 30% of 1-Pentanol blend, 30% of 1-Hexanol showed improved combustion characteristics at all loading conditions. The BTE and BSEC of 30H70D were 5.29% higher and 5.3% lower compared to 30P70D. Compared to the 30% of 1-Pentanol blend, the 30% 1-Hexanol blend showed lower HC and CO emissions. Whereas, the NO_X emission of the engine is higher for 1-Hexanol blend compared to 1-Pentanol.

- Compared to standard and retarded injection time, advanced injection time showed improved combustion, performance, and emission characteristics for 30% 1-Hexanol blend.

- Up to 30 lpm of hydrogen, enrichment showed improved combustion, performance, and emission characteristics compared to diesel alone operation. Whereas, NO_X emission was increased with hydrogen enrichment due to higher LHV of hydrogen.

- The hydrogen enrichment with 30% of 1-Hexanol improves the combustion, performance and emission characteristics of the engine at all loading conditions compared to the 1-Hexanol/diesel alone operation. The hydrogen enrichment reduces the ignition delay caused and improves the combustion.

- The addition of EGR negatively affects the combustion, performance, and emission characteristics of the engine. The unavailability of oxygen for complete combustion with EGR addition was the main reason.

- The 1wt% Pd-Pt SCR catalyst did not affect combustion and performance characteristics. SCR catalyst effectively reduces the HC, CO, and NO_X emissions from the engine.

From the results, it was noted that 1-Hexanol exhibits improved performance compared to 1-Pentanol. Further improvement in 1-Hexanol was noted at advanced injection time and 10 lpm of hydrogen enrichment. Compared to 20% of EGR rate, 10% of EGR rate performs well with a low adverse effect on combustion and performance. The 1wt% of Pt/Pd SCR catalyst effectively reduces the HC, CO, and NO_X emissions. It is concluded that at advanced injection time the 10 lpm hydrogen-enriched 30% of 1-Hexanol/diesel blend at 10% EGR rate with 1wt% of Pt/Pd SCR catalyst improves the performance and reduces the emission compared to diesel fuel operation. It is concluded that 1-Hexanol and hydrogen as sustainable renewable energy sources, which can be considered as promising alternative sources to petro-diesel.

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LIST OF PUBLICATIONS BASED ON Ph.D. RESEARCH WORK

Sl. No	Title of the paper	Authors	Journal Name, Year, Volume Number, Issue, Pages	Month, year of publication	Category*
1	Effect of 1-pentanol addition and EGR on the combustion, performance and emission characteristic of a CRDI diesel engine.	Radheshyam, Santhosh K, G.N. Kumar	**Renewable Energy**, Elsevier, 2019, Volume: 145, Page: 925-936,	June 2019	1
2	Experimental analysis of performance and emission characteristics of CRDI diesel engine fueled with 1-pentanol/diesel blends with EGR technique	Santhosh K, Kumar G. N, Radheshyam, Sanjay P V	**Fuel**, Elsevier, 2020. Volume 267	May 2020	1
3	Experimental analysis of a mini truck CRDI diesel engine fuelled with n-Amyl alcohol/diesel blends with selective catalytic reduction (SCR) as a DeNO$_x$ technique under the influence of EGR	*Santhosh K*, Kumar G N	*Energy Sources, Part A: Recovery, Utilization and Environmental Effects*, *Taylor and Francis,* 2020, page: 1-16	February 2020	1
4	Impact of 1-Hexanol/diesel blends on combustion, performance and emission characteristics of CRDI CI mini truck engine under the influence of EGR	*Santhosh K*, Kumar G N	*Energy Conversion and Management*, *Elsevier,* 2020, Volume 217	August 2020	1

5	Effect of hydrogen and 1-Hexanol on combustion, performance and emission characteristics of CRDI CI engine	*Santhosh K*, Kumar G N	*Fuel*, *Elsevier, 2021,* Volume-285, 119100	February 2021	1
6	Effect of injection time on combustion, performance and emission characteristics of direct injection CI engine fuelled with equi-volume of 1-Hexanol/diesel blends	*Santhosh K*, KumarG N	*Energy*, *Elsevier, 2021* Volume-214, 118984	January 2021	1

Category*   1: Journal paper, full paper reviewed   2: Journal paper, Abstract reviewed

                   3: Conference /Symposium paper, full paper reviewed   4: Conference /Symposium paper, Abstract reviewed

                   5: Others (including paper in workshop, NITK Bulletin, short notes etc.)

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APPENDIX

Mass Emission data for CRDI diesel engine for various fuels in the test.

HC Emission g/kW-hr

Sl. No	Fuel	Load			
		20%	40%	60%	80%
1	100D	0.061	0.049	0.036	0.032
2	30H70D+H_2	0.081	0.061	0.040	0.036
3	30H70D+H_2+10% EGR	0.085	0.065	0.044	0.040
4	30H70D+H_2+10% EGR+SCR	0.069	0.053	0.032	0.032
5	30H70D+H_2+20% EGR	0.093	0.069	0.053	0.049
6	30H70D+H_2+10% EGR+SCR	0.077	0.061	0.036	0.032

CO Emission in g/kW-hr

Sl. No	Fuel	Load			
		20%	40%	60%	80%
1	100D	6.15	2.63	1.75	1.75
2	30H70D+H_2	7.03	2.63	1.75	1.75
3	30H70D+H_2+10% EGR	7.90	2.63	1.75	1.75
4	30H70D+H_2+10% EGR+SCR	7.03	2.63	1.75	2.63
5	30H70D+H_2+20% EGR	8.78	3.51	3.51	3.51
6	30H70D+H_2+10% EGR+SCR	7.90	2.63	1.75	1.75

NO$_X$ Emission in g/kW-hr

Sl. No	Fuel	Load			
		20%	40%	60%	80%
1	100D	3.89	7.24	9.70	10.42
2	30H70D+H_2	2.91	6.4	8.5	9.33
3	30H70D+H_2+10% EGR	2.84	6.02	8.39	9.24
4	30H70D+H_2+10% EGR+SCR	2.54	5.64	7.51	8.42
5	30H70D+H_2+20% EGR	2.82	5.74	8.09	9.13
6	30H70D+H_2+10% EGR+SCR	2.35	5.46	7.34	8.19