COMBUSTION AND EMISSIONS CHARACTERISTICS OF AN ETHANOL AND DIESEL FUELED PARTIALLY PREMIXED CHARGE COMPRESSION IGNITION-DIRECT INJECTION ENGINE

By
SHYAM PANDEY
College of Engineering, UPES, Dehradun

Under the Guidance of
Dr. PARAG DIWAN
University of Petroleum and Energy Studies, Dehradun
&
Co-Guidance of
Dr. PRADEEPTA K. SAHOO
Mechanical Engineering Department, UPES
&
Co-Guidance of
Dr. SUKRUT SHRIKANT THIPSE
Automotive Research Association of India, Pune

Submitted

IN PARTIAL FULFILLMENT OF THE REQUIREMENT OF DEGREE OF
DOCTOR OF PHILOSOPHY
TO
UNIVERSITY OF PETROLEUM AND ENERGY STUDIES,
DEHRADUN

JUNE, 2014

ACKNOWLEDGEMENTS

The last seven years at University of Petroleum and Energy Studies have been a wonderful experience as a faculty and student. I recall being asked a couple of years ago what I would do if I won the lottery. I thought about it for a few moments and replied that as far as where I was, and what I was doing, I would not change a thing. This realization made me very aware of how lucky I was to be surrounded by such great people in an environment where I could learn and grow as a person. Above all it has been the people that I have met along the way that has been the most special part of both my professional as well as personal life from whom I have learned how to live, enjoy and work.

First and foremost, I want to thank my parents, for all of the wonderful things they have done for me. Although my mother, Late Shrimati Kamla Devi, passed away when I was child, I vividly remember her spirituality and simplicity. Likewise, my father, Late Shri Purshottam Pandey, passed away in 2005. I remember him as a highly brave and committed individual; I ever see in my life. My father is always my biggest role model. I also, want to thank my two elder brothers, Umashankar and Ramashankar Pandey, they did a lot for me, sacrificed their happiness to make me happy. I learnt a lot about commitment through my Ramashankar Bhaiya, as I found myself doing nothing as compared to him.

I would like to acknowledge Dr. S. J. Chopra, Chancellor, University of Petroleum and Energy Studies, Dehradun gratefully for his encouragement and guidance. He is my eternal source of inspiration.

I wish to take this opportunity to express my sincere thanks and regards to my supervisors; I express my deep sense of gratitude to the Dr. Parag Diwan, Vice Chancellor, University of Petroleum and Energy Studies, Dehradun. The zeal and dedication which I found in him is immeasurable. I shall always keep him in high esteem. I will remain indebted to him for his excellent guidance in this study. My co-supervisor, Dr. Pradeepta Kumar Sahoo, Associate Professor, College of Engineering, University of Petroleum and Energy Studies, Dehradun. God has given him great perseverance which I admire a lot. His understanding, encouraging and personal guidance have provided a sustained basis for the present thesis. My external supervisor Dr. S. S. Thipse, General Manager (Power Train Engineering ARAI Pune), also given me insight about the combustion and alternate fuel, he helped me a lot whenever I asked for the same.

I am deeply grateful to my senior colleague, Professor D K Gupta, University of Petroleum & Energy Studies, Dehradun, for his encouragement and support throughout this work.

I wish to express my warm and sincere thanks to Professor Mukesh Saxena, PVC, UTM, Shilong for guiding me throughout the work. I owe my most sincere gratitude to Prof. Utpal Gosh, Dr Shrihari, and Dr. Kamal Bansal for their personal guidance and suggestions. Dr. D. N. Saraf, Prof R P Soni and Dr. Pradeep Joshi helped me lot in terms of thesis writing. I wish to thank my colleague Prof. G. G. Sastri, Prof. Narayan Khatri, and Prof. P S Ranjit who always there in the laboratory to catch me before I broke out, they always provided technical and moral support to accomplish the target.

I wish to thank Prof. Sasi Sekhar M., Prof. Rajeshwar Mahajan, and Prof. Sanket Goel, University of Petroleum and Energy Studies, Dehradun for going through the literatures and thesis writing. I wish to thank Mr. Pradeep Singh, for his immense support in laboratory. During this work I have collaborated with many colleagues for whom I have great regard. I wish to extend my warmest thanks to all those who have helped me directly or indirectly.

I am grateful to my family for their continuous encouragement and sacrifice. They are the source of inspiration for me throughout my PhD work. My loving thanks to my sons, Parth and Shiv for their love and support. I owe my loving thanks to my wife Mrs. Anupama Pandey. She sacrificed a lot due to my research work. Without her encouragement and understanding it would have been impossible for me to finish this work.

Shyam Pandey

DECLARATION BY THE SCHOLAR

I hereby declare that this submission is my own work and that, to the best of my knowledge and belief, it contains no material previously published or written by another person nor material which has been accepted for the award of any other degree or diploma of the university or other institute of higher learning, except where due acknowledgment has been made in the text.

(Signature)

CERTIFICATE

This is to certify that the thesis entitled **"Combustion and emissions characteristics of an ethanol and diesel fueled partially premixed charge compression ignition – direct injection engine"** submitted by **Shyam Pandey** to **University of Petroleum and Energy Studies, Dehradun** for the award of the degree of **Doctor of Philosophy (Engineering)** is a bona fide record of the research work carried out by him under our supervision and guidance. The content of the thesis, in full or parts have not been submitted to any other Institute or University for the award of any other degree or diploma.

Internal Research Supervisor:	**Dr. Parag Diwan, Vice Chancellor University of Petroleum & Energy Studies, Dehradun**
Internal Research Co-Supervisor:	**Dr. Pradeepta Kumar Sahoo Associate Professor (Mechanical Engineering Department) UPES**

Place:

Date:

CERTIFICATE

This is to certify that the thesis entitled **"Combustion and emissions characteristics of an ethanol and diesel fueled partially premixed charge compression ignition–direct injection engine"** submitted by **Shyam Pandey** to **University of Petroleum and Energy Studies, Dehradun** for the award of the degree of **Doctor of Philosophy (Engineering)** is a bona fide record of the research work carried out by him under our supervision and guidance. The content of the thesis, in full or parts have not been submitted to any other Institute or University for the award of any other degree or diploma.

External Research Supervisor:

Dr. Sukrut Shrikant Thipse
General Manager
Power Train Engineering,
ARAI, Pune

Place:
Date:

CONTENTS

S.No. *Tiltes* *Page No*

EXECUTIVE SUMMARY

The present work focus on development of partially homogenous charge compression ignition (PCCI) engine fueled with ethanol and diesel. Conventional diesel combustion could be considered to be replaced by PCCI or partial homogenous charge compression ignition (HCCI) combustion. PCCI is the concept of homogeneous charge diesel combustion using port/manifold injection of premixed fuel with high octane number. This is a compromise between HCCI and compression ignition direct injection (CIDI), since enormous amount of effort has been envisaged for HCCI combustion at higher load and speed. Moreover, it is very difficult to control the ignition timing at elevated loads (Yao and Zhaolei Liu 2009). Successful conversion of CIDI engine into PCCI engine is used to study combustion characteristics, emissions and performance at three different engine loads using diesel and ethanol as a fuels. It differs from conventional CIDI in that it can further enhance fuel-air mixing prior to combustion to reduce the peak temperature of combustion and in turn reduces emissions without significantly affecting thermal performance. Thus, modified mode of combustion is successfully used to reduce NO_x and smoke emissions simultaneously.

Chapter one describes alternative fuels used in diesel engines, fundamentals of HCCI combustion and requirement of PCCI combustion. As HCCI combustion mode allows premixing of air-fuel mixture and high compression leading to auto-igniting the charge. The combustion phenomenon offers the potential of nearly zero NO_x and soot emissions, while still providing diesel-like efficiencies. Subsequently it explained about pioneering research work done by Onishi et al (Onishi and Jo 1979), Noguchi et al. (Noguchi and Tanaka 1979), (Najt PM and DE 1983) and Thring (Thring 1989). R-&-D barriers such as combustion phasing control at elevated load, narrow operating range, and cold start capability are some of the restraints. The techniques which are adopted to overcome above mentioned barriers are multiple fuel injections, early direct injection, late direct injection along with high rate of EGR, and port/manifold fuel injection (fumigation) along with direct injection of diesel. PCCI mode of combustion requires two fueling system: one is high pressure fueling for diesel injection directly into the combustion

chamber and other low pressure fuel injection into the air intake manifold. Since last two decades, premixed mode of combustion is being considered to reduce emission rather to use as a power booster technique. This is due to stringent emission norms and scarcity of fossil fuel, poses challenges to the oil engine manufacturers, thus lot of research work is undergone to reduce emission and increase the use of alternate fuels. Thus, four objectives for the thesis are explicitly defined. Firstly, modification of an existing single-cylinder four stroke direct injection diesel engine to operate on the partially premixed charge compression ignition combustion mode. Secondly, study the effects of varying mixture compositions of premixed ethanol combustion at different engine loads. Thirdly, comparative performance, emission and combustion analysis of the developed PCCI engine fueled with ethanol and diesel in different proportions. Fourthly, achieve simultaneous reductions of NO_x and smoke with the help of premixed ethanol and EGR.

Chapter two describes detailed literature review. Combustion phenomenon in diesel engine and formation of diesel exhaust pollutants like NO, NO_2, NO_x, particulate matter, unburned hydrocarbon and carbon monoxide are reviewed. Subsequently emission reduction techniques, fuel injection timing, and exhaust gas re-circulation (EGR) which can reduce in-cylinder NO_x generation. Diesel exhaust after-treatment technologies like Diesel Oxidation Catalyst (DOC), Diesel Particulate Filter (DPF), Selective Catalytic Reduction (SCR), and lean NO_x trap also showed strong potential to cut-down tail pipe emissions. Extensive review is done on the role of oxygen-containing compounds used as a fuel oxygenates to reduce soot emission or to supply oxygen to form CO_2 and reduce carbon rich particles. Considerable efforts has been done on oxygenates such as alcohol, methyl or ethyl esters or bio-diesel, ethers and ethylene glycol monoacetate. However, alcohol is one of the most promising alternative fuels. Probably the reason behind enormous research work carried out in the different part of world could be due to availability of alcohol from bio-sources/bio-mass and their excellent physico-chemical properties. However, especially for ethanol it is explained hereunder in slight more details.

The global fuel crises in the 1970s jolted policy-makers across the globe to thought about depletion of fossil fuel. Considerable attention was focused on the development of alternative fuel sources, with particular reference to the alcohol. A lot of research work has been carried out to examine feasibility of ethanol along with gasoline and diesel in the last three decades. Ethanol is an attractive alternative fuel because it is a renewable bio-based resource and it is oxygenated, thereby providing the potential to reduce particulate emissions in compression ignition engines. However, when considering ethanol as an alternative fuel for use in diesel engines, a number of issues are important. Includes blend properties such as stability, viscosity and lubricity, safety and materials compatibility. The effect of the fuel on engine performance, durability and emissions is also considered. Solubility of ethanol in diesel is one of the major concerns, and it is affected by water content in the blend and temperature. Therefore to prevent separation two methods are commonly used: adding an emulsifier which acts to suspend small droplets of ethanol within the diesel fuel, or by adding a co-solvent that acts as a bridging agent through molecular compatibility and bonding to produce a homogeneous blend (Letcher 1983). Pure Energy Corporation (PEC) have developed additive package that allow ethanol to splash-blended with diesel fuel using a 2-5% dosage with 15% anhydrous ethanol and proportionally lees for 10% blend. PEC is US based company and they are not supplying their product in India, it was communicated by the company way back 2012. So, above mentioned characteristics of ethanol impedes usage of ethanol in diesel engines especially remotely located villages for decentralized power generation, irrigation and in automotive. Nevertheless, to avoid difficulties associated with blending, lubricity and viscosity of ethanol-diesel fuel alternative mode of combustion known as a PCCI is one of the promising way for effective utilization of ethanol. Since it allow to inject diesel and ethanol separately in combustion chamber and intake air manifold respectively. In addition to this hydrous ethanol can also be used without any difficulties and it can overcome most of the limitations of ethanol-diesel blending.

Countries like China (Tsang and Zhang 2010; Zhang Z.H. and Tsang 2011), Jordan (Abu-Qudais and Haddad 2000; Padala and Woo 2012) and in India (Ajav E. A. and Bachchan Singh 1998; Bhupendra Singh Chauhana and Naveen Kumar 2011; Hebbar and Anantha 2012) researchers have worked on premixed ethanol along with diesel. They showed NO_x and smoke could be reduced by using premixed ethanol but BTE decreases at low load and no or small positive effect at high loads. However, only one work is reported on premixed ethanol along with EGR (Hebbar and Anantha 2012), moreover this did not includes combustion study, effect of diesel/ethnaol injection timing, and has lack of clarity in ethnaol injection mechnaism. As per the reported literatures using ethanol in most of the cases leads to reduce brake thermal efficiency. All reported work showed reduction in NO_x and smoke emision with the help of premixed ethnaol. So this study is focused to understand effect of mixture compositions of premixed ethanol with and without EGR in conjunction with late direct injection strategy. As per all available literature the effect of late diesel injection timing along with premixed ethanol-EGR is not studied in single-cylinder direct injection diesel engine.

Chapter three describes development of test-rig to perform PCCI combustion study, which is the major objective of this thesis. Accordingly, Kirloskar single-cylinder naturally aspirated, four strokes, constant speed, water cooled DI diesel engine of rated power 3.5 kW and displacement volume of 0.661 liter is used for the studies. Test-rig is enabled to provide with necessary instruments for in-cylinder pressure, fuel injection pressure, airflow, fuel flow, temperatures and load measurements. However, fuel injection pressure sensor gives precisely fuel line pressure at each crank angle position. As the injection starts shortly after the line pressure begins to rise or it can overcome spring force applied on injector needle. Thus, maximum fuel line pressure is considered as a reference to determine the injection event. This information is required to find out ignition delay period. Various sensors and instruments used for measurements are integrated with computerized data acquisition system. A panel box consisting of air box, fuel tank, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and engine indicator. Rotameters are provided for cooling water and calorimeter water flow measurement. The thermal performance study of engine like brake power, BMEP, brake thermal efficiency and specific fuel consumption. Lab view based engine performance analysis software package “EnginesoftLV” is used for on-line performance evaluation. Emissions measurement is been done using AVL-DIGAS- 444 analyzer and smoke meter (437-c) make AVL.

The main challenge is to carry out combustion, performance and emission study on PCCI test rig, requires so many adaptations and it took lot of time. The design and development has been done on six different components and aspects, which includes, electronically controlled ethanol fuel injection into the air intake manifold, separate direct diesel fuel injection mechanism, development of mechanical EGR system, design and fabrication of EGR cooler, and modification of piston geometry and development of air preheating system. Nevertheless, later two components haven't shown remarkable impact to achieve objective therefore not reported in the thesis, they may be considered as a failure.

First and far most important thing is ethanol injection system. It includes electric fuel pump and low pressure (3 bar) solenoid operated injector. Electronic control ECU to control timing and quantity of ethanol fuel being injected into air intake manifold. Second is the development of standalone in-line fuel injection pump (2-cylinder engine) for diesel fuel, this is basically cylinder-and-piston pumping unit driven by a shaft with eccentrics connected with respective intake and delivery valves to inlet and outlet means for the fuel. Fuel delivery from one high pressure line is connected to the injector and second one is redirected to intake line. The drive for this in-line fuel pump is supplied from the camshaft to toothed pulley and via toothed belt it is fed to the pump. Electronic EGR valve is also tried but could not works out with existing engine therefore, mechanical EGR system is developed.

Chapter four explained about design and development of small EGR cooler. As the available EGR cooler are designed for automotive applications where the flow rate and inlet outlet temperature are completely different compared to small capacity diesel engines. D Q Kern (Kern 1997) method is used to design shell and tube type heat exchanger.

Chapter five describes analysis of results and discussions. Experiments were performed at a constant engine speed of 1500 revolutions/min (±10) and three different loads corresponding to BMEP of 2.1 bar (50% load), 3.1 bar (75% load), and 4.2 bar (100% load), or 1.75 kW, 2.6 kW and 3.5 kW were selected to examine the effect of premixed ethanol with and without EGR (exhaust gas re-circulation). One set of analysis was done keeping constant EGR ratio with varying PFR and in another case PFR is kept fixed whereas EGR ratio was varied. Initially engine is started by using diesel alone thereafter manifold fuel or ethanol is injected into air intake manifold. Ethanol injection is controlled by ECU that precisely maintains its injection timing and quantity of fuel being injected. Thus, soon after the beginning of ethanol injection engine rpm starts increasing due to over-fueling and that could be brought down by reducing quantity of diesel being injected, governor rack provided in the in-line fuel injection pump is adjusted accordingly. EGR ratio is determined by using CO_2 ratio in the intake manifold and in the exhaust. Physico-chemical properties analysis of fuels like alcohol (ethanol and methanol), bio-diesel and diesel are carried out. calorific value, viscosity coefficient, flash point, and density of fuels are measured in the laboratory. Combustion and performance parameters line net heat release rate, and mass fraction burnt, premixed combustion duration, ignition delay, premixed fuel ratio, BSFC, BTE and EGR ratio is explained. So as to achieve simultaneous reduction of NO_x and smoke optimizing injection timing of both diesel and ethanol was done before performing tests.

The effect of PFR and EGR ratio at BMEP of 2.1 bar and 3.1 bar showed reduction in in-cylinder pressure and temperature, decrease in net heat release rate, increase in mass fraction burnt and increase in ignition delay due to premixed ethanol and EGR. However, small increase in peak in-cylinder pressure and temperature is found at BMEP of 4.2 bar. Also, fraction of fuel burnt in the diffusion phase is improved. Brief descriptions of results obtained are presented in the subsequent paragraphs.

At BMEP of 2.1 bar (50% load), it was found that significant reduction in-cylinder pressure is caused by premixing of ethanol and EGR. This leads to increase in ignition delay period and adversely affects thermal performance but significantly reduce NO_x emission. In case of PFR18EGR0 about 50% reduction in NO_x emission was found due to premixed ethanol compared against diesel (PFROEGR0). Also, it increased to 90% and 95% while operating on intake mixture of PFR18EGR10 and PFR18EGR20 respectively. Effect of EGR alone could be seen and compared to diesel (PFR0EGR0) showed about 77% and 89% reduction in NO_x emissions in case of PFR0EGR10 and PFR0EGR20 respectively. At PFR18EGR 20, UHC and CO emissions is increased by

6 and 10 times respectively compared to PFR0EGR0. However, BSFC increased by 15% in case of PFR18EGR20 compared to PFR0EGR0. The maximum reduction in brake thermal efficiency is recorded in case of PFR18EGR20 of about 2.4% compared to pure diesel combustion. However, EGR did not showed significant impact on BSFC and BTE. Positive change in BTE is found with premixed ethanol (PFR18EGR0), it was increased by 4.48% compared to PFR0EGR0.

At BMEP of 3.1 bar (75% load) premixed ethanol and EGR ratio leads to increase ignition delay period. The effect of increasing premixed fuel ratio in all conditions results in decrease in in-cylinder pressure and net heat release rate. Intake mixture PFR0EGR20 showed about 92% reduction in NO_x emission compared to PFR0EGR0. However, combined application of premixed ethanol and EGR increases it up to 95%. PFR15EGR0 lead to reduce NO_x emission by 33% compared to PFR0EGR0. It was found that compared to 2.1 bar (BMEP) engine load cooling effect produced by ethanol is weakened leading to lower reduction in NO_x emission. Also, UHC and CO emissions increases with increase in premixed ethanol and EGR. Maximum increase is found while operating with PFR15EGR20, UHC and CO emissions is increased by 7.6 and 4.7 times respectively compared to PFR0EGR0

At BMEP of 4.2 bar (100% load) premixed ethanol and EGR showed better results compared to other two loads. In case of PFR0EGR10 and PFR0EGR20, NO_x emissions decreases to about 55% and 88% respectively compared to PFR0EGR0. Similarly, PFR13EGR10 and PFR13EGR20 showed reduction of about 58% and 91% respectively compared to PFR13EGR0. Smoke opacity corresponds to PFR13EGR10 and PFR20EGR10 decreased by 45% and 56% respectively compared to PFR0EGR0. Therefore simultaneous reduction of NO_x and smoke was observed. Also, at PFR20EGR0 about 34% reductions in smoke were found compared to PFR0EGR0. Abu-Qudais et al. (Abu-Qudais and Haddad 2000) also showed similar results, they have found maximum decrease in smoke opacity of 33 to 51% corresponds to 20% premixed ethanol at various engine speeds. UHC and CO emissions are increases with increase in premixed ethanol and EGR. Maximum increase in UHC and CO is found in case of PFR20EGR20, about 4.85 and 2.25 times respectively compared to PFR0EGR0. Significant improvement in the combustion characteristics is found at high EGR (20% EGR). BTE showed negative trends with increase in premixed fuel ratio; however there is a marginal gain in BTE was found in case of PFR20EGR10 and PFR20EGR20. It was found that BTE is increased by 2% and 1.67% respectively compared to PFR0EGR0.

Nevertheless, because the addition of oxygen into diesel fuel produces a shift in the NO_x-smoke trade-off behavior, once the smoke emissions had been reduced by the presence of an oxygenate the emissions of NO_x can be indirectly decreased by further utilization of EGR. Hilden et al. (Hilden and Eckstrom 2001) also showed utilization of oxygenated fuels combined with higher EGR ratio without any adverse effect.

It could be concluded that premixed ethanol cools the air/fuel mixture because of its higher latent heat of vaporization (840 kJ/kg) compared to diesel fuel (270 kJ/kg) which reduces in-cylinder temperature and reduces compression work done. However, slow burning and flame quenching leads to increase UHC emissions. Premixed phase combustion increases either by increasing premixed ethanol or by increasing EGR ratio. Effect of high EGR with premixed ethanol at BMEP of 4.2 bar, showed no adverse effect on performance. This could be due to higher in-cylinder temperature coupled with raised intake charge temperature (EGR) reduces the cooling effect produced by ethanol. Thus, it results into better premixing within and outside the cylinder, hence, in-cylinder pressure, net heat release rate and performance remains unaffected.

It was found that there is significant effect of injection timing of ethanol and diesel on engine combustion, performance and emissions parameters. It was established that effect of EGR at full engine load enhances premixing of air-ethanol-EGR and in turns showed marginal gain in brake thermal efficiency. However, EGR does not shown any gain in brake thermal efficiency at BMEP of 2.1 bar (50% load) and at BMEP of 3.1 bar (75% load) but it lead to decrease NO_x emission significantly. Four holes nozzle which improves the atomization of diesel and spray pattern. So, it could be used in small diesel engines as an alternate mode of combustion with bio-ethanol to reduce harmful pollutants (NO_x and smoke) at three-fourth and full engine loads without affecting fuel economy even marginal gain in brake thermal efficiency.

LIST OF SYMBOLS

μ	Dynamic viscosity
c_p	Specific heat at constant pressure
k	Thermal conductivity
ρ	Density
°C	Degree Celsius
$\theta_{p,EtOH}$	Premixed phase combustion in crank angle degrees for PCCI combustion mode
$\theta_{p,diesel}$	Premixed phase combustion in crank angle degrees for diesel
$\theta_{p,EGR}$	Premixed phase combustion in crank angle degrees with EGR
%	Percent
cc	Cubic centimeter
cm	Centimeter
m	Meter
$\dot{m}$	Mass flow rate
Q	Heat transfer
C’	Clearance
P_T	Triangular Pitch
B	Baffle spacing
Gs	Mass velocity of shell side fluid
Gt	Mass velocity of tube side fluid
a_s	Flow area for shell side fluid
D_e	Equivalent diameter for the shell
k	Thermal conductivity
J_H	Factor to determine heat transfer coefficient
h	Heat transfer coefficient
h_i	Heat transfer coefficient of fluid flowing through tube side
h_o	Heat transfer coefficient of fluid flowing through shell side
a'_t	Flow area from a single tube
D_{is}	Inside diameter of shell

D_{it}	Inside diameter of tube
D_{ot}	Outer diameter of tube
a_t	Flow area though tubes
G_t	Mass velocity of tube side fluid
U_D	Design overall heat transfer coefficient
U_D	Design overall heat transfer coefficient without fouling
hr	Hour
kg	Kilogram
kW	Kilowatt
L	Stroke length
min	Minutes
ml	Milliliters
N	Newton
rpm	Revolutions per minute
Re	Reynolds Number
s	Seconds
V	Volume of cylinder

LIST OF ABBREVIATIONS

ATAC	Active thermo atmosphere combustion
aTDC	After top dead centre
bTDC	After top dead centre
BDC	Bottom bead centre
BTE	Brake thermal efficiency
BSFC	Brake specific fuel consumption
BHP	Brake horse power
BMEP	Brake mean effective pressure
BSCO	Brake specific carbon monoxide
BSHC	Brake specific unburned hydrocarbon
$BSNO_x$	Brake specific nitrogen oxides
BTE	Brake thermal efficiency
BTX	Benzene, Toluene and Xylene
CA	Crank angle
CAI	Controlled auto-ignition
CAA	Clean air act
CAI	Controlled auto ignition
CFR	Cooperative fuels research
CI	Compression ignition
CIDI	Compression ignition direct injection
CN	Cetane number
CO	Carbon monoxide
COV	Coefficient of variation
Cyl	Cylinder
cSt	Centistokes
DI	Direct injection
DEE	Diethyl ether
DME	Dimethyl ether
DOC	Diesel oxidation catalyst

DPF	Diesel particulate filter
ECU	Electronic control unit
EGR	Exhaust gas re-circulation
EPA	Environmental protection agency
EVO	Exhaust valve opening
FAME	Fatty acid methyl ester
FIE	Fuel injection equipment
UHC	Unburned hydrocarbons
HCCI	Homogeneous charge compression ignition
HCCI-DI	Homogenous charge compression ignition-direct injection
HPCR	High pressure common rail
HTR	High temperature reaction
IC	Internal combustion
ID	Ignition delay
IMEP	Indicated mean effective pressure
IDI	In- direct injection
IVOP	Inlet valve opening pressure
INCA	Integrated calibration and acquisition system
ISFC	Indicated specific fuel consumption
JSR	Jet stirred reactor
LTR	Low temperature reaction
LMTD	log mean temperature difference
LTC	Low temperature combustion
MCC	Mixing-controlled temperature combustion
MFB	Mass fraction burned
MTBE	Methyl tertiary butyl ether
MON	Motor octane number
NHRR	Net heat release rate
NMHC	Non-methane hydrocarbon
NO_x	Nitrogen oxides

NO	Nitric oxide
NO_2	Nitrogen dioxide
NDIR	Non dispersive infrared method
PAH	Polycyclic aromatic hydrocarbon
PAN	Proxy acetyle nitrate
PFI	Port fuel injection
PFR	Premixed fuel ratio
PCCI	Partially-premixed charge compression ignition
PM	Particulate matter
RON	Research octane number
RME	Rapeseed oil methyl ester
SCR	Selective catalytic reduction
SOC	Start of combustion
SOI	Start of injection
SI	Spark ignition
SMD	Sauter mean diameter
SN	Smoke number
SVO	Straight vegetable oil
TS	Toyota soken
TDC	Top dead centre
TSP	Total suspended Particle
WVO	Waste vegetable oil
BWG	Birmingham wire gauge

LIST OF FIGURES

Figure No		***Page No***

LIST OF TABLES

Table No *Page No*

CHAPTER 1: INTRODUCTION

Increasingly stringent emissions regulations and environmental concerns have caused interest in the development of alternative combustion technologies together with application of renewable fuels for internal combustion engines. However, dwindling fossil fuels and continuously increasing demand of diesel fuel is another reason behind significant research work is being carried out in this direction. As it is evident from the literature, since last three decades ethanol is widely investigated as a partial substitution of diesel fuel. Therefore, two major concerns could be addressed here like, reduce dependence on fossil fuels and to control harmful pollutants, including smoke and NO_x.

Compression ignition engines possess excellent fuel economy and drivability is the motivation behind extensive research and development to reduce harmful emissions such as particularly NO_x and PM. Diesel engines are widely used in all spheres such as agriculture, commercial transportation, automotive and industry. Recent innovations have considerably improved their fuel economy and thermal efficiency. Development of efficient common rail fuel injection systems and application of supercharged or turbocharged engines satisfy the requirements for higher power output. However, new stringent emission norms further raise the bar and recent published literature is focused on the reduction of these pollutants to new lower values. In exhaust after-treatment technologies such as Diesel Particulate Filter (DPF), Selective Catalytic Reduction (SCR), Particulate Matter (PM) and NO_x reduction could be used. This is important since PM could result in enormous concentrations of smaller diameter particles, which has a negative effect on human health.

Moreover, the effects of climate change forces the use of alternative, renewable fuels. In the case of diesel engines, the Straight Vegetable Oil (SVO) or Fatty Acid Methyl Ester (FAME) can be used as a fuel.

The major obstacle in their application is their high viscosity and tendency to increase NO_x emissions. Alcohol has come out as one of the most promising options to displace diesel. This could be done by blending, emulsification and fumigation (injecting secondary fuel into the induction manifold). Diesel exhaust emissions can be improved by either developing new combustion technology such as homogenous charge compression ignition (HCCI) which is capable of reducing emissions and gives better performance while operating with the alternate fuels or using after-treatment technologies reduce emissions.

1.1 HOMOGENEOUS CHARGE COMPRESSION IGNITION (HCCI)

Homogeneous charge compression ignition (HCCI) combustion mode is considered as an improved combustion technique for diesel engines and has the potential to reduce emissions without adversely affecting fuel efficiency (Zhao F and Asmus TW 2003).

Homogenous charge compression ignition is a cutting-edge research area which provides the advantages of both Spark Ignition (SI) and Compression Ignition (CI) engines. In conventional diesel engines, combustion is the result of atomization and vaporization of fuel which is injected at the end of compression, whereas in conventional SI engine combustion is initiated by a spark (high voltage between the electrodes and ionization of air leads to a discharge of current). The HCCI combustion mode allows premixing of air-fuel mixtures and high compression leading to auto-igniting the charge. The combustion phenomenon offers the potential of nearly zero NO_x and soot emissions, while still providing diesel-like efficiencies. The reason behind for low NO_x emissions from HCCI is that it allows the combustion of very lean mixtures combustion, resulting in low combustion temperatures which directly affects NO_x production. Compared to conventional diesel combustion, in HCCI combustion the charge is well mixed, so soot emissions can be very low.

1.1.1 Pioneering Research of HCCI Combustion

HCCI combustion was first suggested as an alternative combustion mode for two stroke IC engines by Onishi et al. (Onishi and Jo 1979) in 1979. This combustion system, designated as “Active Thermo-Atmosphere Combustion” (ATAC), differs from conventional gasoline and diesel engine combustion processes. The drawbacks of two stroke engines are the high level of residual at light loads, and the tendency of run-on combustion when the engine is stopped. Onishi and co-workers turned these deficiencies into strengths by devising a combustion mode (ATAC) that relied on both high level of internal residual and high initial charge temperature. It was found that significant reductions in the UHC emissions and an improvement in fuel economy could be obtained by creating conditions that led to spontaneous ignition of the in-cylinder charge. It is believed that lower UHC emission and lower fuel consumption compared with conventional combustion are obtained by the elimination of misfire and by regular combustion (less cycle-to-cycle fluctuation). ATAC is easily adaptable to two-stroke cycle spark-ignition for stable operation and the absence of misfiring in the engines is obtained in a wide region of the part throttle operation. The shift from the ATAC region to the ordinary spark- ignition region is easily accomplished without any problem. Stable HCCI combustion could be achieved between low and high load limits with gasoline at a compression ratio of 7.5:1 over the engine speed range from 1000 to 4000 rpm.

Noguchi et al. (Noguchi and Tanaka 1979), conducted experimental investigations on reversed crank case scavenging two-stroke cycle engines, employing the special self-ignited combustion process for the purpose of detecting temperatures and pressures in the combustion chamber. This self-ignited combustion was named “TS (Toyota-Soken) combustion”. It was found that self-ignited combustion occurs under relatively low cylinder temperatures and pressures, compared to diesel engine combustion. From optical investigations they noted that ignition took place at numerous points throughout the cylinder with trace of discernible flame front during the combustion. Using spectroscopic methods to detect the intermediate species, they measured high levels of CH_2O, HO_2 and O radicals within the cylinder before auto-ignition. These species are characteristic of low temperature auto-ignition chemistry of larger paraffinic hydrocarbon fuels.

Subsequently, Najt and Foster (Najt PM and DE 1983) extended this work to four-stroke engines and attempted to gain additional understanding of the underlying physics of HCCI combustion. Experiments were performed with blends of paraffinic and aromatic fuels over a range of engine speeds and dilution levels. Since the engine was operated without the benefit of internal residuals like in a two stroke engine, the intake air had to be heated to high temperature in order to obtain HCCI combustion. They concluded that HCCI auto-ignition is controlled by low temperature (below 1000 K) chemistry and the bulk energy release is controlled by high temperature (above 1000 K) chemistry dominated by CO oxidation. Based on chemical kinetics alone they developed a correlation for the energy release rate that explained the observed response of HCCI combustion to changes in compression ratio, equivalence ratio, dilution level; engine speed and fuel type.

Based on these results and previous work of Onishiet al. (Onishi and Jo 1979) and Noguchi et al. (Noguchi and Tanaka 1979), they concluded that HCCI combustion is a chemical kinetic combustion process controlled by the temperature, pressure, and composition of in-cylinder charge. This is unlike traditional SI combustion that relies on flame propagation and diesel combustion that is heavily dependent on the fuel/air mixing.

Najt and Foster (Najt PM and DE 1983) noted that HCCI combustion suffers from lack of control of the ignition process and a limited operating range. These were the same problems identified by both Onishi and Naguchi in two stroke HCCI engines. Furthermore, since external exhaust gas re-circulation (EGR) was used to dilute the mixture, there was little possibility of active radicals being re-cycled into the cylinder that would initiate auto-ignition. While radicals may play a role in HCCI combustion in two-stroke engines, this work demonstrated that they are not required to achieve HCCI auto-ignition.

The studies were later extended by Thring (Thring 1989) to examine the effect of external EGR and air/fuel ratio on the engine's performance. In this work, Thring introduced the terminology homogeneous charge compression ignition (HCCI) that has since been adopted by many other workers to describe this type of combustion process both in gasoline and diesel engines. The limitations of HCCI operation were again noted. The operating regime is restricted to part- loads operation and control of the auto-ignition timing is problematic. It was suggested in this work that the HCCI combustion mode could potentially be integrated into an engine operating strategy, where at part loads the engine operates on HCCI combustion with a transition into SI flame propagation at higher engine loads.

In 1992, Stockinger et al. (Stockinger and Schapertons 1992) showed for the first time that a four-cylinder gasoline engine could be operated on the HCCI mode within a very limited speed and load range by means of higher compression ratio and pre-heating of the intake air. However, in order to develop a production viable gasoline auto-ignition combustion engine for automotive applications, it is necessary to operate it without heating of external charge or extremely high compression ratio or special fuel blends.

As discussed above, a lot of the early efforts were invested into HCCI involved gasoline-fueled engines, and this technology continues to be strongly pursued today. New terminology was also introduced to explain gasoline-fueled HCCI combustion as Controlled Auto-Ignition (CAI) combustion. However, the need for substantial reductions in both NO_x and PM emissions led to investigations into the potential of diesel-fueled HCCI which began in the mid-1990s.

1.1.2 R-&-D Challenges for HCCI

Based on the aforesaid said pioneering research on HCCI combustion, several R-&-D challenges that must be met before the potential benefits of HCCI combustion can be fully realized in terms of high thermal efficiency and low engine tail pipe emissions. HCCI combustion is achieved by controlling the temperature, pressure and composition of the air/fuel mixture so that it auto-ignites near the top dead center (TDC) as it is compressed by the piston.

1.1.2.1 Combustion Phasing Control

In conventional SI engines, combustion timing is easily adjusted by the engine control module by changing the spark timing and perhaps fuel delivery, whereas in case of CI engine fuel, injection event dictates auto-ignition timing. Unlike conventional combustion, a direct control over ignition timing is not available. Instead combustion is achieved by controlling the temperature, pressure and chemistry of air-fuel mixture which, in turn, it can be controlled by residual or EGR rate (Zhao and Peng 2001), air fuel ratio, compression ratio, fuel properties or fuel blends, injection timing of DI gasoline engine and coolant temperature.

Researchers have proposed several methods to control combustion phasing. Some of the most promising methods are varying the amount of hot EGR (Au and Girard 1894) introduced into incoming fresh charge and using variable compression ratio to change the temperature at the end of compression. The Variable valve timing (VVT) technique serves two purposes (Shi and Cui 2006)-it can change the amount of hot residue called ‘internal EGR’ and also the effective compression ratio, at the same time. Fuel additives, variable compression ratio and VVT are particularly attractive because their time response could be made sufficiently fast to handle rapid transient (Mahrous and Potrzebowski 2009). Although, these techniques have shown strong potential yet they are not yet fully proven and, cost and reliability issues are amongst other things to be addressed.

1.1.2.2 Extending the Operating Range for HCCI Engine

One of the major barriers to realizing the potential benefit of HCCI combustion is its very limited operating load range, while maintaining the HCCI mode of operation. Clearly there is a need to extend their range of operation. At higher loads control on auto-ignition of charge is very difficult and combustion can become very rapid and intense, causing unacceptable level of noise, potential engine damage, and eventually unacceptable levels of NO_x emissions. Also, it becomes difficult to control the rate of combustion at higher engine loads since there is no direct means to control combustion, which is possible in SI engine by means of varying the spark timing, and in CI engines by means of fuel injection timing. To overcome the difficulties, an engine can be used in the dual mode, where SI and CI modes of combustion can be used for one-third to one-half of the operating load range and the remainder can be used with the HCCI mode of operation.

Diesel engines have proved to be an efficient and reliable combustion technology, as well as have low maintenance–cost advantages. Currently, all heavy commercial vehicles use diesel fuel with the CI engine. Ideally, it is best to operate the engine in the SI mode at low loads and in the CI mode at high loads with the HCCI mode in the mid-range. SI (Spark Ignition) requires lighter fuel such as gasoline. This call for provision of keeping two fuels on board, this is not feasible. This necessitates that HCCI must be accomplished with diesel fuel on these heavy commercial vehicle engines.

Research work (Lim and Taeck Iida 2012) indicates the operating range can be extended significantly by partially stratifying the charge (temperature and mixture stratification) at high engine loads to stretch out the heat-release event. Some of the methods proposed to control HCCI combustion phasing at elevated loads, includes variable compression ratio (VCR), additives and thermal management. These methods are explained in subsequent discussions.

HCCI combustion is strongly affected by the compression ratio of the engine. Therefore, a VCR engine has the potential to achieve satisfactory operation in HCCI mode over a wide range of conditions because the compression ratio can be adjusted as the operating conditions change. Considering dynamic conditions of vehicular application a fast control system that modifies the compression ratio in fractions of a second is necessary. Several options have been studied to obtain VCR engines. One option is to mount a plunger in the cylinder head whose position can be varied to change the compression ratio (Christensen and Hultqvist 1999). The plunger was controlled by a hydraulic system allowing its position to be varied during engine operation. The data show that the VCR system is capable of controlling HCCI ignition timing to maintain optimal combustion phasing across a very wide range of intake temperatures and fuel types of varying octane number. The compression ratio could also be varied by using an opposed-piston engine design having variable phase-shifting between the two crankshafts (Flynn 1999). Another method that is based on a hinged, tilting cylinder arrangement (Sharke and Paul 2000) to change compression ratio was developed.

Variable valve timing (VVT) can be used to change the trapped compression ratio of the engine, i.e., the amount of compression after the gases are trapped by intake-valve closure and therefore VVT can achieve a similar effect on HCCI combustion as varying the geometric compression ratio of the engine. An engine could be built with a high geometric compression ratio, with lower trapped compression-ratios being obtained by delaying the closing of the intake valve during the compression stroke. Engines with VVT have the added benefit of allowing changes in the temperature and composition of the incoming charge by retaining hot residual gases from the previous cycle in the cylinder. By varying the amount of hot residual, the temperature and mixture of the new charge can be adjusted. Increasing the temperature of the charge in this manner can be used to induce HCCI combustion even with relatively low geometric compression ratios or under cold-engine conditions. In addition, altering the charge composition with partial mixing of the residual could benefit combustion rate control. VVT could be implemented in an engine with mechanical, magnetic, or hydraulic valve actuators (Theobald and Henry 1994). Researchers showed that an electro-hydraulic VVT system, could be successfully used to induce HCCI combustion in an engine with a relatively low (10:1) compression ratio (Kaahaaina and Simon 2001).

Ignition-enhancement by blends and additives they help to achieve stable HCCI combustion, which requires two fuels of different octane ratings. The system could be designed to have a main fuel with a high octane number, while the secondary fuel, with a low octane number, is injected as needed to advance combustion. This procedure has been studied for a combination of methane and dimethyl ether (Flowers and Aceves 1999). For example, according to the reports of Caterpillar Inc in 2004 (Caterpillar 2004), HCCI operation ranges in heavy-duty diesel engines have been extended to 80% loads of the original conventional diesel engine (about 1.6 MPa) by combining the techniques of high boost, improving the mixing rate of fuel and air, EGR and fuel modification. An experimental and chemical analysis carried by Lu et al. (Lu and Ji 2007). A four-cylinder four-stroke high speed diesel engine, reformed to operate on HCCI mode of combustion, two electronic injectors were mounted on the intake pipe at a distance 0.35 m from the intake valve. N-heptane was selected as a baseline fuel along with four suppression additive were used i.e., methanol, ethanol, isopropanol, and methyl tert-butyl ether (MTBE). The results showed that CO emissions strongly dependent on the maximum combustion temperature, UHC emission found to be varied with ratio of ethanol fuel to the total fuel injected.

Simulation results showed that ethanol addition results in retardation of ignition timing, also misfire occurs due to incomplete oxidation of intermediate compounds.

Thermal control is another method to control HCCI combustion by controlling the temperature, pressure, and composition of the mixture at the beginning of the compression stroke. In this methodology, thermal energy from exhaust gas re-circulation (EGR) and compression work from a supercharger are either recycled or rejected to obtain satisfactory combustion (Joel Martinez-Frias and Salvador M. Aceves 2000).

In the HCCI mode of operation, very light load operation is also limited as, there is insufficient thermal energy to trigger auto-ignition of the mixture at the appropriate time. Delay in auto-ignition results in excess CO and UHC emission in combination with low exhaust temperature at close to idle operation. It makes this combustion mode less appealing from a combustion efficiency and emission perspective.

1.1.2.3 Cold-start Capability

After completion of compression there exists a temperature gradient between the gases and the engine components. The compressed-gas starts losing heat very rapidly to the combustion chamber wall and engine cylinder head, since no preheating is done before the charge enters into the cylinder during a cold start. This results in low temperature of gases at the end of the compression stroke. This poses another challenge for initiating the HCCI mode of combustion, and some compensating mechanism is required to overcome this problem. Possible solutions are using glow plugs, using a different fuel or fuel additive, and increasing the compression ratio using a variable compression ratio (Machrafi and Simeon Amouroux 2008). From a practical view point, perhaps one can start the engine for a few cycles (warming up period) in SI or CI mode and then switching over to the HCCI mode. For engines equipped with VVT, it may be possible to make this warm-up period as short as a few fired cycles, since high levels of hot residual gases could be retained from the previous spark-ignited cycles to induce HCCI combustion. Although the above solutions appear feasible, significant research and development will be required to advance these concepts and prepare them for production engines. After a cold start maintaining HCCI mode of combustion at light load imposes a penalty in terms of increased emissions of CO and UHC and deterioration of efficiency. To overcome this problem, fuel needs to be directly injected into the cylinder to achieve appropriate charge stratification.

1.1.2.4 Homogeneous Mixture Preparation

Effective mixture preparation and avoiding fuel/wall interactions is crucial for achieving high fuel efficiency, reducing UHC and PM emissions, and preventing oil dilution. Fuel impinging on the surfaces of the combustion chamber has proven to be disadvantageous to UHC emissions even for moderately volatile fuels such as gasoline (Stanglmaier and Matthews 1999). Mixture homogeneity has an effect on auto-ignition reactions that control the HCCI combustion phasing (Harada and Shimazaki 1998), and there is significant evidence that low NO_x emissions can be produced even with some degree of mixture in-homogeneity within the combustion chamber. Homogeneous mixture preparation is most difficult for fuels with low volatility such as diesel, which requires elevated intake air temperatures for low-smoke operation when port-injected.

Based on pioneering research, HCCI combustion has been attracting growing attention in recent years owing to its potential for significantly reducing NO_x and particulate emissions whilst achieving high thermal efficiency at part loads with the development of electronic control technology, numerical simulations and optical technology. It has been observed that significant amount of research work has been carried out in three main areas of HCCI study; fundamental theory, gasoline fueled HCCI combustion and diesel-fueled HCCI combustion.

Due to the difficulty of high-load operation, most initial concepts involve switching to traditional SI or CIDI (compression ignition direct injection) combustion for operating conditions where HCCI operation is more difficult. This configuration allows the benefits of HCCI to be realized over a significant portion of the driving cycle but adds the complexity of switching the engine between operating modes.

1.2 REQUIREMENT OF PCCI ENGINE

The results from the pioneering investigations indicate the strong potential of HCCI to improve thermal efficiency of gasoline fueled engines and substantially reduce NO_x and soot emissions of diesel-fueled engines. Furthermore, these results confirmed the dominating role of chemical kinetics in HCCI combustion, which has significance for further studies. However, persistence efforts of researcher across the globe have realized this technology for commercial application. Thus, at present two engines have been commercialized based on the HCCI mode of combustion. First of them is developed by Nisan Motors (Kimura and Aoki 1999) and named as Modulated Kinetics (MK) combustion, second is developed by Honda (Ishibashi and Asai 1996) called Active Radical (AR) combustion system.

MK technique is based on the concept of increasing ignition delay period which in turn provides greater time for air-fuel mixing. This is achieved through late direct injection of diesel in a regular CIDI engine, as the fuel injection timing is retarded from 7° bTDC to 4° aTDC together with high swirl ratio and EGR ratio is applied at light loads. Therefore, under low load conditions time required for complete homogenous mixing of air and fuel is shorter than ignition delay period (auto-ignition of fuel), nevertheless due to low equivalence ratio it results extremely low NO_x and PM emissions. The MK 2.5 and 3.0 ccm engines are commercially available in Japan since 1998.

Honda demonstrated two stroke SI production engine to operate on HCCI mode. Active Radical (AR) is also a dual-mode engine. It operates as a spark ignition engine at high loads, at idle, and for cold starts, in range of part loads working under HCCI mode. The engine has low compression ratio (CR=6.1:1), HCCI operation is obtained by throttling the exhaust. With such solution, engine operates with high fraction of hot residual gases, which is enough to obtain HCCI combustion, even at a very low compression ratio. It has demonstrated considerable fuel economy which is 27% better and UHC emissions reduced by 50% than a normal two-stroke engine.

It is also established from the previous discussion that complete HCCI mode of combustion has several challenges. So there is a need to obtain a mixed mode of combustion in which two fueling system needs to be placed one which creates homogenous charge by injecting a small amount of fuel into the air intake manifold and second which directly injected other fuel into the cylinder. Thus, small amount of fuel being injected into the intake air and diesel is injected to ignite the mixture of air-fuel in the cylinder. This is a compromise between HCCI and CIDI known as a PCCI or partial HCCI, since enormous amount of effort has been envisaged for HCCI combustion at higher load and speed. Moreover it is very difficult to control the ignition-timing at elevated loads (Yao and Zhaolei Liu 2009). Therefore, this concept is known as partially premixed charge compression ignition (PCCI).

Diesel engines have faced significant regulatory challenges with regard to emissions by the United States environmental protection agency (EPA) as well as by international regulatory bodies include 'Euro-norms' which are also applicable in India. In the United States, the Clean Air Act (CAA) authorized the EPA to develop emissions standards. The EPA provides maximum allowable emissions for Non-Methane Hydrocarbons (NMHC), Nitrogen Oxides (NO_x), Carbon Monoxide (CO) and Particulate Matter (PM). These emission components are limited as they have been found to be hazardous to the atmosphere and to human health. The emissions of NO_x, UHC, and CO were studied in this research. Emissions regulations norms becoming stringent for engines used for off-road and on- road applications. As discussed earlier, exhaust gas after-treatment is also used for the emission reductions, however, these techniques are not cost-effective for users in the villages and they can also alcohol. Therefore, new combustion system needs to be developed for small capacity engines which could reduce emissions and improve fuel economy along with enhanced utilization of alcohol. It has been accepted by the large number of researchers that alcohol along with diesel, could be used to reduce emissions despite of small reduction in fuel economy at small engine loads. However, advancement in fuel injection mechanism and EGR could also help to improve the brake thermal efficiency.

2. CHAPTER 2: LITERATURE REVIEW

This chapter is describes literature review especially related to diesel combustion, emission formation, diesel exhaust after-treatment technologies and role of additives used along with diesel fuel. Alcohol as a fuel and new combustion technologies like early direct injection, PREDIC and premixed techniques are discussed in great details.

2.1 INTRODUCTION

The purpose of internal combustion engine is to transform chemical energy in fuel into thermal energy by combustion or oxidation inside the engine. The working fluid is air-fuel mixture before combustion and combustible gases after combustion. Since the combustible gases only undergo an expansion of work generation, therefore, it is named as an internal combustion engine. Nevertheless, combustion product and fluid undergoes expansion during energy transformation is different in case of external combustion engine, i.e., closed-cycle gas turbine and steam sterling engine.

J. J. E Lenoir developed first engine in 1860 which uses coal-gas air mixture without any compression mechanism. Induction of charge and air takes place in the first and then charge was ignited by the spark, subsequently expansion produces work output. This was the first commercial engine; nearly 5000 engines were sold during 1860 to 1865.

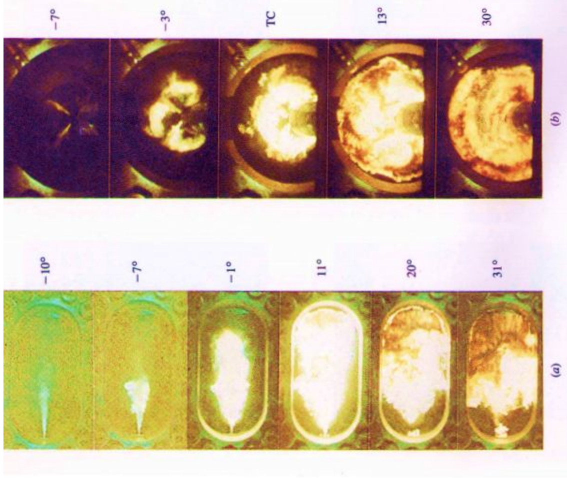
Nicolaus A. Otto and Eugen Langen invented an atmospheric engine operated up to 11% efficiency in 1867, which is considered as a more successful attempt than previous one. Later on, to overcome the low efficiency and excessive weight shortcomings, four stroke prototype engines was built in 1876 by Otto, this is considered as one of the great breakthroughs in the engine invention, almost 50,000 units were sold till the end of 1890.

Rudolf Diesel a German Engineer had come out with a totally different concept of initiating combustion, injecting liquid fuel into the hot air environment results into the start of combustion, no spark-plug is required.

The advantage was demonstrated in terms of efficiency, it was double than other engine. Later on it was named as a diesel engine and becomes inevitable as an engine for all sphere of applications.

2.1.1 Combustion in Diesel Engine

The liquid fuel is injected at very high velocities into the combustion chamber through one or more orifices or nozzles in the injector tip. The fuel injection mechanism transforms the large pressure energy of the fuel into higher kinetic energy and the liquid fuel is atomized into small drops. The next stage of combustion begins with entertainment of fuel particles with air, the vaporization of the atomized fine fuel particles started. The latent heat of vaporization is given off by compressed air in the cylinder which has already attained high pressure and temperature due to compression. However, the cylinder pressure and temperature are above the fuel's auto-ignition point. The three stages of combustion, atomization, and vaporization, fuel air mixing and burning are repeated till the last portion of fuel injected into the cylinder. High speed photography studies was carried out under normal operating conditions and is shown in Fig. 2.1 (Heywood 1988).



Conditions: T=1000K, p=100 bar

Figure 2.1: Combustion of four sprays in DI diesel with swirl (Heywood 1988)

From the above discussions it is clear that the fuel enters the cylinder as a liquid and ignition does not start until a portion of it has vaporized and mixed with air. As a consequence, the flame develops under extremely non uniform conditions of fuel in the air. Even though the overall fuel-air ratio is always lean in a diesel engine, there are large variations in the equivalence ratio across the combustion chamber at both a large and small scale (Stone 1999). This heterogeneous distribution of fuel within the cylinder and the corresponding heterogeneous burning process are the hallmark of diesel combustion. They are also the main reasons why diesel engines have a tendency to emit high levels of their most prominent pollutants, namely PM and NO_x.

2.1.1.1 Stages of DI Diesel Engine Combustion

Fig 2.2 shows a typical heat-release diagram which depicts four different stages of DI diesel combustion.

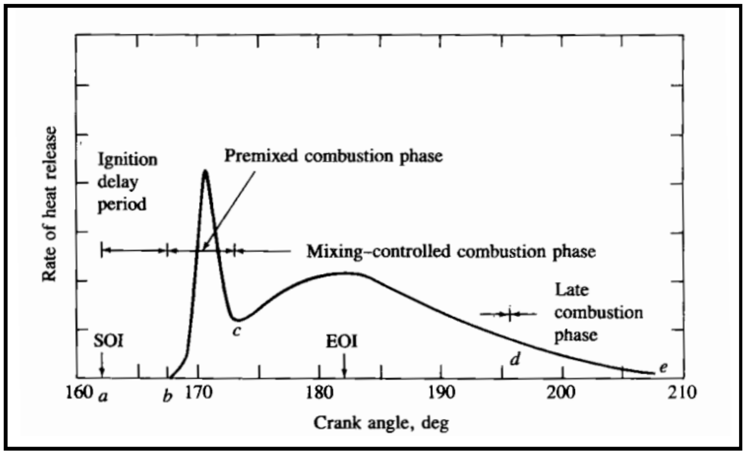


Figure 2.2: Typical DI engine heat-release-rate diagram (Heywood 1988)

Ignition delay (ab): The liquid fuel is injected into the cylinder at the end of compression. Injection is done at high pressures which lead to form a fuel jet and it breaks liquid fuel into small droplets and starts vaporizing in the hot-air environment. The period between the start of fuel injection into the cylinder and the start of combustion is known as an ignition delay. However, combustion does not start until a visible flame or a measured in-cylinder pressure rise can be detected (Tayor 1985). Ignition delay depends on several parameters like injection pressure, injection timing, and cetane number of fuel. Further, fuels with poor auto-ignition properties exhibit longer delay periods.

Rapid combustion (pre mixed combustion (bc): The mixture after completion of the delay period, mixing of fuel vapor and air, and subsequent chemical reactions within the charge lead to ignite spontaneously. It results in a rapid rise in the in-cylinder pressure and temperature due to combustion. Therefore, the rate of burning and duration of the premixed phase are closely related to the extent of the delay period. The cloud of charge, which ignites spontaneously, results in a very sharp increase in the cylinder pressure and may lead to knocking sound. This depends on the other parameters in diesel engines.

Thus, the premixed phase is characterized by high rates of pressure increase and heat release. It is in the high temperature, fuel-rich regions inside this premixed flame, thus soot precursors are originated before the soot itself is nucleated and grown in the richer regions of the subsequent diffusion flame.

Mixing-controlled combustion (cd): After completion of the premixed combustion phase, the remaining unburned portions of the fuel get compressed and preparation time (ignition delay) gets shortened. Therefore in the mixing controlled phase, the combustion reactions are controlled by the rate at which fuel-air mixture is formed, or diffusion of fuel into air or vice-versa takes place. This phase is also known as the diffusion combustion phase. The diffusion flames are also responsible for the formation and growth of soot particles.

Late combustion phase (de): As described by Heywood (Heywood 1988), due to a fall in the temperature of the in-cylinder gases, the reaction rates are slow down during this phase. As the last stage of expansion there is a possibility that a small fraction of energy yet to be released from fraction of fuel energy associated with the soot particles and fuel-rich gases. Therefore a small fraction of fuel is burnt, and oxidation of soot and fuel-rich combustion products can also take place. This is also known as the late combustion phase.

2.1.2 Diesel Engine Exhausts Emissions

In spite of the advantages of using compression ignition engines (high thermal efficiency and durability) there are drawbacks in the case of emissions. As mentioned previously, NO_x emissions still a one of the major problem from diesel engines. Generally, NO_x consists mainly of nitrogen monoxide or nitric oxide (NO) and nitrogen dioxide (NO_2). Compared to spark ignition engines, diesel engines produce more nitrogen oxides (NO_x) and Particulate Matter (PM). However, diesel engines emit relatively lower carbon monoxide (CO) and unburned hydrocarbons (UHC). Due to the hazardous effects on the environment and on human health of these pollutants, legislation norms have been proposed for all passenger cars in countries like Japan, USA (EPA 2012) and EU for these pollutants.

2.2 NO FORMATION IN CI ENGINES

In diesel engines combustion starts after completion of the ignition delay. Fuel and air that are mixed together form flammable mixtures around stoichiometric compositions. In the premixed combustion phase the net heat release becomes positive and the first peak is observed, in the mixing-controlled combustion phase, also combustion occurs in the region where the local equivalence ratio is close to stoichiometric. NO formation rates depend on the local equivalence ratio and temperature.

Chemical pathway for no formation

NO is formed during combustion. The mechanism of its formation is explained below (Mellor and Mello 1998):

- Thermal
- Prompt
- Nitrous oxide

2.2.1 Thermal Mechanism

NO_x formation occurs in the hot combustion gases at sufficiently high temperatures (>1800 K). It describes the NO formed as atomic oxygen reacts with nitrogen in the presence of free radicals (O, N, H, and OH). The Zeldovich mechanism (1946) clearly explains this by the following reactions (Heywood 1988; Stone 1999)

$\mathrm{O + N_2 \leftrightharpoons NO + N}$ -- Eqn. (1.1)

$\mathrm{N + O_2 \leftrightharpoons NO + O}$ -- Eqn. (1.2)

$\mathrm{N + OH \leftrightharpoons NO + H}$ --- Eqn. (1.3)

Lavoie et al. (Lavoie and Heywood 1970) had introduced the third reaction (Eqn. 1.3) whereas the first two were suggested by Zeldovich.

The formation of NO mainly occurs in the high temperature zone, in excess oxygen regions through reaction (Eqn. 1.1), leaving one free atom of nitrogen. It can be later combined with oxygen (Eqn. 1.2) or with the OH radical (Eqn. 1.3), already present in the combustion process, to form nitrogen monoxides.

The already formed NO during combustion can be further converted to NO_2 by reaction (Eqn. 1.4) as given below. Afterwards, it can be also be converted back to NO by reaction- Eqn. 1.5 (Heywood 1988)

$$\mathrm{NO} + \mathrm{HO_2} \leftrightharpoons \mathrm{NO_2} + \mathrm{OH} \text{ -- Eqn. (1.4)}$$

$$\mathrm{NO_2} + \mathrm{O} \leftrightharpoons \mathrm{NO} + \mathrm{O_2} \text{ --- Eqn. (1.5)}$$

From the environmental and health point of view, NO_x are harmful. They can destroy the ozone in the atmosphere. Moreover, favorable conditions (presence of ammonia or moisture) lead to the formation of nitric acid vapour which is affecting human health, mainly because of respiratory and heart diseases (EPA 2013).

2.2.2 Prompt Mechanism

NO is formed in the flame as CH, CH_2 etc. The CH, CH_2 radicals react with N_2 to give intermediate species, HCN and CN, by the reactions (Mellor and Mello 1998):

$$\mathrm{CH} + \mathrm{N_2} \leftrightharpoons \mathrm{HCN} + \mathrm{N} \text{ --- Eqn. (1.6)}$$

$$\mathrm{CH_2} + \mathrm{N_2} \leftrightharpoons \mathrm{HCN} + \mathrm{NH} \text{ ------------------------------------ Eqn. (1.7)}$$

The thermal NO formation in the burnt gas behind the flame front is much higher as compared to any NO in the flame front. Therefore, prompt NO formation is small in the case of CI engines. However, prompt NO may be significant (Hochgreb 1998) in diesel engines with EGR.

Nitrous oxide mechanism: In diesel engines, high pressure combustion involves decomposition and formation reactions. Therefore, the Zeldovich mechanism alone does not predict exact NO formation. N_2O formation and decomposition may also lead to significantly amounts of NO generation. This could be generated in both premixed and diffusion-controlled combustion (Mellor and Mello 1998) phase.

$N_2O + O \rightleftharpoons 2NO$ ---Eqn. (1.8)

$N_2 + O + M \rightleftharpoons N_2O + M$------------------------------------Eqn. (1.9)

By adding the reaction:

$O_2 + M \rightleftharpoons 2O + M$ --- Eqn. (1.10)

All the above reactions can be balanced so that the overall reaction is:

$2O_2 + 2N_2 \rightleftharpoons 4NO$--- Eqn. (1.11)

2.2.3 NO_2 Formation

NO_2 emissions from SI engines are almost negligible. Since, NO is of the order of several thousand of ppm while NO_2 is approximately 60 to 70 ppm. However, in case of CI engines the fraction of NO_2 emissions typically varies from 10 to 30% of the total NO_x emissions. Hilliard et al. (Hilliard and Wheeler 1979) and Merrymanet al. (Merryman and Levy 1975) presented the following mechanism for the formation of NO_2 in diesel engines, in which NO is converted to NO_2 by the reaction of NO with HOO^{-} radicals. The reactions are as follows

$NO + HO_2 \rightleftharpoons NO_2 + OH$ ---------------------------------- Eqn. (1.12)

In the post flame region, NO_2 reacts with atomic oxygen and is converted into NO and O_2 (produces NO again):

$NO_2 + O \rightleftharpoons NO + O_2$ ---------------------------------------Eqn. (1.13)

If the high turbulence prevails in the engine, mixing of the burnt gases with colder air or air fuel mixture may quench the decomposition reactions, so it leads to suppression of the conversion of NO_2 into NO. In diesel engines at low loads, the ratio of NO_2/NO becomes high due to the freezing of the decomposition reactions (Hilliard and Wheeler 1979).

The fuel and air first react in the fuel-rich mixture, leading to soot formation, and then this rich mixture burns out in a high-temperature diffusion flame at the jet periphery, leading to NO_x formation. The combustion model proposed by Dec (Dec 1997) suggested premixed combustion occurs in the rich mixture (ϕ= 2 to 4). Thus, neither *thermal* NO nor *prompt* NO is likely to be formed in significant concentrations during premixed combustion. Most of the NO is formed during mixing controlled combustion phase. The NO formation region is around the periphery of the spray.

2.3 Particulate Matter

PM is defined as any solid or liquid (excluding water) that can be collected on a filter when filtering the diluted exhaust. Also the temperature of the filtering is not higher than 52 °C. Particulate matter besides NO_x emissions are a major pollutant from combustion in compression ignition engines. Particulate matter results from incomplete combustion of fuel hydrocarbon. Also some of it is contributed by the lubricating oil (Heywood 1988). Diesel particulates consist principally of combustion generated carbonaceous materials (soot) on which some organic compounds have been absorbed. They are basically a complex mixture of elemental carbon, hydrocarbons, sulfur compound and ash. And they may vary greatly in size, composition, solubility and in toxicity (Burtscher 2005). The model of soot formation in diesel spray has been proposed by Dec (Dec 1997) and is illustrated in Figure 2.3.

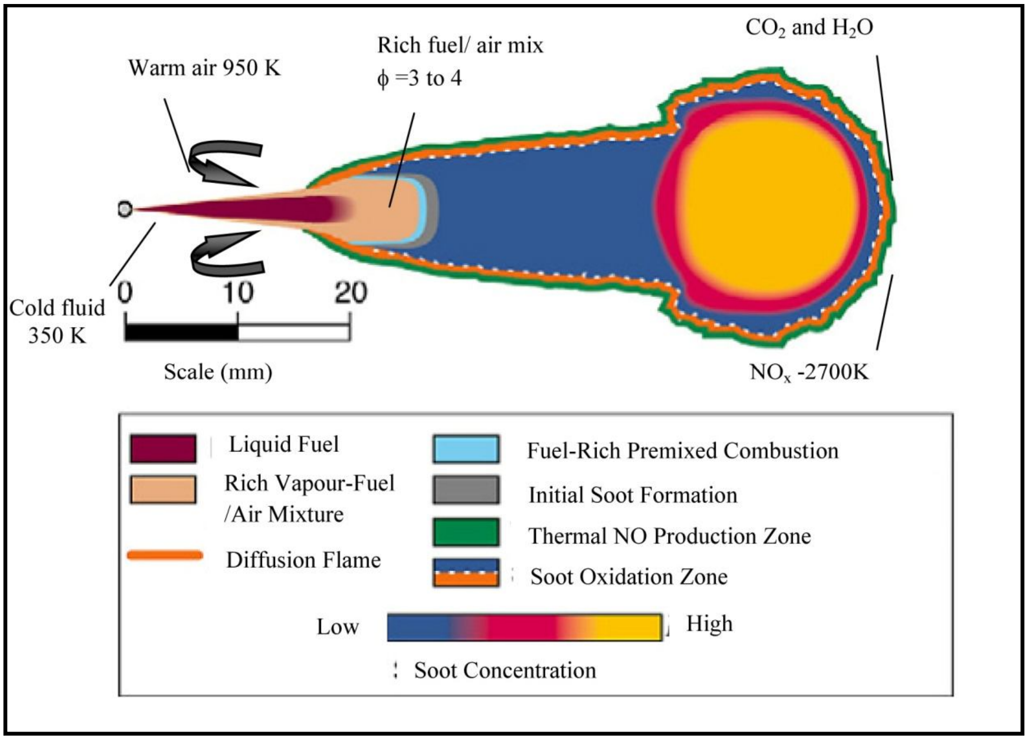


Figure 2.3: Dec combustion model for DI diesel engine. (Dec 1997)

Generally, the particles from diesel combustion can be divided as carbonaceous (soot particles), and volatile fraction (SOF/VOF).

- The solid fraction consists of:
  - carbonaceous agglomerates (soot)
  - inorganic ash, while
- The volatile fraction consists of:
  - organic compounds (hydrocarbons)
  - inorganic compounds (sulfuric acid and sulfates)

In the later part of combustion, sulfur usually reacts with water vapor to form sulphuric acid, and together with SOF/VOF may start to condense on the solid carbon particles at lower temperatures. In Figure 2.4 a detailed diagram of particulate matter composition is presented accordingly to the work done by Kittelson (Kittelson 1998).

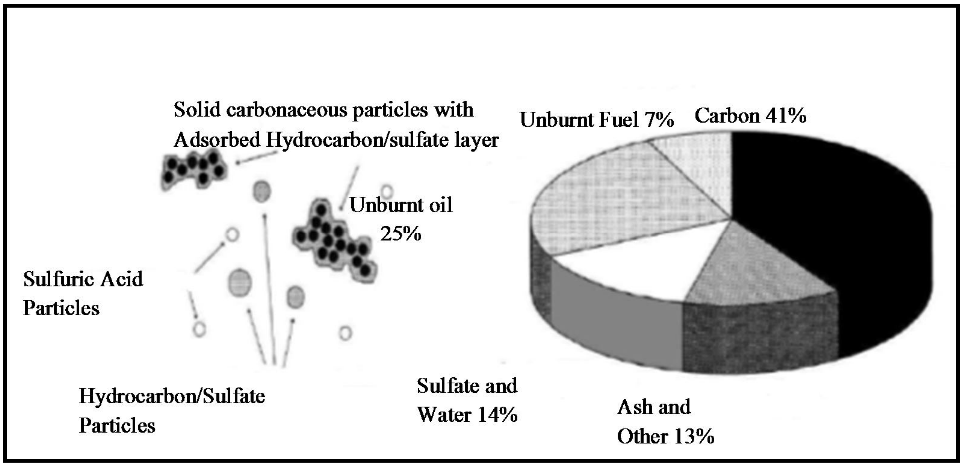


Figure 2.4: Soot particles composition (Kittelson 1998).

The solid fraction is responsible for most of the total particle mass, and its constituents (soot and ash) are formed directly by the combustion process. Soot has been the foremost and also the most characteristic pollutant emitted by diesel engines. Soot formation process starts when the fuel and air first react in the fuel-rich mixture of the premixed flame. Rich mixture regions have large hydrocarbon molecules from the fuel undergoing thermal decomposition (cracking) due to the high temperatures and the relative absence of oxygen. This process is analogous to a pyrolysis process and produces compounds that are precursors to soot formation. During combustion, the formation process starts with a fuel molecule (12 to 22 carbon atoms and an H/C ratio of about 2) and ends up with particles of about hundred nanometers in diameter, composed of spherules 20 to 30 nm in diameter each containing some 10^5 carbon atoms and having an H/C ratio of about 0.1 (Heywood 1988). Soot precursors include unsaturated hydrocarbons, such as ethylene, acetylene and polycyclic aromatic hydrocarbons (PAHs). Typical conditions are characterized by temperatures between about 1000 and 2800 K, and pressures of 50 to 100 atm. (Heywood 1988). As on the periphery of the spray, soot particles start to oxidize and it depends on the diffusion of reactants to and products from the surface as well the kinetics of the reaction. Estimates have suggested that over 90% of the soot eventually oxidizes. A higher rate of oxidation decreases tail pipe soot emissions.

The measurement of particulate matter can be done in the exhaust tail pipe by

- Smoke meter for smoke opacity measuring or the filtration method, smoke number (SN) - Bosch Scale
- Transmission electron microscopy (TEM)-particle size, aggregate size and morphology of soot particles

However, increasing concern due to the hazardous effect of PM emission on human health (causes asthma, lung cancer, cardiovascular issues, and in extreme cases, premature death) various means are adopted to cut it down.

- Common rail and unit injection system – high pressure, electrically controlled fuel injection system capable to precise control of the fuel and air mixing process in the cylinder which, in turn, reduces PM (smoke in particular) from the tail pipe
- Oxidation catalysts and particulate traps – effective only at high temperature (500-600°C)
- Fuel oxygenates-
    - fuel-borne oxygen shifts the products of thermal cracking by displacing the long carbon chains that exist in regular diesel fuel

- during the premixed combustion phase, oxygenates can dramatically increase the concentration of free radicals such as O, OH, and HCO. These radicals help oxidize carbon to CO and CO_2. As a result, the availability of carbon to form soot precursors is decreased
- high concentrations of those free radicals (OH in particular) can also oxidize soot precursors in the diffusion flame, limiting the formation and growth of PAHs and inhibiting the inception of soot particles

Benvenutti et al. (Benvenutti and Marques 2005) reported that the methyl radical (CH_3) is also an important precursor for the formation of soot-oxidizing radicals, such as OH.

2.4 UNBURNED HYDROCARBONS AND CARBON MONOXIDE

The unburnt hydrocarbon emissions from compression ignition engines are comparatively much smaller than the spark-ignition engines. As the boiling points of hydrocarbon compounds associated with diesel fuel are higher, they have higher molecular weights than gasoline. This results in complex unburned hydrocarbon products. Diesel engine fuel spray shows wide variation in the equivalence ratio which occurs from the spray cone to the outer boundary. The fuel close to the spray boundary has low equivalence ratio (ϕ=0.3) which is leaner than the lean limit of combustion. Therefore, it will not auto-ignite or sustain a fast reaction front. The magnitude of UHC depends on the quantity of fuel injected during ignition delay, mixing rate with air, and in cylinder conditions are favorable to auto-ignition. Therefore, over mixing (over leaning) is one of the important cause of UHC formation during normal operation and it becomes more predominant at light-load operation when the ignition delay is longer (Greeves and Khan 1977; Yu and Shahed 1981).

However, undermixing which is resulting in over-rich mixtures also considered as a one of the major causes of UHC emission. The fuel that leaves the injector at low velocity relatively in the later part of combustion process is another important source of UHC (Heywood 1988). This mechanism occurs more frequently especially when injection of fuel is being done after the ignition delay period and during over-fueling conditions, such as while accelerating at high loads. Advancing fuel injection timing tends to cause higher combustion temperatures and pressures, thus promoting the oxidation of hydrocarbons.

Carbon monoxide is a product of incomplete combustion, usually when the engine is operating at very rich air-fuel mixture condition. Accordingly, the CO emissions from IC engines are controlled by the fuel-air equivalence ratio. As the spark-ignition engines operate close to stoichiometric fuel-air equivalence ratio at part load, but fuel rich at full load. Since CO concentrations in the exhaust increases with increasing fuel-air equivalence ratio, as the excess fuel increases (Harrington and Shishu 1973). Thus, it needs to be controlled at high engine loads. To control CO emission in case of spark- ignition engines, exhaust gas treatment is required, such as oxidation catalysts. In contrast, diesel engines operate with excess air. Thus, overall lean combustion ensures very low levels of CO emissions. In the case of well-designed diesel engine, CO emission is low enough and can be considered not relevant.

2.5 NO$_X$-REDUCING TECHNIQUES

Some commonly-used techniques for decreasing the emissions of NO$_x$ in diesel engines are:

2.5.1 Injection Timing

This technique is used to control NO$_x$ emission. Retarding the injection timing of fuel is an effective method used to decrease NO$_x$ emissions from a diesel engine. Thus, combustion events are shifted towards the expansion stroke, causing a reduction in peak flame temperature and in-cylinder pressure that results into slowing down the reactions of the NO$_x$ formation. However, this technique also causes an increase in PM emissions and fuel consumption. The reason could be due to delaying the fuel injection the duration of the diffusion combustion phase is extended and the resulting lower temperatures during the expansion stroke leads to impair the oxidation of the soot. The change in fuel injection timing provides a classic example of the NO$_x$-PM trade-off which is a characteristic of diesel engines.

2.5.2 Exhaust Gas Re-circulation (EGR)

In this technique re-circulating part of the exhaust gases helps in reducing NO_x emission. However, appreciable particulate emissions are observed at high loads, thus there is a trade-off between NO_x and smoke emission. Hence, this technique is mainly used in low-load and low-speed conditions. When a part of this exhaust gas is re-circulated to the cylinder, it acts as diluent to the combusting mixture. This also helps to reduce O_2 concentration in the combustion chamber

Thus, the challenge is to minimize the pollutants by manipulating the thermodynamic properties and the oxygen concentration of the cylinder charge whilst keeping minimum degradations in the power and efficiency, which is the principal reason to apply EGR in diesel engines.

At stoichiometric combustion, very small fraction of oxygen is left as a combustion product while at idle load conditions about 20% oxygen is available at the tail pipe of modern diesel engines. Thus, with the increase in the engine load, concentration of CO_2 and H_2O and specific heat of exhaust gases increases. The flame temperature and the maximum temperature of the working fluid will be lowered with the increase in CO_2 and H_2O. Therefore, at low engine load operation, higher rates of EGR can be tolerated by the engine whereas small rate of EGR is sufficient at high engine load operation.

The effect of increasing EGR leads to lower combustion temperatures and reduce oxygen availability. The decrease in oxygen concentration and combustion temperatures reduces the rate of oxidation of species including soot, UHC and CO (Jacobs and Bohac 2005).

Studies conducted on diesel engines have shown that the EGR can reduce NO_x substantially (Lapuerta and Hernandez 2000)

Ladommatos et al. (Ladommatos and Abdelhalim 1996a; Ladommatos and Abdelhalim 1996b; Ladommatos and Abdelhalim 1997) worked extensively on the effect of EGR and have published three papers which suggested the following mechanism responsible for NO_x control:

i. Dilution mechanism: Increased level of inert gas concentration in the charge increases ignition delay or greater time for premixing and long burn duration which is known as EGR's dilution effect. This results in lowered flame temperatures. Moreover, the partial pressure of oxygen reduces at low temperatures and availability of less oxygen is expected to suppress the NO_x formation.

ii. Thermal Mechanism: Higher specific heat capacity and thermal mass of re-circulated CO_2 and H_2O compared to oxygen and nitrogen present in the fresh air. Thus, it decreases the temperature rise in the combustion chamber after the same heat release. This also helps to reduce heat transfer losses, which occurs during and after the combustion event, at that time the surface area and volume ratio of the engine is a maximum. Thus, the heat loss to the coolant reduces due to low peak combustion temperature.

iii. Chemical Mechanism: As the combustion process progresses water vapor and carbon dioxide molecules undergo dissociation (endothermic dissociation of water vapour), and as the rate of heat release is modified. Thus, it leads to affects NO_x formation and heat absorbed during dissociation result in a decrease in the flame temperature.

2.6 DIESEL EXHAUST AFTER-TREATMENT TECHNOLOGIES

Available technology and measures used to control combustion inside the cylinder and hence to cut-down emissions could not provide large reduction in harmful pollutants. So, to meet the stringent emissions norms after-treatment of exhaust gases has been used for efficient conversion of harmful pollutants into harmless substances. Therefore, in compression ignition engines to reduce emissions of harmful pollutants like carbon monoxide, unburned hydrocarbons and nitrogen oxides and also soot or particulates different techniques are used. Thus, diesel oxidation catalyst (DOC), the diesel particulate filter (DPF) as well as highly efficient NO_x after-treatment technologies will become mandatory in the future.

2.6.1 Diesel Particulate Filter

This method is based on filtration of particulate matter by using ceramic materials such as cordierite ($2MgO_2.Al_2O_3.5SiO_2$), aluminum titanate and silicon carbide (SiC).

Fig. 2.5 shows honeycomb cermaic monolith that traps the particulate matter as the gas flows thorugh porous walls of its cells, thus it is also known as cermaic wall flow filterrs. The flow channles are arrnaged in such a manner that alternate ends of these cells plugged at one end and kept open at opposite end. Theroefore, exhaust gas enters from the upstream end while flowing along the length of the channel flow direction is chnaged and it eneters into adjacent channel to leave from the filter. It is evident that long time application of DPF lead to produce back pressure due to blockage of passage by particulate matter and in trun it manifests in temrs of fuel panelty. So there is need to clean accumulated particulate matter by burning or oxidizing it also know as regenration. Hot burn gas raise the temperature of the DPF which converts soot into CO_2.

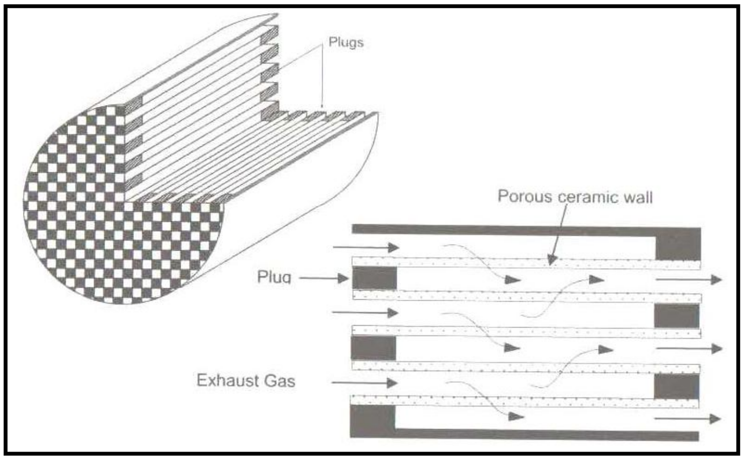


Figure 2.5: Ceramic wall flow filter for diesel particulates.

2.6.2 Diesel Oxidation Catalysts

Since early 1990's diesel oxidation catalysts (DOC) have been widely used in the light-duty diesel commercial vehicles in the European countries. Diesel oxidation catalysts consists of ceramic honeycomb monolith which has a large number of parallel and straight open channels for flow of exhaust gases. The flow through these channels is laminar. The inorganic material (Al_2O_3, SiO_2, $TiO_2{}^{)}$, ceramic or metal with dispersed Pt, Pd or Rh are used on its surface (Handbook 2004). It allows a large reaction surface within a small space. The reactions of the oxidation of UHC and CO are as follows

$C_yH_n + \left(1 + \frac{n}{4}\right)_y O_2 \rightarrow yCO_2 + \frac{n}{2}H_2O$---------------------------Eqn. (1.14)

$CO + \frac{1}{2}O_2 \rightarrow CO_2$---Eqn. (1.15)

$CO + H_2O \rightarrow CO_2 + H_2$--Eqn. (1.16)

2.6.3 NO$_x$ Reducing After-treatment Technologies

Two after-treatment methods could be used to reduce NO_x emissions from compression ignition engines. As the diesel engines operate with excess air, thus exhaust gas is oxygen rich. Therefore in the oxygen rich environment conversion of NO_x needs additional reducing agents termed as reductants. So, selective catalytic reduction and lean NO_x trap are discussed hereunder.

Selective catalytic reduction of NO_x employed anhydrous ammonia or urea as reducing agent. Urea is injected into the exhaust, which undergoes thermal decomposition and hydrolysis to get converted into ammonia.

Urea Hydrolysis

$(NH_2)_2CO + H_2O \rightarrow CO_2 + 2NH_3$--------------------------------Eqn. (1.17)

NO_x conversion

$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O$ ------------------------------Eqn. (1.18)

$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O$ ----------------------------------Eqn. (1.19)

Lean NO_x trap is used to reduce NO_x emission also known as NO_x storage converter (NSC) or a NO_x absorber. This converter mostly catalyzes the reduction of nitrogen oxides to nitrogen. Hence, platinum is used as catalyst; the NSC converter also contains storage components with high affinity to NO_x like Barium compounds.

2.7 EMISSIONS AND THEIR HEALTH EFFECT

Diesel engine exhaust is a complex mixture of hundreds of constituents in either a gas or particle form. Gaseous components of diesel engine exhaust include carbon dioxide, oxygen, nitrogen, water vapor, carbon monoxide, nitrogen compounds, sulphur compounds, and numerous low-molecular-weight hydrocarbons. Among the gaseous hydrocarbon components some have toxicologic relevance are the aldehydes (e.g., formaldehyde, acetaldehyde, acrolein), benzene, 1,3-butadiene, and polycyclic aromatic hydrocarbons (PAHs) and nitro-PAHs.

PM consists of fine particles (fine particles have a diameter <2.5 μm), including a subgroup with a large number of ultrafine particles (ultrafine particles have a diameter <0.1 μm). Collectively, these particles have a large surface area which makes them an excellent medium for adsorbing organics. Also, their small size makes them highly respirable and able to reach the deep lung. Available evidence indicates that there are human health hazards associated with exposure to diesel exhaust. The hazards include acute exposure-related symptoms; chronic exposure related non cancer respiratory effects, and lung cancer.

NO_x react with ammonia, moisture, and other compounds to form small particles. These small particles penetrate deeply into sensitive parts of the lungs and can cause or worsen respiratory disease, such as emphysema and bronchitis, and can aggravate existing heart disease, leading to increased hospital admissions and premature death. Some reported works are given hereunder to discuss the health hazard due to diesel engine exhaust.

Study (Nightingale and Cullinan 2000) found that sulphur oxides (SO_x), nitrogen oxides (NO_x) when mixed with water vapour (H_2O) present in the air forms sulphuric-acid and nitric-acid results in acid rain. Thus, it cause air pollution and adversely affects the respiratory system prone to diseases like chronic Obstructive Pulmonary disease (Pulmonary ventilation provides the air flow function between atmosphere and lungs), Asthma, Pneumonia, Tuberclosis and lung Cancer**.** Kahn et al. (Kahn and Weeks 1988) found severe effects from the NO_x and sulphur compounds on the nervous system, increase in chronic bronchitis and asthma. Also, in another study (Gamble and Jones 1987) breathing problem, nausea, increase in sensation and burning of eyes observed**.**

Study (Edling and Axelson 1984) revealed that PM with the metal composition of Iron, Zinc, Vanadium and Nickel has the direct effect on the pulmonaries, cardiovascular, variations in the heart rate, formation of arrhythmias that may leads to ultimately death due to blockage of atrioventricular. Ultrafine particles (<100 nm) present in the PM cause severe effects on inflammation, it damages epithelium and phagocytosis inhibition.

2.8 ADDITIVES FOR DIESEL ENGINE

Oxygen-containing compounds have been used as fuel oxygenates to reduce soot emission or to supply oxygen to form CO_2 and reduce carbon rich particles. Various oxygenates generally used along with diesel are

- Alcohol (Atcilla and Orhan 2002; Huang and Lu 2004; Zuohua Huang and Hongbing Lu 2004)
- Methyl or ethyl esters or bio-diesel (Sharp and Howell 2000; Jiménez-Espadafor and Francisco J. Torres 2012; Pandey Shyam and Sharma Amit 2012)
- Ethers (Liotta and Montalvio 1993; Youn and Su Han Roh 2011)
- Ethylene glycol monoacetate (Lin CY and JC 2003)

Oxygenates capability to reduce smoke opacity, PM, CO and NO_x emissions depend on the chemical structure and amount of oxygen in the fuel.

Bio-diesel: Straight chained vegetable oils have long been considered as alternative fuel to replace diesel (Shahid and Jamal 2008). The beginning was made by Dr. Rudolf Diesel when he demonstrated use of 100% peanut oil to run his engine in an exhibition at Paris. Since Vegetable oils are usually triglycerides (having branched hydrocarbon chain of fatty acids), they possess some unfavourable physico-chemical properties which impede their path to be directly used as fuel of compression ignition engine. Viscosity of vegetable oils (35 cSt to 60 cSt at 40 °C) is much higher as compared to that of diesel (4 cSt at 40°C) and this is due to large molecular weight and complex molecular structure of vegetable oils.

To overcome above difficulties to use it successfully in diesel engines vegetable oils or triglycerides undergo dilution, pyrolysis, microemulsions, and transesterification process (Ali and Hanna 1994). Transesterification is the process of splitting vegetable oils into light molecules. Therefore bio-diesel is derived from vegetable oils or animal fats made up of (m) ethyl esters. These vegetable oils are renewable biological sources. The merits of using bio-diesel instead of conventional diesel are its comparable energy density, cetane number heat of vaporization, higher combustion efficiency, low sulfur and aromatic content, renewability, and stoichiometric air /fuel ratio (Agarwal 2007). Bio-diesel is also non-toxic and rate of its biodegradation is much faster than conventional diesel. Greenhouse gas effects are least observed in case of bio-diesel (Alat and Balat 2008; Murugesan A and Umarani C 2009).

Although oxygenates are usually used in the form of blends with diesel fuel, alcohol for instance, can also be used in other forms. Depending on the level of engine modifications required alcohol can be used through the fumigation, dual fuel injection and even by neat alcohol. However, last option requires extensive engine modification includes injection system, fueling and injector.

It may be noted that fumigation is a particularly attractive method because it allows the utilization of alcohol in engines with little modification and without the issues associated with the limited miscibility of alcohol (methanol and ethanol in particular) with diesel fuel.

2.8.1 OXYGENATES AND DIESEL COMBUSTION

Literature showed that extensive investigation has been done to understand the effect of oxygenates on the performance and emissions of diesel engine. Researcher (MiŠosŠaw Kozak and Merkisz 2008; Miłosław Kozak and Merkisz 2009) have evaluated potential of oxygenates to reduce PM emissions. Investigators have found that molecular structure and oxygen content of oxygenate primarily control the PM emissions. Ironically, there is still lack of clarity regarding the parameter (molecular structure or oxygen content) plays dominating role to reduce PM emissions. The actual mechanisms through which oxygenated compounds affect pollutant formation in diesel engines are still widely debated by researchers.

Nabi et al. (Nabi and Minami 2000) investigated the effect of oxygen content on smoke emissions. Six different oxygenates have been studied to understand their effect on emissions of single cylinder diesel engine. It was found that smoke emissions decrease linearly with oxygen content of oxygenate and it reaches almost zero level at oxygen contents of 38% (wt.) or more. Investigators have (Miyamoto and Ogawa 1998; Cheng and Dibble 2002) reported that the reduction in PM level was largely influenced by the oxygen content of the blends rather than the chemical structure of the oxygenates. However, several investigators believe that in addition to oxygen content, the chemical structure of an oxygenate plays significant role in determining its effectiveness in reducing PM emissions.

Yeh et al. (Yeh and Rickeard 2001) examined the effect of molecular structure on PM emissions by using fourteen different oxygenates, the blends were prepared keeping same oxygen content. Oxygenates found to be more effective were alcohol and more volatile compounds showed superior performance than heavier ones. They found that ethers, esters, and carbonates outperformed in PM reductions.

The effect of fuel bound oxygen on NO_x emissions is not very clear. Some studies report a small increase in NO_x whereas others conclude that the emissions of oxides of nitrogen remain essentially un-changed. Again, the addition of oxygen modifies the fuel as a whole, and these changes in NO_x formation cannot be attributed solely to oxygen content. In addition to that, testing conditions and engine characteristics probably have a significant influence on NO_x emissions.

2.9 COMBUSTION CONTROL STRATEGIES IN HCCI

The research and development of HCCI diesel engines has been pursued along three main technical routes, depending on the mixture preparation process involved: the first approach involves injecting the fuel into the intake air, upstream of the intake valve, similar to a conventional port-fuel injection (PFI) SI engine. However, it was not until the mid-1990s that systematic investigations began on the potential for diesel fueled HCCI engines for automotive applications, due to the need for substantial reductions in both NO_x and PM emissions. The second approach was early direct injection into the cylinder which increases ignition delay, thus improving premixing the third technique, developed by Nissan Motors, is late direct injection or modulated kinetics.

The above discussion lays emphasis on the hurdles to accomplish advanced combustion technology. Recent publications showing increased interest amongst the scientific community have appeared especially in India, China, Indonesia and Brazil to explore premixed techniques to reduce emissions from diesel engines.

2.9.1 Early Direct Injection HCCI

In this technique homogenous mixture can be prepared by injecting fuel during the early part of compression so that the time available between the start and end of compression can be utilized effectively. Homogenous mixture is prepared during the time interval between the end of the injection and the on-set of combustion, which necessitates injection to be timed meticulously. The advantage of this technique is that in this process air is slightly compressed, as compared to when it is inducted into the cylinder, and the pressure and temperature are higher compared to inlet conditions. Although, wall wetting due to the over penetration of the fuel, less time available for fuel-air mixing, and also combustion phasing control would be critical issues.

Compared to premixing by air intake manifold fuel injection, this technique offers some inherent advantages, namely once the compression begins, movement of piston increases in-cylinder pressure and temperature of the air which helps in vaporization of the fuel being injected, as well as, it improves the mixing prior to initiation of combustion. Moreover, there is no additional requirement for inlet air preheating, which often requires electric heating system. However, researcher also attempted to explore early direct injection along with conventional direct injection (dual injection), and a brief review of that work is presented hereunder.

2.9.1.1 PREDIC

One of the most significant contributions in early direct injection HCCI was done at New ACE institute in Japan by Yoshinaka et al. (Yoshinaka and Nakagone 1996) and

Nakagome et al. (Nakagome K and Shimazaki N 1997). Three alternate fuel injection techniques were tried to better distribute the fuel, and to minimize over penetration and wall wetting that can occur when the fuel is injected well before the top dead center. Since the in-cylinder pressure and densities are low as compared to TDC conditions, conventional injectors designed for TDC conditions will not be able to produce good results. The three major techniques adopted were follows

i) In this first technique, central injector with 6 holes of size 0.17 mm and spray angle (155°) were replaced by 16 holes of size 0.08 mm and the same spray angle without changing position of the injector.

ii) In this technique, three stage spray angle nozzle with 32 holes, were staggered in the three sets, i.e., two sets of 0.08 ×12 at spray angle of 155° and 105° respectively and another with 0.08 ×6 at 55°.

iii) In this technique, to reduce possibility of wall wetting, two injectors were mounted diametrically opposite, with two sprays each, positioned in such a way that they collide at the middle of the chamber.

The reported work used the base fuel have higher cetane number (CN) 62 , at compression ratio of 16.5, it resulted in over advanced combustion, therefore new fuel blends with CN 19 and 40 were developed and used along with changes in intake temperature and compression ratio. Charge dilution was also tried, CO_2 was found to be the most effective diluent gas to control the combustion phasing. Harada et al. (Harada and Shimazaki 1998) carried out follow-up study and examined the effect of swirling flow pintle-nozzle injector that helps to produce more uniform mixture which in turn reduces wall wetting (evaluated by reduction in lube-oil dilution), thus improving combustion efficiency.

The early HCCI-DI, work further expanded by Hashizume et al. (Hashizume and Miyamato 1998). Dual injection strategy was adopted to accomplish this objective, side mounted injectors were used to produce lean premixed fuel while the central injector was used to inject the additional fuel near TDC. Various combinations of injection timing and fuel cetane number were tried to optimize performance and emissions. As the second injection lead to produce diesel like combustion, therefore

NO_x and PM emissions were much higher as compared to pure HCCI. However, NO_x emissions were reduced to about half without substantial increase in BSFC. However, PM decreases but UHC emission increases compared to pure diesel combustion.

Akagawa et al. (Akagawa H and Miyamoto T 1999) further extended the work mainly to solve the shortcomings of PREDIC, i.e., high UHC and CO emissions. They found that UHC and CO emissions could be decreased by adopting a pintle type injection nozzle, or a reduced top-land-crevice of piston. Fuel consumption was also improved by application of EGR, or addition of an oxygenated component to the diesel fuel. Following techniques were presented broadly

i) Crevice volume was reduced by raising top piston ring – shown to be beneficial to reduce UHC and by small extent CO.

ii) Cooled EGR was used- it is known to be beneficial in retarding the HCCI combustion phasing, hence it occurs near TDC.

iii) Several fuel additives like dimethyl carbonate, diethylene glycol dimethyl ether, methyl tertiary butyl ether (MTBE) and ethanol were blended with diesel fuel to determine their effects on combustion phasing. MTBE and ethanol, which have low Cetane number, were found to be advantageous for preventing overly advanced HCCI combustion.

Significant reduction of NO_x emission was recorded but the maximum IMEP could be achieved is limited to about 50% of that of conventional diesel combustion.

2.10 PREMIXED TECHNIQUES

Port Fuel Injection (PFI) is perhaps the most straightforward approach while to obtain a pre-mixed charge. Experimental study conducted by Osses et al. (Osses and Andrews 1998) on diesel fumigation to achieve partial premixed charge of vaporized fuel with air and remaining fuel is directly injected into the combustion chamber. It was observed by using this technique, 20% of port injected diesel lead to lower the NO_x and soot emissions but at the same time BSFC, CO, UHC and VOF (volatile organic fraction of the PM) was increased.

Conventional diesel combustion could be considered to be replaced by partial HCCI or PCCI-DI combustion. Horng-Wen Wu et al. (Horng-Wen Wu and Ren-Hung Wang 2011) investigated combustion and emission of a partial HCCI engine using ethanol and gasoline as a premixed fuel and diesel as a directly injected fuel in a single cylinder compression ignition engine. Concentrations of NO_x and smoke were lower in case of premixed ethanol as compared to premixed gasoline. They have found NO_x concentration decreases with increase in premixed fuel ratio. Nevertheless, unburned HC emission was increased with increase in premixed fuel ratio.

The effect of bowl shape on PCCI combustion was studied using 3D-CFD software and ECFM-3Z combustion model (Juttu and Thipse 2008). Both, simulation and experimental results were compared and found close resemblance. It was concluded that performance characteristics of engine not much dependent on geometry of bowl and at part load operations turbulence zone inside the bowl plays critical role in control of soot and CO emissions. In an another study effect of injection timing and injection pressure was studied (S Juttu and S S Thipse 2011) by using four cylinder diesel engine equipped with CRDI fuel injection system. As per the results reported NO_x emissions and low soot emissions were observed with high injection pressures and combined PCCI-DI could be economical solution to meet Euro-IV emission norms. Moreover, results from visulisation study showed higher injection pressures promotes PCCI combustion compared to lower injection pressures.

Kaneko et al. (Kaneko and Ando 2002) conducted study using pintle nozzle injector (12 MPa injection pressure) mounted on the intake manifold. In this study, the intake air temperature was maintained at 20 ºC, i.e., no heat was supplied to assist vaporization. When diesel fuel was used, UHC emission increased dramatically and the lubricating oil become diluted, presumably due to diesel fuel adhering to the cylinder wall. Some NO_x and notable smoke emissions were also present with diesel fuel, indicating that the mixture was not sufficiently homogenous.

Olsson et al. (Olsson and Tunestal 2001) presented a strategy for closed-loop control of a multi-cylinder turbo charged HCCI engine. A dual fuel port injection system was applied to control the combustion timing and load individually for each cylinder. For a better range of auto-ignition properties, isooctane and n-heptane were used as a fuel. Cylinder pressure sensors was used for feedback and for information regarding combustion. The crank angle of 50% heat release was calculated in real time for each cycle and used for timing feedback. Also, inlet air preheating was used at low loads to maintain high combustion efficiency.

Kim et al. (Kim and Lee 2006) experimented partial HCCI combustion as an effective way to control HCCI combustion. A DI (Direct injection) diesel engine was modified to operate on partial HCCI, port fuel injection mechanism was used to inject either diesel or gasoline or n-heptane. The combustion and emission study showed reduction in NO_x and soot emission simultaneously. NO_x and soot emission decreases when diesel were sued as a port fuel at room temperature, however, at elevated temperature it results in deterioration of combustion, lead to narrowed operating limit at higher premixed ratio. Gasoline as a port fuel showed remarkable improvement in the emissions as compared to other fuels.

In an experimental study conducted based on the control of fuel ignition timing and suppression of rapid combustion in a partially premixed charge compression ignition (PCCI) engine. They attempted injection of water directly into the cylinder as a reaction suppressor. The possible engine operating range with ultra-low NO_x and smokeless combustion was extended to a higher load range with the water injection (Naoyam and Hirokazu 2002).

2.10.1 Alcohol and HCCI Combustion

Application of alternative fuels, e.g., alcohol, natural gas, and bio-diesel considered to be effective method to control the PM and NO_x emissions (Wang and Clark 1997). Because of limitations on the development of dedicated compression ignition engine operate on alcohol. Researchers have shifted their focus on the partial replacement of diesel fuel, which could be done by two ways: either alcohol premixed with diesel or injected into air intake manifold. As we know, alcohol has higher self-ignition temperature, therefore, it is difficult to utilize in HCCI combustion mode, Yap et al. (Yap and Megaritis 2004) utilizes exhaust gas trapping techniques along with moderate intake air heating of bio-ethanol fueled HCCI engine, the combustion results showed very low NO_x emissions.

It has been shown that removal of water from bio-ethanol found to be very energy-extensive process therefore authors (Megaritis and Yap 2007) have thought of converting this demerits into merits. A diesel engine was modified to operate on HCCI mode, fueled with bio-ethanol and bio-ethanol and water blends (up to 20%) together with residual gas trapping was done by negative valve overlap. Earlier investigators (Iwashiro and Tsurushim 2002) have shown blending water with diesel successfully reduces NO_x emissions. Addition of water tends to cool down the in-cylinder gas at inlet valve, hence for stable combustion fueling rate need to be increased and, lambda decreases with increase in water blending. Thus, increase in maximum in-cylinder pressure rise rates was observed which in turn significant rise in NO_x emissions was found at more than 10% of water content in the fuel. Thus, fixed rate of water bending found to be counter productive to control the NO_x emissions.

2.10.2 Alcohol and Fumigation

The process used to atomized the fuel into the air, prior to induction in the engine cylinder, is known as fumigation (Abu-Qudais and Haddad 2000). Initially, fumigation was thought of as a means to enhance the power. German engineers employed this concept on numerous aircraft, which were used in Second World War during 1940's. (Gunston and Bridgman 1994).

The researchers worked on fumigation, prior to 1980's, did not emphasize on the emission characteristics; they rather focused on the usage of alternate fuel and power boosting system. However, later on 80's more than three decades fumigation have been widely explored by researchers to reduce NO_x emissions (Jiang and Ottikkutti 1990). Perhaps, it helps in two ways-one it can reduce the temperature of the flame, which, in turn, increases the density of the air that improves the performance, one more advantage of using alcohol as an alternative fuel is that it requires minimum modifications in the engine (Abu-Qudais and Haddad 2000). It could be possible to use widely available bio-fuels, like ethanol and methanol and, recently butanol showed relatively better alcohol after Butamx (ButamaxTM 2013) has come out manufacturing facility from bio-resources. It has much better physico-chemical properties as compared to other counter parts; therefore it's also considered as another alternative out of alcohol which is available as fumigants.

2.10.2.1 Methanol Fumigation

Dedicated methanol fueled buses were tried by USA, however, due to operational problem later on they were phased out. Gradually R-&-D interests have been shifted towards partial replacement of diesel by methanol. Two possible methods were studied: one by preparing methanol and diesel blends and, another by injecting directly into the intake air.

Baranescu (Baranescu 1986) experimentally examined the effect of fumigation of alcohol (methanol and 190, 160 proof ethanol) on six-cylinder, turbocharged, direct-injection, diesel engine's (7.1 L) combustion and emission pattern. The results showed fumigation of alcohol causes considerable increase in peak in-cylinder pressure and maximum rates of in-cylinder pressure rise as compared to pure diesel combustion. The maximum replacement of diesel fuel by methanol and ethanol together was about 30%, in terms of fuel energy liberated. Considerable increase in UHC and CO emissions were reported due to fumigation of alcohol, especially, at low engine loads. Moreover, NO_x emission was decreased.

Since the fumigation of alcohol significantly changes combustion characteristics (high latent heat of vaporization by cooling effect) so it is advantageous to optimize the injection timing to produce best results from the fumigation combustion mode. To explore complete harnessing possibility by improvised injection mechanism, system complexity needs to be increased, in terms of injection timing and injection duration. The effect of varying injection timing with the help of electronically controlled, multipoint ethanol injection system was studied by Savage et al. (Savage and White 1986). A turbocharged six cylinder (7.1 L Navistar DT-436B) diesel engine was selected to operate in fumigation mode. Experimental results revealed that by varying fumigants (methanol and 190-proof ethanol) injection timing (different crank angle position in the cycle-via induction air) makes remarkable difference in alcohol tolerance level and knock threshold. Therefore, up to 90 and 35% replacement of diesel fuel energy was possible by alcohol at low and high engine loads respectively, without any significant change in thermal efficiency.

Blending of diesel and methanol needed additive to prepare a stable blend (Chao and Lin 2001; Huang and Lu 2004). Moreover, it does not require any engine modification. Fumigation or injecting methanol into the intake air requires minor modification (Cheng and Cheung 2008; Yao CD and Cheung CS 2008) in the induction manifold, low-pressure gasoline injectors along with electronic controller was used. However, the major drawbacks of methanol fumigation were considerable increase in UHC and CO emissions, which have been reported by Yao et al. and Cheng et al. (Yao and Cheung 2007; Cheng and Cheung 2008). Hosoya et al. (Hosoya and Shimoda 1996) used diesel oxidation catalyst (DOC) to reduce UHC, CO and PM emissions.

Cheng et al. (Cheng and Cheung 2008) examined the performance and emission analysis of in-line four cylinder water cooled diesel engine (4.334 L) with fumigation methanol into the air intake at each cylinder. The experiments were done on 1800 rev/min engine speed and 0.08 MPa, 0.19 MPa, 0.38 MPa, 0.56 MPa and 0.67 MPa brake mean effective pressure (BMEP). For instance fueling the engine with diesel until 90% load and fumigating methanol until the engine reaches 100% load, creating a 10% replacement by load. Fumigation of methanol is done in such a way that it can top up the load by respective percentage (10%, 20% & 30%) at various loads. They have found that maximum amount of diesel could be replaced by methanol was about 43% of the total mass of fuel consumed. At all loads except, 0.67 MPa, brake thermal efficiency decreases with increase in fumigation methanol, however, it increases in the case 0.67 MPa load with an increase in fumigation methanol. UHC and CO emissions were increased, considerably, at all loads. CO_2 emissions increase at low and medium engine loads but no significant change was found at high engine loads. However, it was found that at high loads, smoke opacity and particulate mass concentration decreased considerably with increase in fumigation methanol.

Cheung et al. (Cheung and Chuanhui Cheng 2008) investigated effect of methanol fumigation along with bio-diesel on performance and emissions, they have used all equipment as discussed above. It was found brake thermal efficiency decreases with increase in fumigation ratio at low engine load (0.19 MPa), it changes from 27% to 23.2% which corresponds to fumigation ratio 0 to 0.55. At medium engine load (0.38 MPa), initially slight increase in brake thermal efficiency was seen but after 0.26 fumigation ratio it starts decreasing, however the changes were very small only 2%. At higher engine load (0.56 MPa), only 1% variation in BTE was found and no deterioration was observed within 0.2 fumigation ratio. The variation in thermal efficiency was found due to cooling effect of methanol, since methanol has a much higher latent heat of vaporization (1178 kJ/kg) as compared with that of bio-diesel (250 kJ/kg), they have found this phenomenon dominating at low engine load (due to low gas temperature) whereas, at high engine load higher gas temperature overcome cooling effect up to a higher extent. Moreover, homogenous mixture of air and ethanol burns faster, which increases brake thermal efficiency at high engine load. Therefore, not an obvious reduction in the brake thermal efficiency was observed at high engine load. At all engine loads increase in fumigation ratio resulted into drops in CO_2 concentration. It ranged from 3.47% to 3.21%, 5.55% to 4.99% and 7.96 to 7.59% associated with engine load of 0.19 MPa, 0.38 MPa and 0.56 MPa, respectively. At any fixed engine load (0.19 MPa, 0.38 MPa, 0.56MPa) both CO and UHC increased with fumigation ratio. For three engine loads CO increased by ratio 4 and UHC increased by a ratio 2.5, also the rate of increase was significantly higher at low engine loads. All fixed engine load (0.19 MPa, 0.38 MPa and 0.56 MPa) NO_x emission decreases with increase in fumigation ratio. The relatively significant reduction was found at low engine load. However, they explained three distinct mechanisms which control the NO_x emissions (oxygen content of fuel has been used, intake charge cooling due to methanol and ignition delay) and these factors compete with each other. The maximum reduction in NO_x emission was reported at low engine load (0.19MPa/0.38 MPa) about 20% and at high engine load (0.56 MPa) it reduces to about 10%. The particulate number concentration and size distribution showed the total no of particles decreases by about 30% (at 0.19 MPa and 0.38 MPa) but it reduces less than 10% in the case of 0.56 MPa. PM mass concentration reduces significantly due to increased oxygen content in the fuel, thus higher amount of fuel was burnt in the premixed phase and diffusion combustion phase becomes cleaner due to presence of methanol.

In spite of biodegradable and nontoxic nature of bio-diesel, researcher (Sharp and Howell 2000) found that at higher loads application of bio-diesel in diesel engine may increase the NO_x emissions. Engine experiments were performed at constant engine speed of 1800 rpm and at three different engine loads (0.19 MPa, 0.38 MPa and 0.56 MPa). In this study, the fumigation ratio defined as the ratio of volume of methanol to the total volume of methanol and bio-diesel being burnt, the values selected were 0 (pure bio-diesel), 0.27, 0.37, 0.43 and 0.55.

Cheng et al. (Cheng and Cheung 2008) compared emissions from ULSD, bio-diesel , bio-diesel and 10% methanol blends and bio-diesel and 10% methanol fumigation in the 4 cylinder direct injection diesel engine same as used for methanol fumigation (Cheng and Cheung 2008). In the blending mode, methanol tends to increase ignition delay therefore higher percentage of fuel was burnt in premixed phase together with increase in peak net heat release rate. Fumigation mode, also, showed longer ignition delay (cooling effect produced by methanol) period and hence greater amount of bio-diesel and methanol burns in the premixed mode. Thus, shorter diffusion controlled combustion and bio-diesel combust in the environment of air and methanol instead of air alone. At 10% fumigation or blending minimum BSFC was associated with ULSD, whereas, fumigation mode has lower value compared to blending but, higher than that associated with bio-diesel. ULSD showed lowest brake thermal efficiency at all loads, contrary to the lowest BSFC, whereas maximum brake thermal efficiency (39.6%) was attained with 10% methanol premixed mode at 0.67 MPa. Both ULSD and bio-diesel showed almost similar BSCO2 emissions, however, 10% methanol, either in blending or premixed mode, resulted into marginal reduction of about 2.5%. Nevertheless, UHC and CO emissions increased significantly when engine was operated under fumigation mode, NOx emissions reduces by about 5% due to cooling effect of methanol. Compared to ULSD both Bio-diesel and methanol are able to reduce particulate emission. Relatively, methanol seems to be more effective due to oxygen content in the fuel and relatively longer premixed mode of combustion. Nevertheless fumigation mode, compared to blending mode, has less potential to reduce particulate. The reason behind is higher hydrocarbon in fumigation mode condenses into particulate. Submicron size of particles was found to reduce when engine was operated with 10% methanol.

In an another study (Cheung C S and Zhang Z H 2009) authors have analyzed performance and emissions of fumigation methanol under different engine speeds (1280, 1920 and 2560 rpm). Fumigation % represents portion of engine loads takes up by the methanol. The brake thermal efficiency decreases with engine speed but increases with engine load for diesel fuel and fumigation methanol. At each engine speed corresponding to the 20% and 40% engine load, the brake thermal efficiency decreases with increasing level of fumigation methanol. At high engine loads, no significant change was found with level of fumigation methanol. At high engine speed BTE deteriorates due to incomplete combustion. UHC and CO emissions increased to more than two times, whereas, NO_x emissions decreased from 5.6% to 27.5% (averaged at different loads) as compared to baseline fuel. At high engine loads CO_2 emission decreases with increase in fumigation methanol whereas no significant change was recorded at low loads. Maximum reduction in particulate mass concentration was found at 2560 rpm ranges from 19-48% (different level of fumigation methanol and engine loads) compared to baseline fuel. Significant increase in unburned hydrocarbon was the result of higher unburnt methanol. However, unburnt methanol decreases with increase in engine loads but less dependent on engine speed. DOC helps to reduce further PM. At all engine speeds, BTE decreases with increase in premixed methanol

The experimental investigation (Zhang and Cheung 2009) was carried out to understand the performance and emission characteristics of naturally aspirated, 4-cylinder direct injection diesel engine fueled with diesel and methanol used as fumigants. To evaluate the full range of operating conditions, five loads (0.08 MPa, 0.19 MPa, 0.39 MPa, 0.58 MPa& 0.70 MPa) were selected and engine speed was maintained constant at 1800 rpm with and without DOC. The methanol accounted for 10% to 30% of the engine loading. At low to medium loads brake thermal efficiency decreases with increase in fumigation, however, small positive change was observed at higher loads. The maximum reduction and increase in BTE was observed at 30% (methanol displacement ratio) fumigation, it was about 11.2% and 1.8% at BMEP of 0.08 MPa and 0.7 MPa respectively. BSCO and BSHC decrease with increase in load but increases with increase in fumigation. As compared to baseline results, at 30%

fumigation the maximum increase in BSHC was 7 times (at 0.08 MPa) and 3 times respectively (at 0.7 MPa), whereas, the maximum increase in BSCO was observed about 5.5 times. $BSNO_x$ and $BSNO_2$ decrease with increase in engine load, despite an increase in NO_x concentration in the engine exhaust with increase in engine load. The maximum increase in $BSNO_2$ was 7.8 times at 30% fumigation methanol. DOC helps to reduce $BSNO_x$ with a very small magnitude of only 2% for the entire range of fumigation methanol. CO_2 concentration increases with engine load but the $BSCO_2$ decreases with engine load. The maximum decrease of $BSCO_2$ occurred at high engine load with 30% fumigation methanol of about 7.2 times. After DOC, there is a slight increase in $BSCO_2$ was observed, which is the result of conversion of CO into CO_2. The smoke opacity increases with load, however considerable reduction in smoke opacity and particulate mass concentration was discovered by fumigation methanol, the relatively higher reduction was found at medium to high engine load and very small at low engine load. Also, the particulate mass concentration could be reduced further by DOC. Thermal efficiency was found to be reduced at low engine load whereas there was no substantial change observed at medium to high engine load.

To explore further, authors (Zhang and Cheungb 2010) carried out fumigation methanol study on 4 cylinder direct injection diesel engine aimed to know regulated and un-regulated (benzene, toluene, formaldehyde and methanol) emissions and performance. The experiments were performed at constant speed of 1920 rpm and five engine loads (BMEP) of 0.13 MPa, 0.27 MPa, 0.40 MPa, 0.53 MPa and 0.63 MPa. At low engine loads of 0.13 MPa and 0.27 MPa they observed there was reduction in brake thermal efficiency by 11% and 10% corresponds to 30% fumigation methanol. However, at high engine loads (0.40 to 0.63) slight increase in brake thermal efficiency were recorded. At 0.13 MPa and 0.63 MPa engine load with 30% fumigation methanol, BSHC emissions increased by 9.6 and 3 times respectively as compared to pure diesel. Moreover, under the similar conditions BSCO emissions increased by 4.5 and 6.2 times. At all engine loads except 0.63 MPa there was a reduction in NO_x emissions with increase in level of fumigation, maximum reduction were recorded at 0.4 MPa (about 20.8%) at 30% fumigation methanol. At 0.13 MPa engine the brake specific unburned methanol increased from 0.1 $gkW^{-1}h^{-1}$ (at baseline engine or pure diesel) to 25.3 $gkW^{-1}h^{-1}$(at 30% fumigation methanol). Moreover, with increase in engine loads the unburnt methanol decreases but still higher than the pure diesel combustion. Similar to brake specific formaldehyde emission increases with increase in fumigation methanol, the maximum rise was about 3.4 -4.5 times at 30% fumigation methanol. It was concluded that fumigation methanol decreased the emission of ethylene, ethane, 1-3 butadiene, as a drawback there was significant increase in unburnt methanol, formaldehyde and BTX (Benzene, toluene and xylene) was reported.

In an another investigation (Zhang Z.H. and Tsang 2011) they had analyzed the effect of fumigation methanol and ethanol on the gaseous and particulate emission on the similar set-up and experimental condition as discussed above. The results showed that BTE decreases by 2-5% and 3 -8% in the load range of 0.08 MPa- 0.39 MPa, whereas at higher engine load (0.7 MPa) it was increases by 10% and 9% when engine was operated at 10% and 20% fumigation methanol respectively as compared to Euro-V diesel. At 10% fumigation ethanol in the load range (0.08 MPa - 0.39 MPa) showed reductions within 2-4%, whereas at 20% ethanol fumigation level it increased to 7%. Similar to fumigation methanol at higher engine load (0.7 MPa) and 20% ethanol fumigation brake thermal efficiency increased by 3% and no significant change was observed at 10% ethanol fumigation. Both UHC and CO emissions increase with increase in level of fumigation. At low engine load range (0.08 MPa-0.39MPa) BSHC and BSCO emissions were higher in case of fumigation methanol compared to fumigation ethanol, however at high engine loads no significant difference was observed from two alcohols. $BSNO_x$ emission decreases with increase in level of fumigation methanol or ethanol. At all the loads methanol fumigation leads to higher NO_x emissions reduction compared to ethanol. Moreover, at low engine loads NO_x emissions reduction was higher as compared to high engine loads. BSPM decreases with increase in fumigation of either alcohol, similar to $BSNO_x$ it also showed higher reduction when methanol was used as a fumigants. The maximum reduction corresponds to 20% methanol fumigation of about 20-41% compared to pure diesel combustion. Particulate number concentration also decreases with increase in level of fumigation methanol or ethanol.

2.10.2.2 Ethanol Fumigation

Historically, Havemann et al. (Havemann and Rao 1954) thought of utilization of ethanol in diesel engine. Initially, they tried blending of ethanol with diesel, due to poor miscibility of ethanol in diesel it could not produce satisfactory results. Therefore, they attempted to utilize it by fumigation technique. The fumigation tests were carried out on a Ricardo research engine and on a Petter AV1 Series II production engine. They found at full load the maximum fuel energy that could be provided by ethanol was limited up to 36% (Ricardo engine) and 70% (Petter engine), in addition to it, the maximum limit of ethanol fumigation strictly controlled by knocking (Ricardo Engine) and misfire (Petter engine). Results from the study showed, in both engines ethanol fumigation led to high air utilization, which made it to possible to use ethanol to provide a power boost (over fueling) without sacrificing the smoke limit, moreover, smoke density was reduced considerably also it was in proportion of amount of ethanol fumigation. In case of Ricardo engine the thermal efficiency was improved slightly at higher engine loads.

Ethanol fumigation was explored by Alperstein et al. (Alperstein and Swim 1958) in 1957. They examined combustion in single cylinder (swirl-chamber and open-chamber) naturally aspirated diesel engines. Results reveal that smoke reduction up to 70%, increase in smoke-limited power output up to 18.5%, and decrease in specific fuel consumption up to 9.8%.

Barnes et al. (Barnes and Kittelson 1975) examined alcohol (methanol, ethanol and isopropanol) fumigation on six cylinder (5.1 L), turbocharged direct injection diesel engine. Single atomizing nozzle was fitted in the upstream of turbocharger compressor instead of six (one each for cylinder). The results revealed that engine efficiency decreased with increasing flow of alcohol. The loss of efficiency was the result of insufficient compression heating and evaporation of alcohol which leads to considerable reduction in the intake manifold temperature. Smoke density decreases with increase in alcohol fumigation, whereas, other emissions largely remain unaffected. The rate of in-cylinder pressure rise was increased sharply at high engine loads, without any indication of engine knocking.

Bro and Pedersen (Bro and Pedersen 1977) examined the effect of ethanol fumigation on combustion, performance and emissions of single cylinder natural aspirated direct injection diesel engine. The results revealed improved thermal efficiency and reduction in smoke at high engine loads, however, UHC emissions increased with ethanol fumigation. Uncontrolled combustion or engine knocking limits the maximum amount of ethanol fumigation.

Chen et al. (Chen and D. Gussert 1981) performed experimental investigation in order to study combustion, performance and emission of 4-cylinder (5.5 L, turbocharged) direct injection diesel engine which was modified and simple atomizing nozzle fitted in the induction air channel downstream of the compressor to inject alcohol. Ethanol (200 and 160 proof) was selected as fumigants. Results showed 160 proof ethanol mixture was able to substitute up to 25% of diesel. They observed variation in the exhaust port temperature at different conditions affect vapourization of ethanol and in turn results in to non-uniform distribution of ethanol in different cylinder. Combustion study showed that rates of cylinder pressure rise and ignition delay increased due to ethanol fumigation, whereas, thermal efficiency fluctuates at high and low engine loads in the positive and negative direction respectively. However, performance of the engine could not be changed considerably by different ethanol proofs. Smoke and NO_x emissions were decreased with increase in ethanol fumigation, perhaps UHC and CO increased.

Heisey and Lestz (Heisey and Lestz 1981) performed comparative study on performance and emissions of single cylinder diesel engine operated on fumigation ethanol and methanol. Thermal efficiency under ethanol fumigation mode showed marginal improvement compared to methanol at 1/3 of maximum engine load, however, between 2/3 to full maximum load condition no significant difference was observed. The ethanol in different proofs (200, 180, 160, and 140) was used and simple atomizing nozzle was used to inject in the intake air. They concluded that lower proofs helps to reduce NO_x emissions without significantly affecting thermal efficiency and rate of cylinder pressure rise. However, relatively longer ignition delay was found in case of lower ethanol proofs.

Authors (Broukhiyan and Lestz 1981) examined effect of fumigation ethanol on engine fuel efficiency, gaseous emissions, smoke, raw particulate and engine knocking by using indirect injection, naturally- aspirated light-duty automobile diesel engine. Simple air-atomizing nozzle was used and fitted with specially designed mixing chamber to fumigate ethanol among eight cylinders. The results showed that thermal efficiency increases with ethanol fumigation at higher engine loads, however, the quantity of ethanol fumigated limits by engine roughness (occurrence of knock). Particulate matter (mass basis) and NO_x emissions decrease from the reference diesel level with increase in ethanol fumigation.

Sullivan and Bashford retrofitted fumigation kit on an Allis-Chalmers 7040 diesel tractor (Sullivan and Bashford 1981). Ethanol was sprayed into the upstream side of turbocharger compressor and flow rate was regulated by boost pressure. Reported results showed improved diesel fuel economy but very high rotational speed of compressor blades could not sustain impingement of ethanol fuel droplets that leads to considerable damage on the blades within thirty hours of operation.

Goering and Wood (Goering and Wood 1981) investigated the effect of ethanol (100- and 160-proof) fumigation on performance and emission of 3-cylinder, naturally-aspirated tractor engine. The aim of this investigation was to understand how ethanol fumigation helps in augmentation of the total engine power (over-fueling) instead of replacing diesel fuel. Nevertheless, intake air was needed to be heated to vaporize lower ethanol proofs. Results showed that at idle condition and full engine load ethanol was able to add up to 40% and 16% of the total energy, respectively. It was found that engine power output increased, however, thermal efficiency decreased. The emissions of smoke, UHC and CO increased.

Ethanol fumigation tests were performed (Shropshire and Goering 1982) on a four - cylinder, DI diesel engine. Three different ethanol proofs (100, 150, and 190) were selected for the study and four injection nozzles was mounted on each induction manifold of the engine. They showed up to 66% and 30% of energy replacement could be achieved by ethanol (190 proof) fumigation at high and low engine loads, respectively. Thermal efficiency (BTE) was increased slightly at full engine loads, however, adverse effect was observed at lower engine loads. Nevertheless, premixed of ethanol results in to reduction in smoke levels at all engine loads but, on the other hand, UHC and CO were increased.

A long-term endurance and tribological testing was done by Allsup (Allsup 1983) on three engines for five houndred hours. Two engines were tested with fumigation technique and one with ethanol diesel blends (E-diesel). The results showed remarkably clean combustion zones, piston rings and exhaust valves were found for engines running on fumigants whereas other engine using an emulsified fuel had a premature engine failure.

A comparison of various fumigation systems was done (Shropshire and Bashford 1984). Abnormal combustion was found when using air atomizing nozzle which creates fine spray. Thermal efficiency deteriorated at low loads but maintained at high loads. Single nozzle could not produce better results compared to multiple nozzles.

Lower proof ethanol could be produced from distillation facility without much effort, whereas higher proof ethanol was more cost extensive due to dehydration process which consumes lots of time. It is always desirable to use lower proof ethanol. To understand and analyze the effect of different ethanol proofs, Hayes et al. (Hayes and Savage 1988) conducted experimental investigation on six cylinder, turbocharged, direct injection diesel engine (7.1 L Navistar DT-436B) fitted with multipoint electronic ethanol injection system. Engine testing was performed at constant engine speed of 2400 rpm and 0.2, 0.5 and 0.8 MPa BMEP. Optimum ethanol proof which produces better results are fall within 100 to 150 proof. They observed reduction in peak cylinder pressure and the maximum rate of in-cylinder pressure rise was reduced up to 20% as compared to absolute ethanol (200 proof). NO emissions reduced as compared to diesel fuel for hydrous ethanol (150 proof), however, UHC and CO emissions were increased several times as compared to diesel only combustion.

Griffith et al. (Griffith and Savage 1988) examined the effect of ethanol fumigation on Harvester farm tractor engine (6-cylinder, turbocharged, direct injection, 7.6 L diesel engine) by both laboratory and field testing. Ethanol fumigation was achieved by installing electronically controlled multi-point fuel injection system. Experimental results revealed that maximum 28% of fuel energy could be supplied by ethanol at low engine loads and 13% at full load. Lubricating oil and cylinder walls optical borescopic analysis indicates greater degree of engine wear due to fumigation. The reason of abnormal wear could be dilution of lubricating oil due to ethanol which inducted with air and in turn affected the lubricating oil film at the cylinder walls.

A study was conducted (Jiang and Ottikkutti 1990) to explore potential of ethanol fumigation to address the problem of NO_x emissions with the help of reducing adiabatic flame temperature. A four cylinder, turbocharged, direct injection diesel engine (4.5 L John Deere 4276T) was selected to perform investigation of performance, emission and adiabatic flame temperature. Minor modification was done in the induction manifold for ethanol fumigation. They showed stoichiometric adiabatic flame temperature reduces as the water in the mixture vaporized, however thermal efficiency was not affected till the water flow reaches 2.5 times the diesel flow rate. UHC and CO emissions tend to increase due to ethanol fumigation, whereas substantial reduction in the NO_x emissions was found. Using lower ethanol proofs was found to be more beneficial, since higher water content reduces peak combustion temperature. Hence, as the thermal NO_x emissions largely depend on the peak combustion temperature tends to reduce drastically.

Ethanol fumigation was studied at G. B. Pant University of agriculture and Technology, Pantnagr India (Ajav E. A. and Bachchan Singh 1998). A single cylinder direct injection, constant speed, water cooled diesel engine, 1.102 L was used to study their performance and emissions under fumigation mode at 20 °C and 50 °C inlet temperatures of mixture. A solex down drought carburetor was used to vapourize ethanol, EGR device was employed to preheat air-ethanol mixture. Small variation in engine speed was found at maximum brake power developed by the engine under pure diesel and fumigation mode. BSFC decreases with increase in load but small increase was reported after 75% of the engine load. At low loads (0-50%) fumigation ethanol had higher BSFC than diesel. The reason could be explained by reduction in quantity of diesel displacement when operation moved towards higher engine loads. At 66% engine load both diesel and fumigation mode had equal BSFC. The maximum percentage replacement (mass basis) of diesel under preheated/unheated condition was about 33.6% at low loads (0-50%), whereas at full load it reduces up to 15%. The maximum brake thermal efficiency was found at 75% load and fumigation without preheating mode showed relatively higher (27%) value as compared to pure diesel combustion (25.7%). Exhaust gas temperature and lubricating oil temperature increased with increase in load. CO and NO_x emission increases slightly at full load. However, no detailed study was conducted on emissions.

Abu-Qudais et al. (Abu-Qudais and Haddad 2000) conducted comparative analysis of ethanol fumigation technique and ethanol as a blend with diesel fuel. A single cylinder, naturally aspirated, four stroke, direct injection, 0.582 L diesel engine, and for ethanol fumigation at low pressure single hole spraying nozzle was selected for the study. The testing was done in the engine speed range of 1000 rpm to 2000 rpm, total energy kept fixed (baseline fuel) and ethanol supplied in terms of energy flow rates 5%, 10%, 15% and 20%. Both fumigation and blending showed slight gain in brake thermal efficiency, however, relatively more favorable results were achieved by ethanol fumigation which produces 7.5% gain while operating at 20% fumigation. Whereas, optimum percentage for ethanol diesel blends (15%) produces an increase in thermal efficiency was about 3.6%. The reason could be explained by the increase in ignition delay period that results in rapid heat release rates and hence less time was available to transfer the heat from the engine to coolant which saves loss of energy. Results showed optimum percentage of ethanol fumigation rate was 20%, and blending ethanol with diesel fuel was 15%. UHC emission ranges from 20% to 36% under different operating condition when using 20% ethanol fumigation, however, it increased from 25% to 49% under ethanol diesel blends. Moreover, at 20% ethanol fumigation leads to produce a reduction of engine smoke by 48%. CO emissions were increased by 55% and 43.4% under fumigation (20%) and blending (15%) condition respectively.

Leahey et al. (Leahey and Jones 2007) adopted several strategy to use oxygenates (bio-diesel and ethanol) with diesel in a single-cylinder CFR engine. They used different injection strategy (bio-diesel & diesel with advanced injection timing and ethanol diesel blends with late injection timing). Blending of bio-diesel and ethanol with diesel showed small reduction in the soot emissions. However, fumigation of ethanol successfully reduces up to 50% of soot emissions without adversely affecting lubricity which was one of major concerns when ethanol was used in the form of blend. Thermal efficiency improved by about 3% with all combination of biofuel and diesel as a fuel as compared to pure diesel.

K. S. Tsang et al. (Tsang and Zhang 2010) studied fumigation of ethanol on 4-cylinder direct injection diesel engine (4.334 L, Isuzu 4HF1) fueled with Euro-V diesel. They examined various performance, combustion and emission parameters on five different BMEP (0.08, 0.19, 0.39, 0.58 and 0.70 MPa) at constant engine speed of 1800 rpm. BSFC decreases with increase in engine load but increases with increase in fumigation ethanol. BTE was reduced at low and medium loads but there no significant change was observed at high engine loads. At low load (0.08 MPa) peak in-cylinder pressure decreases with increasing fumigation ethanol but at medium and high load (0.39 and 0.70 MPa) the trend was exactly reversed. The peak heat release rate increases with increase of fumigation ethanol irrespective of engine load. Start of combustion was retarded due to fumigation ethanol, thus it increases ignition delay. Maximum increase in ignition delay corresponds to 20% fumigation ethanol of about 2° CA. Premixed combustion phase increases with increase in fumigation ethanol at high engine loads. At engine load of 0.08 MPa and 0.70 MPa, the maximum increase in BSCO was associated with 20% fumigation ethanol were 1.3 and 2.7 times as compared to diesel fuel respectively. Moreover, BSHC increased by 3.3 and 2.4 times in the similar condition as stated above. The reduction in BSNO$_x$ with reference to pure diesel combustion was more significant at low engine loads compared to high engine loads. At engine load of 0.39 MPa and 0.70 MPa the maximum reduction associated with 20% fumigation ethanol was 14.7% and 3.9% respectively with reference to diesel fuel. Smoke opacity decreases significantly at medium and high engine loads with increase in fumigation ethanol. The maximum reduction was reported at 0.58 MPa was about 56% associated with 20% fumigation ethanol. PM concentrations were reduced up to 27% under different percentage of fumigation ethanol and it was more obvious at medium and high engine loads. The maximum reduction in particle number concentration was about 30% associated with 20% fumigation ethanol at 0.39 MPa engine load.

Ethanol fumigation was studied at the Delhi Technical University, Rohini New Delhi India in 2008 (Bhupendra Singh Chauhana and Naveen Kumar 2011). Ethanol was fumigated by using a carburetor. Quantity of ethanol was controlled by butterfly valve of CV (constant velocity) carburetor and quantity of diesel injected was regulated by speed governor of the engine. Interestingly air box has two outlets-one was directly connected with the induction manifold and another with carburetor which further rejoined to the induction manifold. The emissions study was performed on single cylinder (10 HP), constant speed direct injection, and water cooled Kirloskar diesel engine with displacement volume of 0.78 L. At no load CO emissions increases, however at low and high load it decreases with increase in fumigation ethanol. The maximum reduction in CO was recorded at 20% and 45% of full load that was associated with 20% fumigation ethanol. Similarly, CO_2 emission decreases with increase in fumigation ethanol at 20% and 45% of full loads and remains constant at no load condition. At no load NO_x emission increases with increase in fumigation ethanol, however at 20% and 45% of full load it decreases up-to certain level of fumigation ethanol (20-22%) and again starts increasing. Considerable amount (up to 7 times) of UHC emissions was observed at no load and small load with increase in fumigation ethanol, nevertheless at full load the maximum substitution was limited up to 20% fumigation ethanol and it showed relatively lower emissions. Reduction in smoke opacity was higher in the medium and high engine load conditions. At 70% and full load maximum reduction in smoke opacity was found at 14% of fumigation ethanol.

Ethanol fumigation with hot and cold EGR was studied at the Christ University, Bangaluru Karnataka India (Hebbar and Anantha 2012). A small DI, water cooled four-strokes Kirloskar diesel engine (0.624 L, AV 1) was used for the study. At 50% EGR brake thermal efficiency drops by 20% and 40% with fumigation and without fumigation respectively. However at low level of fumigation (10%) and EGR (30%) measured loss was very low around 5% and further increase in fumigation leads to enhance brake thermal efficiency by around 20%. CO_2 emissions showed insignificant change with increasing EGR under fumigation ethanol associated with EGR mode, whereas, it increases with EGR without fumigation ethanol. Pure diesel combustion along with EGR emits higher smoke compared to fumigation ethanol mode, also considerable reduction they had measured when fumigation ranges from 5% to 15%. NO_x emission reduced from 10% to 15% by fumigation ethanol. UHC emissions increase with EGR rate and level of fumigation ethanol.

The role of injection timing on performance of dual fueled operation was demonstrated by Padala et al. (Padala and Woo 2012) by using small single cylinder common rail DI engine. In-cylinder pressure decreases at TDC whereas net heat release rate increases with increase in fumigation ethanol. Brake thermal efficiency increased during fumigation ethanol as compared to diesel fuel for the same combustion phasing. They showed stable combustion could be achieved up to 40% fumigation ethanol, also engine delivers higher IMEP while running on fumigation ethanol compared to diesel fueling. One interesting observation was made that increase in fumigation ethanol leads to reduce premixed phase and late cycle burning duration.

Surawski et al. (Surawski and Miljevic 2010) conducted a study in order to investigate the effect of ethanol fumigation on PM emissions, volatility, and toxicity. Their experiments were carried out on a four-cylinder Ford diesel engine (2.7 L) and the fumigation system featured an electronically-controlled ethanol injector. A Scanning Mobility Particle Sizer (SMPS) was used to analyze particle-size distributions. They showed reductions in total particle mass concentrations compared to baseline diesel. However, the particle size distributions also exhibited a shift towards smaller particle diameters (nucleation-mode particles), with a corresponding increase in total particle number concentrations with ethanol fumigation.

2.11 ETHANOL PRODUCTION FROM RENEWABLE SOURCES

Ethanol is derived from renewable sources feedstock, locally grown plants such as wheat, sugar beet, corn, straw, and wood (Gray and Zhao 2006). It is possible that wood, straw and even house-hold wastes may be economically converted to alcohols. Nevertheless, largely ethanol is produced by fermentation; it is an anaerobic biological process in which the sugars are converted to alcohol by the action of microorganisms, usually yeast (Demirbas 2007). It is envisaged that long-term, economical and environmentally friendly production of bio-ethanol is via carbohydrates (hemicelluloses and cellulose) in lingocellulosic materials. The lignocellulose is subjected to delignification, steam explosion and dilute acid pre-hydrolysis, followed by enzymatic hydrolysis and fermentation into bio-ethanol (Satgé de Caro and Mouloungui 2001; Fahd and Ebna Alam Wenming 2013). In

India, ethanol is produced from conventional feedstocks, such as sugar cane/beet juice and molasses or from starch based feedstocks like wheat, corn or even tubers like cassava.

As the agricultural research demonstrated that tropical sugar beet and sweet sorghum are cost effective feedstocks, recently sweet sorghum also accepted commercially (Praj Industries) to produce bio-ethanol that would cater to the growing need of ethanol as a fuel. India is the world's largest sugar consumer and currently producing 4% of global bio-ethanol from sugar cane. Praj industry at Pune in India (Praj Industries) emerged as the first company in South Asia to set up an integrated second generation (2G) cellulosic ethanol plant. Table 2.1 (Imran and Varman 2013) shows increasing ethanol production to justify the ethanol demand as a fuel.

Table 2.1: Bio-ethanol production annually (millions of U.S. liquid gallons per year)

Country or region	**2007**	**2008**	**2009**	**2010**	**2011**
United States	6485	9235	10,938	13,231	13,900
Brazil	5019.2	6472.2	6577.89	6921.54	5573.24
European Union	570.30	733.60	1039.52	1176.88	1199.31
China	486.00	501.90	541.55	541.55	554.76
Canada	211.30	237.70	290.59	356.63	462.3
Thailand	79.20	89.80	435.20	270.13	289.29
India	52.80	66.00	91.67	110	135
Colombia	74.90	79.30	83.21	73.96	79.26
Australia	26.40	26.40	56.80	66.04	87.2

2.12 EFFECT OF EGR ON HCCI COMBUSTION

Despite the large differences in the auto-ignition characteristics of gasoline and diesel fuels, most applications of HCCI for SI and diesel engines use some form of exhaust gas re-circulation (EGR) to help control the combustion phasing.

The (Sjöberg and Dec 2011) results have showed that the single-stage ignition fuel ethanol is quite sensitive to the reduction of compression heating that occurs with EGR due to the higher heat capacity of the EGR gases compared to air. This high sensitivity to the cooling effect of EGR is similar to that of gasoline and iso-octane, which also are single-stage ignition fuels under these conditions. On the other hand, ethanol is very insensitive to the reduction of O_2 concentration associated with the addition of EGR. Both of these characteristics relate to ethanol's molecular stability – it does not react much until just before the hot-ignition point is reached. Consequently, ethanol has a low intermediate-temperature heat-release rate, which leads to a low temperature-rise rate prior to hot ignition, and therefore a high sensitivity to the cooling effect of EGR. Also, the relative lack of intermediate-temperature heat release prevents $[O_2]$ from having much influence on the temperature rise prior to hot ignition, leading to a low sensitivity of the auto-ignition timing to changes of $[O_2]$. Finally, both H_2O and trace species have significant ignition-enhancing effects for ethanol that to some degree counter the retarding effect of EGR.

Study (Ryan III and Callahan 1996) carried out on a single-cylinder, direct-injection diesel engine, which was modified to operate on compression ignition of homogenous mixtures of diesel fuel and air. The limitations of HCCI mode of combustion which happens to be misfire and knock could be overcome. To control the auto-ignition, the variables examined in this study included air-fuel ratio, compression ratio, fresh intake air temperature, exhausts gas re-circulation rate, and intake mixture temperature. The results suggested that controlled homogeneous charge compression ignition is possible. It is found that satisfactory power settings are possible with high EGR rates and stoichiometric fuel-air mixtures. Successful HCCI operation results in near zero smoke emissions.

Follow-up research on the same engine (Gray III and Ryan III 1997) it was observed that for diesel fueled HCCI combustion, premature ignition and knocking occurs if normal diesel compression ratios were used. Compression ratios in the range of 8:1 to 13:1 produces satisfactory results, together with intake temperatures and the amount of EGR also plays important role. Compression ratios of 9:1 to 11:1 provide acceptable combustion over the largest range of fueling rates. However, combustion phasing was still advanced even at low compression ratios. Also, relatively high intake temperatures (135 ºC to 205 ºC) were required to minimize the accumulation of liquid fuel on the surface of intake system. When intake temperatures were reduced below 130 ºC, smoke emissions was increased significantly. Authors suggested that this owing to combustion occurring in scattered diffusion flames around large droplets. Further, UHC emission tended to be very high. As a result of poor combustion efficiency, reduced compression ratios, and non-optimal combustion phasing, fuel consumption increased by an average of about 28% over normal direct-injection (DI) diesel combustion.

Mingfa Yao et al. (Mingfa and Zheng Chen 2006) experimentally examined the HCCI combustion fueled by DME and methanol, two injectors were mounted near the port of water cooled diesel engine modified to operate on HCCI mode. Quantity of DME and methanol were regulated by electromagnetic valve and electric controller respectively. The percentage of DME was expressed in terms of amount of energy released by DME to the total energy released by DME and methanol together. It was found that HCCI combustion could be controlled by regulating percentage of DME and EGR. However EGR is not able to enlarge the maximum IMEP but it can be increased the higher % of DME at normal combustion. Ultra-low NO_x emission was reported at normal combustion, CO emissions found to be closely related with the peak mean temperature in the cylinder. UHC emissions found to be reaching a lowest value by adjusting percentage of EGR and DME.

Simescu et al. (Simescu and Ryan 2002) carried out partial HCCI combustion in heavy duty engine along with cooled and hot EGR. The lowest weighted $BSNO_x$ achieved was 2.55 g/kW-hr using cooled EGR and 20% port fuel injection (PFI). This corresponds to 54% reduction compared, however, BSHC and BSCO emissions increased by a factor of 8 and 10, respectively, compared to the stock engine.

Qiang et al. (Qiang and Junhua 2012) investigated the influence of pilot injection and exhaust gas re-circulation (EGR) on combustion and emissions in a HCCI-DI combustion engine and observed that, NO_x emission decreased with increase in pilot quantity. In addition, substantial reduction in NO_x emission was found with low level of EGR as compared to without EGR. Nevertheless, smoke opacity was increased because of EGR.

Xing-Cai et al. (Xing-Cai Chen and Lü 2005) used single-cylinder four stroke high speed diesel engine to investigate the effect of RON (research octane number) fuels, EGR rate, intake charge temperature, equivalence ratio, coolant temperature, and engine speed on the HCCI ignition timing, combustion duration, cycle-to-cycle variation, and emissions. Moreover, minor modifications were done in the intake, exhaust and fuel injection system to successfully operate on HCCI mode of combustion. The fuels selected for different set of operating conditions were n-heptane and iso-octane and their blends with RON25, RON50, RON75, and RON90. The results revealed that the cumulative heat release in the first-stage combustion was strongly dependent on the concentration of n-heptane in the mixture. The start of ignition of the second-stage found to be linear with the start of ignition of the first-stage.

In continuation to the first part Xing-Cai (Xing-Cai Chen and Lü 2005) author studied the effect of EGR rate, intake temperature, coolant temperature and engine speed on the HCCI combustion characteristics of a single-cylinder HCCI engine. The ignition timing for the both first and second stage of combustion was found to be retarded along with combustion duration, and it increases with the use of cooled EGR. High octane fuel showed considerable change in the emissions of CO and UHC whereas moderate effect was found in case of n-heptane and RON25.

Shi et al. (Shi and Cui 2006) used combined internal EGR with the help of variable valve timing (VVT) and external cooled EGR. Early direct injection of diesel fuel was done to enhance the fuel vaporization by the hot EGR trapped into the cylinder by negative valve overlap. This helps to prepare homogenous mixture, which in turn reduces NO_x and soot emissions. Cooled external EGR was effective to control the start of ignition; this basically delayed the start of combustion, and as the diesel has high cetane number shows early start of ignition (SOC) at elevated temperature.

Machrafi et al. (Machrafi and Simeon Amouroux 2008) performed parametric study by using mono-cylinder cooperative fuel research engine on the emission of HCCI combustion fueled with primary reference fuel. The parameters selected for this purpose were inlet temperature, it changed from 30 to 70 ºC. In addition to that, equivalence ratio (0.28 to 0.41), compression ratio (6 to 14), together with thermal and chemical effects of exhaust gas re-circulation was analyzed. A higher dilution due to EGR, decreases the peak temperature, and incomplete combustion can take place when the dilution is too strong, increasing the CO concentration.

The experimental work (Miguel Torres Garcı and Francisco Jose´ Jime´nez-Espadafor Aguilar 2009) conducted on single cylinder HCCI engine fueled by diesel. They have studied the effect of injection timing and EGR on start of combustion and emissions. Results were revealed that an increase of inlet temperature at constant EGR rate has a large effect on the start of combustion in HCCI mode, and advances it. Soot emissions are negligible in the HCCI combustion mode, and are independent of the EGR rate. Engines running in the HCCI combustion mode with EGR reach ultra-low NO_x emissions.

2.13 EMISSION STANDARDS

The emissions and fuel regulations for four-wheeled light-duty, heavy-duty vehicles are one of the motivation to improve diesel combustion. Table 2.2, shows emissions standards for light-duty vehicles propelled by diesel engines. Initially, Euro-I norms was implemented first in the year 2000 (DieselNet 2013) and named as a India 2000. Progressively the emission limits becomes more stringent and currently a mixed of BS-III and BS-IV are applicable across the nation. However, BS-IV is still applicable in National capital regions and eleven major cities of the country.

Table 2.2: Emission standards for Light-Duty vehicles (DieselNet 2013)

		Exhaust Emissions (g/km)				
	Standard	**CO**	**UHC**	**UHC+NO_x**	**NO_x**	**PM**
1992	-	17.3-32.6	2.7-3.7	-	-	-
1996	-	5.0-9.0	-	2.0-4.0	-	-
2000	India 2000	2.72-6.90	-	0.97-1.70	-	0.14-0.25
2005	Bharat Stage II	1.0-1.5	-	0.7-1.2	-	0.08-0.17
2010	Bharat Stage III	0.64 0.80 0.95	-	0.56 0.72 0.86	0.50 0.65 0.78	0.05 0.07 0.10
2010 (Only in selected regions)	Bharat Stage IV	0.50 0.63 0.74	-	0.30 0.39 0.46	0.25 0.33 0.39	0.025 0.04 0.06

Nevertheless, advanced combustion systems with increased injection pressure, EGR, improved piston-bowl geometries, and improved in-cylinder flows have resulted in substantial reductions in emissions, and research efforts continue. Despite these efforts, it appears highly unlikely that conventional jet-mixing controlled diesel combustion can meet future emission requirements without being fairly expensive after-treatment systems.

2.14 PHYSICOCHEMICAL PROPERTIES OF ALCOHOL AS FUEL

It has been established by various studies (Alan and Hansen 2005; Demirbas 2007) that alcohol along with diesel fuel produces satisfactory results with compression ignition engine. Alcohol fuels, methanol and ethanol have similar physicochemical properties and emission characteristics as that of petroleum fuels.

i) The role of fuel viscosity and lubricity is very vital for the injection system, especially, when the engine is operating on rotary distributor injection pumps or common rail injection system, however, it is not very serious when inline fuel pumps and unit injectors are sued. Addition of ethanol to diesel lowers the fuel viscosity and lubricity (Wrage and Goering 1980) that results into increase the leakage from the pump and injector and it leads to reduce the maximum fuel delivery and ultimately power output. Fuel viscosity also affects the atomization and spray characteristics (smaller Sauter mean droplet diameter) in the combustion chamber.

ii) Power output from the engine directly depends on the energy content of the fuel, since alcohol has low heating value as compared diesel fuel. Therefore, addition of each 5% ethanol decreases the energy content of E-diesel blends by approximately 2%. This indicates that to get the same power output from the E-diesel compared to diesel more alcohol is required by mass and volume.

iii) Solubility ethanol-diesel blends is mainly affected by two factors, temperature and water content (small trace amount of water also leads to separation of blends) of the blend. Therefore, a higher percentage of alcohol is not allowed for they could not mix with diesel fuel homogeneously. The aromatic content of the diesel fuel also affects solubility of ethanol in diesel (Gerdes 2001). Prevention of separation can be possible by two ways either by adding emulsifier (to suspend small droplet of ethanol in diesel) or by adding co-solvent that acts as a bridging agent through molecular compatibility (Letcher 1983).

iv) As per ASTM standard D975-02, the specified minimum cetane number for No. 2 diesel fuels is 40. Typically no. 2 diesel fuels have cetane number of 45-50. Since ethanol has higher octane no which is inversely related with cetane no therefore ethanol has extremely low CN (between 5 and 15). Higher cetane number leads to increase ignition delay period which provides more time for fuel vapourization before combustion starts. However, it increases initial burn rate (constant volume heat release) which is more efficient conversion process of heat to work. Sometimes poor ignitability may lead to rapid burning of vapourized alcohol and sever knock which may damage the engine.

v) Alcohol has higher stoichiometric fuel/air ratio (methanol -0.15393 and diesel - 0.06924) than diesel this is due to the partially oxidized state or it is an oxygenated fuel. Thus, blending alcohol into diesel fuel leads to leaner operation. Also, alcohol has negligible sulfur content and high H/C ratio.

vi) As alcohol (methanol and ethanol) has higher latent heat of vapourization as compared to diesel fuel, therefore E-diesel leads to longer ignition delay period. Also, when alcohol is injected into the intake manifold (fumigation) it cools down the air which is inducted and increased volumetric efficiency (reduces compression work done) and similarly this cooling effect reduces temperature of air-fuel after compression.

vii) Alcohol burning produces higher laminar flame propagation speed which completes the combustion process faster and thus improves engine thermal efficiency (Adelman H 1979; Wagner and Zarah 1979)

viii) As the high flammability of alcohol is one of the major concerns during handling and storage while implementing their application in existing infrastructure.

ix) In addition to lower calorific value, very high latent heat of vaporization, there is a serious health hazard associated with methanol which limits use of methanol over ethanol as an automotive fuel.

- Methanol is a highly toxic substance used as an alternative fuel, industrial solvent and automotive antifreeze. Moreover, it is extremely dangerous in case of human consumption, since it leads to sever visual disturbance.
- Methanol is more toxic than ethanol. The key reason behind this when methanol converted in to aldehyde, it leaves the by product of formaldehyde which is toxic.
- Robert and Nauss examined methanol toxicity on humans (Robert Kavet and Nauss 1990) specially due to automotive methanol vapour. They reported that consequences of acute exposures of methanol leads to un-compensated metabolic acidosis with superimposed toxicity to the visual system.

2.15 CONCLUSIONS FROM THE LITERATURE REVIEW

Based on the literature review above, it is clear that lot of emphasis is given on development of alternate diesel combustion technology and application of alternate fuel in diesel engine has increased significantly. Alcohol increasingly being used as a replacement to diesel fuel compared to other fuels.

After reviewing the literature few gaps are clearly visible and need extended research work. These are mentioned hereunder:

Partially premixed charge compression ignition study which was carried out in China (Tsang and Zhang 2010; Zhang Z.H. and Tsang 2011) and other parts of world is being done on multi-cylinder engines. Only few researcher (Abu-Qudais and Haddad 2000; Padala and Woo 2012) worked on single cylinder engine and especially in India (Ajav E. A. and Bachchan Singh 1998; Bhupendra Singh Chauhana and Naveen Kumar 2011; Hebbar and Anantha 2012) and reported premixed ethanol.

i) All the reported work does not includes optimization of injection timing for premixed fuel (ethanol) and DI (diesel) fuel.

ii) Reported studies (Ajav E. A. and Bachchan Singh 1998; Bhupendra Singh Chauhana and Naveen Kumar 2011; Hebbar and Anantha 2012) were used carburetor or single hole atomizing nozzle to create premixing of ethanol with air. So it was felt to study the effect of electronically controlled multipoint fuel injection system for ethanol which creates premixing of ethanol with the air.

iii) Hebbar et al. (Hebbar and Anantha 2012) is only the single work reported effect of EGR on premixed ethanol. However, it could not include combustion study and effects of injection timing. So there is a need to explore premixed ethanol on small capacity diesel engines which are used for agricultural and decentralized power generation purpose in rural India.

iv) Effect of intake air-preheating is not explored in great details.

v) Effect of combustion geometry (toroidal shape) on emissions is not reported.

vi) Effect of altering the injection spray pattern also not reported. So in this case instead of three holes nozzle is replaced with four holes nozzle.

2.16 OBJECTIVES OF THESIS

1. Modification of an existing single-cylinder four stroke direct injection diesel engine to operate on the partially premixed charge compression ignition combustion mode.
2. Study the effects of varying mixture compositions of premixed ethanol combustion at different engine loods.
3. Comparative performance, emission and combustion analysis of the developed PCCI engine fueled with ethanol and diesel in different proportions.
4. To achieve simultaneous reductions of NO_x and smoke with the help of premixed ethanol and EGR.

CHAPTER 3: EXPERIMENTAL SET-UP

The experimental set-up and methodology for the PCCI engine testing is described in this chapter. That includes PCCI test set up, development of various components to achieve required combustion.

3.1 DESCRIPTION OF EXPERIMENTAL TEST RIG

Experiential set-up consists of single cylinder diesel engine which is coupled with eddy current dynamometer as loading system, modified fuel injection system for diesel and ethanol. Set-up is enabled to provide with necessary instruments for in-cylinder pressure, fuel injection pressure, crank angle, airflow, fuel flow, temperatures and load measurements. Various sensors and instruments used for measurements are integrated with computerized data acquisition system. The test rig has stand-alone panel box consisting of air box, fuel tank, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and engine indicator. Rotameters are provided for cooling water and calorimeter water flow measurement. The thermal performance study of engine like brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, Mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio and heat balance. Labview based engine performance analysis software package “EnginesoftLV” is used for on line performance evaluation.

However, design and development of various components was one of the major challenges to achieve PCCI combustion that includes, ethanol injection system, in-line diesel injection pump, mechanical EGR and EGR cooler. Subsequent section provides details about each of them. Moreover, other trials also been carried out which are not used in the study, like modification of combustion geometry and air preheating system development.

3.2 EXPERIMENTAL TEST –RIG

The schematic of experimental engine setup is presented in Figure 3.1 and comments are labeled. Table 3.1 gives name of the components.

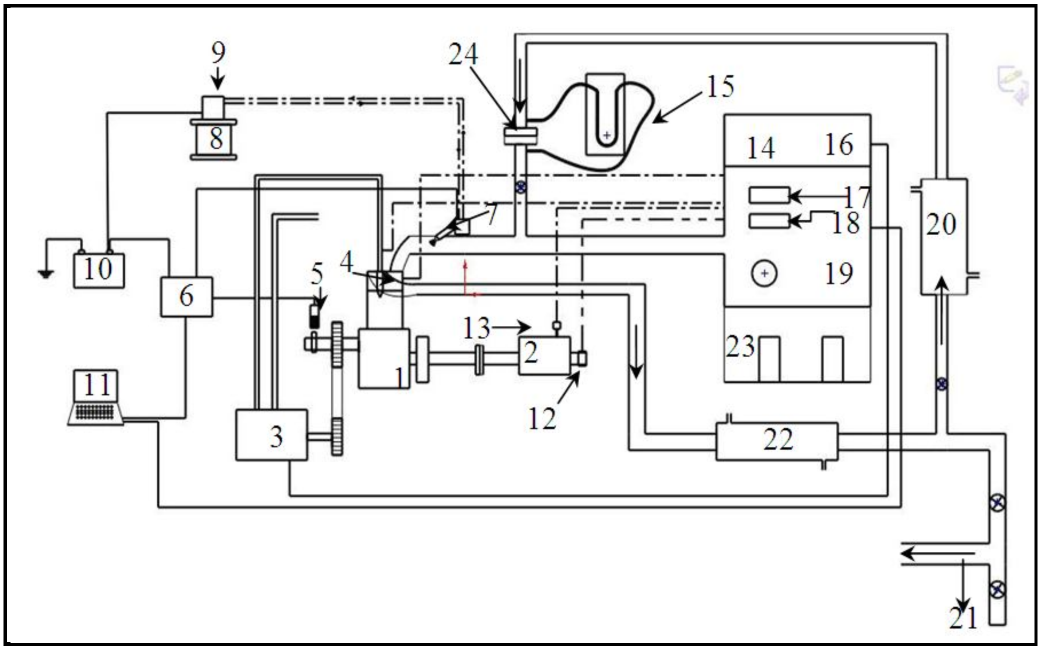


Figure 3.1: Schematic of experimental test rig.

Table 3.1: List of different components in the set-up.

1: Engine	9: Low pressure fuel pump	17: RPM indicator
2: Dynamometer	10: Battery	18: Load controller
3: Extracted fuel pump (inline)	11: Computer for DAQ	19: Air box
4: DI injector	12: Angle Encoder	20: EGR cooler
5: Proximity Switch	13: Load cell	21: Exhaust-to - analyzer
6: ECU (PFI)	14: NI (DAQ)	22: Calorimeter
7: Port fuel injector and common rail	15: U-tube manometer	23: Rotameters to measure flow of water
8: Ethanol fuel tank	16: Fuel tank (Diesel)	24: Orificemeter

Table 3.2: Technical specification of Experimental test rig

Engine	Make Kirloskar, Type 1 cyl., 4 stroke diesel, water cooled, power 3.5 kW at 1500 rpm, stroke 110 mm, bore 87.5 mm.
Dynamometer	Type eddy current, water cooled, with loading unit
Piezo sensor	Range 5000 PSI, with low noise cable
Crank angle sensor	Resolution 1 Deg, speed 5500 RPM with TDC pulse.
Data acquisition device	NI USB-6210, 16-bit, 250 kS/s.
Piezo powering unit	Make-cuadra, model AX-409.
Temperature sensor	Type RTD, PT100 and thermocouple, type K
Temperature transmitter	Type two wire, Input RTD PT100, range 0–100 Deg C, output 4–20 mA and type two wire, input thermocouple
Load indicator	Digital, range 0-50 Kg, supply 230VAC
Load sensor	Load cell, type strain gauge, range 0-50 Kg
Fuel flow transmitter	DP transmitter, range 0-500 mm WC
Air flow transmitter	Pressure transmitter, range (-) 250 mm WC
Software	“Enginesoft LV” engine performance analysis software
Rotameter	Engine cooling 40-400 LPH; calorimeter 25-250 LPH
Exhaust gas analyzer	AVL DIGAS-444
Smoke meter	AVL 437 c
In-line fuel pump	Robert bosch type VE diesel injection pump
Electric fuel pump	Denso
ECU	Ethanol injection timing
Proximity switch	Inductive type proximity switch
EGR cooler	Shell and tube type

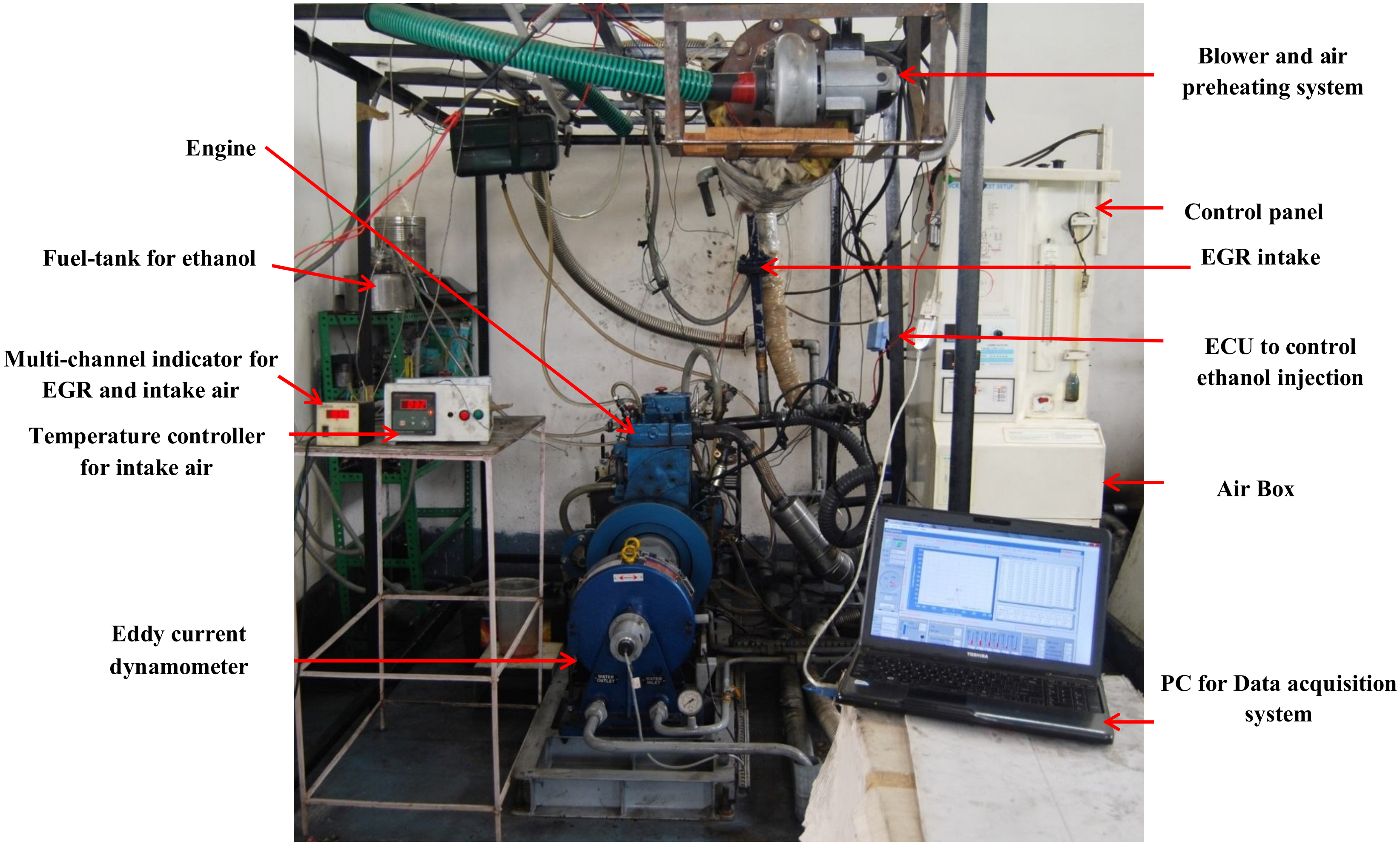


Figure 3.2: Actual experimental test rig

3.2.1 Single Cylinder Engine

The setup consists of small capacity Kirloskar single-cylinder diesel engine. This is widely used in rural India for irrigation and gen-set applications. The detailed technical specifications of the engine are given in the Table 3.3.

Table 3.3: Detailed engine specifications.

Model	**TV1**
Rated bake power (BHP/kW)	5/3.5
Rated speed (rpm)	1500
Bore - Stroke (mm)	87.5-110
Compression ratio	18:1
Cooling system	Water cooled
Lubrication system	Forced feed
Cubic capacity	0.661 L
Inlet valve open (degree)	4.5° bTDC
Inlet valve closed (degree)	35.5° aBDC
Exhaust valve open (degree)	35.5° bBDC
Exhaust valve closed (degree)	4.5° aTDC
Fuel injection timing for standard engine	23° bTDC
Fuel injection pressure (bar)	200
Lubricating oil pump	Gear type
Lubricating oil pump delivery	6.50 lit/min
Sump capacity	2.70 litres
Lubricating oil sump	1.5% normally exceed of fuel
Connecting rod length	234 mm
Overall dimensions	617L × 504W × 877H
Weight	160 kgs

3.2.2 Engine Load Measurement

To perform PCCI study at various engine loads therefore a reliable and accurate system is needed for engine loading and measurement. Engine loading and measurement system comprises of eddy current dynamometer load cell of strain gauge type, which is done by using SAJ AG Series eddy current dynamometer. The working principle is based on Fleming's law of right hand rule. Fig. 3.3 shows schematics of eddy current dynamometer.

The dynamometer unit consits of a stator comprises of number of electromagnte and a rotordisc which is fixed on the shaft, moreover in turn it is coupled with output shaft of the engine. The shaft is supported on deep groover ball bearing which is most precise method of mouting.

On the outer periphery of the casing, there are two field coils connected in series. To prodcue magnetic field in the casing across the air gap at either side of the rotar a direct current is supplied to these coils. Thus, when the rotor starting turning under the influence of this magnetic field it prodcues breaking effect between the rotor and casing.The torque exerted on the casing is measured by by a strain gauge load cell incorporated in the restraining linkage between the casing and dynamometer bed plate.

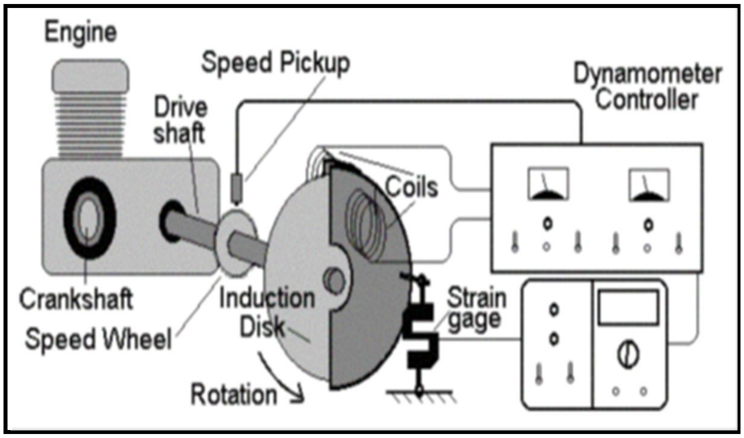


Figure 3.3: Schematic of eddy current dynamometer.

To prevent overheating of dynamometer cooling water is circulated using centrifugal pump through the passages provided in the casing. However, it is required to maintain the minimum water inlet pressure, as mentioned in the specification. Loading of the Eddy current Dynamometer is done by regulating the current in electro magnets.

Table 3.4: Technical specifications of eddy current dynamometer

Model	**AG-10**
Make	SAJ Test Plant Pvt. Ltd
End flanges both side	Carbon shaft model 1260 type A
Water inlet	1.6 bar
Minimum kPa	160
Pressure	23lbf/in$_2$
Air gap mm	0.77/0.63
Torque Nm	11.5
Hot coil voltage max	60
Continuous current amps	5
Cold resistance ohms	9.8
Speed max.	10000 rpm
Load	3.5 kg
Bolt size	M12×1.75
Weight	130 kg
Arm Length mm	185

A load cell is a transducer that is used to convert a force into electrical signal. Fig. 3.5 shows load cell (strain gauge type) used in the test rig, and Table 3.5 gives detailed technical specifications.

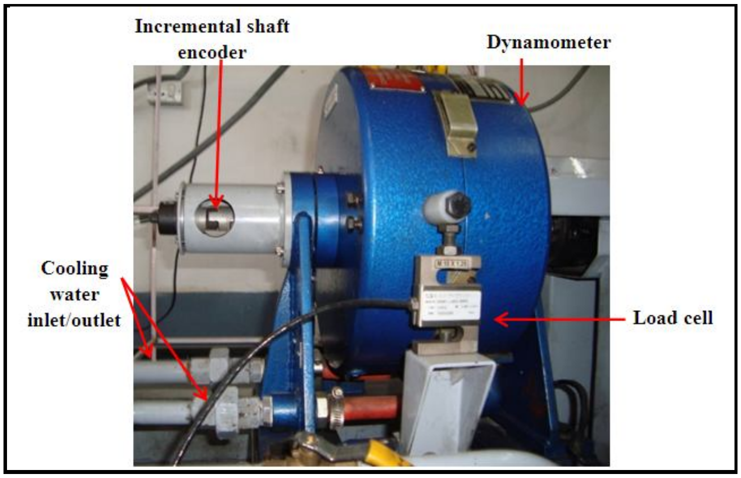


Figure 3.4: Eddy current dynamometer, load cell and angle encoder.

Whenever electrical conductor is subjected to stress, induces strain and this leads to change the resistance of metallic conductor. Therefore, a strain gauge is usually used as sensing part of the load cell, as the torque on the dynamometer casing exerts force on load cell, the foil is deformed it results into change in electrical resistance.

Therefore provision is made to measure this change in resistance by using a Wheatstone bridge. Usually fours train gauges are used in Wheatstone bridge configuration in a typical load cell. However, output electrical signal is very small in the order of millivolts thus suitable instrumental amplifier is used to amplify it before sending to the measuring system.

Figure 3.5: Load cell type strain gauge, make sensortronics

Table 3.5: Technical specifications of the load cell used

Make	**Sensortronics**
Model	60001
Type	S – Beam universal
Capacity	0 – 50 kg
Mounting thread	M10 × 1.25 mm
Full scale output (mV/V)	3.00
Tolerance on output (FSO)	+/-0.25%
Zero balance (FSO)	+/-0.1mV/V
Non-linearity (FSO)	<+/-0.025%
Hysteresis (FSO)	<+/-0.020%
Non-repeatability	<+/-0.010%

Creep (FSO) in 30 min	<+/-0.020%
Operating temperature range	-20 0C to +70 0C
Rated excitation	10V AC/DC
Maximum excitation	15V AC/DC
Bridge resistance	350 Ohms (Nominal)
Insulation resistance	>1000 Meg ohm @ 50VDC
Span / 0C (of load)	+/-0.001%
Zero / 0C (of FSO)	+/-0.002%
Combined error (FSO)	<+/-0.025%
Safe overload (FSO)	150%
Ultimate overload (FSO)	300%
Protection class	IP 67
Overall dimensions	51 L x 20 W x 76 H mm
Weight	380 gm

Engine Loading System

Fig. 3.6 shows loading unit, Apex make model AX-155. The loading unit consists of dimmerstat, step down transformer and rectifier. Dimmersstat is the common name of variable auto transformer or variac. Variable autotransformer is basically comprises of single layer winding on toroidal core of high grade steel and smoothly running carbon brush is sliding on the winding which is attached to the handle or knob. So, when a part of the winding coils are exposed and the secondary connection is made through a sliding brush, a continuously variable turns ratio will be obtained, which allows a smooth control of voltage.

Figure 3.6: Engine loading unit

Fig. 3.7 shows the circuit diagram of dimmerstat. The voltage can be smoothly varied from maximum voltage to minimum of 0 V to a between turns because the brush is having a relatively high resistance as compared with a metal contact and then the actual output voltage. To avoid any possible damage to the rectifier circuit a step down transformer is connected with the output of dimmerstat and then it is fed to rectifier. Thus, alternating current (AC) could be converted into direct current (DC) and this process is known as rectification and the electrical device by which it is done is known as rectifier.

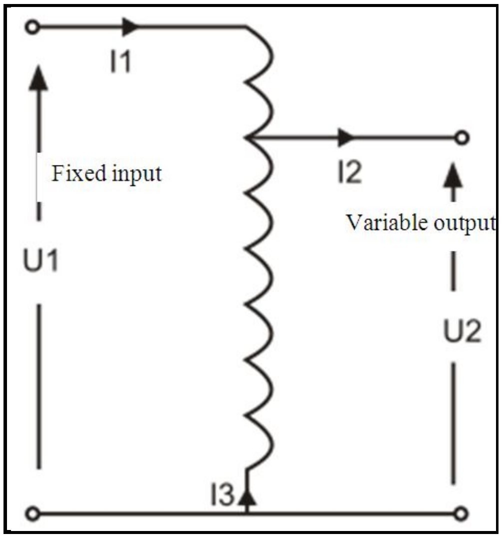


Figure 3.7: Circuit diagram of dimmerstat

3.2.3 Data Acquisition System

To study the engine performance such as brake power, BMEP, brake thermal efficiency, volumetric efficiency, and specific fuel consumption. The data acquisition system which is used in the experimental work is from National instruments (Bus-Powered M Series multifunction DAQ for USB 16-Bit, up to 250 kS/s, up to 32 Analog Inputs, Isolation). It has 16 analog inputs, 8 digital I/O lines and four programmable input ranges. It provides flexibility to use portable computer such as laptop with a USB port and connect with the system. As discussed above Labview based Engine performance analysis software package “EnginesoftLV” for on-line performance evaluation. The software scans the signals received from all the channels as shown in Fig. 3.8 and uses for display and analysis. The components of the data acquisition system are pressure Pick up, charge amplifier, TDC position sensor, A/D card and a portable computer.

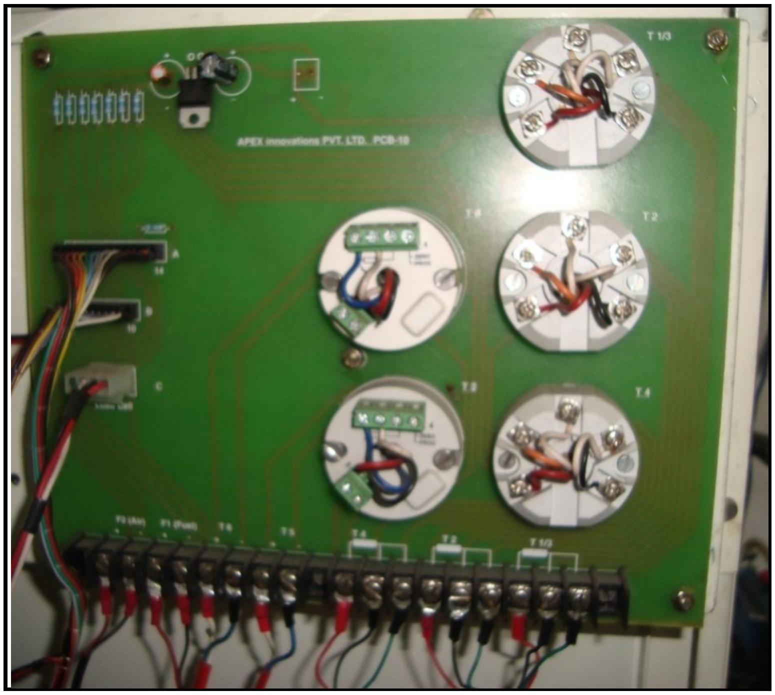


Figure 3.8: PCB and Sensor interface

The in-cylinder pressure varies with respect crank angle or piston displacement, thus in-cylinder pressure and crank angle is logged into computer require high speed data acquisition system.

Further, to get the valuable information from experimentation, the data is to be analyzed after verifying whether the data is properly recorded or not. Therefore, processing data is an important step in the experimental investigation. The large data which is collected during the experimentation has to be systematically processed as mentioned below. The cylinder pressure with respect to crank angle generally has large cycle to cycle variations and hence one such cycle data cannot be used to represent the particular operating condition. Due to hardware limitations only 50 cycles' data were acquired and analyzed during the entire experimentation. Plate A1, shows, user interface of "EnginesoftLV".

To measure temperatures six two types of temperature sensors are used. First is RTD type, they are used for water temperature measurement. RTD sensors have platinum resistances which changes with temperature. This change in resistance is calibrated to temperature variation. This is connected to transmitter and proportional to signal scanned by PC through National Instruments unit. Second sensors for exhaust gas and intake temperature of mixture measurements are thermocouples. Principal of operation is based on Seebeck effect as the junction of dissimilar metal when subjected to elevated temperature this generates millivolt which is proportional to temperature.

The air flow is measured by installing pressure (vacuum transmitter) on the air box with orificemeter. The software solves orifice equation for flow measurement where all other parameters are constant except the transmitter signal which is proportional vacuum in the air box.

3.2.4 Gas Analyzer and Smoke Meter

Five gas analyzer (AVL DIGAS-444) as Shown in Fig 3.9 is used to measure UHC, CO, CO_2, O_2 and NO_x emissions. UHC, CO and CO_2 emissions are measured by using non dispersive infrared method (NDIR) and NO_x and O_2 is measured by electrochemical method.

Figure 3.9: AVL DIGAS-444 analyzer.

Fig. 3.10 shows architecture of NDIR method. A single wave band within the infrared spectrum is selected for each gas to be measured. When the sample cell is filled with sample gas, the IR detector measures the resultant reduction of transmitted IR energy within the wave band of each gas. However, measurement of oxygen and NO_x electrochemical sensor is used, when chemical comes with contact of gas being measured its property gets changes. Measurment resolution is given in Table 3.6.

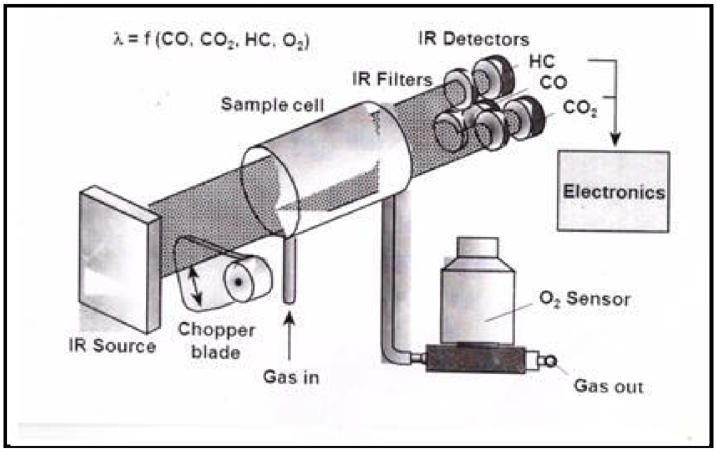


Figure 3.10: Gas analyzer architecture-adapted from AVL-India (AVL).

Table 3.6: Measurement resolution of Exhaust gases.

UHC	1ppm
CO	0.01%
CO_2	0.01%
NO_x	1ppm
O_2	0.01%

When the probe is inserted into the exhaust pipe of the engine, the exhaust gas is passed through a metal mesh screen to filter the soot and dust particles after which it is allowed to pass through a fine fiber element which filters the entire gas for any foreign particles and also cold traps are provided to prevent moisture from entering the exhaust gas analyzer. After this, the cleaned and cooled gas sample enters the direct sensor measurement through filter arrangement and the readings are displayed on the screen. The emission measurements are done on dry basis. Exhaust gas analyzer was calibrated by AVL Gurgaon Ltd.

Smoke meter is used to measure smoke opacity as shown in Fig 3.11 (AVL-437c). It measures the opacity of polluted air, in turn; this also has indication of particulate matter (PM) in diesel exhaust gases. The opacity is defined as the extinction of light between light source (incandescent bulb) and receiver (photocell). The exhaust gas to be measured is fed into a chamber with non-reflective inner surfaces by using flexible hose. The effective length of the light absorption track is determined by taking into consideration possible influences of devices used to protect the light source and the photocell.

Figure 3.11: AVL 437 c smoke meter.

3.2.5 Fuel Consumption Metering Devices

The fuel metering device differential pressure transmitter is used for the quantity of diesel being injected, as given in Fig 3.12. This is manufactured by YOKOGAWA. This differential pressure transmitter is uses a silicon resonant sensor formed from mono crystal silicon, an ideal material free from hysteresis in pressure or temperature changes. The fuel metering is based on principle of hydrostatic law. The fuel transmitter measures hydrostatic head in the fuel measuring unit. The difference between two signals in the interval of one minute is converted in to fuel consumption. Technical specifications are given in Table 3.7.

Table 3.7: Technical specifications of differential pressure transmitter

Make	**Yokogawa**
Model	EJA110A-DMS5A-92NN
Output signal communication (Linear)	4 – 20 mA DC with digital
Measurement span 1 to	100kPa (100 to 10000 mm H_2O)
Calibration range	0 – 200, 0 – 500 mmH_2O
Wetted parts material	Body – SCS14A, Capsule – SUS316L
Process connections	Without process connector (1/4BSP body connection)
Bolts and nuts material	SCM 435
Installation	Horizontal impulse piping left side high pressure
Electrical connection	½ NPT female
Cover 'O' rings	Buna-N
Supply	10 to 24 VDC
Process temperature limit	-40 to 120 °C
Housing	Weather proof
Weight	3.9 Kg

Figure 3.12: Differential pressure transmitter.

3.2.6 PRESSURE MEASUREMENTS

Cylinder pressure is usually measured with piezoelectric sensor or commonly known as a piezo sensor. This type of sensors consists of a quartz crystal, diaphragm (invar) and stainless steel casing. Hence the sensing part is diaphragm which is exposed to the combustion chamber. In this study the Piezo sensors are used for cylinder pressure and fuel injection pressure is manufactured by PCB Piezotronics pressure division. Fig. 3.13 shows sensor which is used.

Fig. 3.14 shows mounting position of piezo sensors. To measure combustion or cylinder pressure piezo sensor is mounted on the top of the cylinder head and to measure the fuel injection pressure it is mounted on fuel high pressure fuel line. Working of Piezo sensor involves the fundamental principal of piezoelectric effect. As per this any crystal can produce definite charge output under the application of mechanical load. However, the charge output is very small, therefore a charge amplifier is employed to produce an output voltage proportion to this charge.

Thus, the increase in in-cylinder pressure compresses the crystal and it generates electric charge which is proportional to the pressure. The dynamic voltage signal from piezoelectric sensor is scanned with respect to crank angle signals received from rotary encoder fitted on the engine crankshaft. Table 3.8 gives detail technical spefications.

Figure 3.13: Piezo sensor (PCB) fuel injection and in-cylinder pressure.

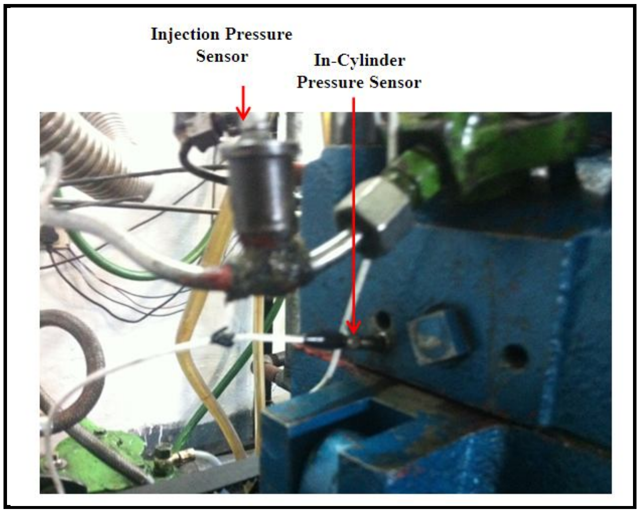


Figure 3.14: PCB sensors mounting in the cylinder head and fuel line.

Table 3.8: Technical Specificaition of pressure sensor

Make	PCB Piezotronics
Model	HSM111A22
Measurement Range (for ±5Voutput)	34,475 kPa
Useful Over range (for ±10V output)	68,950 kPa
Sensitivity (±0.1mV/psi)	0.145 mV/kPa
Maximum Pressure	103.425 kPa
Resolution	0.69 kPa
Resonant Frequency	≥400kHz
Rise Time (Reflected)	≤1.5µsec
Low frequency response	.001Hz
Non-linearity	≤2% FS
Acceleration sensitivity	<0.0014 kPa

Make	**PCB Piezotronics**
Operating Temperature Range	-73 to +135 °C
Temperature coefficient of sensitivity	≤0.36%/°C
Maximum Vibration	19,624 m/s^2 pk
Maximum shock	196,140 m/s^2 pk
Output Polarity (Positive pressure)	Positive
Discharge time constant (at room temp)	≥500 sec
Excitation voltage	20 to 30 V DC
Constant current excitation	2 to 20 mA
Output impedance	≤100 ohm
Output bias voltage	8 to 14 V DC
Sensing geometry	Compression
Sensing element	Quartz
Housing material	17-4 stainless steel
Diaphragm	Invar
Sealing	Welded hermetic
Electrical conductor	10-32 coaxial jack
Weight	6 gm

3.2.7 Angle Encoder

Fig 3.15 shows incremental shaft encoder used for the measurement of rotation speeds. Electrical power input requires is minimum 5 V DC to maximum 30 V DC. An incremental encoder works on the principle of generating specified amount of pulses in one rotation of the encoder. Output from the encoder can be single line or two lines of pulses which are offset in order to determine rotation. Phasing between the two signals is called quadrature. An incremental encoder consists of a spindle, PCB and cover. Crank angle encoder sends two different pulse trains to the computer. Therefore, it consists of one for every rotation (marker pulse) and one for every crank angle. These signals are used for plotting P-θ diagram. The PCB contains a sensor array that creates just two primary signals for the purpose of position and speed. Table 3.9 gives technical specifications of angle encoder.

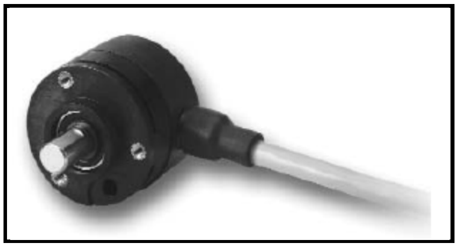


Figure 3.15: Angle encoder

Table 3.9: Technical specifications

Make	Kubeler
Model	8.3700.1321.0360
Supply voltage	5-30 VDC
Output Push pull	(AA,BB,OO)
PPR	360
Outlet Cable	type axial

3.3 DEVELOPMENT OF TEST-RIG TO ACHIEVE PARTIAL HCCI COMBUSTION

The main challenge to carry out combustion study on PCCI test rig requires so many adaptations and, it took lot of time. Two major modifications need be done in the fueling system, as the one fuel being injected into the manifold and other directly into the cylinder. Carburetor does not able to provide precise and varying quantity of fuel at different loads under un-throttled condition. Therefore advanced multipoint fuel injection system is used. So, following design and development have been carried out to achieve partial HCCI combustion.

- Manifold fuel injection system
- Separate DI injection mechanism
- Development of Mechanical EGR system
- Design and fabrication of EGR cooler
- Modification of piston geometry
- Development of air preheating system

3.3.1 Development of Manifold Fuel Injection System

The preparing homogenous mixture of air and fuel is very critical for achieving PCCI combustion to deliver high efficiency and reduce NO_x and smoke emissions. To prepare homogenous mixture, manifold injection technique is chosen for this experimental setup.

The fuel injection system comprises of fuel pump, fuel accumulator, injector, injector controller as a programmable ECU and fuel tank. The schematic diagram of fuel injection system is shown in figure 3.16.

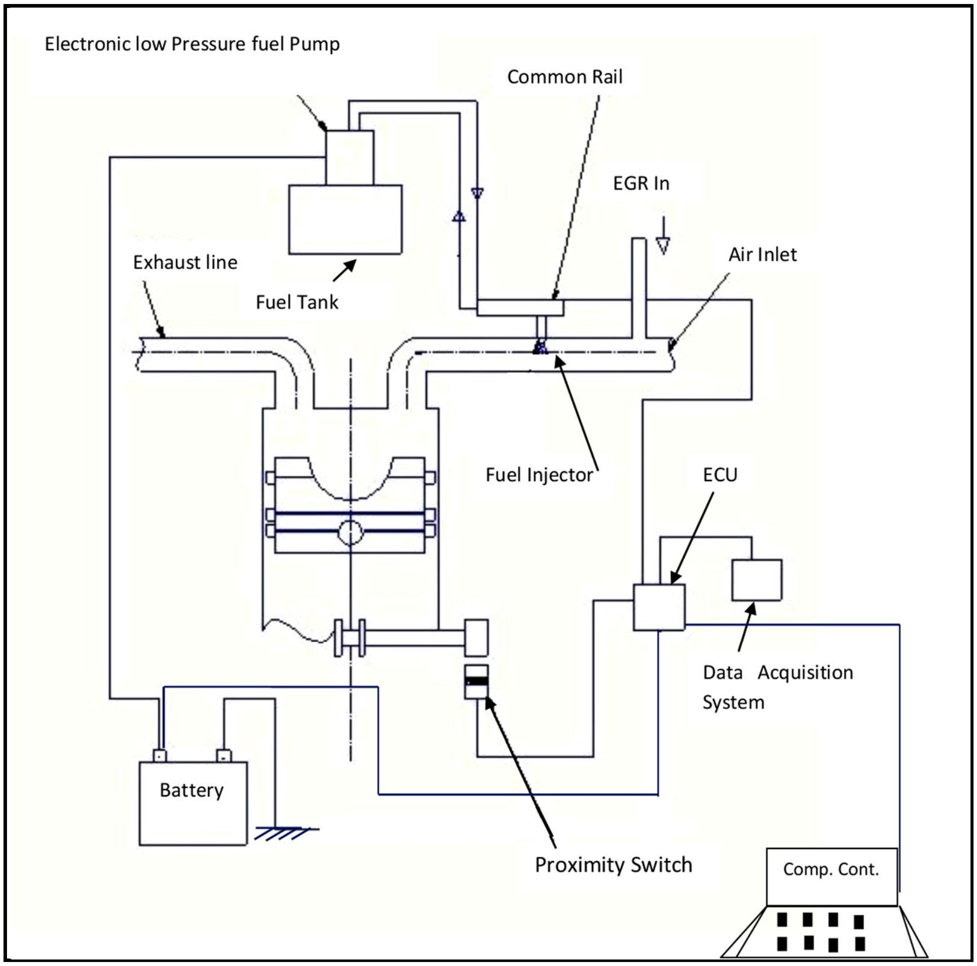


Figure 3.16: Manifold fuel injection system.

3.3.1.1 Fuel Pump

An electric fuel pump was used to supply the fuel from fuel tank to the accumulator. The fuel pump operates on 12 V DC, using electrical power supply through a battery. This pump make is Denso (model no-1500M844M1) mounted on specially designed stainless steel sheet fuel tank, shown in Fig 3.17. As the pump can produce an injection pressure of about 3 bar, and a fuel filter is also attached at intake side, could be seen in Fig. 3.18 to prevent from entering any particulates in the fuel line.

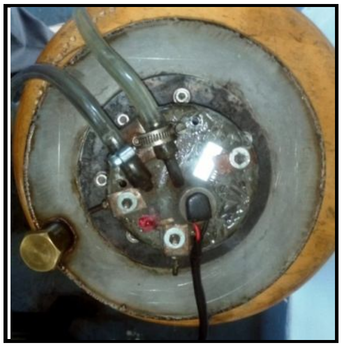


Figure 3.17: Fuel pumps mounting on tank.


Figure 3.18: Fuel pump and fuel filter assembly

3.3.1.2 Fuel Accumulator and Fuel Injector

Fuel accumulator used for ethanol injection is the one which is used in automotive three cylinder engines. So, there is provision to mount three injectors, however in this application only one injector is needed. Thus, rail is modified and two holes were blocked which is visible in Fig. 3.19. The rail is connected with two fuel lines in which one is used to supply the fuel to the accumulator and another return line that fed back surplus fuel to the tank.

Ethanol was injected into the air intake manifold at about 30 cm upstream from the air intake valve by using solenoid operated fuel injector as shown in Fig. 3.20. Fig. 3.21 shows a solenoid operated injector it is mounted with fuel accumulator, and also showing fuel spray pattern. The working of this injector is based on the solenoid actuated mechanism, when electrical current pulses is allowed to pass through the injector coil, it set up a magnetic field that lead to draw the plunger and valve away from the nozzle seat. Hence the fuel which is held under pressure in the accumulator can now pass through the injector into the intake air. Moreover, spray also breaks the liquid fuel into small droplets and elevated temperature at intake results into vapourization of ethanol.

Ethanol injection was going through so many iterations to optimize injection timing, mounting position and orientation. Thus, it was concluded that the ethanol needs to inject just a few degrees after when intake valve opens (25 °CA aTDC) during intake stroke. The orientation of fuel injector is made such that so as to it avoid manifold's wall wetting, therefore fuel injector is inclined at 45° with the centerline of the manifold.

Figure 3.19: Modified Fuel Accumulator.

Figure 3.20: Solenoid operated fuel injector.

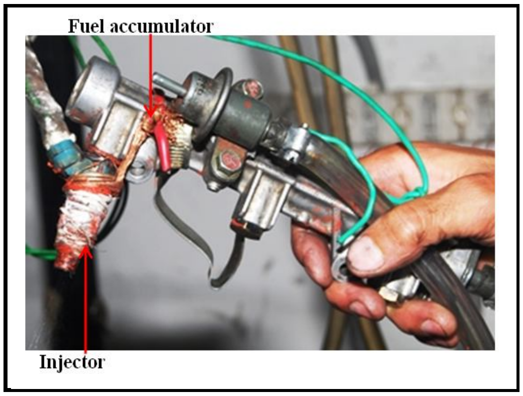


Figure 3.21: Spray pattern of ethanol via solenoid operated injector.

Inductive Proximity Sensor

As the fuel injection timing has significant role to prepare air-fuel mixture. An inductive type position sensor is used as a fuel injection sensor in the set-up. Fig. 3.22

shows the inductive proximity sensor used to detect metallic targets. It consists of coil, oscillator, detector and the output circuit. The inductive coil generates high frequency magnetic field in front of the face and when the metallic target crosses this magnetic field it absorb some of the energy. Thus, oscillator field is got affected and it is detected by detector. If the oscillation amplitude reaches a certain threshold value the output switches.

This proximity sensor operates on voltages 10-30 V DC. This sensor gives output current of 250 mA, however output voltage is same as input. Since in case of four stroke engine there is single power stroke in each revolution of the crankshaft. Therefore only one pulse is required for fuel injection, so detector is mounted on the camshaft. Since camshaft rotates at half speed than the crankshaft, thus proximity sensor trigger once in each two revolution of crankshaft, Fig. 3.23 shows the mounting position of proximity sensor. The technical specifications of inductive proximity switch (NPN-NO. M-18, NFM) is given in the Table 3.10.

Table 3.10: Technical specifications of proximity sensor

Operating Voltage	5-40 VDC
Sensing Distance	8.0 mm
Max. Laod Current	300 mAmp.

Figure 3.22: NPN type inductive proximity sensor.

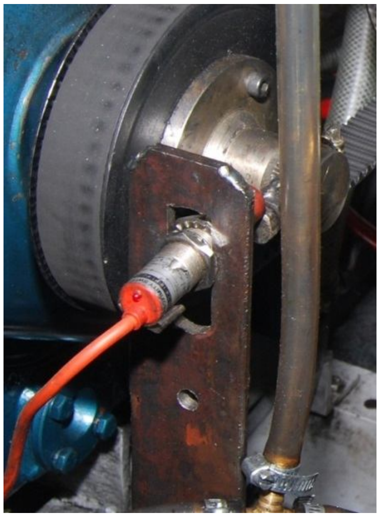


Figure 3.23: Proximity sensor mounting.

PROGRAMMABLE ECU

Fig. 3.24 shows block diagram to illustrate working of programmable ECU. As the proximity sensor and engine rpm sensor are the two input variable. So, based on these inputs, and the logic it can change the output pulse for injector and delay. So the quantity and timing of fuel injection could be controlled very precisely. Injection period or quantity of fuel injected can be changed by changing the pulse duration given to solenoid operated injector. This can be more than two milliseconds and fraction of milliseconds pulse is not converted into signal. Fig. 3.25 showing detailed wiring diagram for programmable ECU.

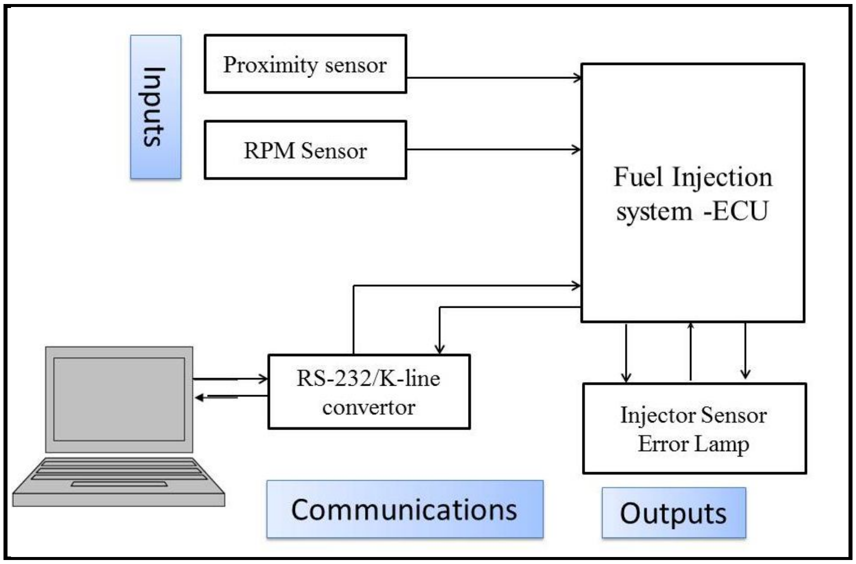


Figure 3.24: Block diagram showing working of ECU.

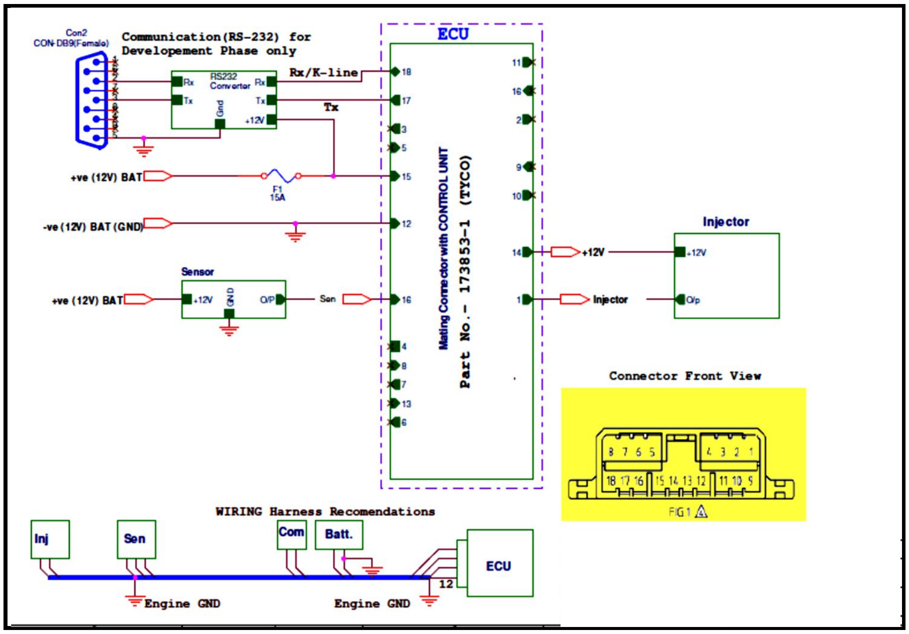


Figure 3.25: Circuit diagram of fuel injection system.

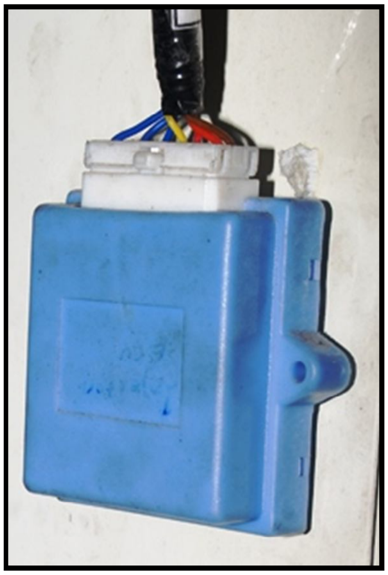


Figure 3.26: ECU for Ethanol injection system.

Therefore, pulse duration selected for the study is 2, 3 and 4 milliseconds (ms). While taking measurements of quantity of ethanol being injected corresponds to 2 ms pulse duration is 4ml/min at 1500 rpm. Therefore, quantity of fuel injected needs to be correlated with pulse duration and measurements have been done. Fig. 3.26 and 3.27 shows ECU and measurement of ethanol fuel with pulse duration. Table 3.11 gives technical specifications of ECU.

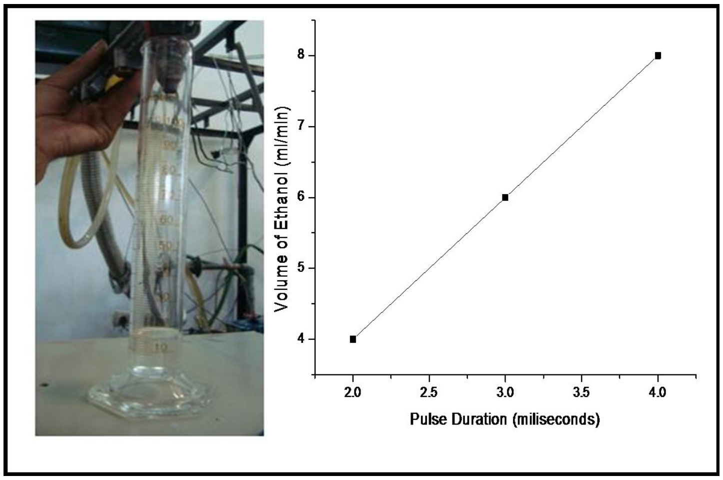


Figure 3.27: Measurement of ethanol fuel volume with pulse duration.

As fuel accumulator fitted with injector are fixed on the intake manifold. Fig. 3.28 showed deatials about intake manifold which is modified to accommodate ethanol injector, EGR intake and CO_2 measurement system to calculate percentage of EGR.

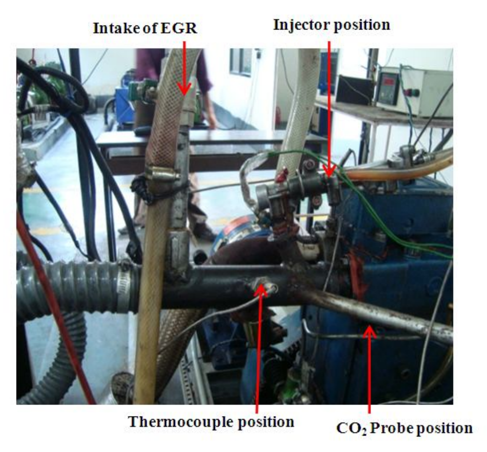


Figure 3.28: EGR intake, ethanol injection and CO_2 measurement.

Table 3.11: Detailed technical specifications of ECU

Supply Voltage	11 – 16 V DC (type.: 13.5V DC)
Reference Voltage	5V ± 0.020V DC

Operating Temperature		-30 to 75 °C	
Storage Temperature		-40 to 85 °C	
Tightening Torque		10±2Nm (Mounting screws)	
INPUT		OUTPUT	
TPS-ADC	0-5V DC	Motorized	12V @ 6A max
Feedback- ADC Injector	0 – 5V DC	Error lamp	12V @ 2W to 10W max
Battery sense (inside ECU)	0- 12V DC		
RPM (Digital)	10Hz – 2000Hz		
Proximity sensor	0- 12V DC		

3.3.2 Development of Fuel Injection Mechanism

As per literature, it was realized that injection timing has profound effect on combustion and performance of a premixed mode of combustion. So, it was the motivation to develop a fuel injection mechanism which provides flexible injection timing, without changing the quantity of fuel being injected. While optimizing so many iterations were examined by advancing and retarding the same. However, it was observed that retardation of injection timing (from 18 °CA bTDC to 9 °CA bTDC) leads to significant increase in soot emissions. Thus, entire study was conducted at constant injection timing of 12 °CA bTDC. This was decided based on the highest performance and emissions of modified engine associated with pure diesel combustion.

Partially premixed charge compression ignition (PCCI) engine test set requires dual fueling mechanism along with precise control on the timing and quantity of fuel being injected. The fuel injection system equipped with engine does not provide the flexibility to precise control over the above mentioned parameters. So, an entirely new concept brings in is to use an in-line fuel injection pump (used in Ford tractor engine), since this pump gives three advantages- one fuel injection timing does not directly depends on the mechanical component's position (piston position) and four hole injector could be used which provides better atomization. Throttle position will be fixed as per the requirement of premixing ethanol and load on the engine. Fig. 3.29 shows schematics of fuel injection system developed to perform PCCI combustion study.

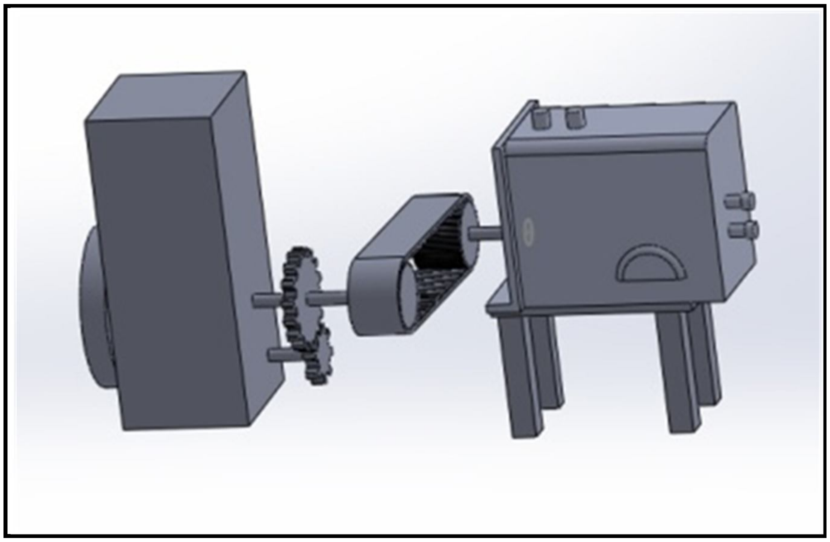


Figure 3.29: Schematics of In-line fuel injection pump with the engine.

Fig. 3.30 shows the actual arrangement of different components of DI fuel injection system. The system is provided with a body containing at least one in-line cylinder and piston pumping unit driven by a shaft with eccentrics connected through the respective intake and delivery valves to the inlet and outlet means for the fuel. For injection timing adjustment, first toothed belt is removed from two sprockets, and then flywheel is rotated either anticlockwise or clockwise for retarding or advancing timing respectively. After setting up injection timing on flywheel, move the pointer on sprocket so that it becomes in line with slotted cut on vertical foundation of in-line pump, at this position plunger is at high lift of camshaft means TDC position and then toothed belt is inserted above two sprocket in such that two sprocket does not move from its position.

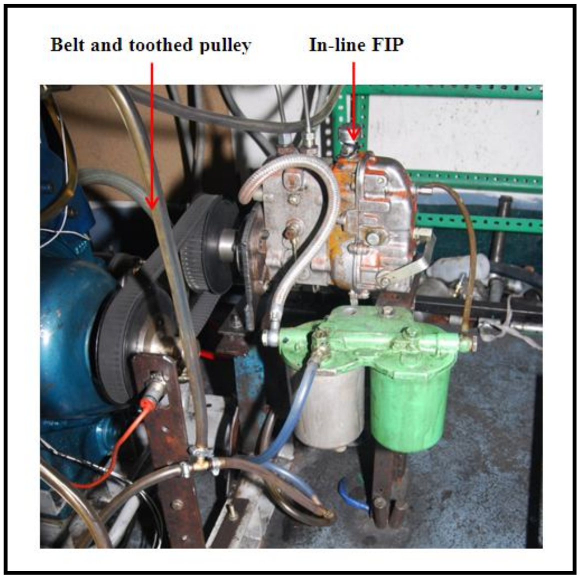


Figure 3.30: Pump mechanism for diesel.

3.3.3 Mechanical EGR System Development

In the EGR system, a part of the exhaust gas was re-circulated to the combustion chamber after mixing with fresh air. Initially, initially it was attempted with electronic EGR valve, since that was not customized to existing engine, after lot of efforts, it was not showed expected outcome. Therefore, finally mechanical EGR system was designed, the flow of EGR into the intake requires depression in the pressure that was done by using small nozzle in the flow, and the diameter of the nozzle reduces to 12.5 mm and 20 cm length. Schematic is shown in Fig. 3.31 illustrate the details about EGR system.

Only exhaust stroke is putting out all the gases out from the engine, it leads to pulsation in the flow that can be reduced by using exhaust gas damper. The corrugated pipe is used from the outlet of the EGR cooler to intake in the orifice meter pipe. NO_x reduction accomplished by both way hot and cold EGR, literature suggest, cold EGR, effectively control NO_x emissions, without adversely affecting BSFC. EGR cooler was designed from the scratch to final fabrication; it reduces exhaust gas temperature from 220 °C to 120 °C.

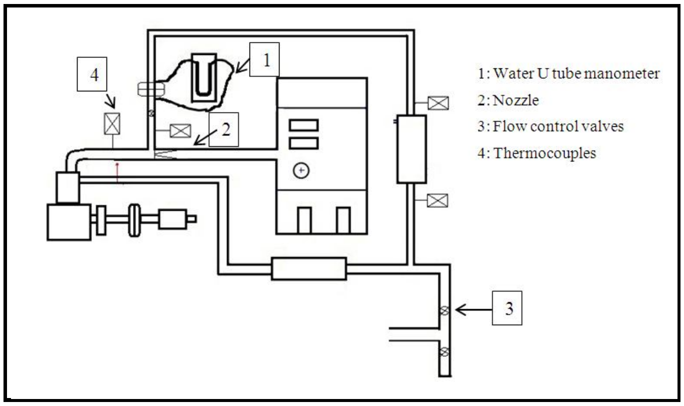


Figure 3.31: Schematic of mechanical EGR system.

3.3.4 Modification of Combustion Geometry

We found that the UHC emission increases, drastically, at elevated temperature. During the experimentation and to improve the premixing, it requires higher tangential velocity (swirl ratio) of flow inside the cylinder. Keeping constant compression ratio, combustion geometry is modified into toroidal shape. This is achieved by measuring the bowl volume very precisely and designing the geometry based on same the same volume (39 cc). This design helped in improving UHC emissions only at elevated temperature; however those findings are not reported in the thesis. However, details of their development are presented in Fig. 3.32 to Fig. 3.34.

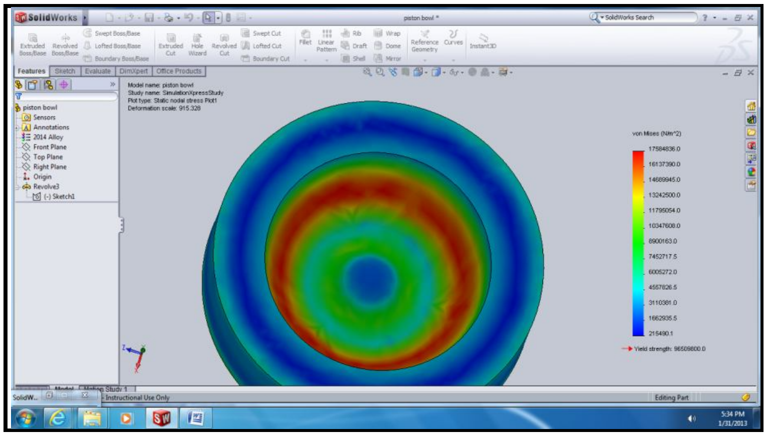

Model name: piston bowl
Study name: SimulationXpressStudy
Plot type: Static nodal stress Plot1
Deformation scale: 915.328

Figure 3.32: Stress analysis on piston geometry.

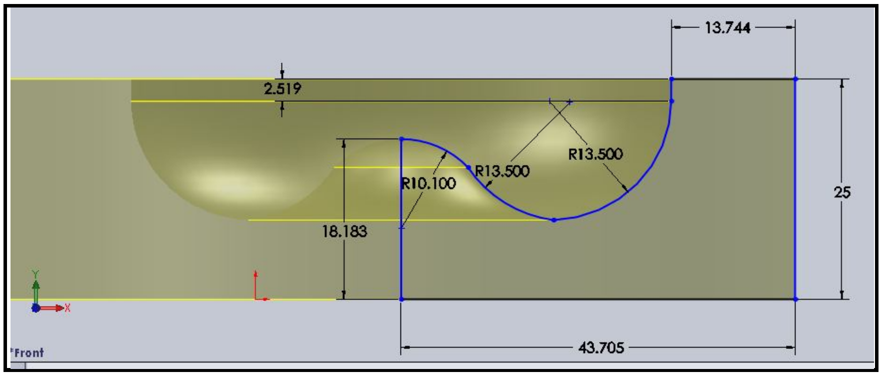


Figure 3.33: Dimensions toroidal piston cavity.

Figure 3.34: Actual and modified piston bowl design.

3.3.5 Air Preheating System

Literature review suggested to examine premixed ethanol at elevated intake change temperature, so air preheating system is developed consists of centrifugal blower, band heaters and temperature controller. It has been observed (Maurya and Agarwal 2011) by air preheating HCCI combustion could result when the intake air temperature is raised up to 140 °C to 150 °C. The results presented in this study do not use the same, since it showed negative effect on performance and emissions. Initially, the simple emulsion rod heaters were used, which takes away around substantial a period of 2 to 3 months' time, ultimately it was one of the failure!

- Blower to create boost pressure
- Heat exchange to preheat intake air
- Temperature controller with the help of solid state relay.

Figure 3.35: Electrically air preheating system.

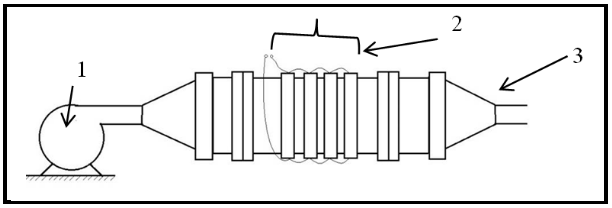


Figure 3.36: Heat exchanger for air preheating.

1: Centrifugal blower 2: Band heaters - connected in series (4 Nos.) 3: 4” Diameter pipe

3.4 DATA ANALYSIS

In this section the processing of the data recorded from the engine is documented. To understand the combustion parameters, net heat release rate, and mass fraction burnt, premixed combustion duration and ignition delay is presented. The definitions and calculations of EGR rate and performance parameter associated with premixed combustion are also discussed in this section.

3.4.1 Theoretical Analysis of In-Cylinder Pressure

The cylinder pressure variation with crank angle degree during the compression and expansion is vital information to analyze the combustion characteristics, fuel burning rate emission formation. Since cylinder pressure changes with cylinder volume change, combustion, heat transfer to the chamber walls, flow into and out of crevice regions and leakage. The first two of these effects are the largest.

To get accurate cylinder pressure following requirement should be met

- The correct reference pressure is used to convert the measured pressure signals to absolute pressure
- The pressure versus the volume phasing is accurate within 0.2°
- The clearance volume should be estimated with sufficient accuracy
- Temperature variation of the transducer which affects the calibration
- The compression and expansion process are well fitted by a polytrophic relation

$$pV^n = constant \text{ -- Eqn. (3.1)}$$

The exponent for the compression and expansion processes is 1.3 (±0.05) for conventional fuels.

3.4.2 Net and Cumulative Heat Release Rate

Generally, the heat release rate calculations are based on the first law of thermodynamics, applied to a control volume in which there is no mass transfer. It's calculated as an energy balance of the combustion chamber.

$$\delta \mathrm{Q}_{hr} = \mathrm{dU}_s + \delta \mathrm{Q}_{ht} + \partial W + \sum h_i dm_i \text{ ------------------------- Eqn. (3.2)}$$

Where:

- $\mathrm{U_s}$ – internal energy
- $\mathrm{Q_{ht}}$ – convective heat transfer to the cylinder walls
- W – work output
- $\mathrm{h_i dm_i}$ – the losses by the crevices

The final formula for the heat release by the combustion ($\delta \mathrm{Q}_{hr}$) is given by:

$$\frac{\mathrm{dQ}_{hr}}{d\theta} = \frac{c_v}{R}\left(p\frac{dV}{d\theta} + V\frac{dp}{d\theta}\right) + p\frac{dV}{d\theta} + \frac{\mathrm{dQ}_{ht}}{d\theta} \text{ -------------------------- Eqn. (3.3)}$$

The $\frac{dV}{d\theta}$ term is defined from the geometry of the engine and the $\frac{dp}{d\theta}$ term has been recorded from the engine. However, c_v and R are function of temperature, and the heat transfer rate cannot be evaluated. Then equation 3.3 can be written as

$$\frac{dQ_{hr}}{d\theta} - \frac{dQ_{ht}}{d\theta} = \frac{1}{(\gamma - 1)}\left(p\frac{dV}{d\theta} + V\frac{dp}{d\theta}\right) + p\frac{dV}{d\theta}$$

After rearranging the terms

$$\frac{dQ_{n}}{d\theta} = \frac{\gamma}{(\gamma-1)}p\frac{dV}{d\theta} + \frac{1}{(\gamma-1)}V\frac{dp}{d\theta} \text{------------------------------------ Eqn. (3.4)}$$

Where $\frac{dQ_{n}}{d\theta}$ = The net heat release rate.

This equation is plotted as differential heat release and as shown in Fig 3.37 (Stone 1999)

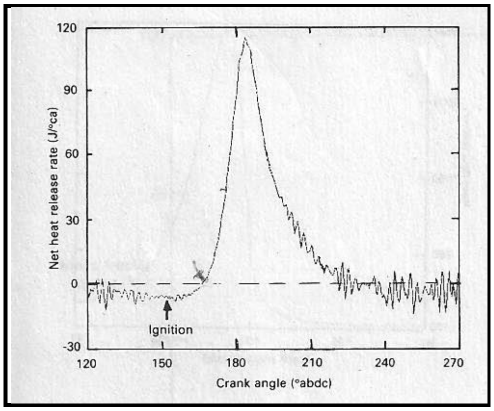


Figure 3.37: The net heat release rate (Stone 1999).

3.4.3 Mass Fraction Burnt

Rassweiler and Withrow (Rassweiler and Withrow 1980) examined combustion in the year 1938, proposed analytical method to define start and end of combustion.

In any crank angle interval $\Delta\theta$, the total pressure change

$$\Delta p = \Delta p_c + \Delta p_c \text{ -- Eqn. (3.5)}$$

Where Δp_c = Increase in pressure due to combustion and

Δp_v = Pressure change due to volume change

The pressure and volume at the start and end of the interval $\Delta\theta$, in the absence of combustion, are related by

$$p_iV_i^n = p_jV_j^n \text{--} \text{Eqn. (3.6)}$$

Hence

$$\Delta p_v = p_j - p_i = p_i\left[\left(\frac{V_i}{V_j}\right)^n - 1\right] \text{------------------------------} \text{Eqn. (3.7)}$$

The mass charge burnt in the interval $\Delta\theta$ is proportional to the pressure rise due to combustion; the mass fraction burnt at the end of the ith interval is given by

$$MFB = \frac{m_{bi}}{m_{btotal}} = \frac{\sum_0^i \Delta p_c}{\sum_0^N \Delta p_c} \text{--} \text{Eqn. (3.8)}$$

The end of combustion occurs after N increments, and is defined by the pressure rise due to the combustion.

Above relation is valid since the volume change is small when the piston is in the region of TDC, the computed mass fraction burnt is insensitive to slight error in the positioning of the TDC.

Mass fraction burnt for premixed phase combustion is determined by using NHRR curve to mark crank angle position at which combustion of this phase is completed. Thus, crank angle position is noted down from NHRR curve and corresponding point is located on mass fraction brunt plot. This can be illustrate by taking an example, if the premixed phase combustion completed at 7.5 °CA aTDC (concluded from NHRR curve), so corresponding vertical line intersected with MFB plot as shown in Fig. 3.38 and it is found that mass fraction of fuel burnt in the premixed phase is about 40%.

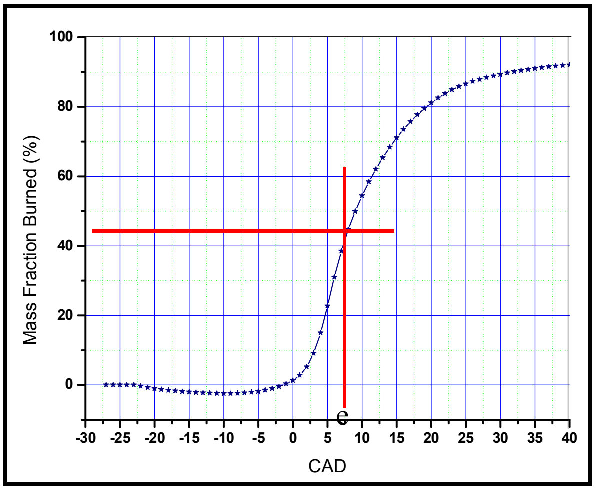


Figure 3.38: Evaluation of mass fraction burnt in the premixed phase.

3.4.4 Ignition Delay and Premixed Phase Combustion Duration

The ignition delay in a diesel engine was as the time (angular displacement of crank) interval between the start of injection and the start of combustion. The start of injection can be determined from the injector needle lift transducer or plot between injection pressure with respect to crank angle shows the peak pressure value at certain crank position. Thus, it corresponds to commencement of injection or SOI (start of injection) is used from the injection pressure sensor. Fuel injection pressure sensor provides information about the fuel line pressure at every crank angle turn of crankshaft, so maximum fuel line pressure is associated with injection event. Start of combustion is very difficult to determine precisely. DI engines the abrupt change in slope of P-θ curve is marked as a point at which combustion begins or SOC (start of combustion).

- This is determined by using NHRR curve, the “ab” shown in Fig. 3.39 gives ignition delay period is: crank angle degree between start of injection (given by pressure sensor) to the point where NHRR curve becomes positive.
- Premixed phase combustion duration can also be determined by using NHRR curve and it is the duration in terms of crank angle between point b and c.

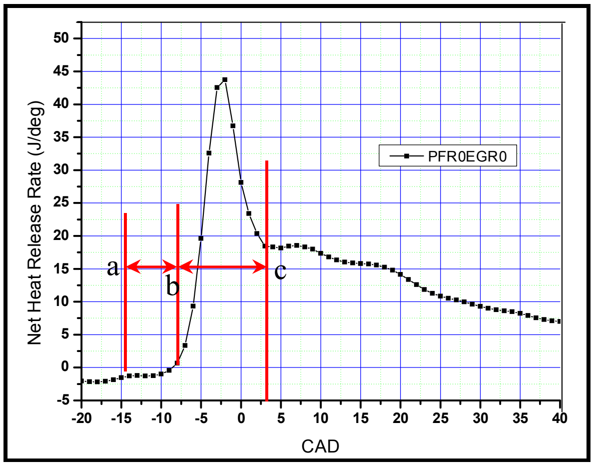


Figure 3.39: Ignition delay and premixed phase combustion duration.

Ignition Delay (ID) is a key parameter in achieving low soot and NO_x emissions, since increasing its duration provides more time for fuel-air mixing, potentially allowing higher fractions of pre-mixed charge.

Ignition delay is expressed in the equation as (Heywood 1988):

$$\tau_{id}=Ap^{-n}exp(E_A|\bar{R}T)$$ --- Eqn. (3.9)

Where, τ_{id} = Ignition delay period

E_A = Apparent activation energy for auto ignition of fuel.

$\bar{R}$ = Universal Gas constant.

A & n = Fuel dependent constants.

3.4.5 Brake Specific Fuel Consumption (BSFC) and Brake Thermal Efficiency (BTE)

For each test, the mass consumption rate of each fuel was calculated. On the basis of the fuel consumption rates, the engine speed was kept constant at 1500 rpm, the brake power, the brake-specific fuel consumption (BSFC), the brake thermal efficiency (BTE), and the BTE change rate ($\Delta\dot{\eta}_{bth}$) can be calculated. The equations for BSFC, BTE, expressions are given below

$$BSFC = \frac{\dot{m}_d + \dot{m}_p}{BP} \times 1000$$

$$BTE = \eta_{bth} = \frac{3600\times BP}{(\dot{m}_d \times h_{ud} + \dot{m}_p \times h_{up})}$$ ---------------------------- Eqn. (3.10)

h_{up} =Lower heating value of premixed ethanol fuel

h_{ud} =Lower heating value of direct injected diesel fuel

$\dot{m}_d$ = Mass consumption rate of directly injected diesel fuel (kg/hr)

$\dot{m}_p$ = Mass consumption rate of premixed ethanol fuel

Brake thermal efficiency change rate $=\Delta\eta_{bth} = \frac{[\eta_{bt,ref} - \eta_{bt,s}] \times 100}{\eta_{bt,ref}}$

$\eta_{bt,ref}$ = Brake thermal efficiency of reference condition

(0% of rate of EGR or % of PFR)

$\eta_{bt,s}$ = Brake thermal efficiency of sample case considered

3.4.6 Premixed Fuel Ratio

To study the combustion suitable instrumentation is required to measure the cylinder pressure, fuel injection timing and quantity of fuel injected.

In this work, the premixed fuel ratio was defined as the ratio of heat released by premixed fuel to the total heat released.

$$\mathrm{PFR} = \frac{\mathrm{m_p h_{up}}}{\mathrm{m_p h_{up} + m_d h_{ud}}} \text{-- Eqn. (3.11)}$$

Where m_p = Mass consumption rate of premixed ethanol

m_d= Mass consumption rate of directly injected diesel fuel

h_{up} = Lower heating value of premixed ethanol

h_{up}= Lower heating value of directly injected diesel fuel

As per pulse duration is fixed, thus constant quantity of ethanol is being injected at all load at constant speed and volume of diesel (in ml) is required to know the premixed fuel ratio. Therefore, to estimate the premixed fuel ratio there is a one excel program is used which takes the input from the equation given below:

$$m_{eth}\text{=}37.95 \times (V_{Diesel}) \times PFR/(1 - PFR) \times 26708 \text{ ----- Eqn. (3.12)}$$

3.4.7 EGR Ratio

EGR cooler is installed to control the exhaust gas temperature and so to control the temperature of the intake charge temperature.

Temperature of the exhaust gases varied from 450 °C to 700 °C in case of turbocharged engine it can go up to 600 °C, the engine which is used in the study has maximum exhaust temperature of about 320 °C, EGR cooler is designed and installed to control the temperature of the charge, since higher temperature of exhaust gases reduces BSFC or loss of efficiency, therefore temperature was brought down up to 110 °C. The CO_2 is used to find percentage of EGR.

Proportion of exhaust gas re-circulation is defined by

$$\%\ \text{EGR} = \frac{\dot{m}_{EGR}}{\dot{m}_{EGR} + \dot{m}_{air}} \times 100$$

EGR can also defined by the concentration of CO_2 in the intake mixture and Exhaust (Baert and Beckman 1999)

$$\text{EGR Ratio} = \frac{[CO_2]_{intake} - [CO_2]_{ambient}}{[CO_2]_{exhaust} - [CO_2]_{ambient}} \times 100$$

OR

$$\text{EGR Ratio} = \frac{[CO_2]_{ambient}}{[CO_2]_{exhaust}} \times 100 \text{----------Eqn. (3.13)}$$

CHAPTER 4: DESIGN OF SMALL EGR COOLER

EGR cooler is required to achieve necessary cooling of exhaust gases. Shell and tube type of EGR cooler is available in the market which was designed for automotive like, Maruti Swift and other vehicles. They are not found compatible with set-up, since the exhaust temperature of these automotive diesel engines is quite high (600 °C) and coolant (oil and water mixture) is flowing in close circuit. However, the condition in case of small capacity single-cylinder engine, which is used for the experimental purpose, is not similar.

As the exhaust flow rate is quite low (20 kg/hr) and maximum temperature of exhaust gases coming out from the cylinder is around 300 °C, also water could be used instead of coolant. Cooling water coming out from the cylinder jacket is used in the shell and tube type heat exchanger that eliminates requirement of separate arrangement. As per the literature exhaust gases should be cooled up to 110 °C to avoid loss of efficiency due to hot exhaust gases. To optimize baffles spacing, clearance, shell and tube size an excel calculation sheet is used is developed, given in Plate A2 and Plate A3 that helps to iterate as per the operating conditions. Kern method (Kern 1997) is used for designing shell and tube type heat exchanger and same is also submitted for design patent to controller general of patents, designs and trademarks India.

4.1 DETAILED CALCULATION FOR SHELL AND TUBE TYPE EGR COOLER DESIGN

As mentioned above after optimizing dimension of EGR cooler, stainless steel (Grade 404) pipe of 14 BWG is selected for shell.

Inner diameter of the shell ID = 59.28 mm

Outer diameter of the shell OD = 63.5 mm

Water is selected as a shell fluid,

Intake temperature of water, T_{ci} = 40 °C

Outlet temperature is unknown, T_{co} =?

Exhaust gases are flowing through tubes

As the copper shows excellent thermal conductivity, therefore it is selected as a tube material.

Number of copper tubes = 13, Length of copper tubes = 39.6 cm

Outer diameter of the copper tube OD = 7.92 cm and BWG = 2.2 BWG

Pitch = 1.153 cm and Number of passes = 1

Heat balance

Heat transfer from exhaust gases

$Q_{ex} = \dot{m}_{\dot{m}\dot{m}_{ex}} \times c_{pex} \times \Delta t_{ex}$ -------------------------------------- Eqn. (4.1)

Inlet temperature of exhaust gases, T_{hi} = 245 °C

Outlet temperature of exhaust gases, T_{ho} = 110 °C

Mass flow rate for exhaust gases, $\dot{m}_{\dot{m}_{\dot{m}\dot{m}_{ex}}}$ = 9.8 kg/hr.

Specific heat C_{pex} = 1.007 KJ/kgK

$$Q_{ex} = \frac{9.8}{3600} \times \ 1.007 \times (245 - 110)$$

$Q_{ex} = 370\ W$ -- Eqn. (4.2)

Heat transfer from water

$$Q_w = \dot{m}\dot{m}_w \times c_{pw} \times \Delta t_w = 370$$

Using Eqn. (4.2)

$$Q_w = \frac{200}{3600} \times \ 4.179 \times (T_{co} - 40\ ) = 0.370$$

$$T_{co} \ = \ 41.59\ °C$$

Log mean temperature difference and true temperature difference (LMTD)

From process heat transfer (Kern 1997), page 89

$$\text{LMTD} = \frac{\Delta T1 - \Delta T2}{\ln(\frac{\Delta T1}{\Delta T2})} \text{--} \text{Eqn. (4.3)}$$

$$\Delta T_1 = T_{hi} - T_{co} = 245 - 41.59$$

$$= 203.41$$

$$\Delta T_2 = T_{ho} - T_{ci} = 110 - 40$$

$$= 70$$

$$\text{LMTD} = \frac{203.41 - 70}{\ln(\frac{203.41}{70})}$$

$$= 125.15$$

4.1.1 Heat Transfer Co-Efficient of Water Side

Since the heat exchanger is designed for single tube and single pass, LMTD for counter flow is true temperature difference. It doesn't require any correction factors

Caloric temperature: If the liquid is neither a petroleum fraction nor a hydrocarbon the caloric temperature cannot be determined.

Heat transfer co-efficient for shell side (water)

The shell side or bundle cross flow area a_s–page no 138 (Kern 1997)

$$\text{Flow area } a_s = \frac{D_{is} \times C' \times B}{P_T} m^2 \text{ ---} \text{Eqn. (4.4)}$$

Where

C' =Clearance between the tubes $P_T =$ Triangular pitch

$B =$ Baffles spacing

$$a_s = \frac{5.918 \times 0.431 \times 3.81 \times 10^{-6}}{1.153 \times 10^{-2}}$$

$$a_s = 8.42 \times 10^{-4} \text{ m}^2$$

From page 138 (Kern 1997) for mass velocity

$$\text{Mass velocity of water Gs} = \frac{\dot{m}_{\dot{m}\dot{m}_W}}{a_s} \frac{kg}{(hr)(m^2)} \text{--------------------------} \text{Eqn. (4.5)}$$

$\dot{m}\dot{m}_w$ = Mass flow rate of water through shell

$\dot{m}\dot{m}_w$ = 200 kg/hr.

$G_s = \frac{200}{8.42\ X\ 10^{-4}} = 237529\ \frac{kg}{(hr)(m^2)}$

Reynolds number

$R_e = \frac{D_e \times G_s}{\mu}$ -- Eqn. (4.6)

μ = Viscosity of water at average temperature

With reference to plate A4 (Appendices A)

From table X = 10.2, Y= 13, and subsequently using plate A5

$\mu = 0.675$ cP -- Eqn. (4.7)

Where D_e = equivalent diameter for the shell

For triangular pitch, the arrangement is shown in Fig. 4.1

Figure 4.1: Schematics of triangular pitch arrangement.

Shell side equivalent diameter:-

Equivalent diameter = 4 × Hydraulic radius obtained for the pattern as layered on the tube sheet.

$D_e = \frac{4 \times \text{Free area}}{\text{Wetted perimeter}}\ m$ -- Eqn. (4.8)

For triangular pitch

$$D_e = \frac{4\times(\frac{1}{2}\times P_T \times 0.86P_T - \frac{1}{2}\times\frac{\pi d_o^2}{4})}{\frac{1}{2}\pi d_o}$$

$$D_e = \frac{4\times\left(1.153\times 0.86\times 1.153 - \frac{\pi(0.792)^2}{4}\right)}{\pi\times 0.792}\times 10^{-2}\,m$$

$$= 1.044\ \times 10^{-2}m$$

Therefore Reynolds number

$$R_e = \frac{1.044\ \times 10^{-2}\ X\ 237529}{0.675\times 10^{-3}\times 3600}$$

$$= 1020$$

As per the Reynolds number obtained, from plate A6, we could get J_H

$J_H = 16.5$ -- Eqn. (4.9)

At average temperature of water, $T_{avg.} = 40.79\ ^0C$, thermal conductivity of water

$k = 0.622\ \frac{W}{mK}$--- Eqn. (4.10)

To determine ‘c’ of water at average temperature (T_{av}) of 40.79 °C, plate A7 is used

$c = 4.179\ \frac{kJ}{kgK}$ --- Eqn. (4.11)

From page 111 (Kern 1997) for heat transfer co-efficient

By using Eqn. (4.7) to Eqn. (4.11)

$h_0 = J_H\left(\frac{k}{D_e}\right)\left(\frac{\mu c}{k}\right)^{1/3} \times \varphi_t$-- Eqn. (4.12)

$$= 16.5\left(\frac{0.622}{1.044\ \times 10^{-2}}\right)\left(\frac{0.675\times 10^{-3}\times\ 4.179\times 10^{3}}{0.622}\right)^{1/3} \times 1$$

$$= 16.5\ \times 59.60\ \times\ 1.654$$

$= 1618.67\ \frac{\mathrm{W}}{\mathrm{m^2-K}}$-- Eqn. (4.13)

4.1.2 Heat Transfer Co-Efficient of Exhaust Gases Side

Flow area from single tube, page 843 (Kern 1997)

$$a'_t = \frac{\pi}{4}\times(0.65\times 10^{-2})^2 = 3.31\times 10^{-5}m^2$$

From page 150 (Kern 1997) , for the flow area from tube side

$a_t = \frac{N_t a'_t}{n} m^2$ -- Eqn. (4.14)

$= \frac{13 \times 3.31 \times 10^{-5}}{1} m^2$

$= 4.31 \times 10^{-4}$

Mass velocity $G_t = \frac{\dot{m}_{\dot{m}\dot{m}_{ex}}}{a_t} \frac{kg}{(hr)(m^2)}$-------------------------------------- Eqn. (4.15)

Where, $\dot{m}_{\dot{m}\dot{m}_{ex}}$ = Mass flow rate for exhaust gases

$G_t = \frac{9.8}{4.31 \times 10^{-4}} = 22737 \frac{kg}{(hr)(m^2)}$

Reynolds number for the tube side fluid or exhaust gases

$R_e = \frac{D_{it} \times G_t}{\mu}$--- Eqn. (4.16)

Viscosity can be calculated by using linear interpolation, therefore

$\mu = 0.031$ cP -- Eqn. (4.17)

Reynolds number

$R_e = \frac{0.65 \times 10^{-2} \times 22737}{0.031 \times 10^{-3} \times 3600} = 1324$

Heat transfer co-efficient

$h_i = h_0 = J_H \left(\frac{k}{D_{it}}\right) \left(\frac{\mu c}{k}\right)^{1/3} \times \varphi_t$------------------------------ Eqn. (4.18)

From plate A8 exhaust gas side co-efficient, $J_H = 5$ ----------------- Eqn. (4.19)

$k = 0.0479 \frac{W}{m\,K}$--- Eqn. (4.20)

$c = 1.055 \frac{kJ}{kg\,K}$ -- Eqn. (4.21)

By using Eqn. (4.16) to Eqn. (4.21)

$h_i = 5 \left(\frac{0.0479}{0.65 \times 10^{-2}}\right) \left(\frac{0.031 \times 10^{-3} \times 1.055 \times 10^3}{0.0479}\right)^{0.33}$

$= 32.38 \frac{W}{m^2K}$-- Eqn. (4.22)

$h_{io} = h_i \left(\frac{D_{it}}{D_{ot}}\right) = 32.38 \times \frac{0.65 \times 10^{-2}}{0.792 \times 10^{-2}} = 26.49 \frac{W}{m^2K}$ ------------- Eqn. (4.23)

4.1.3 Overall Heat Transfer Co-Efficient

Using equations 4.13 and 4.23

Therefore clean overall heat transfer co-efficient

$$U_c = \frac{h_{io} \times h_o}{h_{io} + h_o} = 26.06 \frac{W}{m^2 - K} \text{ --- Eqn. (4.24)}$$

Using heat transfer equation

Now using equation $Q = N_t \times U_D \times \pi \times d_o \times l \times LMTD$ ------------ Eqn. (4.25)

$13 \times U_D \times \pi \times 0.792 \times 10^{-2} \times 0.396 \times 125.15 = 370$

$$U_D = 23.08 \frac{W}{m^2 - K} \text{ -- Eqn. (4.26)}$$

Where U_D = Design overall heat transfer coefficient. When comparing Eq. 4.24 and 4.26. It is established that $U_C > U_D$. Therefore length of shell and tube heat exchanger is 39.6 cm. However to reach this calculation several trial error iterations were made. Fig. 4.2 to Fig. 4.4 shows three dimensional CAD model actual EGR cooler under working condition.

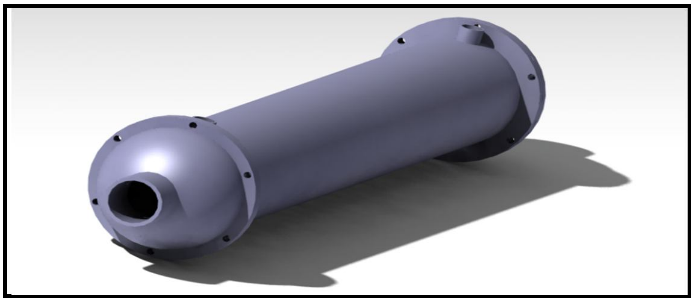


Figure 4.2: 3-D model of EGR cooler.

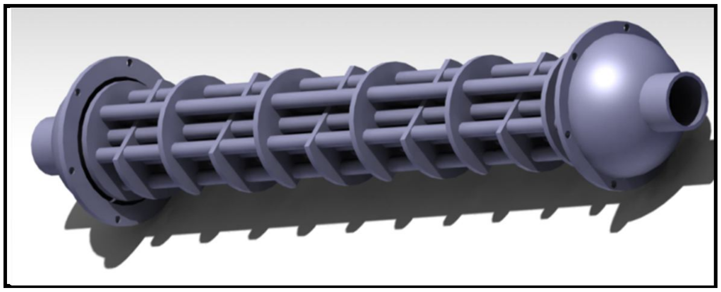


Figure 4.3: Baffles and tube arrangement of EGR cooler.

4.1.4 Number Of Passes

Dirt factor

$R_{d\ calculated} = 0.0305$

$R_{d\ required} = 0.011$

Number passes can be calculated, from page 147 (Kern 1997)

Number of crosses = $\frac{\text{Tube lenght}}{\text{Baffle spacing}} = \frac{\text{L}}{B}$-------------------------------- Eqn. (4.27)

$= \frac{39.6}{3.81} = 10.393$

Hence total 11 numbers of crosses being used through baffles.

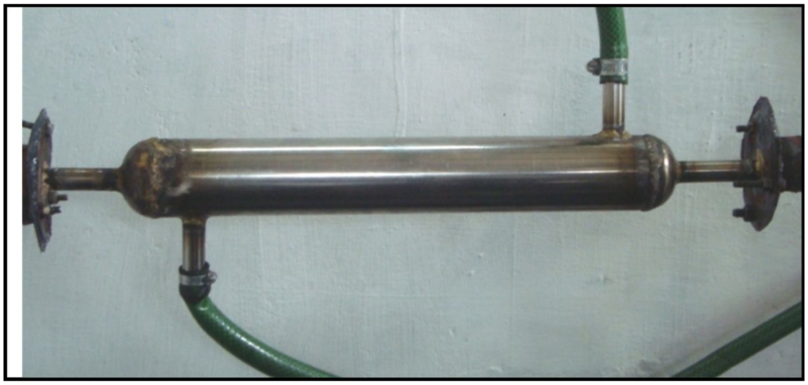


Figure 4.4: Shell and tube type EGR cooler under operation.

CHAPTER 5: RESULTS AND DISCUSSIONS

Successful modifications of diesel engine into partially premixed combustion engine while keeping major objective in the mind of reducing NO_x and smoke emission without sacrificing thermal efficiency. Firstly physico-chemical analysis of the fuel properties is carried out. Subsequently, to examine partially premixed charge compression ignition engine (PCCI) combustion for the entire engine load range three different BMEP was selected. These engine loads correspond to 50%, 75% and 100 % rated power of the engine. This has been reported in the literature and also experienced by experimental observations that at low engine load PCCI combustion could not produce satisfactory results. Nevertheless, at no load and 25% of engine load PCCI combustion yields dramatic rise in UHC emission without showing favourable performance and emissions, so it is excluded from the study.

Thus, experiments were performed at a constant engine speed of 1500 revolutions/min (±10) and three different loads corresponding to BMEP of 2.1 bar (50% load), 3.1 bar (75% load) and 4.2 bar (100% load), or 1.75 kW, 2.6 kW and 3.5 kW were selected to examine the effect of premixed ethanol with and without EGR (exhaust gas re-circulation). At each intake air/fuel mixture condition a detailed analysis was carried out to understand combustion characteristics, performance and gaseous emissions. However, combustion geometry modifications and air preheating, which is discussed in the chapter 3, section 3.3.4 and 3.3.5 not shown any appreciable improvement in the performance and emissions, therefore, is not considered in the reported work. All the experiments were conducted on original combustion geometry (hemispherical bowl) and at ambient air temperature.

To obtain comparative inferences on each engine load the combustion, performance and emissions were measured while engine was fueled with diesel and without EGR.

This condition hereafter referred as a PFR0EGR0, Where, PFR stands for the Premixed Fuel Ratio and EGR stands for Exhaust Gas Re-circulation when both were zero indicates pure diesel combustion. Also, any other number followed by PFR and EGR indicates percentage of premixed ethanol (based on energy addition) and percentage of exhaust gas concentration (based on CO_2 concentration) respectively.

To examine the effects of varying mixture composition of premixed fuel and exhaust gas re-circulation, the results are analyzed by keeping constant EGR and varying PFR and vice-versa. Thus, one set of analysis is done on a constant EGR ratio and other on constant PFR. Thus, the first set of analysis is done on influence of premixed fuel ratio on combustion, emission and performance of PCCI engine; and second on the influence of exhaust gas re-circulation ratio on combustion, emission and performance of PCCI engine

5.1 FUEL PROPERTIESANALYSIS

The use of alcohol, as a fuel for internal combustion engine, is not a new concept. Ethanol diesel blends used in unmodified diesel engine (Hwanam and Byungchul 2008) reported similar or even improved brake thermal efficiency while operating on ethanol diesel blends fuels as compared to that of pure diesel.

As discussed above fuel properties have significant role in combustion and chemical kinetics of PCCI engine. To measure calorific value, viscosity coefficient, flash point, and density of fuels, the facilities are available in our biofuel research laboratory at University of Petroleum and Energy Studies. All the measurement is being conducted as per ASTM standards. The apparatus are shown in Fig. 5.1 to 5.4 used for the measurements. Measured properties of fuels are given in the Table 5.1.

Table 5.1: Comparison of various properties of alcohol fuels with bio-diesel, and diesel

Property	**Methanol**	**Bio-diesel**	**Diesel**	**Ethanol**
Chemical formula	CH_3OH	$C_{16}H_{32}O_2$ - $C_{19}H_{32}O_2$	$C_{14}H_{30}$	CH_3CH_2OH
Molecular weight (g/mol)	32.04	256-292	198.4	46.07
Density (g/cm^3)	0.7866	0.870–0.895	0.810 0.860	0.785
Lower heating value (MJ/kg)	20.1	36.5-38	42.5-44.2	26.708
Viscosity coefficient (mPas)	0.5445	4.6	2–4	1.078
Flash Point (°C)	12	100-170	74	13

Figure 5.1: Flash and Fire point apparatus.

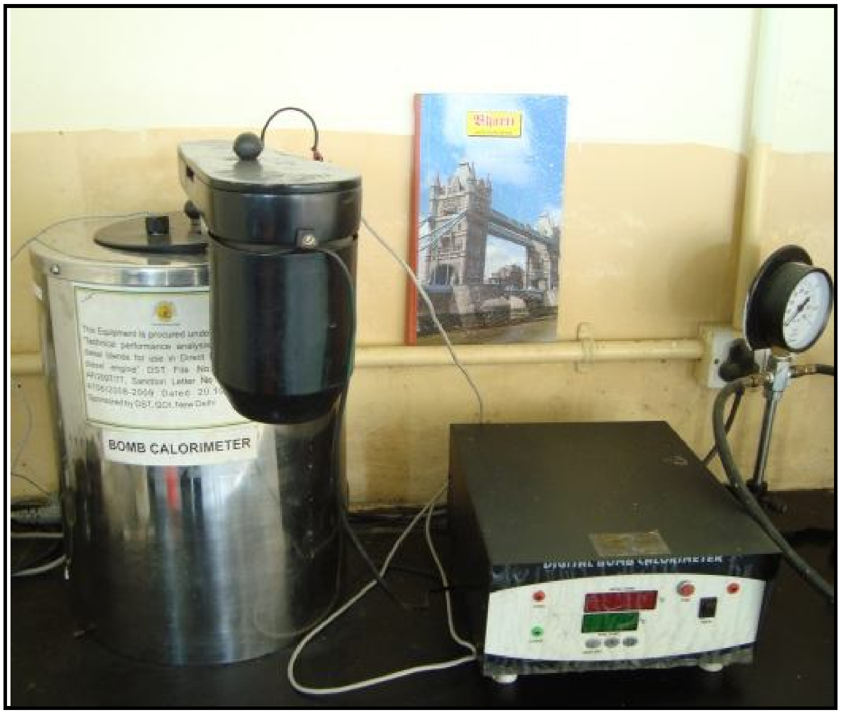


Figure 5.2: Bomb Calorimeter

Figure 5.3: Viscometer

Figure 5.4: Digital density meter.

5.2 INFLUENCE OF PFR AND EGR AT HALF LOAD

The experimental study was carried out at fixed engine speed, injection pressure and injection timing for diesel and ethanol. The operating condition corresponds to BMEP of 2.1 bar (50% load) or 1.75 kW.

While operating without premixed and EGR the diesel fuel consumption was about 598 g/hr and at this load it is associated with lean combustion, thus poor brake thermal efficiency was obtained. To inject ethanol into the air an electronic pulse of 2 ms is given to the controller that results into 190 g/hr flow rate of ethanol. This flow rate corresponds to 18% of energy addition as compared to the total energy supplied by ethanol and diesel together. Nevertheless, increasing pulse duration (2 to 3 ms) leads increases PFR from 18 to 25%, but there is no significant change in emissions trend was observed. Therefore, the results which were presented here at a 2.1 bar BMEP only at 18% of PFR. Table 5.2 show matrix of fuel composition selected for the study.

Table 5.2: Test matrix at BMEP of 2.1 bar

Sr. No.	Abbreviations	Description
1	PFR0EGR0	Premixed ratio = 0% and Exhaust gas re-circulation = 0%
2	PFR18EGR0	Premixed ratio = 18% and Exhaust gas re-circulation = 0%
3	PFR0EGR10	Premixed ratio = 0% and Exhaust gas re-circulation = 10%
4	PFR18EGR10	Premixed ratio = 18% and Exhaust gas re-circulation = 10%
5	PFR0EGR20	Premixed ratio = 0% and Exhaust gas re-circulation = 20%
6	PFR18EGR20	Premixed ratio = 18% and Exhaust gas re-circulation = 20%

5.2.1 Effect of PFR on Combustion

To understand the effect of the premixed ethanol on combustion characteristics, in-cylinder pressure, net heat release rate, and mass fraction burnt, premixed combustion duration and ignition delay was analyzed.

5.2.1.1 Effect of PFR on Combustion Without EGR

In-cylinder pressure analysis is the most effective way to analyze the engine combustion behaviour; also in-cylinder pressure history directly affects the power output and emissions from an engine. An average data of 50 consecutive cycles is used to avoid cyclic variations.

Fig 5.5 shows the effect of premixed ethanol on the in-cylinder pressure. The peak in cylinder pressure decreases from 51.49 bar to 48.74 bar and it is due to increase in ignition delay period and later combustion phasing occurs. Maximum in-cylinder pressure peaks shifted from 369 °CA to 371 °CA when intake mixture composition changes from PFR0EGR0 to PFR18EGR0. Maximum rate of pressure also decreases due to premixed ethanol, this value of pure diesel mode (PFR0EGR0) was about 3.41 and whereas in case of premixed mode (PFR18EGR0) it was dropped to 2.62.

Hayes et al. (Hayes and Savage 1988) conducted study on fuel injection of ethanol at BMEP of 2 bar, they had reported, in-cylinder pressure decreases with increase in premixed ethanol into the intake air. This was due to increase in ignition delay period which results in combustion occurring during the expansion stroke. Kowalewicz (Kowalewicz 2006) also reported that the in-cylinder pressure decreased at low-engine load at different fractions of fumigated ethanol while rapeseed oil methyl ester (RME) was used as a DI fuel.

Fig 5.6 shows variation in net heat release rate for with and without premixed ethanol. As expected, trends are observed that the peak net heat release rate does not changed significantly.

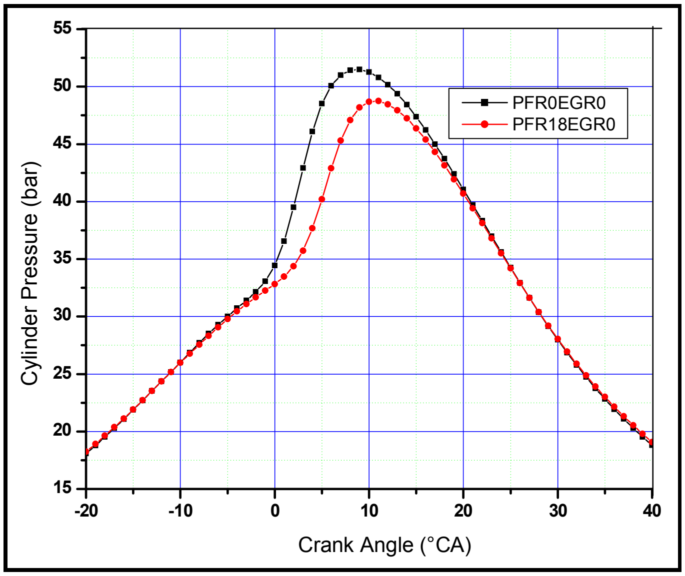


Figure 5.5: Effect of premixed fuel ratio on in-cylinder pressure.

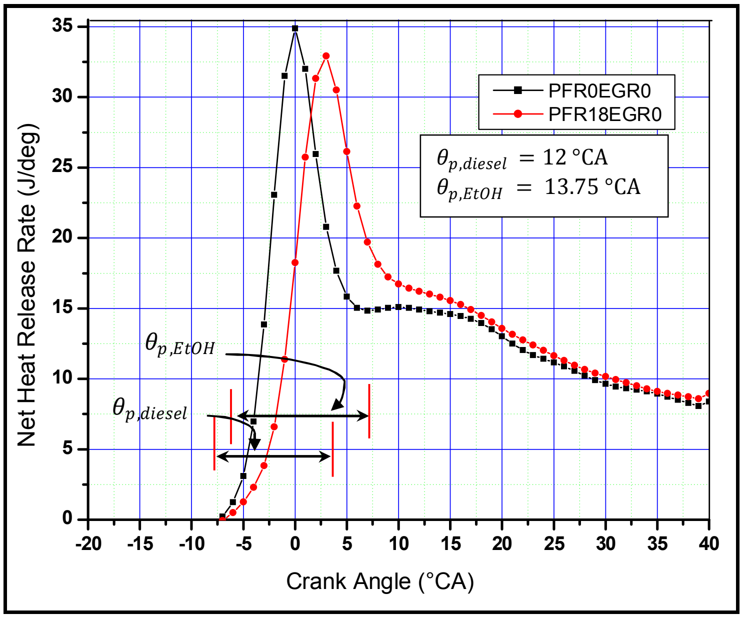


Figure 5.6: Effect of premixed fuel ratio on net heat release rate.

It was recorded 34.88 J/deg and 32.92 J/deg associated with PFR0EGR0 and PFR18EGR0 respectively. Nevertheless, premixed phase combustion duration increases, and rate of burning decreases due to premixed ethanol. This was due to the lower overall specific heat ratio of intake as well as the lower cetane number (CN) of the ethanol associated with PFR18EGR0 compared to PFR0EGR0. Since net heat release rate corresponds to peak in-cylinder pressure (as per first law of thermodynamics), therefore it decreases with decrease in pressure.

Fig. 5.7 shows the influence of premixed ethanol on mass fraction burnt. Mass fraction burnt did not show significant difference in the beginning of combustion and it diminishes as the combustion proceeds. It also indicates slower burning rate in case of premixed ethanol corresponds to PFR18EGR0. The premixed combustion phase completed at 7 °CA aTDC and 7.5 °CA aTDC corresponds to PFR0EGR0 and PFR18EGR0, therefore mass fraction burnt remains almost same in both the cases. Mass fraction burnt was found about 40% correspond to premixed phase combustion.

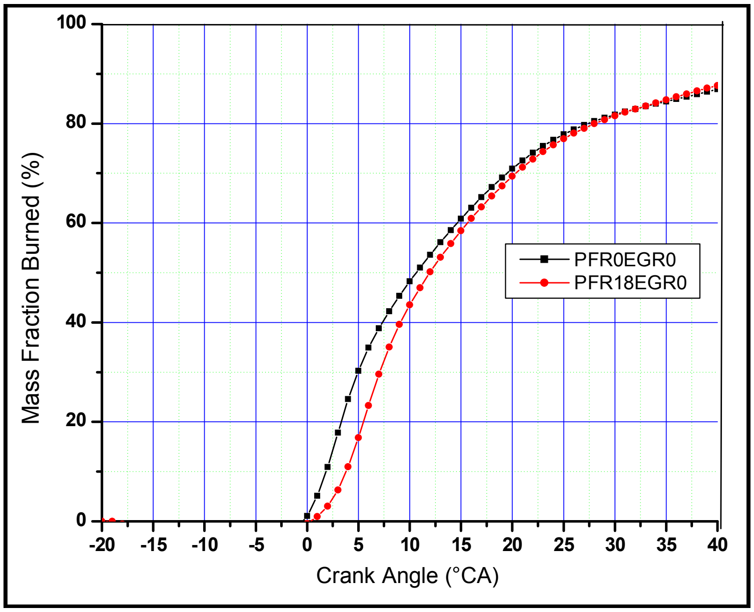


Figure 5.7: Effect of premixed fuel ratio on mass fraction burnt.

5.2.1.2 Effect of PFR on Combustion at 10% EGR

The effect of premixed ethanol with EGR has significant effect on the combustion characteristics. Therefore, in this section premixed ratio which was selected ranges from 0 and 18% and EGR ratio was kept fixed as 10%.

Fig 5.8 shows the in-cylinder variation with crank angle for diesel combustion at 10% EGR (PFR0EGR10) and partially premixed charge compression ignition engine combustion with 18% premixed ethanol or premixed fuel ratio (PFR18EGR10). PFR18EGR10 shows considerable reduction in peak in-cylinder pressure it was about 8% as compared to PFR0EGR10 or pure diesel combustion with exhaust gas re-circulation.

The combined effect of EGR and premixed ethanol are responsible for the dilution of intake air due to higher specific heat of EGR constituents reduces the peak in-cylinder pressure and temperature. However, higher latent heat of vaporization and low cetane number (CN) of ethanol further retarded the peaks of in-cylinder pressure, which were obtained at 373 °CA and 369 °CA associated with PFR0EGR10 and PFR18EGR10. Rate of in-cylinder pressure rise was adversely affected when intake mixture changes from PFR0EGR10 to PFR18EGR10 and it gets reduced from 2.97 to 1.87. However, in case of PFR18EGR10 rate of pressure rise was higher during the latter part of expansion which improves work conversion efficiency.

Fig. 5.9 shows the variation of net heat release rate with crank angle. Combustion phasing was observed due to EGR and premixing of ethanol. The reduction in maximum net heat release rate was about 9.4% observed in case of PFR18EGR10 compared to PFR0EGR10. Peak of NHRR shifted from 2 °CA aTDC to 7 °CA aTDC when intake charge mixture changes from PFR0EGR10 to PFR18EGR10. Thus, significant increase in the premixed phase combustion duration was recorded and it increases from 13.5 °CA to 17.5 °CA. This shows that relatively greater amount of the fuel combusted in the premixed combustion phase, associated with PFR18EGR10 compared to PFR0EGR0.

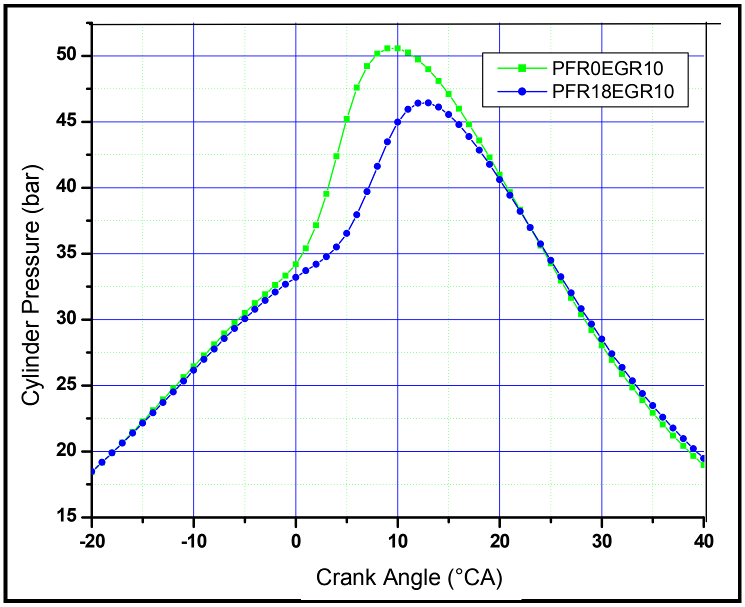


Figure 5.8: Effect of premixed fuel ratio on in-cylinder pressure

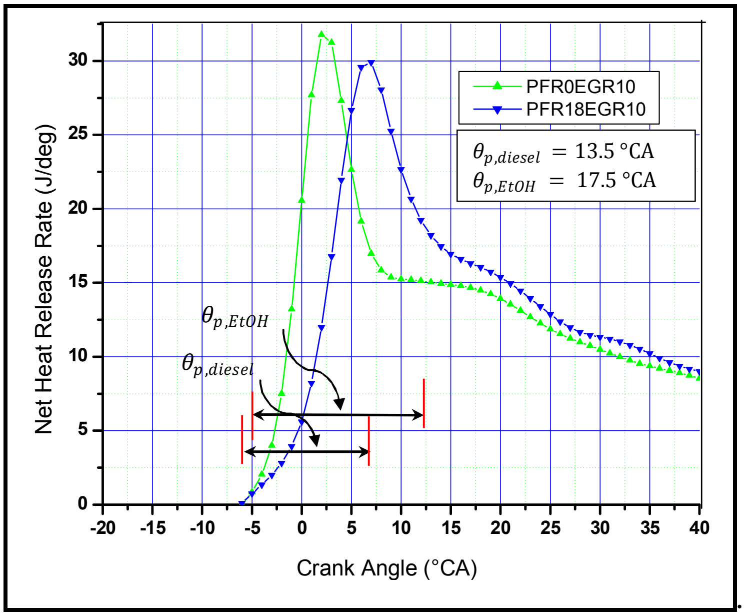


Figure 5.9: Effect of premixed fuel ratio on net heat release rate

Fig. 5.10 shows variation of mass fraction burnt. In case of premixed ethanol, the early stage of mass fraction burn was lower whereas last stage of burning showed higher value as compared to diesel fuel (PFR0EGR0). It happens mainly due to variation in chemical reactivity of the mixtures at different stages of combustion. Fig. 5.9 shows that PFR0EGR10 premixed phase combustion ended at 7.5 °CA aTDC whereas it increased to 15 °CA aTDC for the PFR18EGR10. Keeping these positions in mind mass fraction burnt changes from 27% to 47% associated with PFR0EGR10 and PFR18EGR10. Also, the diffusion combustion has shown significant improvement compared to 10% EGR ratio without premixed ethanol. However, mass fraction burnt increased due to oxygen availability.

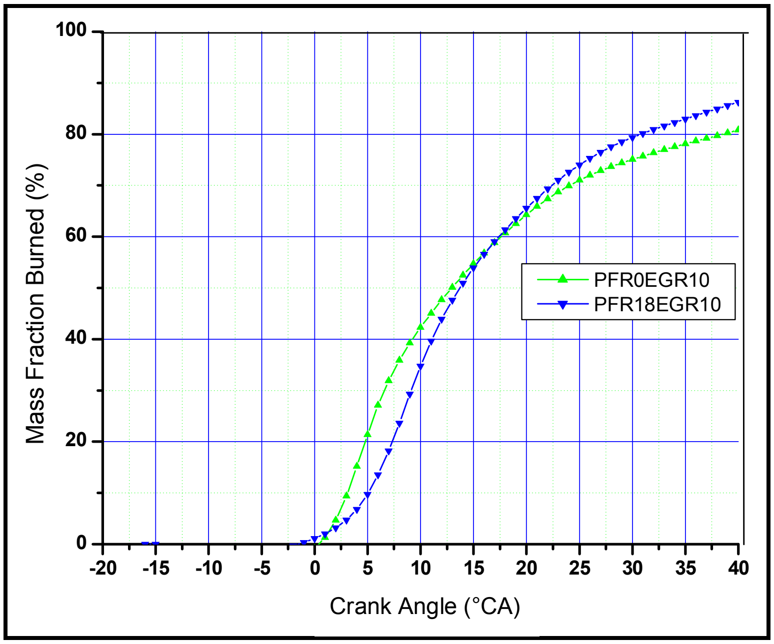


Figure 5.10: Effect of premixed fuel ratio on mass fraction burnt.

5.2.1.3 Effect of PFR on Combustion at 20% EGR

Fig. 5.11 shows the maximum in-cylinder pressure was reduced from 49.79 bar to 45.72 bar associated with PFR0EGR20 and PFR18EGR20 respectively. High rate of EGR leads to significantly decrease in peak in-cylinder pressure as well as rate of pressure rise. The maximum rate of pressure rise was 2.97 and 1.87 for the PFR0EGR18 and PFR18EGR20 respectively. However, PCCI combustion with EGR did not affect considerably the rate of pressure rise and combustion phasing.

Fig 5.12 shows the variation of net heat release rate with crank angle. Combustion phasing was observed due to EGR and premixing of ethanol.

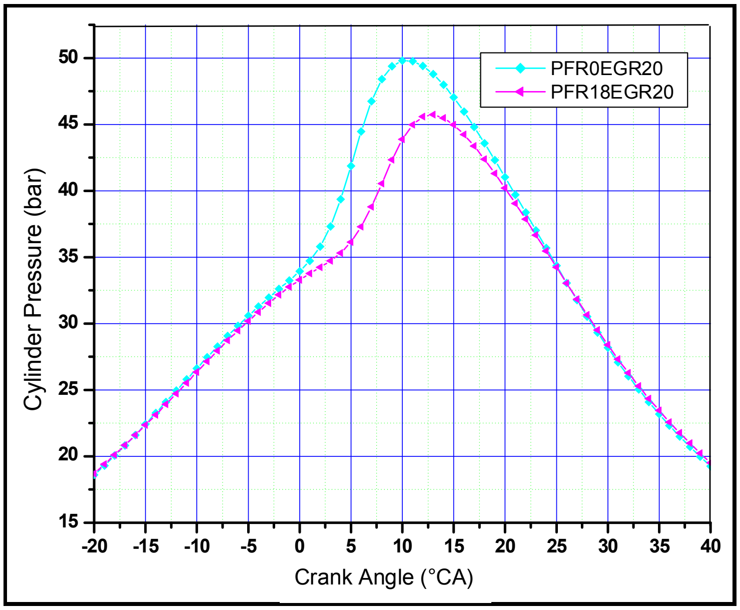


Figure 5.11: Effect of premixed fuel ratio on in-cylinder pressure.

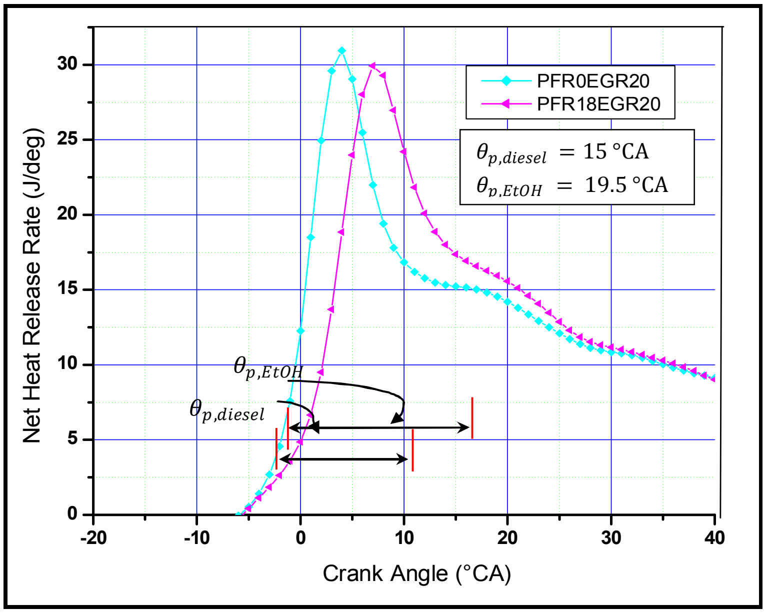


Figure 5.12: Effect of premixed fuel ratio on net heat release rate.

As described in earlier case with 10% EGR ratio (PFR18EGR10), peak net heat release rate decreases due to premixing of ethanol. Due to increase in ignition delay period causing higher amount of fuel being burnt during the premixed combustion phase. However, the reduction in oxygen availability associated with the application of EGR diluents reduces the rate at which the fuel burnt in the premixed phase (Ladommatos and Abdelhalim 1998).

Fig 5.13 shows clearly that premixing of ethanol along with high amount of EGR associated with PFR18EGR20 further extends ignition delay period. Premixed combustion phase extended up to 15.2 °CA aTDC corresponds to PFR18EGR20, whereas, in case of PFR0EGR20 it is only 12.5 °CA aTDC. Therefore, mass fraction burnt was found about 50 and 60% associated with PFR0EGR20 and PFR18EGR20 respectively. However, maximum mass fraction burnt increased with premixing ethanol, it changes from 88% to 98% due to presence of higher oxygen content. This clearly shows diffusion combustion burning rate increases under partially premixed charge compression ignition engine mode.

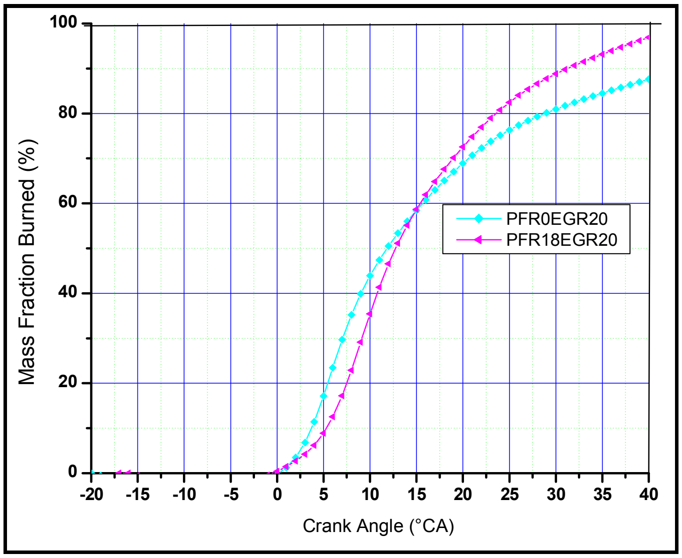


Figure 5.13: Effect of premixed fuel ratio on mass fraction burnt.

5.2.2 Effect of EGR Ratio on Combustion

As per the literature reported using ethanol in most of the cases led to reduce brake thermal efficiency. However, the focus is shifted to reduce emission especially, NO_x and PM. In the previous section the effect of premixed ratio with EGR rate have been discussed. In this section, our aim is to understand impact of EGR on combustion characteristics at a fixed premixed fuel ratio. In addition, it can also provide basis to evaluate outcome of increasing EGR on the combustion and how it depends on premixed fuel ratio.

5.2.2.1 Effect of EGR on Combustion without PFR

Fig. 5.14 shows, in-cylinder pressure traces for varying rate of EGR. Maximum reduction in peak in-cylinder pressure was observed in case of PFR0EGR20, it was decreased from 51.49 bar to 49.79 bar. Nevertheless, due to higher EGR the rate of pressure rise decreased with increase in EGR but the trend was reversed during later part of expansion. Peak in-cylinder pressure was observed at 10 °CA aTDC for PFR0EGR20, showed maximum ignition delay period.

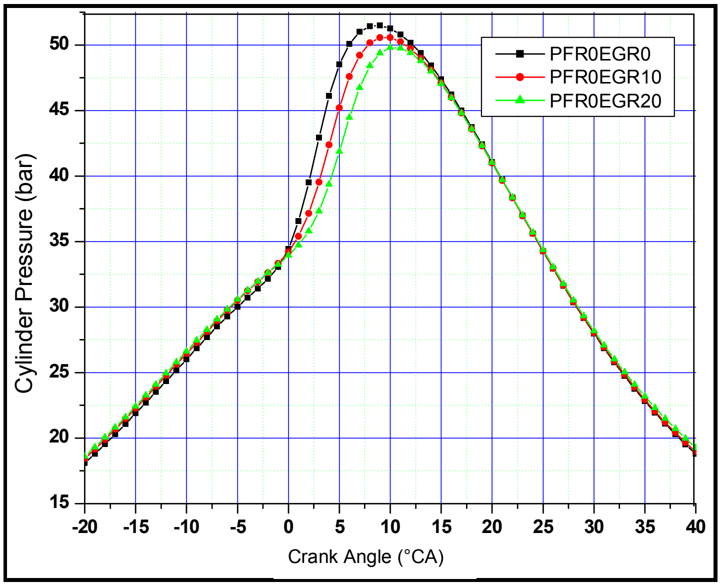


Figure 5.14: Effect of EGR ratio on in-cylinder pressure.

Fig 5.15 shows net heat release rate at 0, 10 and 20% of EGR without premixed ethanol fuel (PFR0EGR0, PFR0EGR10 and PFR0EGR20). A little variation in maximum net heat release rate was observed but combustion phasing was causes due to increase in EGR ratio, which reduces oxygen availability. Thus, dilution effect leads to reduction of local temperature and lower flame temperature (Ladommatos N and Abdelhalim SM 1997; Ladommatos N and Abdelhalim SM 1998). Premixed combustion duration found to be increased with increase in EGR ratio.

Fig. 5.16 shows mass fraction burnt plot. Premixed combustion in case of PFR0EGR0 lasted up to 7 °CA aTDC whereas for PFR0EGR20 it increased to 12.5 °CA aTDC. Therefore, mass fraction burnt during premixed combustion was reaches a values of

42% and 55% for PFR0EGR0 and PFR0EGR20 respectively. However, it also reduces the rate at which the fuel being burnt in the premixed phase, since EGR reduces the oxygen availability which in turn increases soot particles.

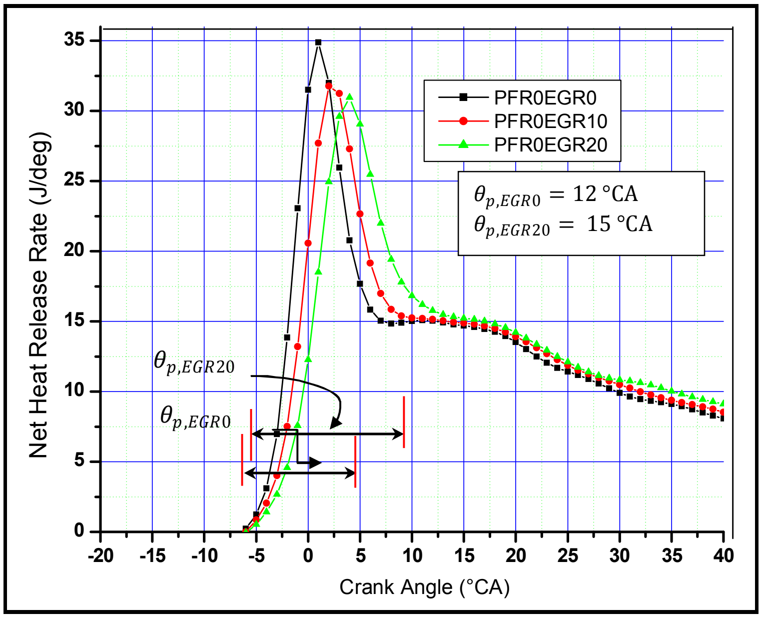


Figure 5.15: Effect of EGR ratio on net heat release rate.

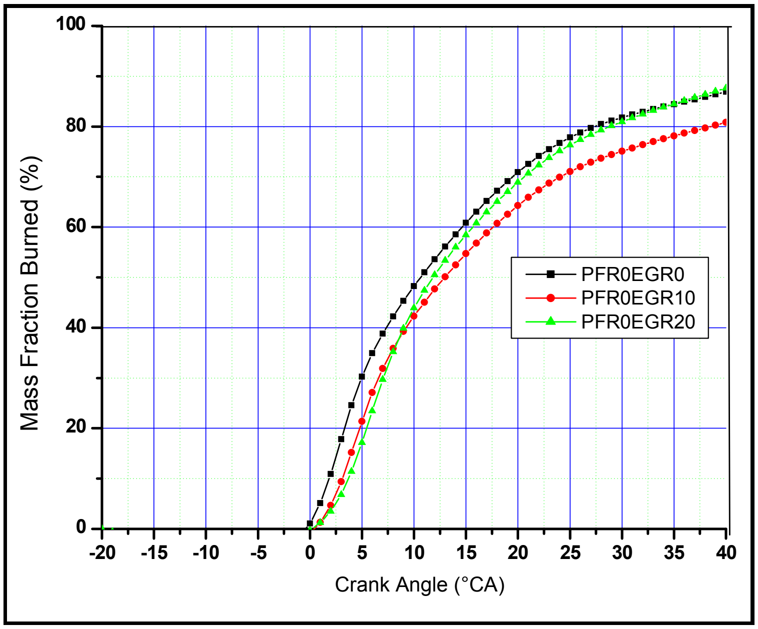


Figure 5.16: Effect of EGR ratio on mass fraction burnt.

5.2.2.2 Effect of EGR on Combustion With 18% PFR

It is observed from the Fig 5.17 shows in-cylinder pressure traces. Increase in EGR reduces peak in-cylinder pressure and increases ignition delay. It is revealed from the later combustion pashing that the peak in-cylinder pressure shifts further towards the expansion stroke. The maximum reduction in peak in-cylinder pressure was found about 6.1% associated with PFR18EGR20 as compared to PFR18EGR0. Maximum rate of pressure rise decreased significantly from 2.62 to 1.83 corresponds to PFR18EGR0 and PFR18EGR20 respectively.

Net heat release rate is shown in Fig. 5.18. The trend depicts that with increase in EGR ratio peak of NHRR decreases due to dilution effect. Maximum premixed combustion duration was found in case of PFR18EGR20, the reason is explained by the cumulative effect of premixed fuel and exhaust gas re-circulation which leads to increase ignition delay period. Thus, combustion phasing and greater portion of combustion shifted towards expansion. There are two mutually countering effect may increase or decreases soot emission. Firstly, low rate of fuel burning in the premixed phase and low temperature combustion increases soot formation in the premixed phase. Secondly, due to availability of oxygenates (ethanol) during diffusion combustion improves soot oxidation. However, due to low temperature soot oxidation rate is low therefore second effect is less dominating.

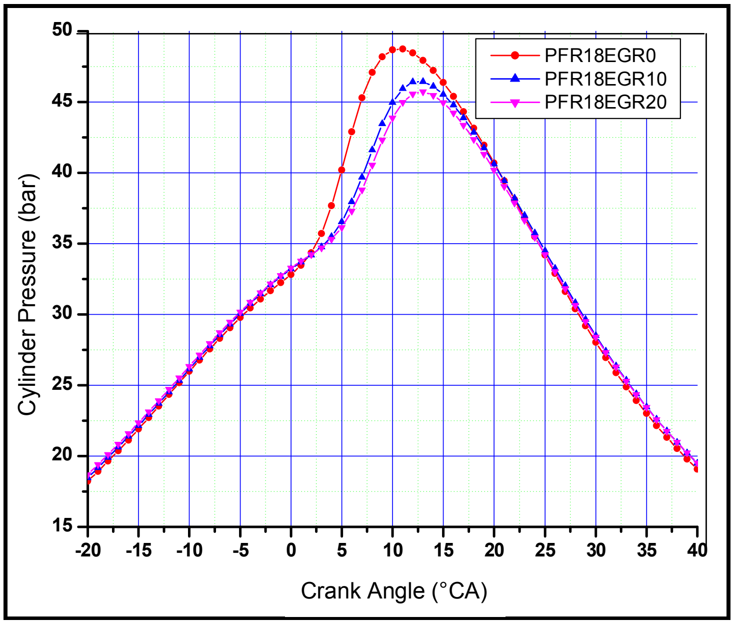


Figure 5.17: Effect of EGR ratio on in-cylinder pressure.

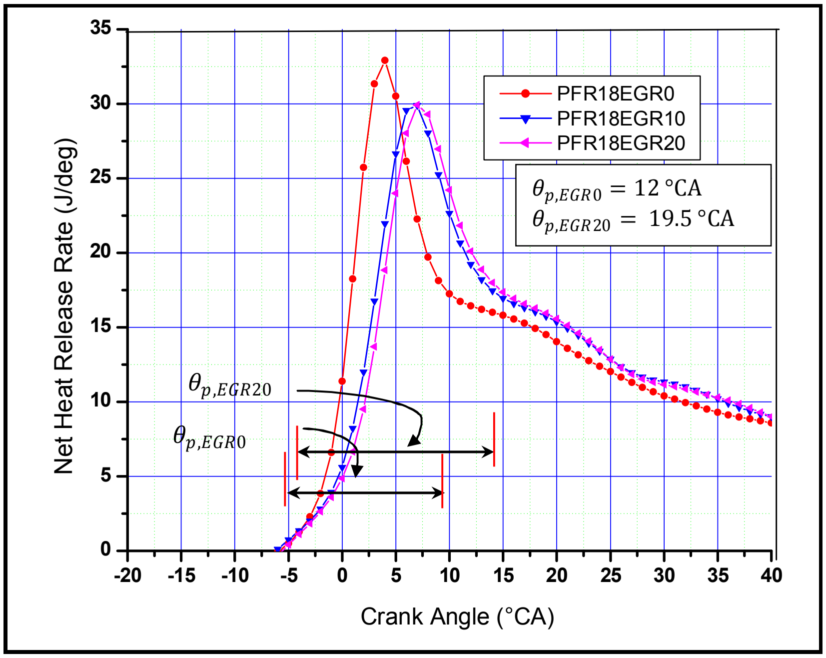


Figure 5.18: Effect of EGR ratio on net heat release rate.

As Fig. 5.19 shows variation of mass fraction burnt with different EGR ratio and constant premixed ratio. Slow burning rate was observed in all the cases except PFR18EGR20 where the last portion of the charge showed slightly faster burning rate. Thus, we compare the portion of charge burnt in premixed phase increases with increase in EGR and it was found only 42% mass fraction burnt in case of PFR18EGR0 and increased to 60% and 55% corresponds to PFR18EGR10 and PFR18EGR20 respectively. Higher rate of EGR also helps in improving vapourization of ethanol in the intake manifold. These values were associated with the position of crank angle at which premixed combustion ended up. Thus, 10 °CA aTDC, 15 °CA aTDC and 15.2 °CA aTDC could be found for PFR18EGR0, PFR18EGR10 and PFR18EGR20 respectively.

It can be conclude that the effect of ethanol and EGR on combustion characteristics. The parameters which are presented here are maximum in-cylinder pressure (P_{max}), rate of pressure rise (RPR), net heat release rate (NHRR) and premixed phase combustion duration (θ_p). Table 5.3 presented below to compare under different operating conditions.

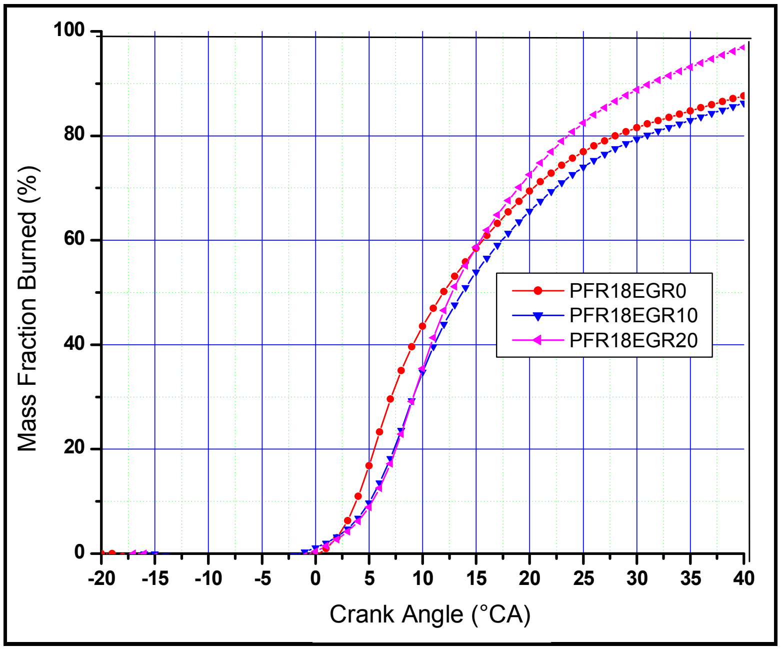


Figure 5.19: Effect of EGR ratio on mass fraction burnt.

Table 5.3: Comparison of combustion variables at BMEP of 2.1 bar

Sr. No.	Abbreviations	P_{max}(bar)	RPR ($\frac{dp}{d\theta}$)	NHRR (J/°CA)	(θ_p) (°CA)
1	PFR0EGR0	51.49	3.41	34.88	12
2	PFR18EGR0	48.74	2.62	32.92	13.75
3	PFR0EGR10	50.56	2.97	31.76	13.5
4	PFR18EGR10	46.43	1.87	29.9	17.5
5	PFR0EGR20	49.72	2.57	30.95	15
6	PFR18EGR20	45.72	1.83	29.92	19.5

5.2.3 Effect of PFR and EGR Ratio on Emissions

Diesel exhaust contains CO_2, H_2O, N_2 and O_2 in thermodynamically significant quantities and CO, UHC, NO_x and soot in thermodynamically insignificant but environmentally harmful quantities. In diesel engines, the combination of the former quantities normally comprise more than 99% of the exhaust, while the latter combination, the pollutants, accounts for less than 1% in quantity.

Fig. 5.20 shows the variation of the nitrogen oxides emissions (NO_x) for all the test conditions. It can be seen that the NO_x emissions with and without premixed ethanol and EGR. NO_x emissions decrease significantly with premixed ethanol and EGR compared to pure diesel combustion. All the data are collected under the same operating conditions. Maximum NO_x emissions correspond to pure diesel combustion (PFR0EGR0) of about 455 ppm. A reduction of more than 50% was achieved by premixing ethanol (PFR18EGR0) when compared against diesel (PFROEGR0).

Similarly, Hayes et al. (Hayes and Savage 1988) also studied fumigation of ethanol at BMEP of 2 bar and 20% of premixed ethanol (200 proof), they reported 40% of reduction in NO_x emissions. Tsang (Tsang and Zhang 2010) reported NO_x emission reduction was more significant at low and medium engine loads under similar conditions. However, similar concentration of premixed ethanol when clubbed with EGR leads to dramatic reduction in NO_x emissions.

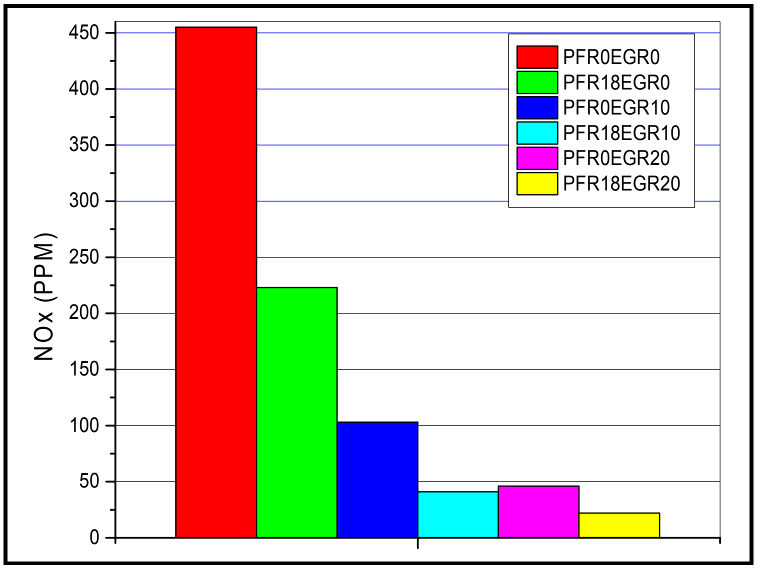


Figure 5.20: Effect of PFR and EGR ratio on nitrogen oxides emissions.

Considerable reduction in NO_x emissions of about 90% and 95% was recorded while operating on intake mixture of PFR18EGR10 and PFR18EGR20 respectively. Ishida et al. (Ishida and Shohei Ueki 2010) also showed similar results using high rate of EGR and ethanol-diesel blends of 30 and 50% (vol. percentage). They report about 70 to 75% reduction in NO_x emission measured at 20% EGR when operating with E-diesel and small variation was reported when pure diesel was used.

It can be seen that without premixing ethanol, NO_x emission could be reduced significantly by using exhaust gas re-circulation. Compared to diesel (PFR0EGR0) about 77% and 89% reduction in NO_x emissions was found in case of PFR0EGR10 and PFR0EGR20 respectively. As previously mentioned, the NO_x emissions from engines are strongly influenced by the amount of heat released in the premixed combustion, the in-cylinder pressure peak and the resultant in-cylinder temperatures. Therefore at low loads simultaneous reduction of NO_x and PM is not observed.

Fig. 5.21 shows the variations in the smoke emissions for diesel with and without EGR and premixed ethanol. It is noticeable that the smoke emissions are not changed significantly except in case of premixed ethanol (PFR18EGR20). Tsang (Tsang and Zhang 2010) examined ethanol fumigation without EGR on 4-cylinder diesel engine, also found that no significant variation in smoke was recorded at BMEP of 2 bar.

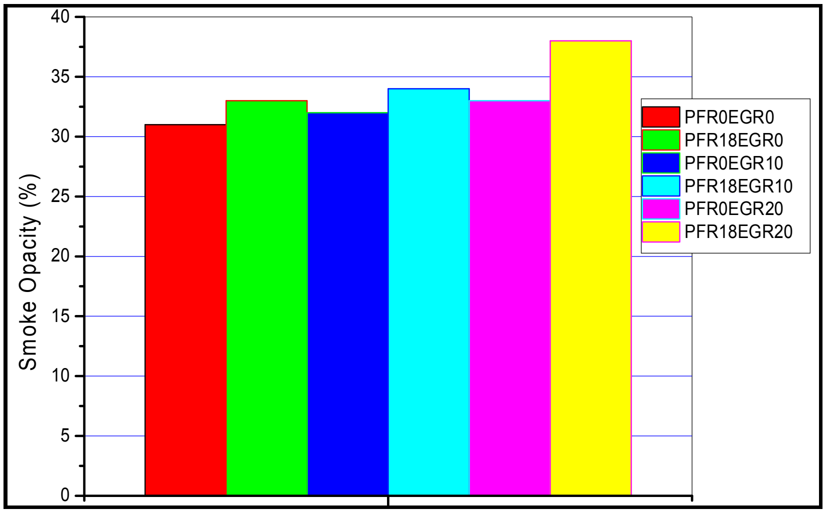


Figure 5.21: Effect of PFR and EGR ratio on smoke emissions.

EGR leads to dilution effect and increases ignition delay period which, in turn, reduces the peak in-cylinder temperature. Therefore, decreased in-cylinder temperature reduces the soot oxidation rate. Nevertheless, at medium load the in-cylinder temperature is already low. Premixed ethanol with EGR produces cumulative effect on the dilution of intake air and fuel rich combustion leads to increase the soot formation, it can be seen in case of PFR18EGR20. However, the effect is not much dominating since total amount of diesel (higher polyaromatic compounds) injected into the cylinder gets reduced due to premixed ethanol, thus lesser soot formation during combustion.

Fig. 5.22 and Fig. 5.23 show variation of UHC and CO emission at different mixture conditions. UHC and CO emissions both are products of incomplete combustion, usually from engine operation at very rich A/F mixtures. It was observed that UHC emission increases with EGR. It was found that UHC increased by 1.43 and 1.56 times corresponding to PFR0EGR10 and PFR0EGR20 compared to PFR0EGR0 respectively. As the exhaust gas re-circulation leads to increase the ignition delay period and cumulative effect of PFR and EGR leads to excessive dilution of air. It was reported that UHC emissions increase at an increasing rate (Merryman and Levy 1975) with ignition delay.

UHC emission increases dramatically with increase in premixed ethanol. In case of PFR18EGR0, it increased by 6.63 times as compared to PFR0EGR0. Dual fuel engine in the premixed conditions are partially similar to spark ignition engine, air-fuel mixture enters into the cylinder and some portion gets trapped in the crevices region and remains unburned during later part of combustion. However, another reason could be the lower in-cylinder gas temperature which might not be able to ignite the unburnt ethanol (last potion comes out from crevices) during expansion stroke. The oxygen concentration further lowered by combined effect of premixed ethanol and EGR results longer ignition delay and lower in-cylinder temperature. Thus, in case of PFR18EGR20, UHC increased by 10 times compared to PFR0EGR0.

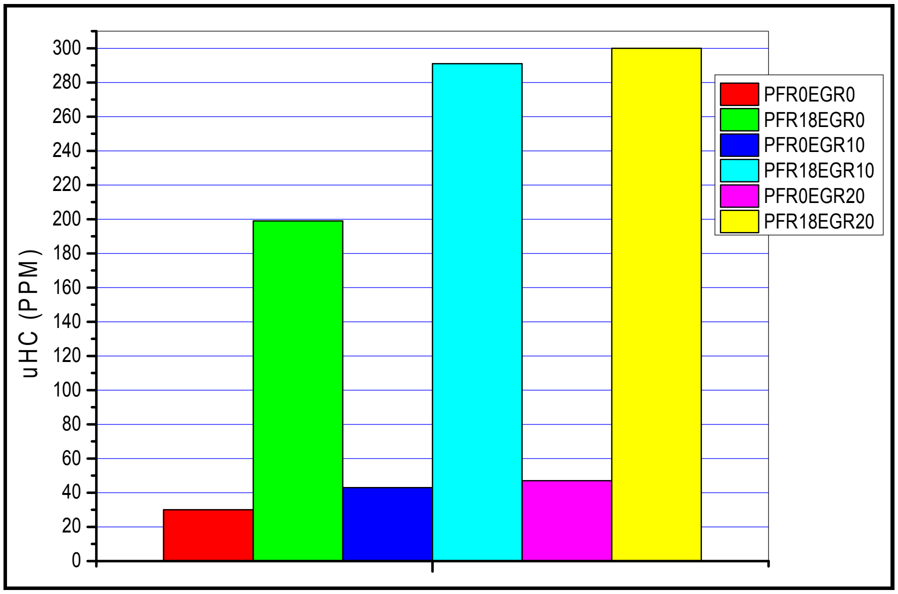


Figure 5.22: Effect of PFR and EGR ratio on unburnt hydrocarbons emissions.

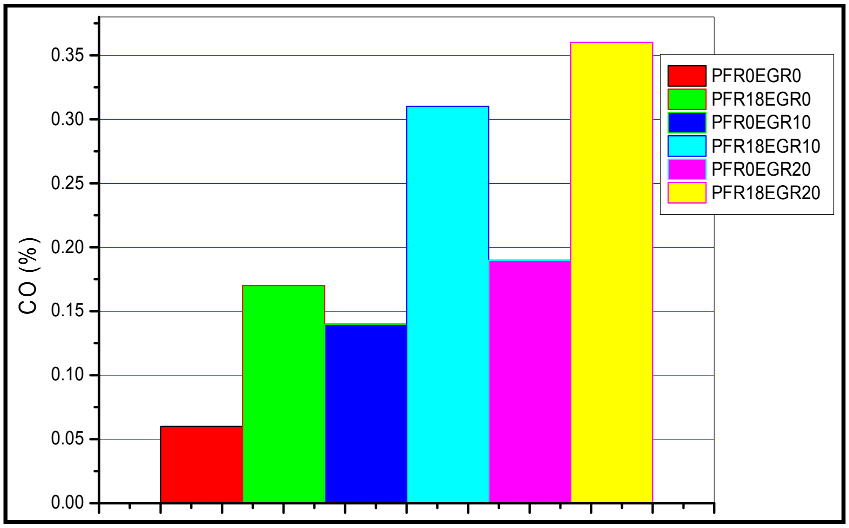


Figure 5.23: Effect of PFR and EGR ratio on carbon monoxide emission.

Similarly, CO emissions increases by 2.83 times and 6 times associated with PFR0EGR10 and PFR18EGR20 as compared to PFR0EGR0. Results obtained for UHC and CO emissions without EGR in this study shows close resemblance with Hayes et al. (Hayes and Savage 1988) reported at BMEP of 2 bar.

5.2.4 Effect of PFR and EGR Ratio on Performance

The effect of PFR on the brake thermal efficiency and BSFC was calculated and presented in the Table 5.4. BSFC increases with PFR and EGR ratio. However, at medium load operation small amount of EGR did not shows any significant adverse effect on BSFC or brake thermal efficiency. Increase in brake specific fuel consumption is the result of lower calorific value of ethanol (26708) as compared to diesel (44208). Brake thermal efficiency associated with PFR affected by several factors. First is the cooling effect produced due to premixed ethanol with air, since latent heat of vapourization of ethanol (840 kJ/kg) is very high compared to diesel (270 kJ/kg). Second reason could be some of the fuel (ethanol) escapes directly into exhaust without taking part in combustion which leads to deteriorate thermal efficiency.

Table 5.4: Performance analysis at BMEP of 2.1 bar.

Fuel	Ethanol (ml)	Diesel (ml)	Power (kW)	q_{EtOH} (kW)	q_{diesel} (kW)	q_{EtOH} + q_{diesel} (kW)	η_{BTH}	BSFC (g/kWhr)
PFR0 EGR0	0	12	1.76	0	7.33	7.33	23.98	339.54
PFR18 EGR0	4	10	1.78	1.40	6.11	7.52	23.66	386.29
PFR0 EGR10	0	12	1.75	0	7.33	7.33	23.85	341.48
PFR18 EGR10	4	10	1.78	1.40	6.11	7.52	23.66	386.29
PFR0 EGR20	0	12	1.76	0	7.33	7.33	23.98	339.54
PFR18 EGR20	4	10	1.76	1.40	6.11	7.52	23.40	390.68

However, to understand more comprehensively, the effect of EGR ratio on brake thermal efficiency a comparison is been made with no EGR condition to 10% and 20% EGR ratio. Table 5.5 shows variation in brake thermal efficiency with respect to no EGR conditions. EGR along with premixed ethanol negatively affects brake thermal efficiency. This is due to considerable reduction in in-cylinder pressure and temperature.

Table 5.5 : Variation in brake thermal efficiency with EGR ratio.

	PFR0EGR0	% Change	PFR18EGR0	% Change
	23.98			
PFR0EGR10	23.85	-0.54211843		
PFR0EGR20	23.98	0		
			23.66	
PFR18EGR10			23.66	0
PFR18EGR20			23.4	-1.0989011

Another method is to compare different EGR and PFR combination with no PFR condition. Table 5.6 shows a negative BTE change rate, it indicates a decreased in BTE at 18% premixed ratio of ethanol. All the test conditions showed that BTE change rates were negative. Maximum reduction was measured in case of PFR18EGR20. It was found that on BTE no effect or negative effect also reported by

Hayes et al. (Hayes and Savage 1988) and (Tsang and Zhang 2010) with premixed ethanol and without EGR.

Table 5.6 : Change in brake thermal efficiency with no PFR condition.

	PFR0 EGR0	% Change in BTE	PFR0 EGR10	% Change in BTE	PFR0 EGR20	% Change in BTE
	23.98					
PFR18EGR0	23.66	-1.3344				
			23.85			
PFR18EGR10			23.66	-0.7966		
					23.98	
PFR18EGR20					23.4	-2.4187

5.3 INFLUENCE OF PFR AND EGR AT THREE-FOURTH LOAD

Similar to previous study, experiments were performed at the engine speed of 1500 revolutions/min and engine load corresponds to 75% of engine load or BMEP of 3.1 bar. At this engine load, experiments were first carried out using diesel fuel alone to build up a data-base for comparison to those obtained with premixed ethanol. Test matrix is given in Table 5.7.

The new fuel injection mechanism provides full flexibility to change the injection timing and quantity of fuel injected at each load. However to achieve consistent results compared to other loads, injection timing for diesel and premixed fuel were kept constant. It is also important to take note here 15% of premixed fuel ratio corresponds to 4 ml of ethanol fuel supplied per minute.

Table 5.7: Test matrix at BMEP of 3.1 bar.

Sl. No.	Abbreviations	Description
1	PFR0EGR0	Premixed fuel ratio = 0% and Exhaust gas re-circulation = 0%
2	PFR15EGR0	Premixed fuel ratio = 15% and Exhaust gas re-circulation = 0%
3	PFR0EGR10	Premixed fuel ratio = 0% and Exhaust gas re-circulation= 10%
4	PFR15EGR10	Premixed fuel ratio = 15% and Exhaust gas re-circulation = 10%
5	PFR0EGR20	Premixed fuel ratio = 0% and Exhaust gas re-circulation = 20%
6	PFR15EGR20	Premixed fuel ratio = 15% and Exhaust gas re-circulation = 20%

5.3.1 Effect of PFR on Combustion

As discussed in previous chapter, the complete study has two facets: one examining the combustion characteristics at constant EGR ratio and another is varying EGR ratio and keeping the premixed fuel ratio as a constant.

5.3.1.1 Effect of PFR on Combustion without EGR

Figure 5.24 compares the peak in-cylinder pressure for pure diesel (PFR0EGR0) and premixed ethanol (PFR15EGR0). The plot is showing values obtained after averaging 50 consecutive cycles. The maximum in-cylinder pressure corresponds to PFR0EGR0 of about 56.54 bar, it decreased to 53.66 bar associated with PFR15EGR0 (5% reduction). The reduction in pressure along with the later combustion phasing was observed due to premixed ethanol. Tsang et al. (Tsang and Zhang 2010) and Hayes et al. (Hayes and Savage 1988) also showed peak in-cylinder pressure reduction with premixed ethanol at similar engine load conditions.

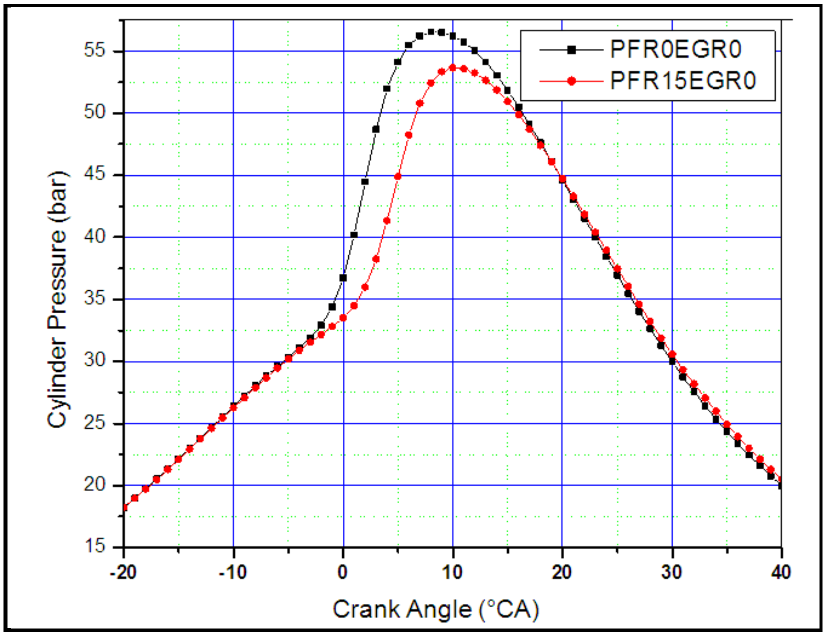


Figure 5.24: Effect of premixed fuel ratio on in-cylinder pressure

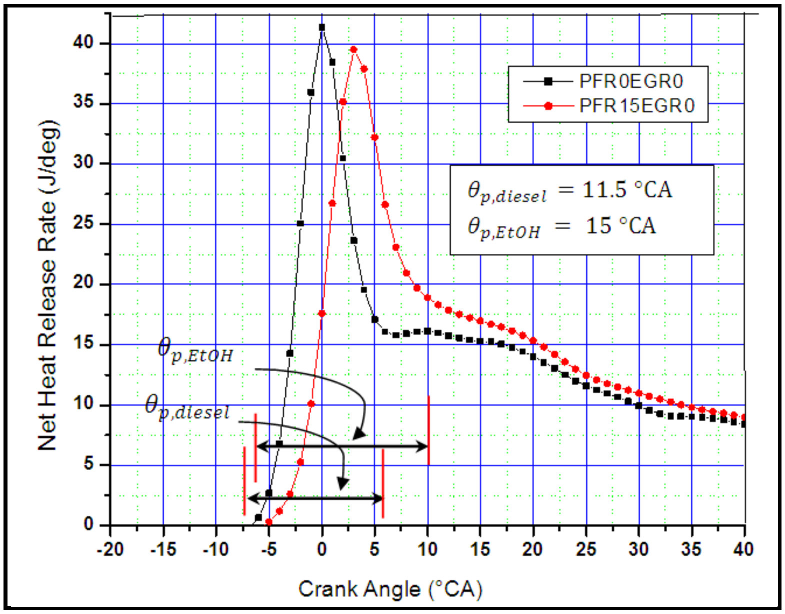


Figure 5.25: Effect of premixed fuel ratio on net heat release rate

Fig. 5.26 shows the influence of premixed fuel ratio on mass fraction burnt, as discussed, this is based on cylinder pressure increment ratio. Only small change in the combustion duration was observed by using premixed ethanol.

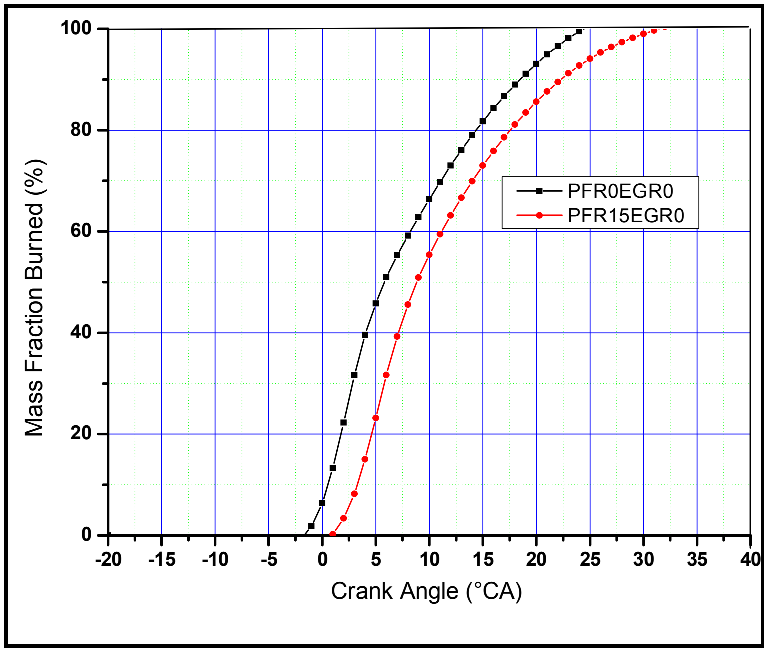


Figure 5.26: Effect of premixed fuel ratio on mass fraction burnt

Premixed phase of combustion continued till 5 °CA aTDC and 10 °CA aTDC which corresponds to PFR0EGR0 and PFR15EGR0 respectively. It shows mass fraction burnt increases while using mixture PFR15EGR0 up to 57.5% in premixed combustion phase, whereas, it was only 42.5% in case of PFR0EGR0. Therefore, relatively higher premixed phase combustion duration was the result of higher ignition delay period in case of PFR15EGR0.

5.3.1.2 Effect of PFR on Combustion at 10% EGR Ratio

As with the use of EGR, ignition delay increases further, and considerable reduction in pressure was found. Fig. 5.27 shows the percentage change in in-cylinder pressure associated with PFR15EGR10 was about 7% compared to PFR0EGR10. This is due to the combined effect of EGR and premixed of ethanol. Thus, both were responsible for the dilution of intake air, higher specific heat of EGR constituents which reduces the peak cylinder pressure and temperature. Also, higher heat of vaporization and low cetane number (CN) further retarded the combustion. Peak of in-cylinder pressure were obtained at 12 °CA aTDC as compared to 9 °CA aTDC associated with PFR15EGR10 and PFR0EGR10, it shows increase in ignition delay period.

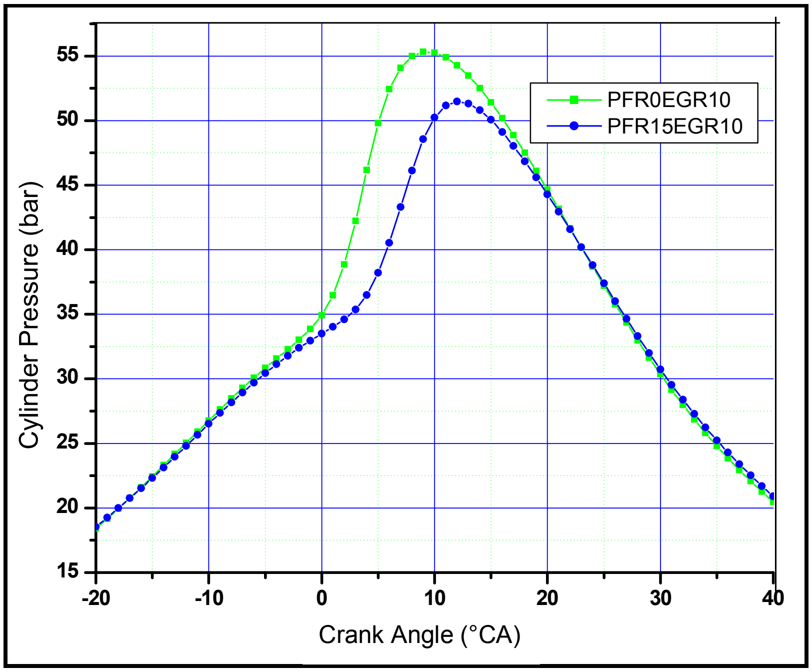


Figure 5.27: Effect of premixed fuel ratio on in-cylinder pressure.

Fig. 5.28 shows the variation of net heat release rate for with and without premixed ethanol. Peak NHRR associated with PFR15EGR10 decreased by 9.4% compared to PFR0EGR10, this was due to higher ignition delay of premixed ethanol. It could be found that compared to pure diesel combustion and with EGR (PFR0EGR10), premixed phase combustion duration increases in case of PFR15EGR10. It was found that about 13 °CA corresponds to PFR0EGR10, whereas in case of PFR15EGR10 it was increased to 20 °CA. Combustion phasing was evident from shift of peak NHRR from at 362 °CA position to 366 °CA associated with PFR0EGR10 and PFR15EGR10 respectively.

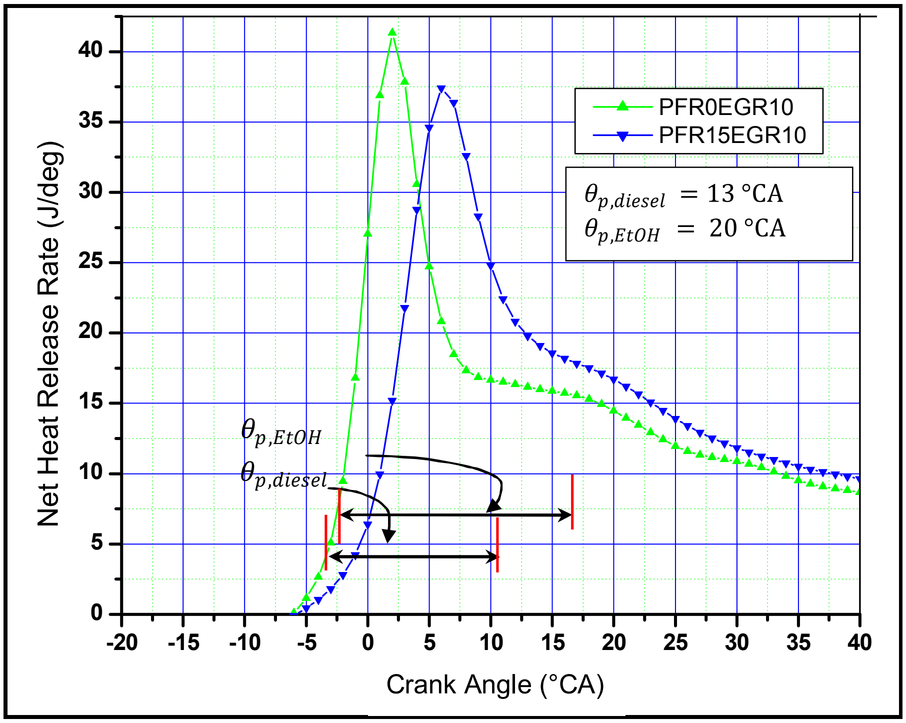


Figure 5.28: Effect of premixed fuel ratio on net heat release rate.

Fig. 5.29 shows mass fraction burnt at different intake mixture condition. The slope of mass fraction burnt curve indicates slow burning with PFR15EGR10 compared to PFR0EGR10. Similarly, it can be concluded that mass fraction burnt correspond to premixed phase combustion in case of PFR0EGR10 was about 45% (7.5 °CA aTDC) whereas it was increased to 68% (15 °CA aTDC) in case of intake mixture PFR15EGR10.

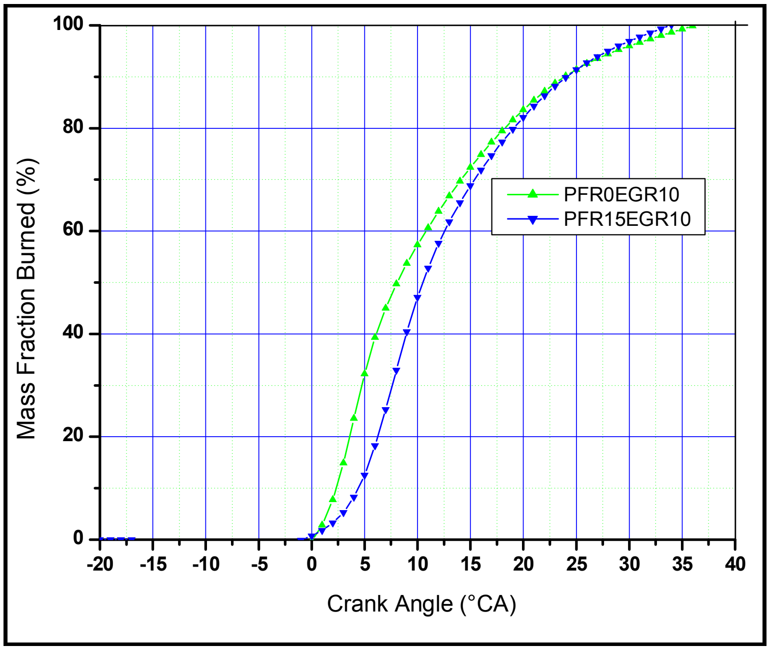


Figure 5.29: Effect of premixed fuel ratio on mass fraction burnt.

5.3.1.3 Effect of PFR on Combustion at 20% EGR Ratio

Fig. 5.30 shows in-cylinder pressure traces obtained for PFR0EGR20 and PFR15EGR20. It shows substantial reduction in peak in-cylinder pressure associated with PFR15EGR20 compared to PFR0EGR20. Peak in-cylinder pressure corresponding to PFR0EGR20 was about 53.7 bar whereas for PFR15EGR20 it reduced to 48.98 bar. There was about 8.7% reduction in peak pressure which was considered as a significant reduction in peak in-cylinder pressure.

Fig. 5.31 shows variation of net heat release rate for with and without premixed ethanol. The peak net heat release rate was about 39.26 J/°CA obtained for PFR0EGR20 and in case of PFR15EGR20 it decreased to about 36.57 J/°CA. Therefore, reduction in maximum heat release rate was about 6.8%. Later combustion phasing indicating increase in ignition delay and subsequently increase in premixed phase combustion duration. Premixed phase combustion duration was found about 15 °CA and 22.5 °CA associated with PFR0EGR20 and PFR15EGR20 respectively. Thus, significant rise in premixed phase combustion duration was the result of combined effect of premixed fuel and EGR ratio.

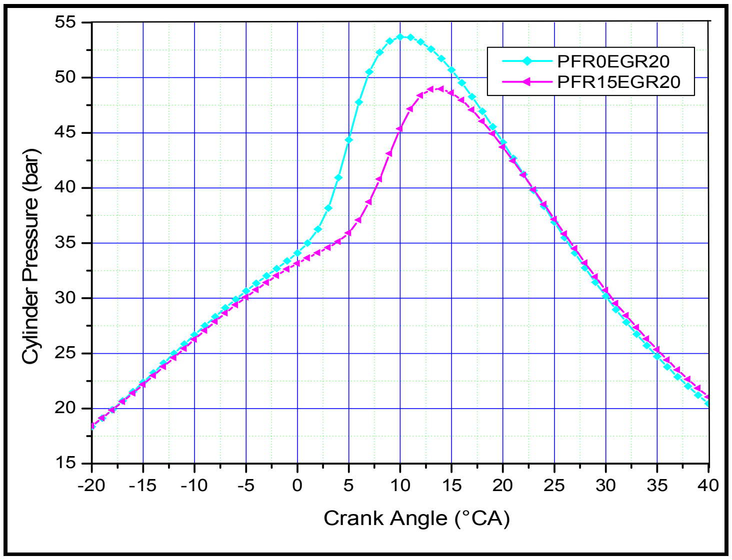


Figure 5.30: Effect of premixed fuel ratio on in-cylinder pressure.

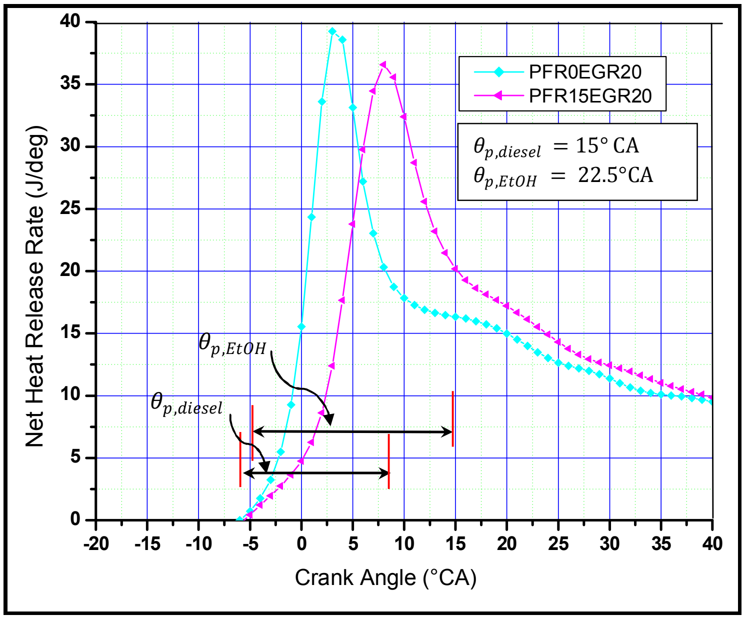


Figure 5.31: Effect of premixed fuel ratio on net heat release rate.

Fig. 5.32 shows mass fraction burnt for with and without premixed ethanol. With reference to Fig 5.31, premixed phase combustion continued till 10 °CA aTDC and 17.5 °CA aTDC associated with PFR0EGR20 and PFR15EGR20 respectively. Therefore, corresponding mass fraction burnt during premixed phase was found to be about 60% and 82.5%, which showed significant increase in premixed combustion. Thus, most of the fuel burnt in premixed phase in case of PFR15EGR20.

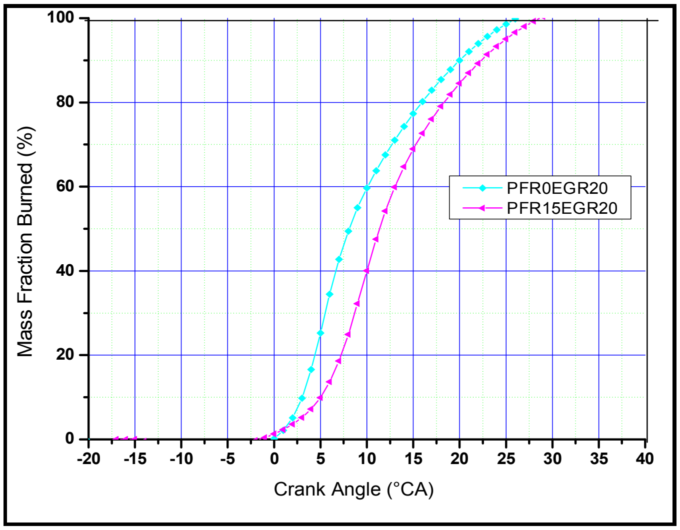


Figure 5.32: Effect of premixed fuel ratio on mass fraction burnt.

5.3.2 Effect of EGR Ratio on Combustion

To understand effect of EGR at constant premixed ratio of fuel, the in-cylinder pressure, net heat release rate and mass fraction burnt were studied. Analysis of this parameter helps to examine relative changes in the combustion characteristics to find best strategy to control combustion, emission and optimize performance.

5.3.2.1 Effect of EGR Ratio on Combustion without PFR

Fig. 5.33 shows in-cylinder pressure traces for diesel combustion at different EGR ratio. Peak in-cylinder pressure decreases with increase in EGR ratio, also it shifts further after top dead position or towards expansion stroke. Maximum reduction in peak in-cylinder pressure was associated with PFR0EGR20 of about 5% as compared to PFR0EGR0. The diluent caused an increase in ignition delay and shift the location of start of combustion which leads to shift the entire combustion process.

Fig. 5.34 shows net heat release rate at EGR ratio of 0%, 10% and 20% while engine was operating on pure diesel with different EGR ratio (PFR0EGR, PFR0EGR10 and PFR0EGR20). Depending on in-cylinder pressure and temperature history, premixed phase of combustion may show fast burning and rapid heat release. However, at moderate engine load lean combustion did not shows rapid heat release and only small variation in peak NHRR was found. Premixed phase combustion duration increases from 11.5 °CA to 15 °CA associated with intake mixture PFR0EGR0 and

PFR0EGR20. Further, longer ignition delay period provide more time for the fuel to penetrate which increases amounts of gases in the flame envelope, resulting into lower the combustion temperature and in turn lower NO_x formation.

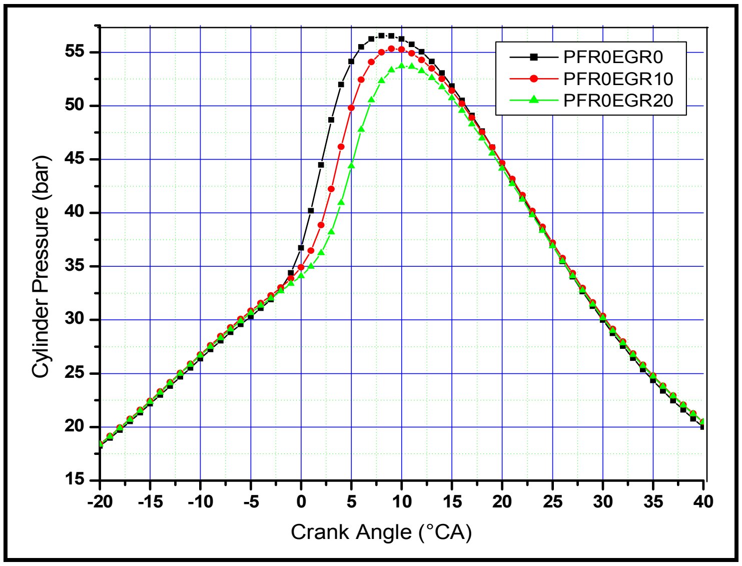


Figure 5.33: Effect of EGR ratio on in-cylinder pressure.

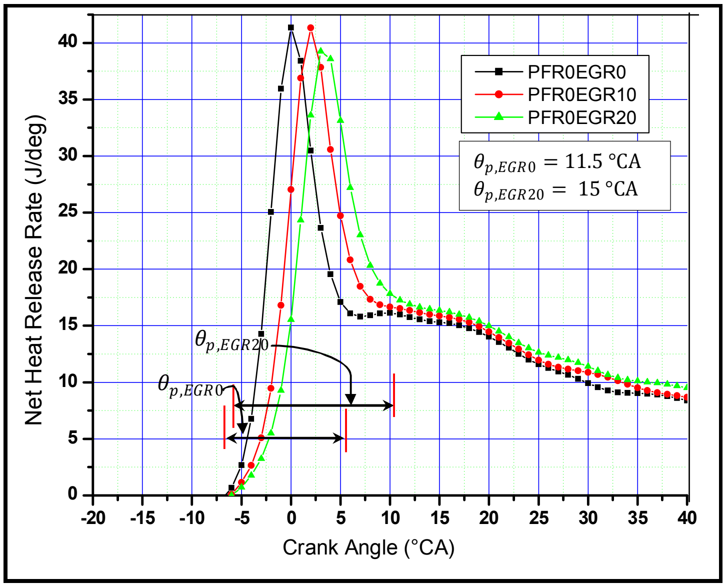


Figure 5.34: Effect of EGR ratio on net heat release rate.

Fig. 5.35 shows mass fraction burnt plot for different EGR ratio. This with reference to Fig. 5.34, it is observed that premixed phase combustion continued till 5 °CA aTDC, 7.5 °CA aTDC and 10 °CA aTDC corresponds to PFR0EGR0, PFR0EGR10 and PFR0EGR20 respectively. Thus, respective mass fraction burnt were 45%, 50% and 60% determined from Fig. 5.35.

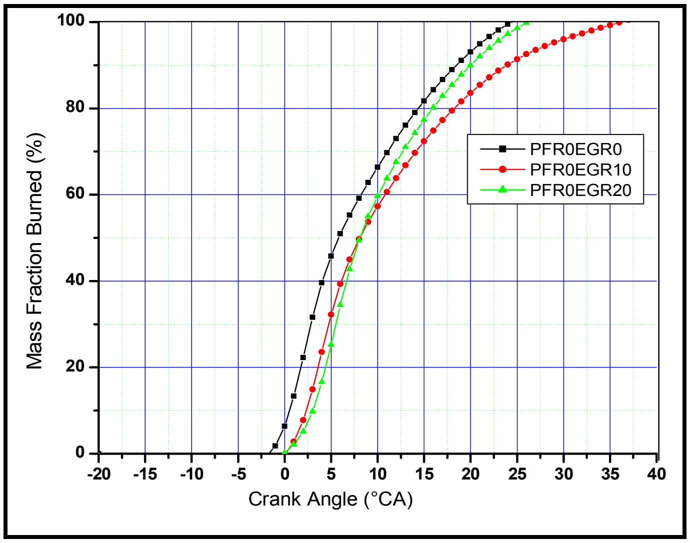


Figure 5.35: Effect of EGR ratio on mass fraction burnt.

5.3.2.2 Effect of EGR Ratio on Combustion with 15% PFR

Fig. 5.36 shows in-cylinder pressure traces, it is found that increase in EGR ratio reduces peak in-cylinder pressure and increases ignition delay period. In case of PFR15EGR0 it was found about 53.66 bar and it reduced to about 48.98 bar corresponding to PFR15EGR20. So, maximum reduction in peak in-cylinder pressure was recorded about 8.7%.

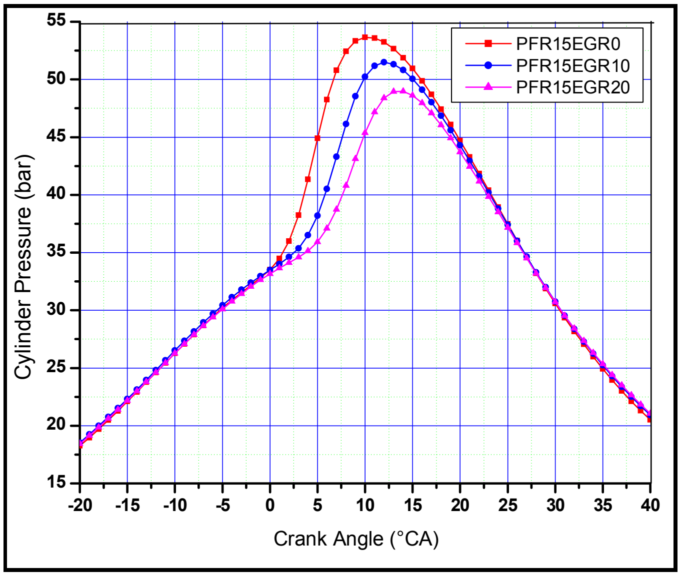


Figure 5.36: Effect of EGR ratio on in-cylinder pressure.

Later combustion phasing is evident from the shift of in-cylinder pressure peaks. The maximum shift in peak in-cylinder pressure was associated with PFR15EGR20 of about by 4 °CA on p-θ curve.

Fig. 5.37 shows variation in NHRR with EGR ratio. The effect of increase in ignition delay with EGR ratio is evident from shift of peak NHRR. Premixed phase combustion duration increases from 15 °CA to 22.5 °CA as the EGR ratio increases from 0% to 20%. However, intake mixture PFR15EGR20 showed maximum reduction in NHRR of about 7.3% as compared to PFR0EGR0. Therefore, in this case in-cylinder pressure reduction is considerable higher compared to without premixed ethanol.

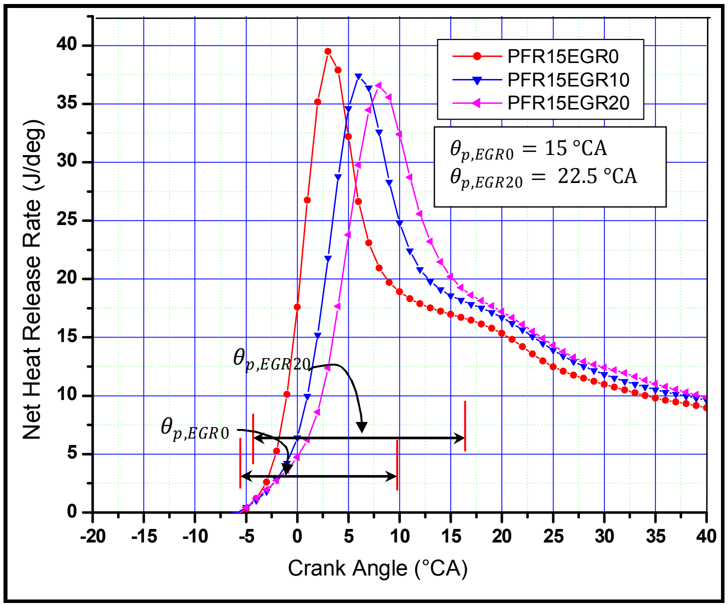


Figure 5.37: Effect of EGR ratio on net heat release rate

Fig. 5.38 shows mass fraction burnt varied little in diffusion combustion whereas, significant variation was observed in premixed phase it could be concluded from significant rise in later part. From Fig. 5.37 it was found that extent of premixed phase combustion associated with PFR15EGR0, PFR15EGR10 and PFR15EGR20 continued till 10 °CA aTDC, 15 °CA aTDC and 17.5 °CA aTDC respectively. Thus, corresponding mass fraction burnt during premixed phase were 55%, 68% and 79% for three mixtures in the respective order.

It can be conclude that the effect of PFR and EGR on combustion characteristics at BMEP of 3.1 bar. Four parameters are compared in Table 5.8.

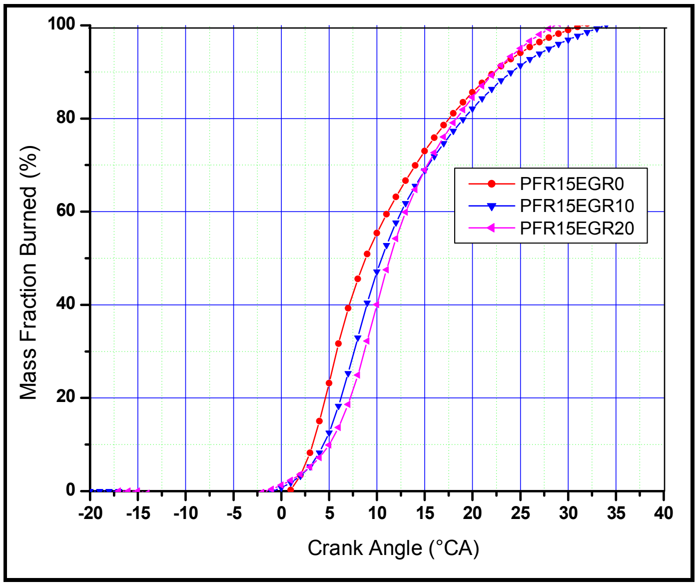


Figure 5.38: Effect of EGR ratio on mass fraction burnt

Table 5.8: Comparison of combustion variables at BMEP of 3.1 bar.

Sr. No.	Abbreviations	P_{max}(bar)	RPR ($\frac{dp}{d\theta}$) (bar/°CA)	NHRR (J/°CA)	(θ_p) (°CA)
1	PFR0EGR0	56.54	4.41	41.36	11.5
2	PFR15EGR0	53.66	3.61	39.49	15
3	PFR0EGR10	55.33	3.88	41.32	13
4	PFR15EGR10	51.48	2.83	37.42	20
5	PFR0EGR20	53.7	3.56	39.26	15
6	PFR15EGR20	48.98	2.24	36.57	22.5

5.3.3 Effect of PFR and EGR Ratio on Emissions

Fig. 5.39 shows NO_x emission for pure diesel combustion and premixed ethanol with and without EGR. Variation in emissions shows dramatic reduction in NO_x emissions could be possible by using EGR alone. Reduction in NO_x emissions is achieved by using PFR0EGR10 and PFR0EGR20 are found about 82% and 92% respectively compared to PFR0EGR0. The reduction in NO_x emission of about 90% and 95% is achieved in case of PFR15EGR10 and PFR15EGR20 compared to PFR0EGR0. Nevertheless, premixed ethanol alone (PFR15EGR0) shows significant reduction in NO_x emissions of about 33% compared to PFR0EGR0.

Fig. 5.40 shows the variations in smoke opacity for different intake mixture condition. As the fuel-air mixing can significantly control the smoke emissions, premixed fuel increases ignition delay period that leads to lower the oxygen-depleted zones and soot concentration is reduced. Smoke opacity increases slightly with increase in EGR while engine is operating without premixed ethanol. Tsang et al. (Tsang and Zhang 2010) and Bhupendra (Bhupendra Singh Chauhana and Naveen Kumar 2011) reported similar results.

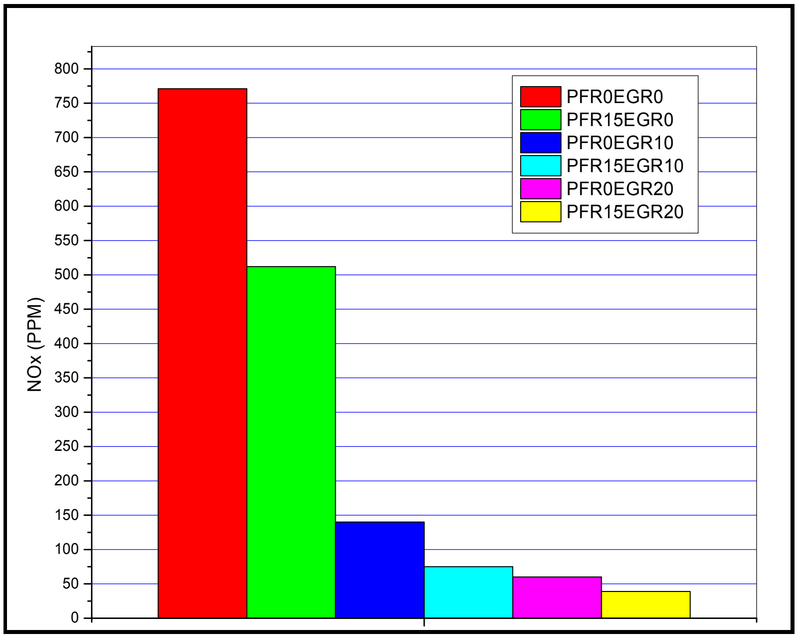


Figure 5.39: Effect of PFR and EGR ratio on nitrogen oxides emissions.

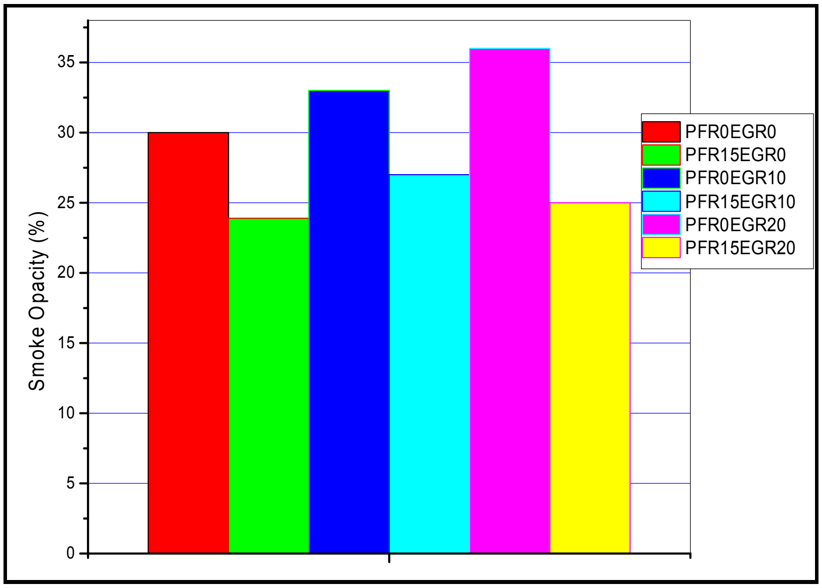


Figure 5.40: Effect of PFR and EGR ratio on smoke emissions.

Compared to pure diesel (PFR0EGR0) combustion premixed ethanol (PFR15EGR0) has showed strong potential to successfully cut down smoke opacity by 20%. The reason could be explained from the combustion of pure diesel and premixed mode. Firstly, from the NHRR analysis it has been found that diesel has relatively long diffusion combustion phase, at the same time this phase lacks sufficient oxygen availability. Whereas, in case of PFR15EGR0, increase in premixed combustion phase and higher oxygen content during premixed and diffusion combustion led to reduce smoke or PM emissions. Secondly, diesel has higher polyaromatic compounds compared to ethanol which results in higher smoke. Premixed ethanol with 10% EGR (PFR15EGR10) also does not show any significant changes in emission. Increase in EGR with premixed ethanol (PFR15EGR20) decreases smoke emissions marginally.

Ishida et al. (Ishida and Shohei Ueki 2010) showed with increase in EGR ratio associated with higher premixed ethanol lead to reduce smoke emission. Reduction is also due to significant increase in premixed combustion phase while using PFR15EGR20. So, simultaneous reduction of NO_x and PM is clearly visible while engine is running is at 75% of rated engine load.

Fig. 5.41 shows UHC emission increases with premixed ethanol and EGR. Compared to PFR0EGR0, UHC increased by 1.8 and 2.3 times when engine is running on intake mixture PFR0EGR10 and PFR0EGR20 respectively.

Premixing of ethanol showed considerable rise in the UHC emissions. Unburned HC increased by 4, 5.8 and 7.6 times associated with PFR15EGR0, PFR15EGR10 and PFR15EGR20 as compared to PFR0EGR0 respectively. At medium load condition, similar levels of UHC emissions was observed by Bhupendra et al (Bhupendra Singh Chauhana and Naveen Kumar 2011).

Fig. 5.42 shows variation of CO emissions. Carbon monoxide (CO) emission also increases with premixed ethanol and EGR similar to UHC. CO emission increases due to incomplete combustion of fuel and steep increase is found when premixed ethanol combusted in presence of EGR. The dilution of intake air increased due to combined effect of premixed ethanol (displaces some amount of air) and EGR (add CO_2). Therefore, maximum CO emission is recorded in case of PFR15EGR20.

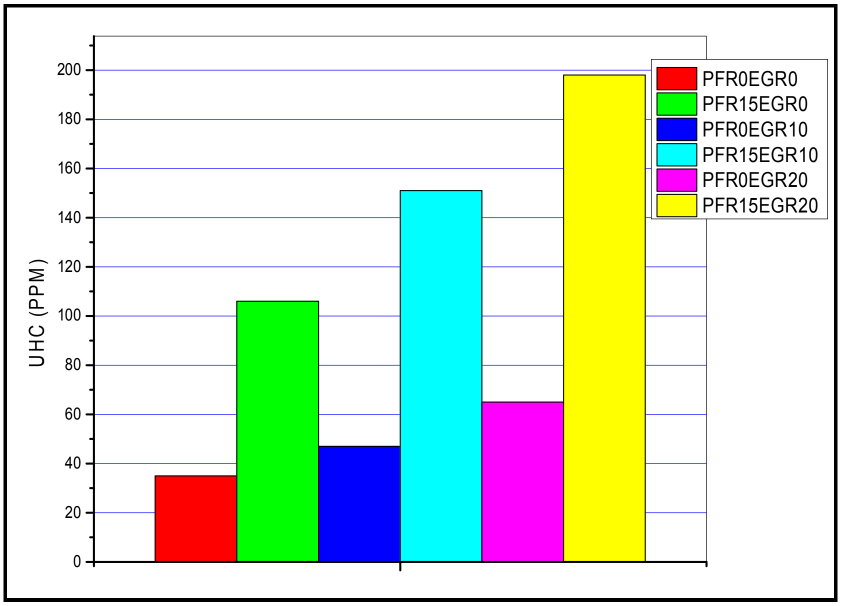


Figure 5.41: Effect of PFR and EGR ratio on unburnt hydrocarbon emissions.

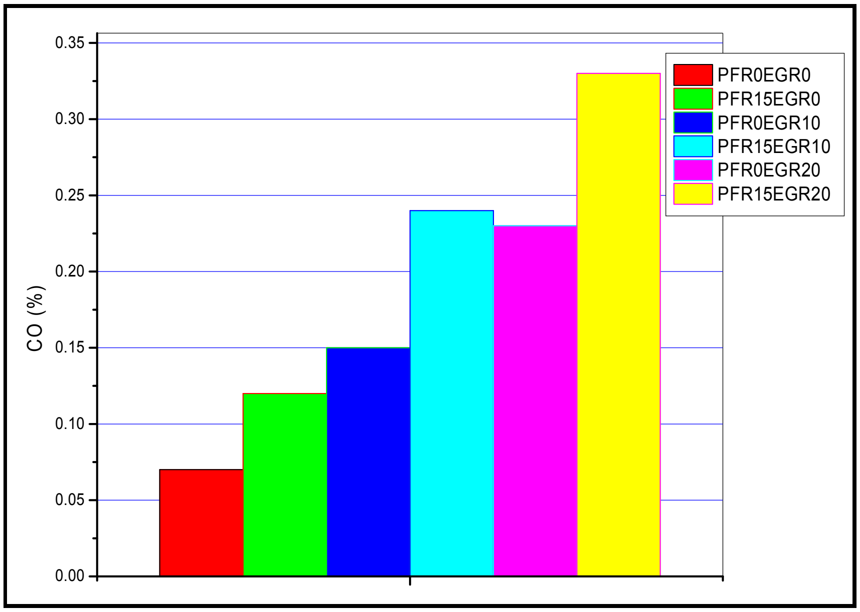


Figure 5.42: Effect of PFR and EGR ratio on carbon monoxide emission.

The rate at which CO increases is higher in case of premixed ethanol with EGR (PFR15EGR10 and PFR15EGR20) compared to pure diesel with EGR (PFR0EGR10 and PFR0EGR20). In case of PFR0EGR10 and PFR0EGR20, CO emission increased by about 2 and 3.3 times, moreover PFR15EGR10 and PFR15EGR20 showed significant rise of about 3.4 and 4.7 times respectively compared to PFR0EGR0.

5.3.4 Effect of PFR and EGR Ratio on Performance

Table 5.9 shows brake specific fuel consumption, energy input and brake thermal efficiency at different intake mixture condition. BSFC increases with premixed ethanol and in-significant change is observed with EGR. However, premixed ethanol along with EGR (PFR15EGR10 and PFR15EGR20) shows dramatic increase in BSFC as compared to pure diesel combustion (PFR0EGR0).

Table 5.9: Performance analysis at BMEP of 3.1 bar.

Fuel	EtOH (ml)	Diesel (ml)	Power (kW)	q_{EtOH} (kW)	q_{Diesel} (kW)	q_{EtOH} + q_{Diesel}(kW)	η_{bth}	BSFC (g/kWhr)
PFR0 EGR0	0	15	2.6	0	9.17	9.17	28.35	287.3
PFR15 EGR0	4	12	2.59	1.4	7.33	8.74	29.62	303.9
PFR0 EGR10	0	15	2.61	0	9.17	9.17	28.46	286.2
PFR15 EGR10	4	13	2.57	1.4	7.94	9.35	27.47	325.7
PFR0 EGR20	0	15	2.59	0	9.17	9.17	28.24	288.4
PFR15 EGR20	4	13	2.59	1.4	7.94	9.35	27.69	323.2

To obtain more appropriate conclusion, it is better to find out percentage change in brake thermal efficiency.

From Table 5.10 it is found that in case of PFR0EGR10 and PFR0EGR20 small variation in thermal efficiency could be obtained compared to PFR0EGR0. However, increase in EGR with premixed fuel (PFR15EGR20) lead to significant reduction in brake thermal efficiency. This behaviour could be seen from the dramatic increase in UHC emissions due to significant reduction in in-cylinder performance together with prolonged ignition delay period. It is also observed that in case of PFR15EGR20 short combustion duration can also be one of the causes.

Table 5.10 : Comparison of change in BTE based on PFR without EGR.

	PFR0EGR0	**% Change**	**PFR15EGR0**	**% Change**
	28.35			
PFR0EGR10	28.46	0.38		
PFR0EGR20	28.24	-0.39		
			29.62	
PFR15EGR10			27.47	-7.2
PFR15EGR20			27.69	-6.51

Table 5.11 shows variation in BTE without PFR. Intake mixture PFR15EGR0 showed small increase in efficiency, as brake thermal efficiency increased from 28.35 to 29.62. This corresponds to about 4.48% increase in BTE. Reported work showed close resemblance with results. Chen et al. (Chen and D. Gussert 1981) reported BTE varies in the positive and negative direction at high and low engine loads respectively. Qudais et al. (Abu-Qudais and Haddad 2000) also shows 3.6% increase in BTE at 15% ethanol-diesel blends.

However, premixed fuel with EGR (PFR15EGR10 and PFR15EGR20) again showed negative effect of brake thermal efficiency. As PFR0EGR20 and PFR15EGR20 does not show much variation (-1.95%), premixed fuel helps to provide oxygen and better combustion could be obtained with reduction in smoke.

Table 5.11 : Comparison of change in BTE based on EGR ratio without PFR.

	PFR0 EGR0	**% Change**	**PFR0 EGR10**	**% Change**	**PFR0 EGR20**	**% Change**
	28.35					
PFR15EGR0	29.62	4.48				
			28.46			
PFR15EGR10			27.47	-3.48		
					28.24	
PFR15EGR20					27.67	-1.95

5.4 INFLUENCE OF PFR AND EGR RATIO AT FULL LOAD

This corresponds to 100% engine load or 4.2 bar BMEP (3.5 kW) while engine is operating under normal diesel combustion and PCCI mode. Rest conditions are kept similar as mentioned in the previous sections. The test matrix is mentioned below in Table 5.12

Table 5.12: Test matrix at BMEP of 4.2 bar

Sr. No.	Abbreviations	Description
1	PFR0EGR0	Premixed fuel ratio = 0% and Exhaust gas re-circulation = 0%
2	PFR13EGR0	Premixed fuel ratio = 13% and Exhaust gas re-circulation = 0%
3	PFR20EGR0	Premixed fuel ratio = 20% and Exhaust gas re-circulation = 0%
4	PFR0EGR10	Premixed fuel ratio = 0% and Exhaust gas re-circulation = 10%
5	PFR13EGR10	Premixed fuel ratio = 13% and Exhaust gas re-circulation = 10%
6	PFR20EGR10	Premixed fuel ratio = 20% and Exhaust gas re-circulation = 10%
7	PFR0EGR20	Premixed fuel ratio = 0% and Exhaust gas re-circulation = 20%
8	PFR13EGR20	Premixed fuel ratio = 13% and Exhaust gas re-circulation = 20%
9	PFR20EGR20	Premixed fuel ratio = 20% and Exhaust gas re-circulation = 20%

5.4.1 Effect of PFR on Combustion

In this section in-cylinder pressure, net heat release, mass fraction burnt and ignition delay period are studied with respect to change in premixed fuel at fixed EGR ratio. At higher engine loads increased fuel consumption allowed to obtain two different premixed fuel ratios (13% and 20%) by injecting 4 ml and 6ml ethanol per minute respectively.

5.4.1.1 Effect of PFR on Combustion without EGR

Figure 5.43 compares the peak in-cylinder pressure for PFR0EGR0, PFR13EGR0 and PFR20EGR0 fuel. The plot is showing average values obtained after 50 consecutive cycles. The maximum in-cylinder pressure during pure diesel combustion (PFR0EGR0) was about 62.67 bar, at PFR13EGR0 and PFR20EGR0 it increases slightly and it reaches to about 63.03 and 63.08 bar, i.e., the percentage increase are 0.57 and 0.65 respectively. However, combustion phasing was observed and the peak in-cylinder pressure for PFR13EGR0 and PFR20EGR0 are obtained at 8 °CA aTDC and 9 °CA aTDC respectively. Hayes et al. (Hayes and Savage 1988) also reported increase in-cylinder pressure as the premixed ratio increases at BMEP of 5 bar.

Fig 5.44 shows the pattern of net heat release rate for different premixed fuel ratio without EGR. An expected trend is observed that the peak net heat release rate increases after introducing ethanol, the maximum net heat release rate was observed at PFR20EGR0, which is about 49.92 J/°CA, maximum percentage change was about 14.12% compared to PFR0EGR0.

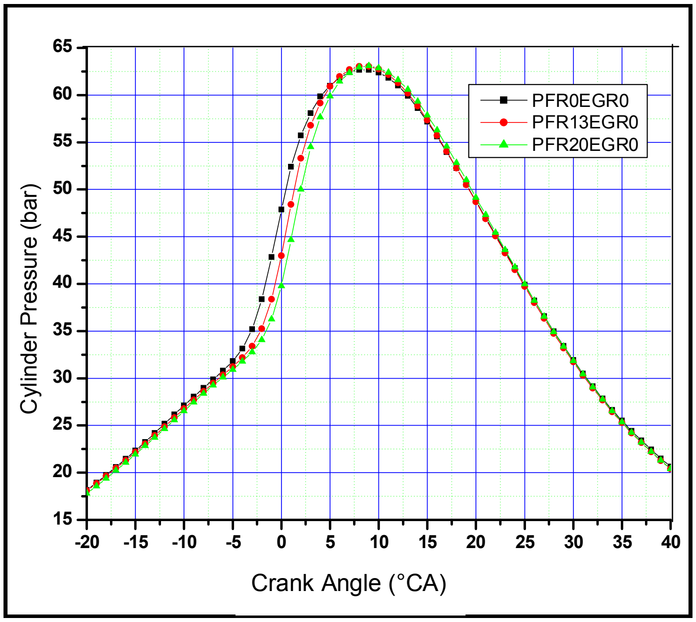


Figure 5.43: Effect of premixed fuel ratio on in-cylinder pressure.

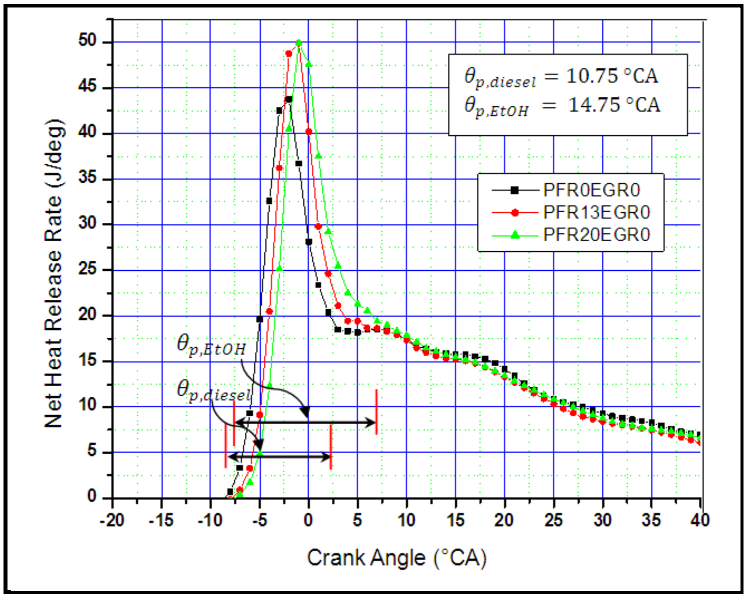


Figure 5.44: Effect of premixed fuel ratio on net heat release rate.

It is also observed that beyond 13% PFR, only marginal change occurred. Pickett et al. (Pickett and Siebers 2005) suggested the ignition delay also increases by lowering ambient oxygen concentration. It is obvious as ambient gas dilution (reduction of oxygen concentration), therefore, premixing of ethanol lowers the oxygen concentration or displaces air by ethanol. From the Fig. 5.44, it is also observed that the premixed combustion duration increases with increases in premixed ratio. At a higher premixed ratio, an obviously slightly faster rise of gas temperature was discovered than that for the case of pure diesel combustion.

Fig. 5.45 shows the influence of premixed fuel ratio on mass fraction burnt for various premixed fuel ratio of ethanol. As the premixed combustion continued till 2.5 °CA aTDC, 4 °CA aTDC and 7.5 °CA aTDC, thus the fraction of fuel burnt in the premixed phase corresponds to PFR0EGR0, PFR13EGR0 and PFR20EGR0 are 40%, 46% and 65% respectively. Similar results also reported by K.S. Tsang et al. (Tsang and Zhang 2010), premixed combustion duration increases with fumigation ratio (5% to 20%) while operating on premixed ethanol mode.

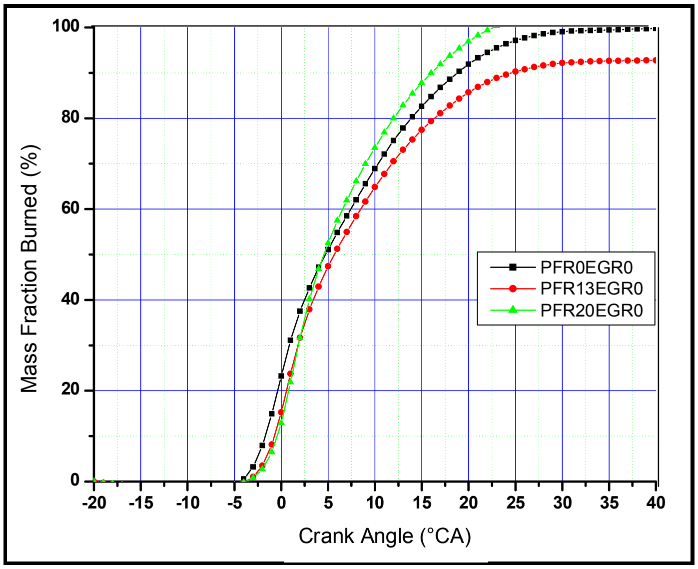


Figure 5.45: Effect of premixed fuel ratio on mass fraction burnt.

5.4.1.2 Effect of PFR on Combustion at 10% EGR

Fig. 5.46 shows that EGR and premixed ethanol (PFR13EGR10 and PFR20EGR10) produces cumulative effect on combustion chemistry and tends to lowers the in-cylinder pressure marginally. It is due to relatively grater rise in ignition delay period and dilution of intake air. In-cylinder pressure decreases with increase in premixed ethanol at fixed EGR.

Fig. 5.47 shows the variation of net heat release rate with different premixed fuel ratio. Peak net heat release rate increases with increase in premixed ethanol.

Maximum increase in peak is about 4.4% for PFR20EGR10. However, only marginal changes were found between PFR13EGR10 and PFR20EGR10. The reason could be explained by the increasing charge dilution which reduces the rate of combustion, at the same time increase in ignition delay period which provides more time for preparation of charge. Premixed combustion duration increased from 11.25 °CA to 17.5 °CA corresponds to increase in premixed ratio from 0% to 20%.

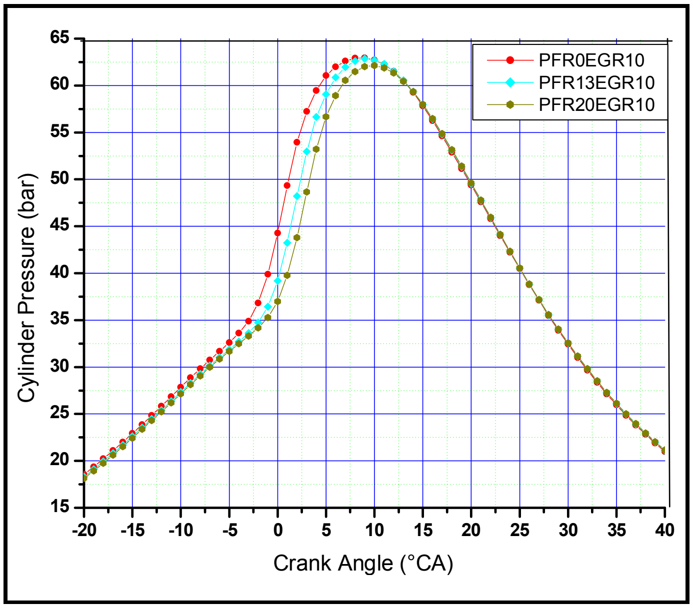


Figure 5.46: Effect of premixed fuel ratio on in-cylinder pressure

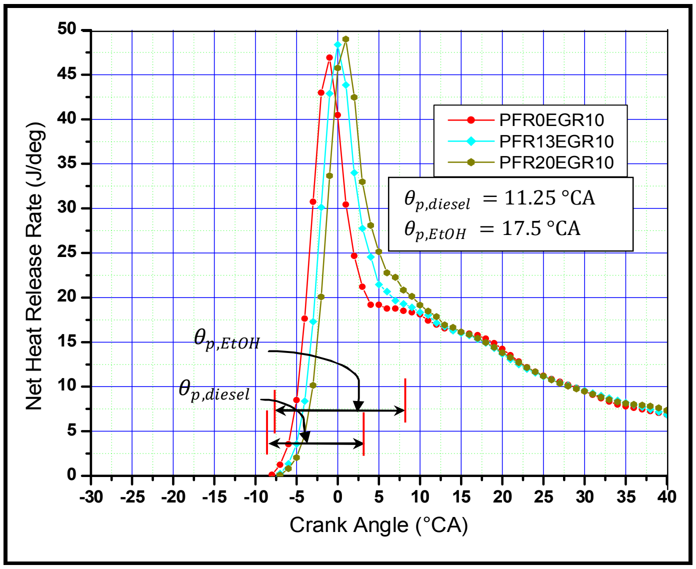


Figure 5.47: Effect of premixed fuel ratio on net heat release rate

Fig. 5.48 shows that maximum mass fraction burnt decreases with increase in premixed fuel ratio. This is due to reduction in air at high engine load causes incomplete combustion which is the result of replacement of greater mass of intake air by both ethanol vapour and EGR. Fraction of fuel burnt during premixed combustion phase associated with PFR0EGR10, PFR13EGR10 and PFR20EGR10 are 47%, 54% and 60% corresponding to 3 °CA aTDC, 7.5 °CA aTDC and 11 °CA aTDC respectively. Crank angle position of completion of premixed phase combustion respectively.

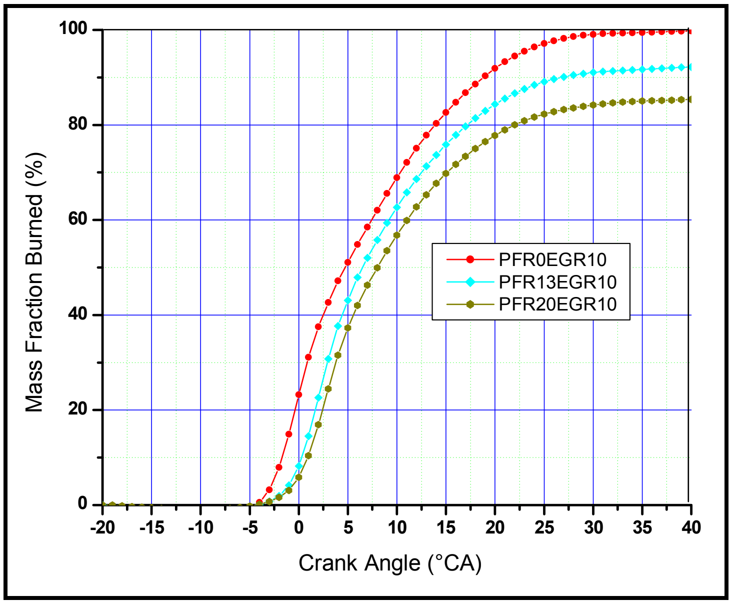


Figure 5.48: Effect of premixed fuel ratio on mass fraction burnt.

5.4.1.3 Effect of PFR on Combustion at 20% EGR

Fig. 5.49 shows the maximum in-cylinder pressure is attained in case of PFR0EGR20 is 61.46 bar, whereas, minimum is associated with PFR13EGR20 (60.34). Retardation in combustion phasing was observed at PFR13EGR20 but any further increase in PFR did not shows any change, peak in-cylinder pressure were obtained at 371 °CA for both PFR 13 and 20%.

Fig. 5.50 depicts the net heat release rate increases with increase in premixed fuel ratio. Maximum net heat release was about 49.93 J/°CA obtained at PFR13EGR20 rather at higher premixed ethanol (PFR20EGR20). Later, combustion phasing indicates increase in ignition delay period and consequently increases in premixed combustion duration with premixed fuel ratio.

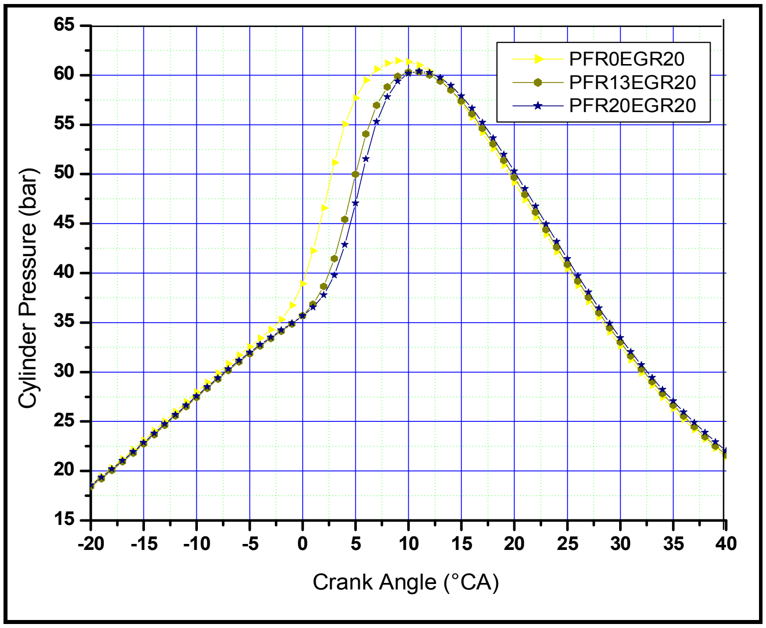


Figure 5.49: Effect of premixed fuel ratio on in-cylinder pressure

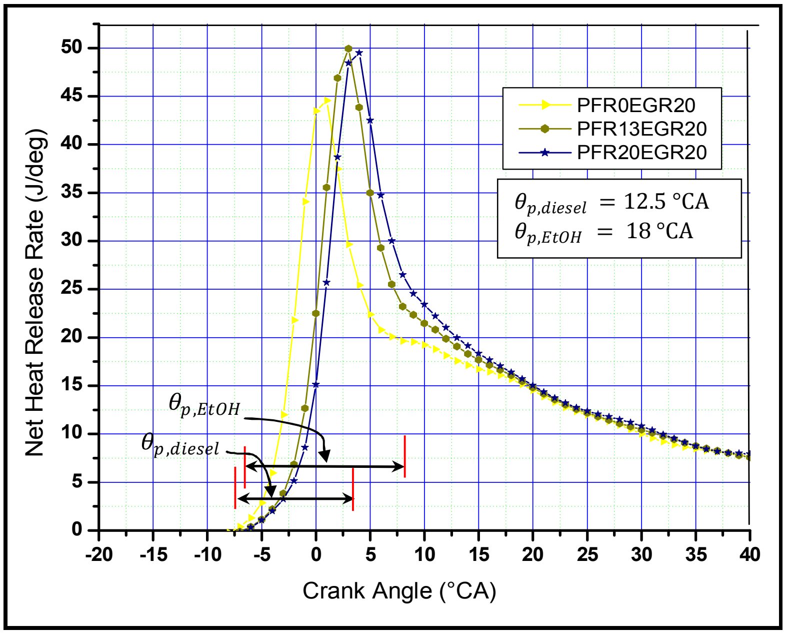


Figure 5.50: Effect of premixed fuel ratio on net heat release rate.

Fig. 5.51 shows mass fraction burnt curve indicates clearly PFR0EGR20 has shortest combustion duration. Premixing of ethanol helps to achieve stable combustion. Fraction of fuel burnt in the premixed phase based on net heat release curve is determined as 62%, 56% and 64% associated with PFR0EGR20, PFR13EGR20 and PFR20EGR20 respectively. As the maximum fraction of fuel burnt is almost same in pure diesel and premixed phase (PFR20EGR20), however, rate of combustion is different and fast burning is found with diesel combustion and slow combustion is resulted due to premixed ethanol.

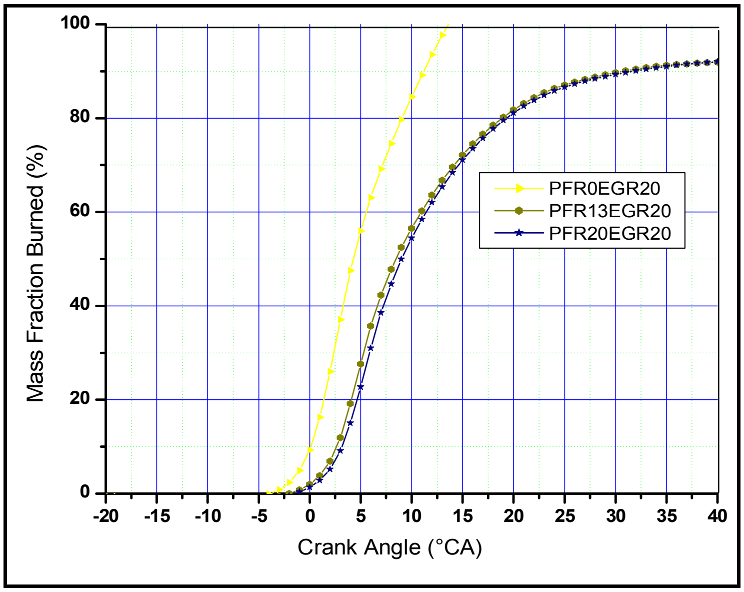


Figure 5.51: Effect of premixed fuel ratio on mass fraction burnt

5.4.2 Effect of EGR Ratio on Combustion

5.4.2.1 Effect of EGR Ratio on Combustion without PFR

Diesel combustion with EGR leads to reduction in-cylinder pressure and temperature which in turn reduces NO_x emissions. However, in this study the advantage is EGR has two folds - one is to control the rate of combustion another which is rather more important is to enhance the vapourization of ethanol.

Fig. 5.52 shows in-cylinder pressure traces for varying rate of EGR at constant PFR. Maximum reduction in peak in-cylinder pressure (about 1.21 bar) was observed in case of PFR0EGR20. However, while using mixture PFR0EGR10, it shows slight increase in peak in-cylinder pressure. Similar to other cases combustion phasing is evident in case of PFR0EGR20.

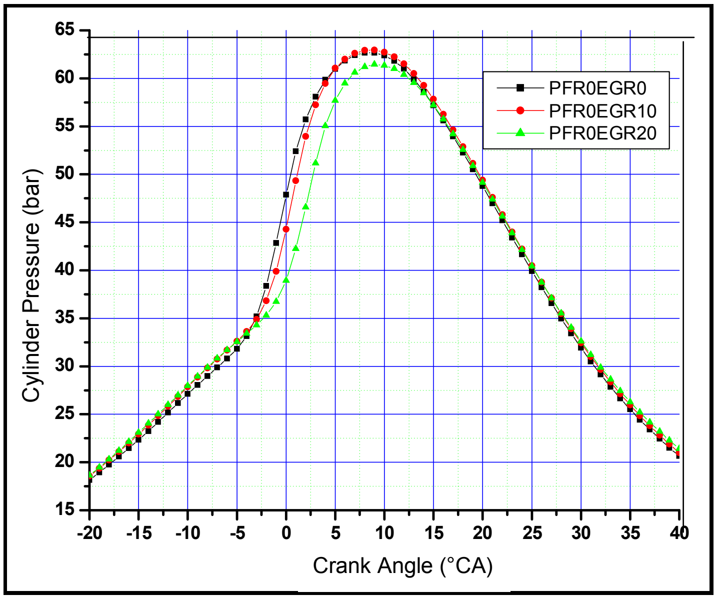


Figure 5.52: Effect of EGR ratio on in-cylinder pressure

Fig. 5.53 shows variation of net heat release rate with increasing EGR and without premixed ethanol. Similar to in-cylinder pressure, in this case also little variation in peak heat release rate was observed. The oxygen concentration changes when the exhaust gas is recycled into the intake. As the oxygen concentration is decreased, the ignition delay periods become longer. Maximum rise in peak heat release rate was about 7.2% found in case of PFR0EGR10 compared to PFR0EGR0. Late combustion phasing and peaks of NHRR for three cases (PFR0EGR0, PFR0EGR10 and PFR0EGR20) were observed at 358, 359 and 361 °CA respectively. Premixed combustion duration increased from 10.75 °CA to 12.5 °CA.

If hot exhaust is directly re-circulated, the cylinder charge temperature will float with the influx of the EGR heat, especially at high loads, which will raise the working fluid temperature. Test results demonstrated that cooled EGR reduces NO_x more effectively than hot EGR (Tomazic and Pfeifer 2002). Another important observation could be regarding increase in premixed combustion, in this it is lesser compared to varying

PFR this could be seen in Fig. 5.47. So, premixed ratio showed dominating effect on ignition delay compared to EGR ratio.

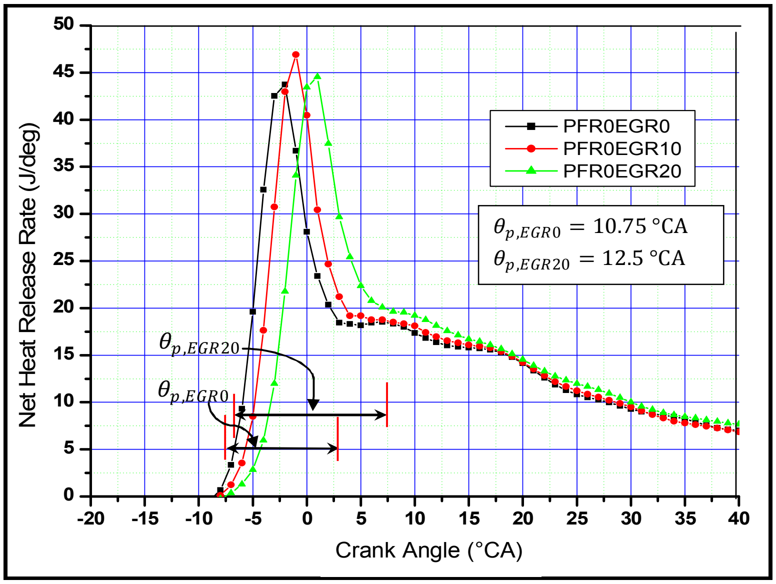


Figure 5.53: Effect of EGR ratio on net heat release rate

Fig. 5.54 shows mass fraction burnt plot. Important observations could be regarding premixed phase combustion duration increases higher with PFR compared to EGR.

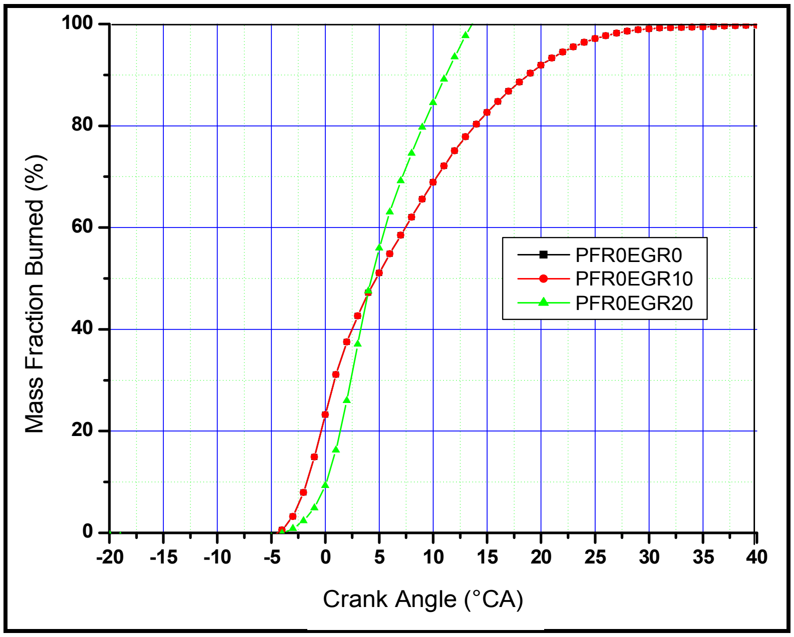


Figure 5.54: Effect of EGR ratio on mass fraction burnt

Fig. 5.53 shows that premixed phase combustion duration continued till 2.5 °CA aTDC, 3 °CA aTDC and 6 °CA aTDC corresponds to PFR0EGR0, PFR0EGR10 and PFR0EGR20 respectively. Thus associated mass fraction burnt were 40%, 47% and 62% determined using Fig. 5.54. However, short combustion duration is clearly visible from the steep slope of mass burning curve corresponds to PFR0EGR20.

5.4.2.2 Effect of EGR Ratio on Combustion with 13% PFR

It is observed from the in-cylinder pressure traces as shown in Fig. 5.55 that with increase in EGR ratio peak in-cylinder pressure decreases, ignition delay period also increases. It's revealed from the later combustion phasing. The maximum peak in-cylinder pressure was found to be about 63.03 bar in PFR13EGR0 and minimum peak were found in the case PFR13EGR20 (60.34 bar). The maximum peak pressure reduces by 2.6 bar.

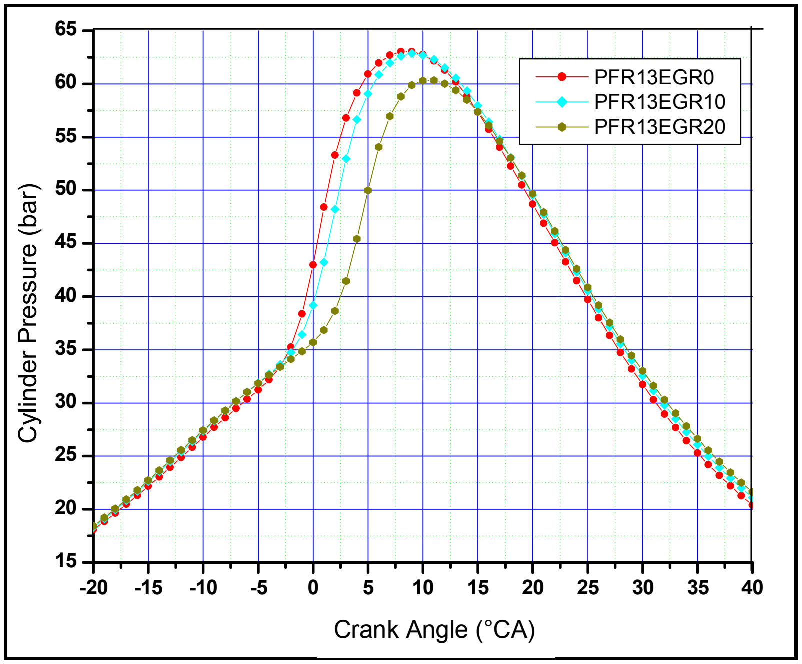


Figure 5.55: Effect of EGR ratio on in-cylinder pressure

It is evident from the slope of p-θ curve, later part of expansion has higher slope as compared to compression. Reduction in peak pressure and later combustion phasing does not affect negatively to the work conversion efficiency.

The study conducted (Kumar MS and Ramesh A 2003) on dual fuel engine, where diesel was replaced with high octane fuel (hydrogen, methane, methanol, gasoline) premixed at the air inlet manifold. This in conjunction with late diesel injection (near TDC) results in better premixed combustion. It leads to reduction or eliminations of the diffusion combustion phase by the spontaneous ignition of the homogeneous mixture preferably at TDC. So that improvement in emissions and fuel consumption could be obtained by reducing the compression work.

Fig. 5.56 shows, increasing EGR, leads to increase the ignition delay and premixed combustion duration, but only a marginal change in peak net heat release rate was observed. It is evident that with increase in EGR, premixed combustion duration also increased without affecting NHRR.

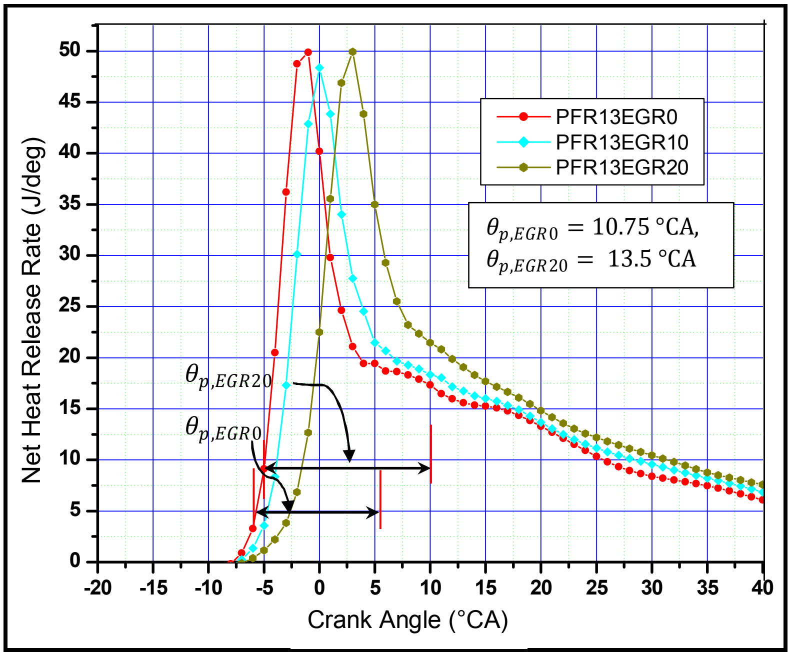


Figure 5.56: Effect of EGR ratio on net heat release rate

Fig. 5.57 shows mass fraction burnt plot. Premixed combustion in case of PFR13EGR0 and PFR13EGR10 continued to 4 °CA aTDC and 7.5 °CA aTDC whereas for PFR13EGR20 it increased to 10 °CA aTDC. Therefore, mass fraction burnt during premixed combustion was reaches a values of about 46%, 54% and 56% for PFR13EGR0, PFR13EGR10 and PFR13EGR20 respectively.

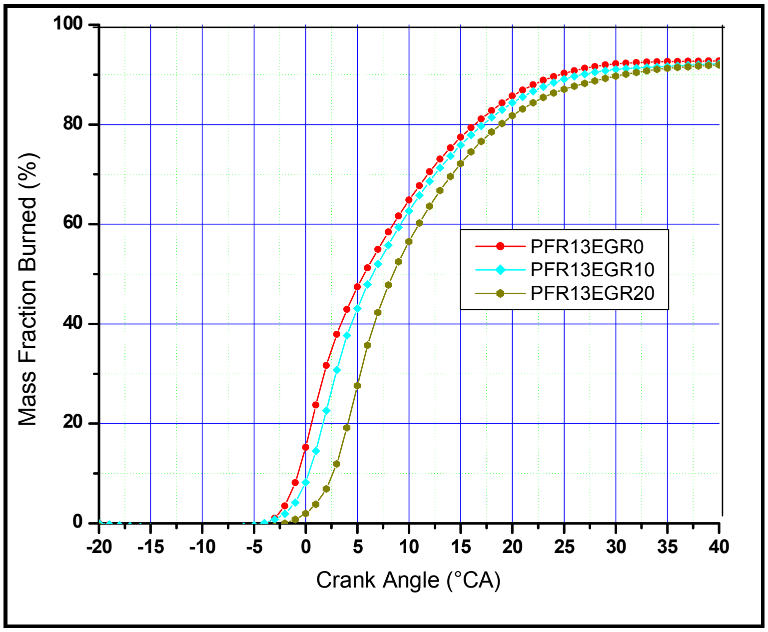


Figure 5.57: Effect of EGR ratio on mass fraction burnt

5.5.2.3 Effect of EGR Ratio on Combustion with 20% PFR

Fig. 5.58 shows in-cylinder pressure traces. It compares the peak in-cylinder pressure and combustion phasing obtained at a different EGR ratio at fixed premixed fuel ratio (20% PFR). Higher latent of vaporization of ethanol coupled with EGR reduces peak in- cylinder pressure. It reduced from 63.08 to 60.40 bar associated with PFR0EGR20 and PFR20EGR20. The maximum reduction in peak in-cylinder pressure was 2.68 bar which occurs at PFR20EGR20.

Fig. 5.59 shows variation of net heat release rate with crank angle. The peak of net heat release rate was found to be almost constant, later combustion phasing and an increase in premixed combustion duration is visible. Therefore, the net effect of the increase in EGR is that it retards the combustion. It is evident from the premixed phase combustion duration that it increases with increase in EGR ratio from 0 to 20% at constant premixed fuel ratio. The premixed phase combustion duration corresponds to PFR20EGR0 is 14.75 °CA, which increases to 18 °CA for PFR20EGR20.

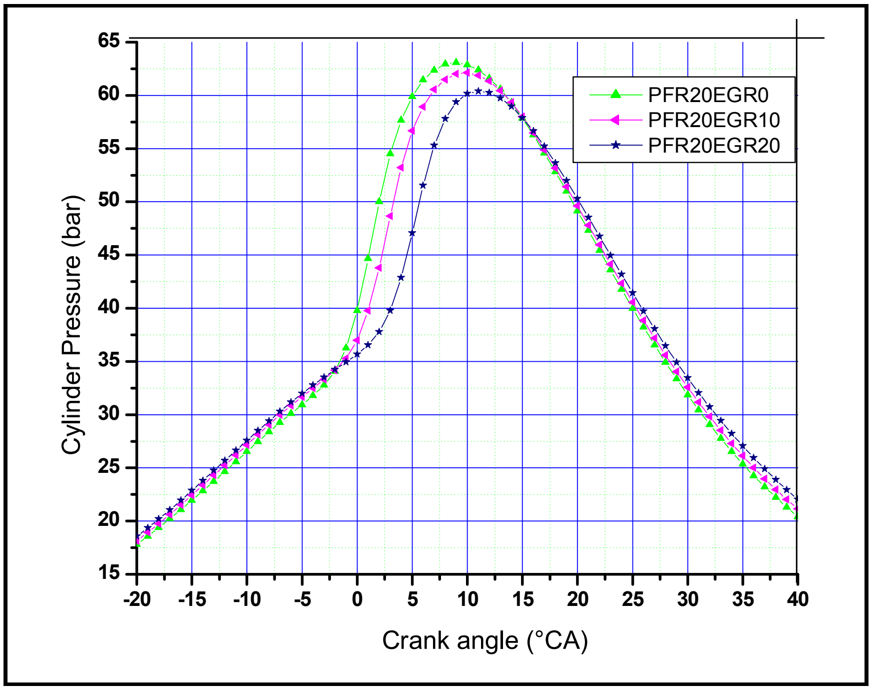


Figure 5.58: Effect of EGR ratio on in-cylinder pressure.

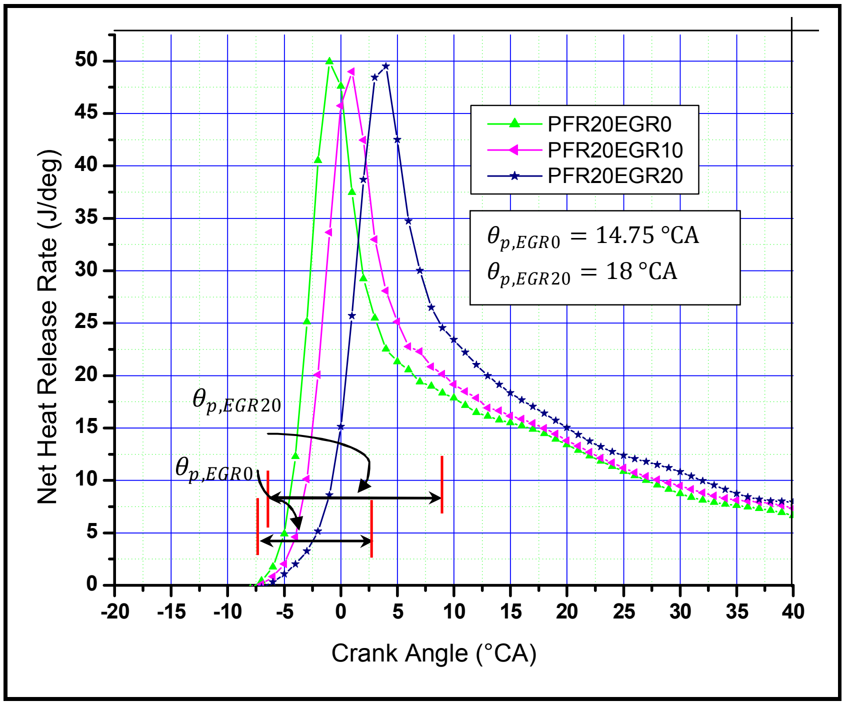


Figure 5.59: Effect of EGR ratio on net heat release rate.

Fig. 5.60 shows variation of mass fraction burnt with different EGR ratio and at constant premixed fuel ratio. Thus, we compare the portion of charge being burnt in premixed phase with EGR and without EGR. In case of PFR20EGR0 it was found about 65% whereas it changes to 60% and 64% corresponding with PFR20EGR10 and PFR20GR20 respectively. These values were associated to the position of crank angle at which premixed combustion ended such as 7.5 °CA aTDC, 11 °CA aTDC and 12 °CA aTDC corresponds to PFR20EGR0, PFR20EGR10 and PFR20EGR20 respectively. Rapid combustion was found without EGR, however with increase in EGR significant rise in ignition delay results which in turn reduces rate of burning during premixed phase of combustion.

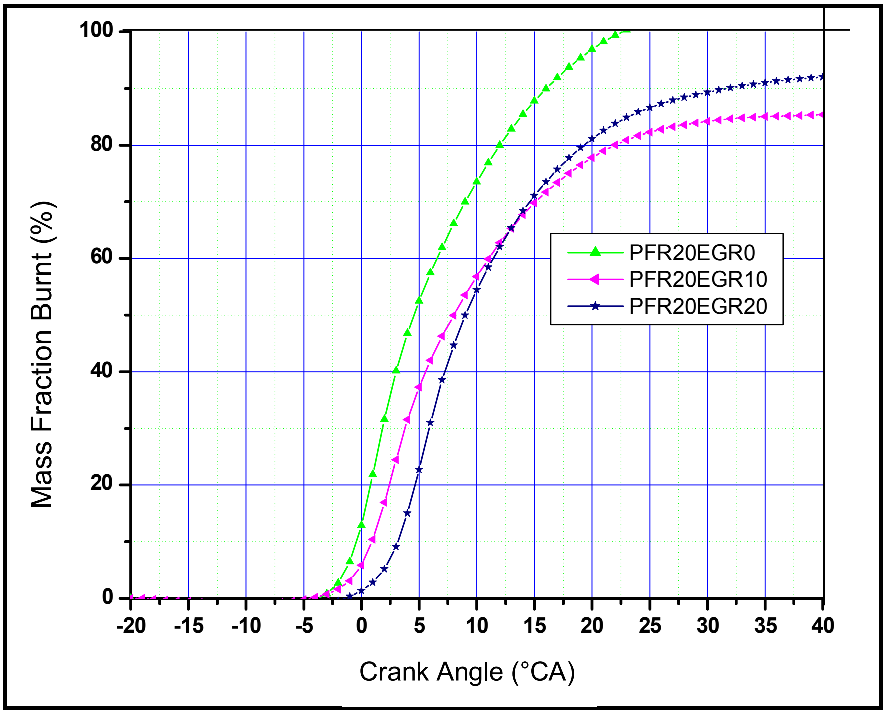


Figure 5.60: Effect of EGR ratio on mass fraction burnt.

It could be conclude the effect of PFR and EGR on combustion characteristics at BMEP of 4.2 bar. As discussed earlier four parameters are compared, this is given in Table 513.

Table 5.13: Comparison of combustion variables at BMEP of 4.2 bar.

Sr. No.	Abbreviations	P_{max}(bar)	RPR ($\frac{dp}{d\theta}$)	NHRR (J/°CA)	(θ_p) (°CA)
1	PFR0EGR0	62.67	4.88	43.74	10.75
2	PFR13EGR0	63.03	5.32	49.87	11
3	PFR20EGR0	63.08	5.51	49.92	15.75
4	PFR0EGR10	62.94	4.98	46.9	11.25
5	PFR13EGR10	64.87	5.03	48.36	13.5
6	PFR20EGR10	62.14	4.91	48.99	17.5
7	PFR0EGR20	61.46	4.48	44.55	12.5
8	PFR13EGR20	60.34	4.79	49.93	13.5
9	PFR20EGR20	60.40	4.46	49.49	18

5.4.3 Effect of PFR and EGR Ratio on Emissions

5.4.3.1 Effect of PFR on Emissions

The NO_x generation is affected by two mutually countering effects. Higher latent heat of vaporization of ethanol produces cooling effect whereas increase in ignition delay and heat release rate in premixed phase lead to increase combustion temperature. Increase or decrease in NO_x emissions solely depends on the relative contribution of attributes discussed above.

Fig. 5.61 shows the variations of NO_x with increasing premixed ratio of ethanol and at constant EGR ratio. A negative trend was observed for constant rate of EGR, further maximum reduction in NO_x emissions was observed at high premixed fuel ratio. In case of PFR20EGR0 and PFR13EGR0, 16% and 5.5% reduction was found compared to PFR0EGR0 respectively.

The experimental work carried by Tsang et al. (Tsang and Zhang 2010) reported similar results which was obtained at BMEP 3.9 bar. They found reduction in $BSNO_x$ as compared to Euro-V diesel, typically 6.6 and 14.7% associated with 10% and 20% premixed fuel ratio respectively.

NO_x emissions decreases correspond to mixture PFR20EGR10 and PFR20EGR20 by about 25% and 36% compared to PFR0EGR10 and PFR0EGR20 respectively. Therefore it shows that maximum reduction in NO_x emissions was recorded with PFR20EGR20 of about 36%, this is the contribution of premixed fuel ratio since EGR was kept constant. Bhupendra et al. (Bhupendra Singh Chauhana and Naveen Kumar 2011) also showed reduction in NO_x emission at similar engine load.

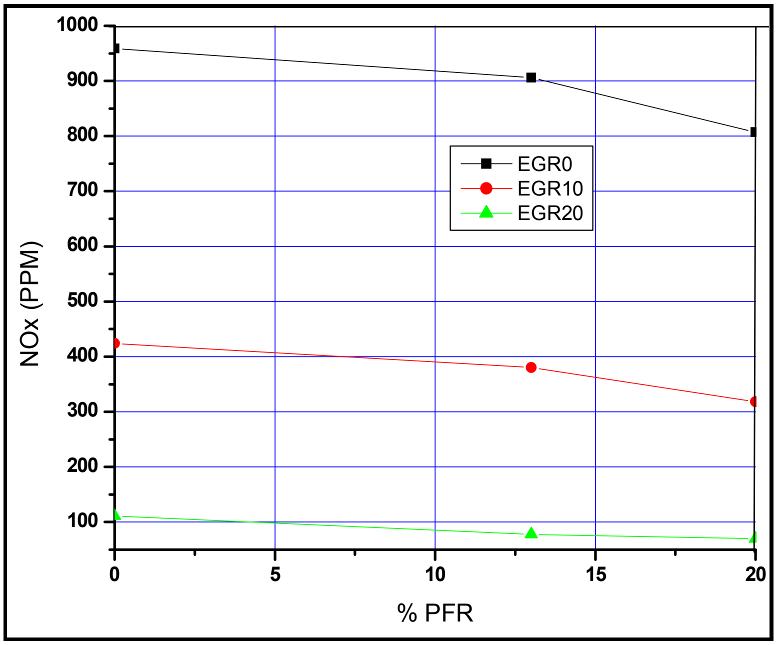


Figure 5.61: Effect of premixed fuel ratio on nitrogen oxides emissions.

Fig. 5.62 shows the variation of smoke opacity presented in terms of percentages with premixed ratio. In general, the smoke opacity decreases with increase in premixed fuel ratio. However, maximum reduction is occurs at 20% PFR for all the cases. At PFR20EGR0 about 34% reduction in smoke was found as compared PFR0EGR0. Abu-Qudais et al. (Abu-Qudais and Haddad 2000) also showed similar results, they have found maximum decrease in smoke opacity of 33% to 51% for 20% PFR at various engine speeds.

Smoke emissions decreases correspond to mixture PFR20EGR10 and PFR20EGR20 by about 62% and 55% compared to PFR0EGR10 and PFR0EGR20 respectively. This indicates higher premixed ratio and EGR when both are applied together are not effective as compared to a combination of high premixed fuel and low EGR (PFR20EGR10). Also, it shows simultaneous reduction of NO_x and smoke.

There are several reasons leading to such reduction of soot and, hence, smoke emission when premixed ethanol is applied. Firstly, increasing premixed ethanol reduces the quantity of diesel fuel being injected. Thus, less diesel fuel is burned in the diffusion mode and hence reduction of soot formation. Secondly, the increase in ignition delay increases the amount of diesel fuel burned in the diffusion mode.

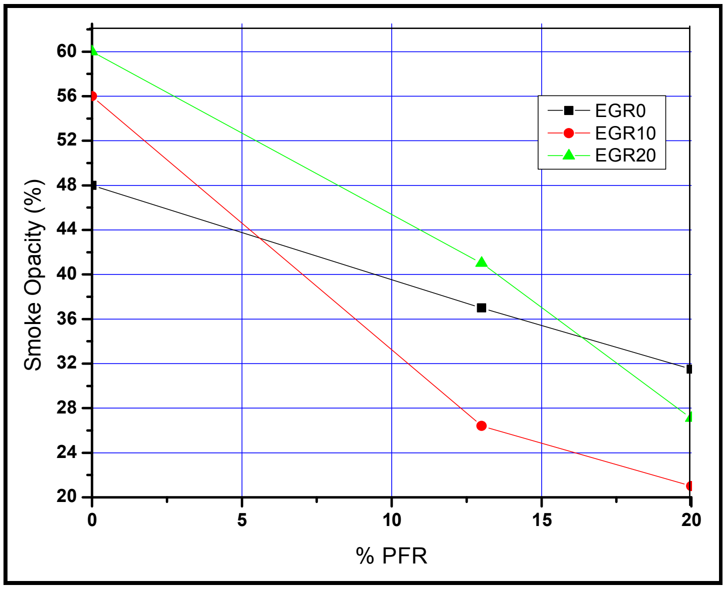


Figure 5.62: Effect of premixed fuel ratio on smoke emission

Fig. 5.63 and Fig. 5.64 show the emission of UHC and CO for different intake mixture conditions. It is found that UHC and CO is increases with increase in premixed fuel ratio. At PFR13EGR0 and PFR20EGR0 UHC emissions increased by 2.9 times and 3.6 respectively times compared to PFR0EGR0. Similarly, with PFR13EGR10 and PFR20EGR10 it increased by 2.3 times and 3 times respectively compared to PFR0EGR10. Whereas in case of PFR13EGR20 and PFR20EGR20 correspondingly values reaches to 2.1 times and 3 times respectively as compared to PFR0EGR20.

At PFR13EGR0 and PFR20EGR0, CO emissions increased by 1.1 and 1.25 times compared to PFR0EGR0. Similarly, with PFR13EGR10 and PFR20EGR10 it changed to 0.9 times and 1.04 times respectively compared to PFR0EGR10. Whereas, in case of PFR13EGR20 and PFR20EGR20 correspondingly values reaches to 1.27 times and 1.44 times respectively compared to PFR0EGR20.

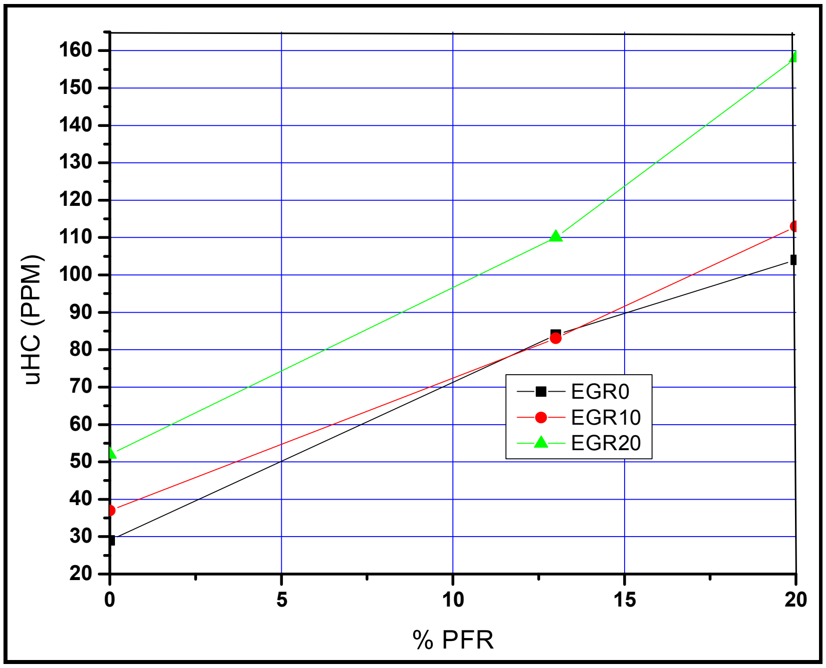


Figure 5.63: Effect of Premixed fuel ratio on unburnt hydrocarbons emissions.

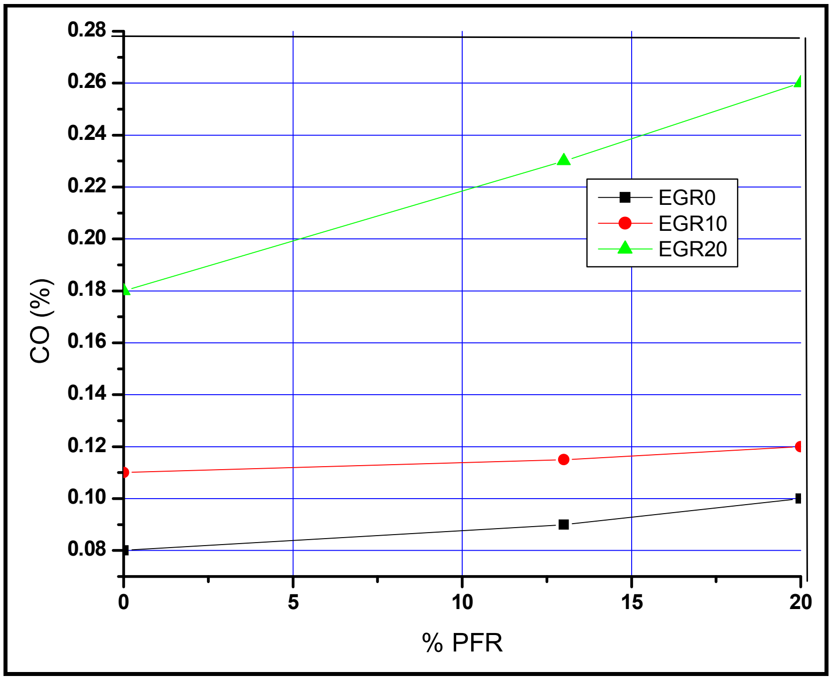


Figure 5.64: Effect of premixed fuel ratio on carbon monoxide emisison.

5.4.3.2 Effect of EGR Ratio on Emissions

As discussed in literature review about NO_x formation. Thermal NO_x is the fraction of peak in-cylinder temperature, so it greatly influenced by temperature. Fig. 5.65 shows a steep negative gradient is observed corresponding to constant premixed fuel ratio and increasing EGR. It is the result of cooling effects along with dilution of intake charge due to EGR.

In case of PFR0EGR10 and PFR0EGR20, NO_x emissions decrease to about 55% and 88% compared to PFR0EGR0. Similarly, with PFR13EGR10 and PFR13EGR20 it decreases to about 58% and 91% compare to PFR13EGR0. Whereas, PFR20EGR10 and PFR20EGR20 shows reduction of about 60% and 91% as compared to PFR20EGR0.

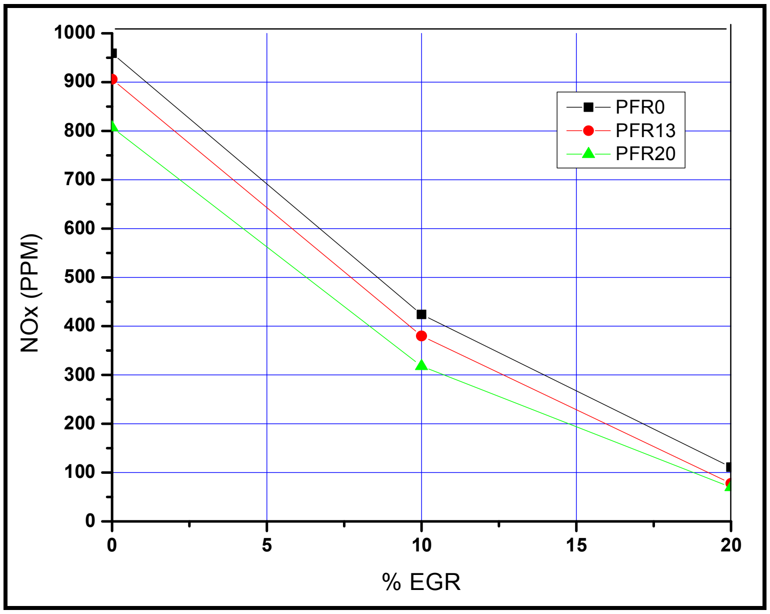


Figure 5.65: Effect of EGR ratio on nitrogne oxides emissions

Study conducted (Pundir 2007) have found that at 10% EGR in case of DI diesel engine could reduce NO_x by 50% and at 15% EGR it further reduces NO_x but UHC and CO increase. It was found that decrease in excess air causes sharp increase in smoke and loss of fuel economy. So, with increase in EGR, it leads to increase in smoke, particulate and unburnt hydrocarbon emissions and an increase in fuel consumptions.

Fig. 5.66 shows variation of smoke opacity. Gradually increasing trend was observed with increase in EGR without premixed ethanol. However, increase in premixed fuel reduces smoke initially (at 10% EGR) and further increase in EGR lead to increase the same. Smoke opacity corresponds to PFR13EGR10 and PFR20EGR10 decreased by 45% and 56% respectively compared to PFR0EGR0. Therefore simultaneous reduction of NO_x and smoke was observed.

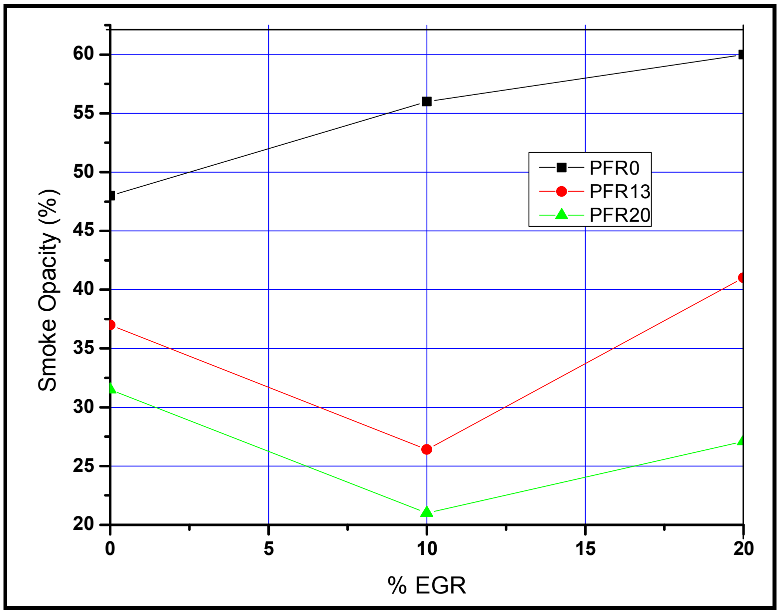


Figure 5.66: Effect of EGR ratio on smoke emission

Fig. 5.67 and Fig. 5.68 show the emission of UHC and CO with EGR ratio. It is found that UHC and CO emission increases with increase in EGR ratio. At PFR0EGR10 and PFR0EGR20 UHC emissions increased by 1.3 times and 1.8 times respectively compared to PFR0EGR0. Similarly, with PFR13EGR10 and PFR13EGR20 it increased by 2.3 times and 2.1 times respectively compared to PFR13EGR0. Whereas, in case of PFR20EGR10 and PFR20EGR20 correspondingly values reaches to 3 times in both the cases compared to PFR20EGR0.

At PFR0EGR10 and PFR0EGR20, CO emissions increased by 1.3 and 2.2 times respectively compared to PFR0EGR0. Similarly, with PFR13EGR10 and PFR13EGR20 it changes to 0.9 times and 1.27 times respectively compared to PFR13EGR0. Whereas, in case of PFR20EGR10 and PFR20EGR20 correspondingly values reaches to 1.04 times and 1.44 times respectively compared to PFR20EGR0. With increase in EGR ratio incomplete combustion tends to increased steeply beyond 10% EGR.

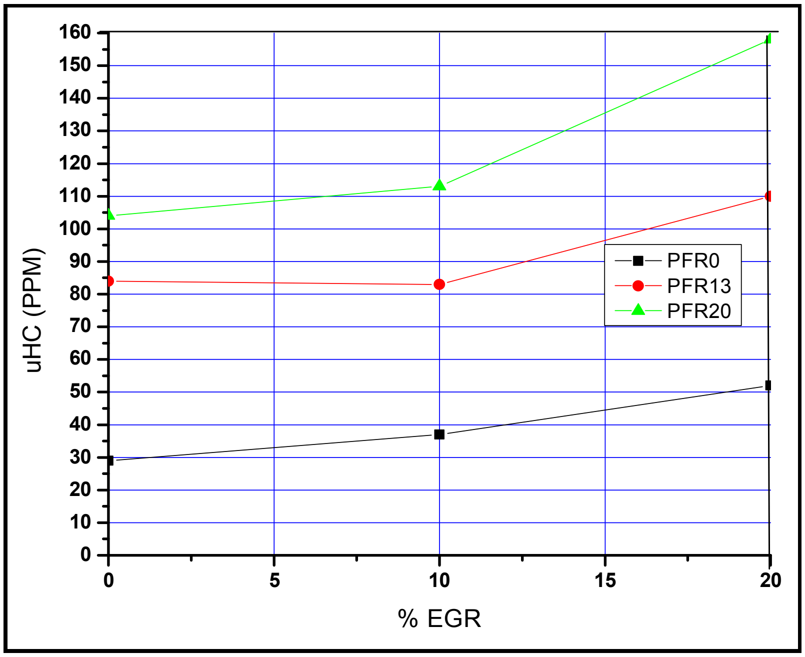


Figure 5.67: Effect of EGR ratio on unburnt hydrocarbons emissions.

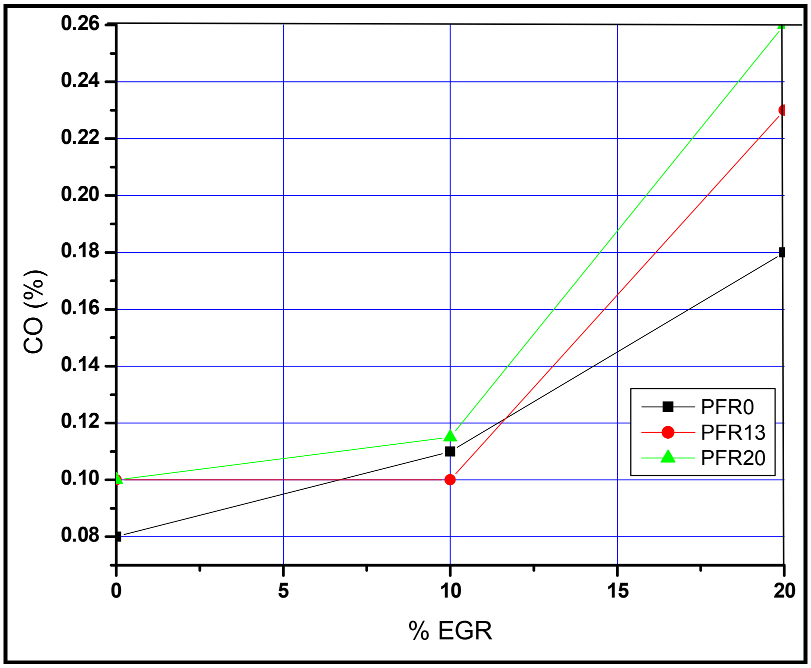


Figure 5.68: Effect of EGR ratio on carbon monoxide emission.

5.4.4 Effect of PFR and EGR Ratio on Performance

5.4.4.1 Effect of PFR on Performance

As discussed performance of PCCI engine is measured in terms of BSFC and BTE at constant BMEP and engine speed. Table 5.14 shows quantity of fuel supplied, heat supplied by ethanol and diesel, BTE and BSFC.

Heisey and Lestz (Heisey and Lestz 1981) found that, with premixed ethanol, the thermal efficiency decreased at low engine loads but increased marginally at medium and high engine loads. Hayes et al. (Hayes and Savage 1988) also showed an increase of thermal efficiency at high engine load with increasing premixed ethanol but no effect or negative effect on thermal efficiency at medium and low engine loads.

Table 5.14: Performance analysis at BMEP of 4.2bar

Fuel	**EtOH (ml)**	**Diesel (ml)**	**Power (kW)**	**q_{EtOH} (kW)**	**q_{diesel} (kW)**	**q_{EtOH} + q_{diesel} (kW)**	**η_{BTH}**	**BSFC (g/kWhr)**
PFR0 EGR0	0	18	3.48	0	11.01	11.01	31.62	257.6
PFR13 EGR0	4	16	3.45	1.4	9.78	11.19	30.83	285.9
PFR20 EGR0	6	15	3.44	2.11	9.171	11.28	30.49	299.8
PFR0 EGR10	0	18	3.45	0	11.01	11.01	31.35	259.8
PFR13 EGR10	4	16	3.42	1.4	9.78	11.19	30.57	288.4
PFR20 EGR10	6	14	3.44	2.11	8.56	10.67	32.24	285.3
PFR0 EGR20	0	18	3.43	0	11.01	11.01	31.17	261.3
PFR13 EGR20	4	16	3.45	1.4	9.78	11.19	30.83	285.9
PFR20 EGR20	6	14	3.43	2.11	8.56	10.67	32.15	286.2

Zhang et al. (Zhang and Cheung 2009) performed similar study with premixed methanol. Their results showed a negative BTE change rate from 0.8 bar to 3.9 bar, negligible change at 5.8 bar, and positive change at 7 bar. Present study also shows that at BMEP of 4.2 bar, negative change in the thermal efficiency was observed. However, due to scarcity of literature and limited research in the area of premixed ethanol with EGR results could not be compared.

Firstly, as ethanol starts vaporizing in intake manifold due to high latent heat of vaporization (840 kJ/kg) compared to diesel fuel (270 kJ/kg) it produces cooling effect which cools the premixed fuel. Secondly, small amount of ethanol escapes without participating in combustion process and may stick with the induction manifold. Thirdly, premixed air-ethanol mixture burns faster, also ethanol tends to increase ignition delay and, hence, provides more premixed combustion; both of them tend to increase the BTE. However, these factors compete with each other and net result would be either increases or decrease in BTE. So, at full load increase in in-cylinder temperature and pressure helps to reduce the cooling effect and hence less adverse effect on BTE. Using EGR helps to reduce the cooling effect of ethanol and lead to increase brake thermal efficiency.

Fig. 5.69 shows BSFC increases with increase in premixed fuel ratio; however rate of increase is slightly higher in case of EGR10 and EGR20 up to 13% PFR. Nevertheless, EGR and premixed fuel shows decrease in BSFC at higher percentage of PFR. Therefore, in case of PFR20EGR10 and PFR20EGR20 BSFC was found lower than PFR0EGR0.

Fig. 5.70 shows variation of BTE with premixed fuel ratio. It can be concluded that maximum increases in BTE was observed in case of PFR20EGR10 by about 6.5% compared to PFR20EGR0. The reason could be explained by EGR that leads to further increase ignition delay and reduces cooling effect due to ethanol, which helps in providing better mixing before ignition. So it could be understood for the given condition as premixed combustion phase and rate of combustion increased, resulting in higher brake thermal efficiency.

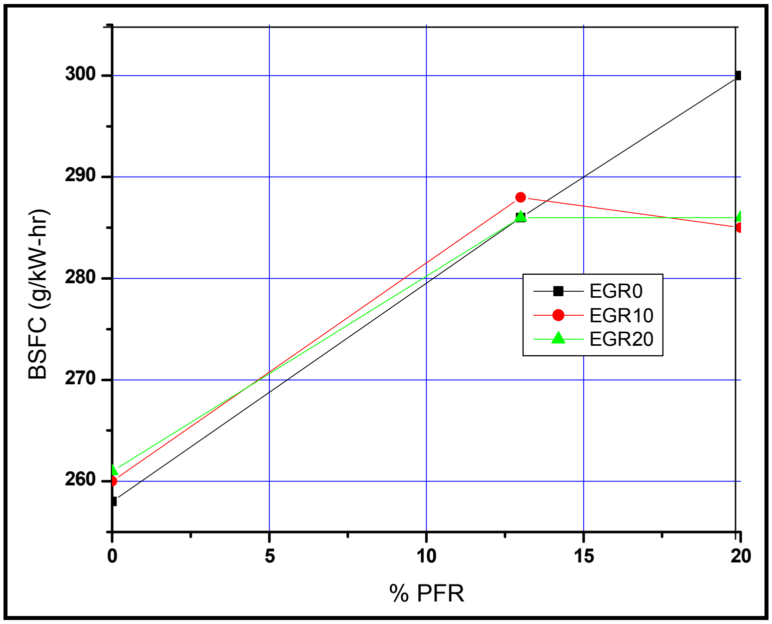


Figure 5.69: Effect of premixed fuel ratio on BSFC.

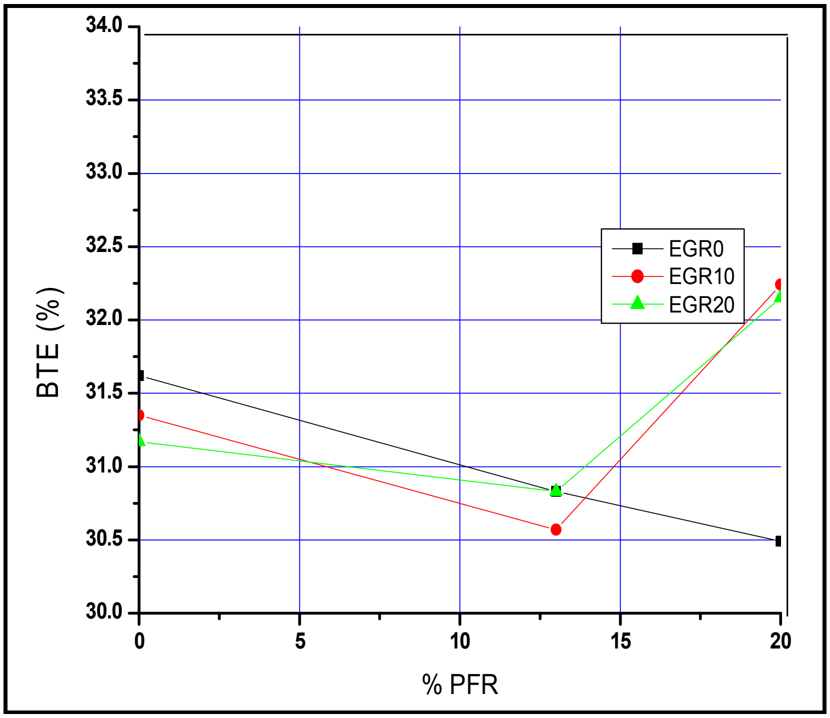


Figure 5.70: Effect of premixed fuel ratio on brake thermal effiecicny.

However, in case of PFR20EGR20, ignition delay period extended further beyond optimum for combustion reaction and, it tends to reduce rate of burning. In case of PFR20EGR20 marginal increase in efficiency was observed (1.66%) compared to PFR20EGR0. PFR20EGR10 shows overall improvement and BTE, and it increased by 2% compared to PFR0EGR0.

Fig. 5.71 shows the influence of premixed ethanol on performance which is reflected by the BTE change rate. Comparison has been done at constant 0% EGR and corresponding percentage of premixed ratio considered as a reference. A negative BTE change rate indicates a decrease in BTE upon increasing premixed ratio of ethanol. Under all test conditions, the BTE change rates are all negative except for the cases PFR20EGR10 and PFR20EGR20.

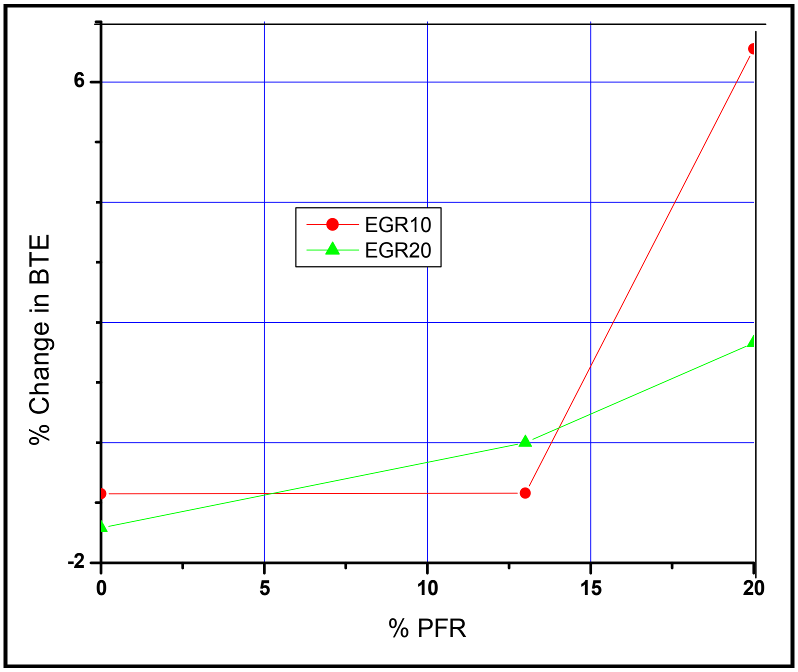


Figure 5.71: Effect of premixed fuel ratio on % change in BTE.

Fig. 5.72 shows variations of ignition delay period with premixed fuel. Minimum and maximum ignition delay period corresponds to PFR0EGR0 and PFR20EGR20. However, optimum ignition delay corresponds to higher BTE is found about 5.26 °CA.

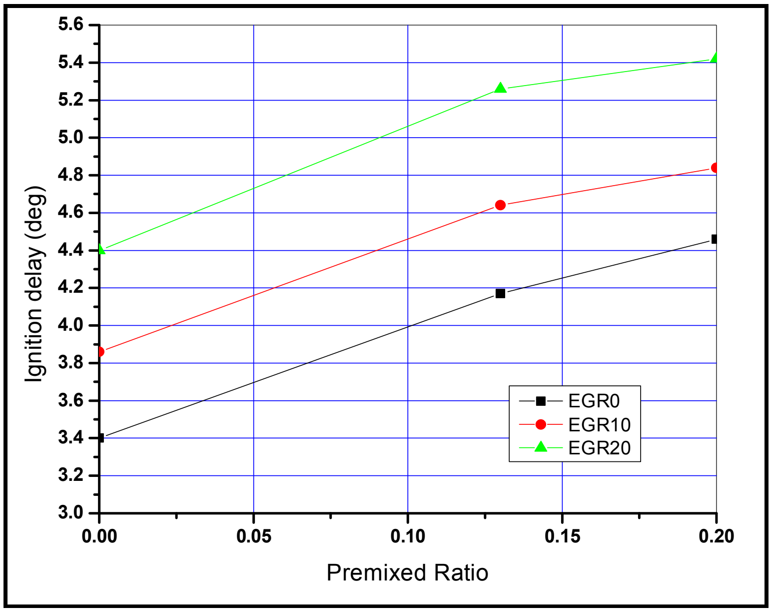


Figure 5.72: Variation of igniton delay period with premixed fuel ratio.

5.4.4.2 Effect of EGR Ratio on Performance

Fig. 5.73 shows brake thermal efficiency decreases with increase in EGR for pure diesel combustion. It was the direct consequence of the increase in BSFC, however, with premixed ethanol an increase in EGR, showed marginal improvement. Lower premixed ratio (13%) also shows initially a negative change in the brake thermal efficiency, but at higher EGR it is improved marginally.

However, BTE in case of PFR20EGR20 increased by 3.1% and 1.67% compared with PFR0EGR20 and PFR0EGR0 respectively. Fig. 5.74 shows BSFC increases with increase in EGR, only PFR20EGR10 and PFR20EGR20 it showed reduction in BSFC with EGR.

Fig. 5.75 shows variation of ignition delay period with % EGR. This shows steep rise in ignition delay period is recorded, the slope of PFR lines are higher as compared to constant EGR line (Fig. 5.72). So it could be concluded that rate of increase of ignition delay with increase in EGR is higher compared to PFR.

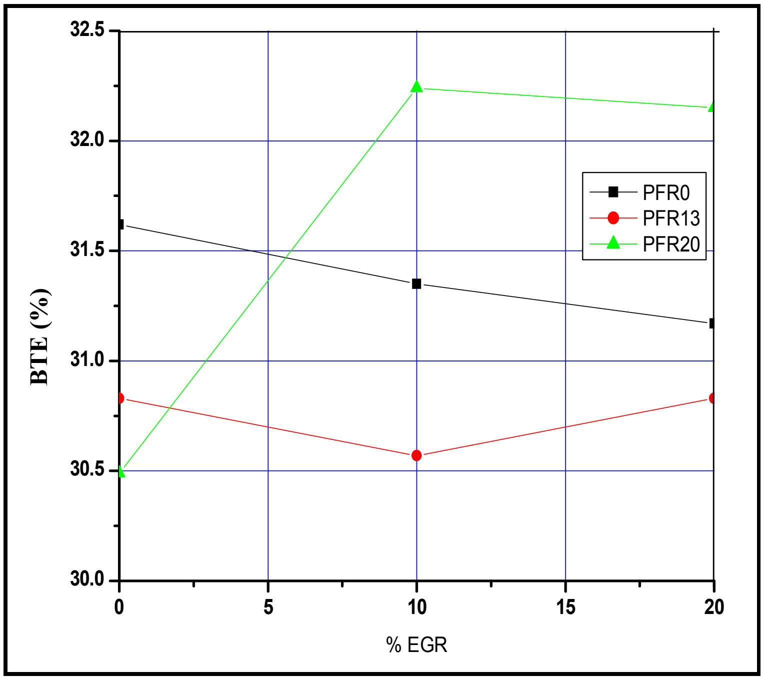


Figure 5.73: Effect of EGR ratio on brake thermal efficiency.

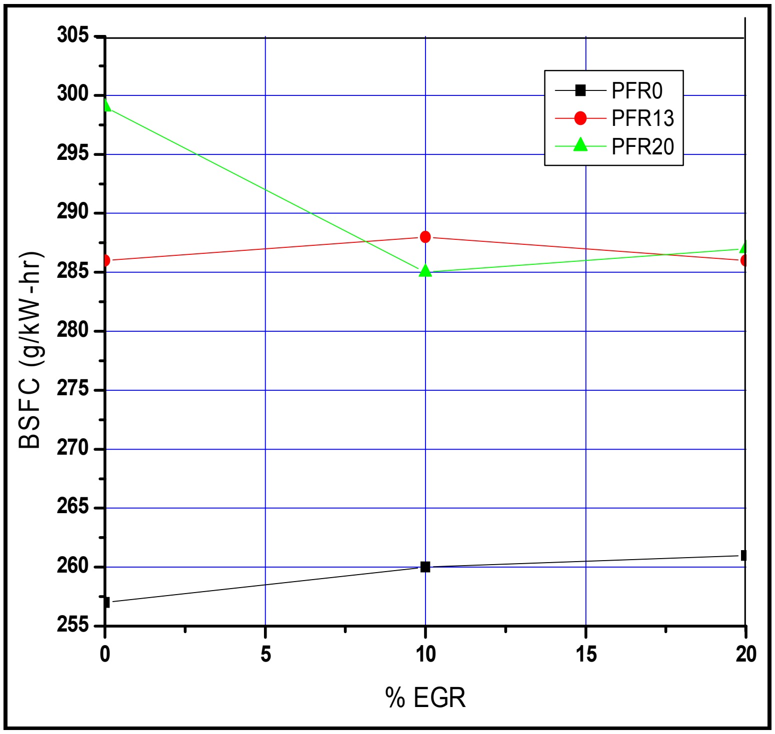


Figure 5.74: Effect of EGR ratio on BSFC.

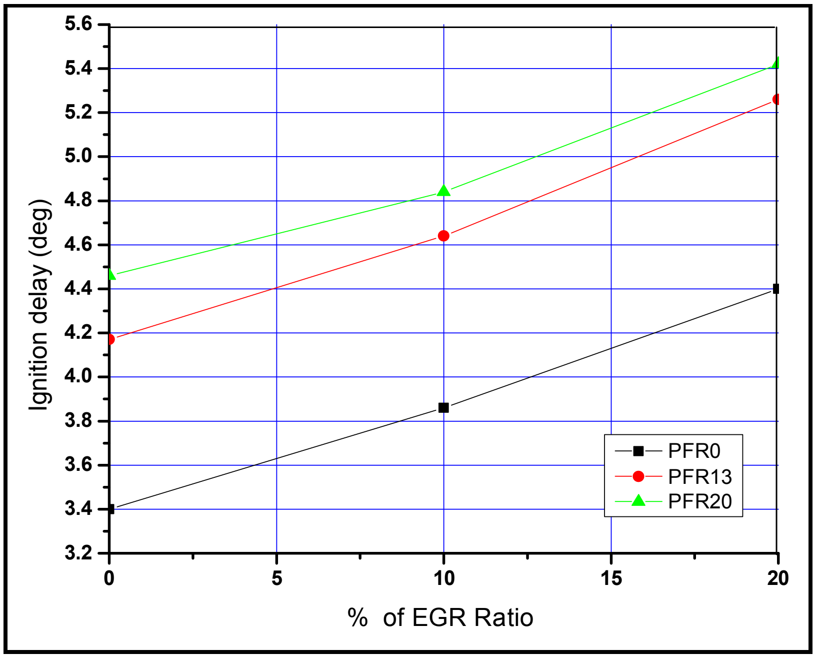


Figure 5.75: Variation of ignition delay period with EGR ratio.

5.5 COMPARISON OF IGNITION DELAY AND EMISSIONS

5.5.1 Ignition Delay Period

Fig. 5.76 shows variation of ignition delay period with BMEP.

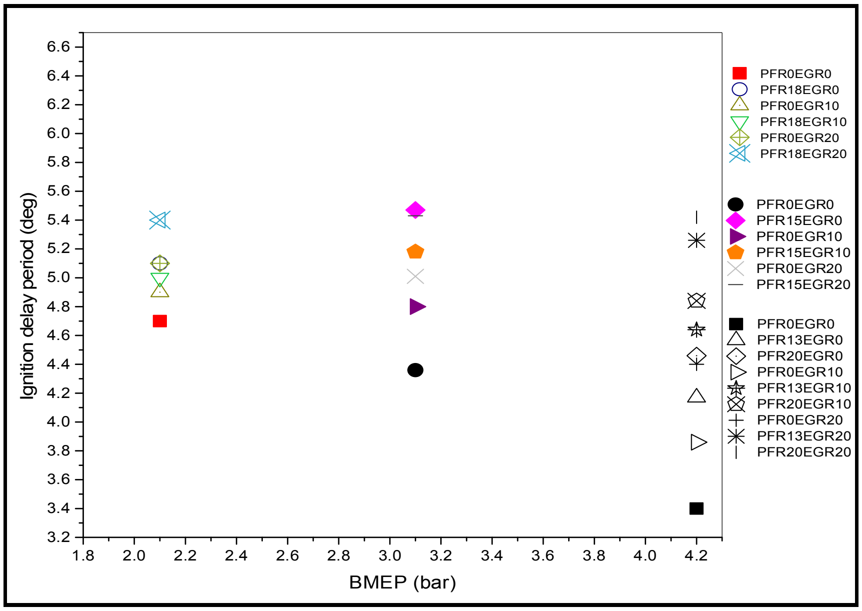


Figure 5.76: Variation of ignition delay period with engine loads.

It is found that in case of diesel combustion ignition delay period decreases with increase in load. However, significant reduction was found at BMEP of 4.2 and only little variation is recorded at BMEP of 3.1 bar. Also, ignition delay period increases with EGR and premixed ethanol by mixing in any proportion with fresh air. The best performance of the engine at BMEP of 3.1 and 4.2 bar is found in the range of 5.25 to 5.5 °CA. Since the maximum BTE is recorded corresponds to 3.1 and 4.2 bar (BMEP) are found at 5.26 °CA and 5.47 °CA. So it is concluded pure diesel combustion under such conditions could not provide sufficient time for charge mixing which can reduce emissions.

5.5.2 Nitrogen Oxides Emission

From Fig. 5.77 it is evident that NO_x emission increases with increase in BMEP. NO_x from diesel engine is mainly formed by the oxidation of nitrogen at high temperature. Therefore, the oxygen content and flame temperature are significant factors affecting NO_x formation. However, in case of PCCI mode oxygen in the fuel might enhance the NO_x formation. However, cooling effect tends to reduce NO_x, hence net effect may increase or decreases depends on which one is more dominating factor.

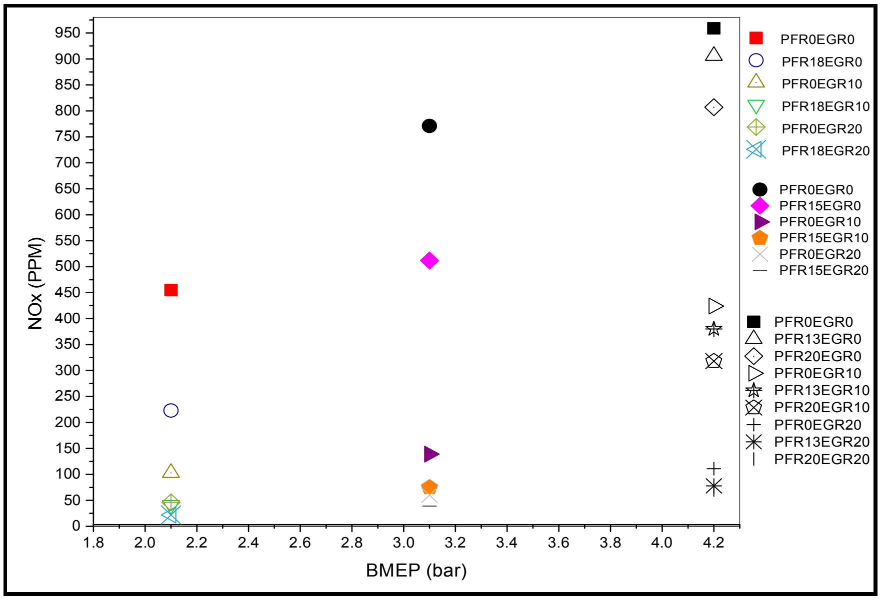


Figure 5.77: Variation of nitrogen oxides emissions with engine loads.

Increase in NO_x with engine load is because of obvious reason of increase in in-cylinder temperature with increase in engine load. Thus, it results into increases in thermal NO_x formation. Nevertheless, at each engine load NO_x emission decreases with premixed ethanol and EGR due to dilution effect and higher latent heat of vaporization of ethanol. The decreases of NO_x emission with premixed ethanol is more significant at low engine loads of BMEP of 2.1 and 3.1 bar compared to 4.2 bar. It is clearly evident that by using premixed ethanol and EGR, NO_x emission can be reduced significantly at BMEP of 4.2 bar and it could brought down up to the level of emissions at BMEP of 2.1 bar.

5.5.3 Smoke Emission

Fig. 5.78 shows variation of smoke opacity with engine load (BMEP). Smoke consists mainly of combustion generated carbonaceous soot on which some organic or hydrocarbon compounds and sulfates have been adsorbed. It is found the smoke opacity increases with increases in engine load corresponds to pure diesel combustion with and without EGR. However the trend with premixed ethanol and EGR is not consistent for different loads. Smoke opacity decreases with premixed ethanol at all engine loads.

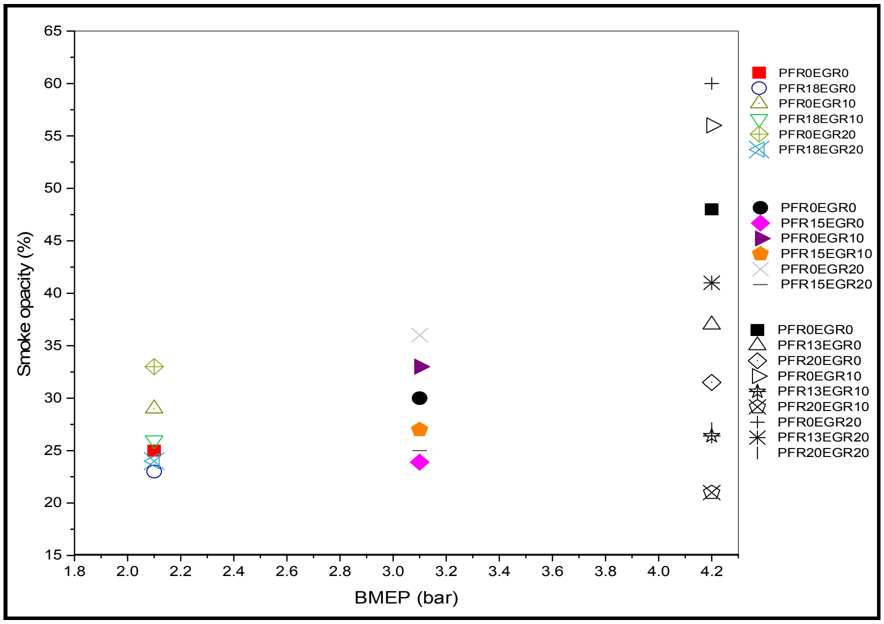


Figure 5.78: Variation of smoke opacity with engine loads

However, at BMEP of 4.2 bar smoke opacity decreases corresponds to premix ethanol with increasing rate of EGR. Therefore is it evident that simultaneous reduction of NO_x and smoke could be obtained by using premixed ethanol and EGR. Moreover, it showed better combustion in terms of BSFC and BTE.

PFR20EGR20 at BMEP of 4.2 bar (100% load) showed simultaneous reductions of NO_x and smoke together with improved thermal efficiency reinforce the merit of PCCI mode over diesel combustion. However in all other mixture of premixed ethanol showed small reduction of smoke emission whereas at high rate of PFR and EGR results into significant reduction. This reason could be the lowest sensitivity of auto-ignition timing of ethanol's due to EGR constituent, it was the result of chemical kinetics study conduct by Magnus (Magnus Sjo¨berg 2010) using CHEMKIN codes.

5.5.4 Unburned Hydrocarbons

Fig. 5.79 shows that under different engine load the UHC emissions. Maximum UHC concentration occurs at BMEP of 2.1 bar (50% load) it decreases with increase in load. However, UHC emission is found to increase dramatically at BMEP of 2.1 bar with premixed ethanol. However premixed ethanol showed gradually declining influence with increase in BMEP from 3.1 to 4.2 bar, on UHC emissions.

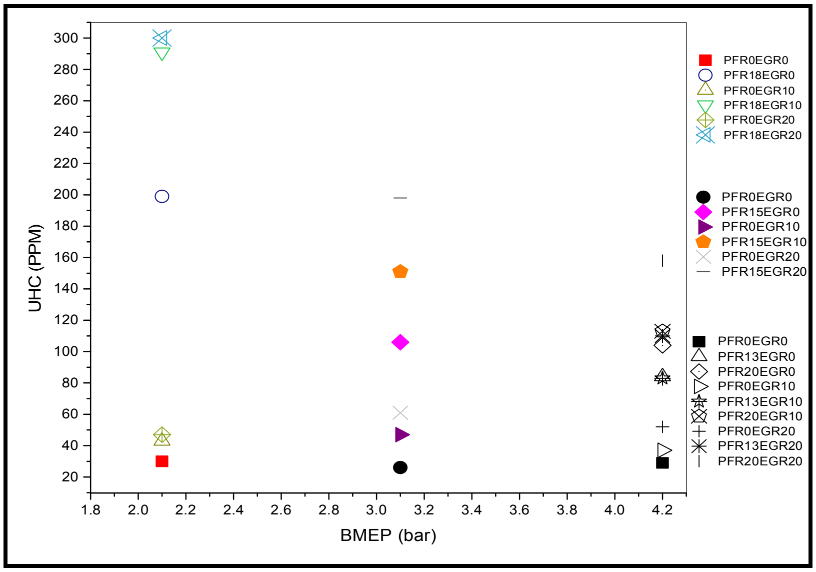


Figure 5.79: Variation of unburned hydrocarbons with engine loads.

The reason could be the low combustion temperature coupled with uniform air/ethanol mixture at BMEP of 2.1 impedes the oxidation of unburned hydrocarbons, however at higher load increase in cylinder temperature increases the oxidation rate. Increases in EGR also increase UHC emission, nevertheless with smaller rate. Formation of UHC during PCCI combustion mode is very much similar as it is formed in the petrol engine. Similar trend also reported (Cheung C S and Zhang Z H 2009) while operating diesel engine under premixed methanol combustion mode.

5.5.5 CARBON MONOXIDE

Fig. 5.80 shows CO emission with varying load and different premixed fuel condition. CO emissions are expected to increase under condition that causes incomplete combustion such as lower combustion temperature or poor mixing and are primarily controlled by the global or local air/fuel equivalence ratio. In the PCCI mode ethanol is burned as a homogenous charge and the flame has to propagate through the charge. The lower flame temperature due to high cooling effect of ethanol and EGR makes it difficult to have complete oxidation of the ethanol/air mixture which results into increase in CO emission. At any load premixed ethanol and EGR both lead to increase CO emission, however effect of EGR alone is more dominating at BMEP of 2.1 and 3.1 bar compared to 4.2 bar.

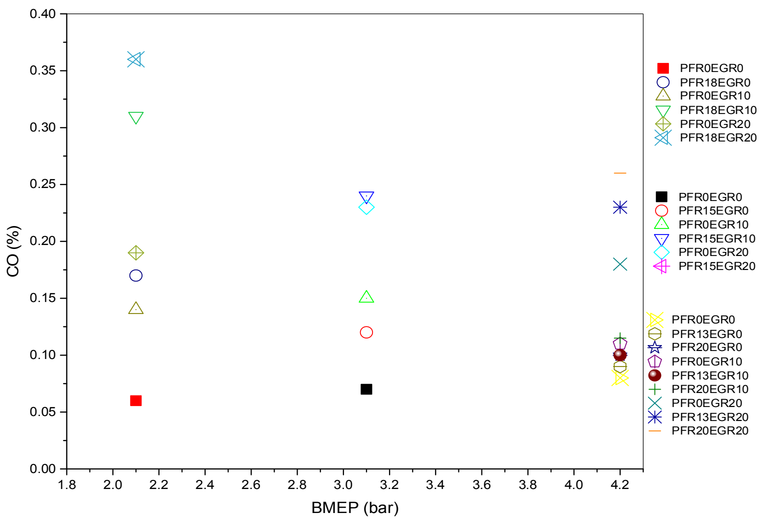


Figure 5.80: Variation of carbon monoxide with engine loads.

5.6 SUMMARY

Overall results showed strong potential of alcohol fuel utilization in the modified diesel engine without any difficulty of blending and emulsification. PCCI mode of combustion can be used successfully at medium and full engine load with and without EGR. At full engine load (at BMEP of 4.2 bar) premixed ethanol and EGR showed marginal gain in brake thermal efficiency and simultaneous reduction of NO_x and smoke emission.

CHAPTER 6: CONCLUSION AND FUTURE WORK

The results of Physico-chemical analysis of the test fuels and combustion, emissions and performance are summarized as under.

(i) When PCCI combustion is applied to the diesel engine without EGR, the BTE is reduced at BMEP of 2.1 bar (50% load) under all operating conditions. So it can be concluded at this load there is poor utilization of ethanol. However, at BMEP of 3.1 bar (75% load) while operating with PFR18EGR0, BTE increased by 4.48% compared to pure diesel. Several researchers (Cheng and Cheung 2008; Cheung C S and Zhang Z H 2009; Tsang and Zhang 2010) also concluded that at low engine load premixed alcohol produce negative effect whereas at high engine load it reversed the trend. The BSFC increases with the percentage of premixed ethanol at all engine loads, which is a consequence of the lower calorific value of ethanol. Nevertheless, percentage increase in BSFC was higher at BMEP of 2.1 and 3.1 bar engine loads compared to BMEP of 4.2 bar.

(ii) When PCCI combustion is applied to the diesel engine with EGR, the BTE is reduced at BMEP of 2.1 bar and 3.1 bar but at 4.2 bar (100 % load) there is slight positive effect was evident when premixed ethanol is used along with 10% and 20% of EGR. Thus, at BMEP of 4.2 bar the in-cylinder gas temperature is higher and there is more ethanol in the air/fuel charge, which tends to reduce the adverse effect on BTE. Results showed that in case of PFR20EGR10 and PFR20EGR20, BTE is increased by 2 and 1.67% respectively compared to pure diesel fuel. The BSFC increases with the percentage of premixed ethanol at all engine loads, which is a consequence of the lower calorific value of ethanol. Nevertheless, percentage increase in BSFC was higher at low and medium engine loads and small reduction is recorded with EGR at high engine load.

(iii) When PCCI combustion is applied to the diesel engine without EGR at BMEP of 4.2 bar, there is small increase in peak in-cylinder pressure, higher heat release rate, and an increase in ignition delay arising from the cooling effect of ethanol. However at BMEP of 2.1 bar and 3.1 bar, with and without EGR in-cylinder pressure decreases with premixed fuel ratio.

(iv) In case of PCCI combustion mode, there is an increase in fuel burnt in the premixed mode and subsequently it results into reduction of fuel burnt in the diffusion mode. Thus, it is one of cause of reduction in soot under premixed ethanol at all engine loads. Also, diesel has higher polyaromatic compounds compared to ethanol which results in higher smoke emission from pure diesel combustion.

(v) In case of PCCI combustion mode, there is an increase in UHC and CO emissions. At BMEP of 2.1 bar, UHC emission increases dramatically, however rate of increase is decreases with increase in engine loads. Moreover, combined effect of PFR and EGR lead to significantly increase in emissions.

- At BMEP of 2.1 bar, PFR18EGR20, UHC and CO emissions is increased by 6 and 10 times respectively compared to PFR0EGR0
- At BMEP of 3.1 bar, PFR15EGR20, UHC and CO emissions is increased by 7.6 and 4.7 times respectively compared to PFR0EGR0
- At BMEP of 4.2 bar, PFR20EGR20, UHC and CO emissions is increased by 4.85 and 2.25 times respectively compared to PFR0EGR0

(vi) In the PCCI combustion mode, there is a decrease in NO_x emission arising from the cooling effect of ethanol and EGR, nevertheless EGR alone effectively reduce NO_x significantly. Simultaneous reduction of NO_x and smoke is evident at BMEP of 3.1 and 4.2 bar.

- At BMEP of 2.1 bar up to 89% of NO_x reduction is achieved by using 20% EGR without premixed ethanol, however, combined application of premixed ethanol and EGR increases it to 95%. And using 18% of premixed ethanol without EGR leads to reduce NO_x emission by 50% compared to pure diesel. Hayes et al. and Ishida et al. (Hayes and Savage 1988; Ishida and Shohei Ueki 2010) also observed similar trends.

- At BMEP of 3.1 bar up to 92% of NO_x emission could be achieved by using 20% EGR without premixed ethanol (PFR0EGR20), however, combined application of premixed ethanol and EGR increases it to 95%. And using 15% of premixed ethanol without EGR (PFR15EGR0) lead to reduce NO_x emission by 33% compared to pure diesel. This is due to at high engine loads cooling effect produce by ethanol is weakened leading to lower reduction in NO_x emission.
- At BMEP of 4.2 bar up to 88% of NO_x emission could be reduced by using 20% EGR without premixed ethanol (PFR0EGR20), however, combined effect of premixed ethanol and EGR increases it to 92%. And maximum reduction of 16% is recorded using 20% premixed ethanol without EGR (PFR20EGR0).

(vii) At BMEP of 3.1 and 4.2 bar premixed ethanol can effectively reduce smoke as compared to diesel fuel. However, at relatively low load (BMEP of 2.1 bar) this could not happened.

- At BMEP of 2.1 bar smoke opacity increases with EGR as well with premixed ethanol. Low in-cylinder temperature reduces rate of oxidation and hence smoke. However only marginal variation is recoded.
- At BMEP of 3.1 bar smoke opacity increases with EGR but premixed ethanol lead to reduce it. In case of PFR15EGR0 smoke opacity is decreased by 20% compared to PFR0EGR0. However, other mixture composition did not show any significant reduction.
- At BMEP of 4.2 bar, smoke opacity corresponds to PFR13EGR10 and PFR20EGR10 decreased by 45% and 56% respectively compared to

PFR0EGR0. Also, at PFR20EGR0 about 34% reduction in smoke was found as compared PFR0EGR0. Therefore simultaneous reduction of NO_x and smoke was observed.

(viii) Ignition delay period decreases with increase in engine load, however PFR and EGR both lead to increase it at any load. Thus, increase in ignition delay period provides more time for fuel evaporation and reduces in-homogeneities in the reactant mixture, thus reducing NO_x formation from local temperature spikes and soot formation from locally rich mixtures.

(ix) It can be concluded that the application of PCCI combustion could lead to pave the way to achieve simultaneous reduction of smoke and NO_x emissions, while there is an increase in CO and UHC. Moreover, there could be a slight reduction in BTE at medium load but it increases at BMEP of 3.1 bar and 4.2 bar corresponds to different intake mixture composition.

6.1 SCOPE OF FUTURE WORK

Toroidal combustion geometry improves mixing under high engine loads and results did not provide any exact correlation of other parameters. Therefore, more detailed analysis is required to understand effect of increasing swirl ratio without changing compression ratio.

Since the experimental test-rig available does not provide flexibility to change the premixed ethanol in any proportion as per demand, therefore controller needs to be upgraded to change the injection duration in fraction of milliseconds. It gives greater range of premixed fuel ratio within the maximum limit. So that even at low engine loads, it will possible to obtain more numbers of different premixed fuel ratios.

As discussed compression ratio have significant effect on combustion kinetics, therefore PCCI combustion fueled with ethanol and diesel should be studied at variable compression ratio.

To reduce unburned hydrocarbons (UHC) and carbon monoxide (CO) emissions further enhancement of combustion geometry is required. Also, further study needs to be done to analyze effect of diesel oxidation catalyst. Moreover, with the help of advanced experimental facility one could see particulate mass concentration, particle number concentration and size distribution.

REFERENCES

Abu-Qudais, M., O. and M. Q. Haddad (2000). "The effect of alcohol fumigation on diesel engine performance and emissions." Energy Conversion & Management **41**: 389-399.

Adelman H (1979). "Alcohols in diesel engine." Society of Automotive Engineers **SAE Paper 790956**

Agarwal, A. K. (2007). "Biofuels (alcohols and bio-diesel) applications as fuels for internal combustion engines." Progress in Energy and Combustion Science **33**: 233-271.

Ajav E. A. and B. Bachchan Singh (1998). "Performance of a stationary diesel engine using vapourized as supplementary fuel." Biomass and Bioenergy **15**(6).

Akagawa H and H. A. Miyamoto T, Sasaki S, Shimazaki N, Hakeshi T, et al. (1999). "Approaches to solve problems of the premixed lean diesel combustion." Society of Automotive Engineers **SAE Paper 1999-01-0183**.

Alan, C. and A. Hansen, Qin Zhang , Peter W.L. Lyne (2005). "Ethanol–diesel fuel blends—a review." Bioresource Technology **96**: 277–285.

Alat, M. and H. Balat (2008). "Critical review of bio-diesel as a vehicular fuel." Energy Convers Manage **49 (10)**: 2727-2741.

Ali, Y. and M. Hanna (1994). "Alternative diesel fuels from vegetable oils." Bioresource Technology **50(2)**: 153-163.

Allsup, J. R. (1983). "Emissions, efficiency, and durability of agricultural diesel engines using low-proof ethanol." Society of Automotive Engineers **SAE Paper 831353**.

Alperstein, M. and W. Swim, Schweitzer, P. (1958). "Fumigation Kills Smoke - Improves Diesel Performance." Society of Automotive Engineers **SAE Paper 580058 (66)**.

Atcilla, B. and D. Orhan, Zehra¸ Sahcin (2002). "The Effects of Diesel-Ethanol Blends on Diesel Engine Performance." Energy Sources **24**.

Au, M. and J. Girard, Dibble R, et al. (1894). "1.9-Liter four-cylinder HCCI engine operation with exhaust gas re-circulation." Society of Automotive Engineers **SAE Paper 2001-01-1894**.

Baert, R. and D. E. Beckman, Veen A (1999). "Efficient EGR technology for future HD diesel engine emission targets." Society of Automotive Engineers **SAE Paper 1999-01-0837**.

Baranescu, R. (1986). "Fumigation of Alcohols in a Multicylinder Diesel Engine Evaluation of Potential." Society of Automotive Engineers **SAE Paper 860308**.

Barnes, K. and D. Kittelson, and Murphy, TE (1975). "Effect of Alcohols as Supplemental Fuel for Turbocharged Diesel Engines." Society of Automotive Engineers **SAE Paper 750469**.

Benvenutti, L. and C. Marques, Bertran, CA (2005). "Chemiluminescent emission data for kinetic modeling of ethanol combustion." Combustion Science and Technology **177(1)**: 1 -26.

Bhupendra Singh Chauhana and S. S. P. Naveen Kumar, Yong Du Jun (2011). "Experimental studies on fumigation of ethanol in a small capacity Diesel engine." Energy **36**: 1030-1038.

Bro, K. and P. Pedersen (1977). "Alternative diesel fuels: an investigation of methanol, ethanol, methane and ammonia in a DI diesel engine with pilot ignition." Society of Automotive Engineers **SAE Paper 770794**.

Broukhiyan, E. and S. Lestz (1981). "Ethanol Fumigation of a Light Duty Automotive Diesel Engine." Society of Automotive Engineers **SAE Paper 811209**.

Burtscher, H. (2005). "Physical characterization of particulate emissions from diesel engines: a review." Journal of Aerosol Science **36(7)**: 896-932.

ButamaxTM. (2013). "The Biobutanol Advantage." 2013, from http://www.butamax.com.

Caterpillar (2004). Heavy Truck Clean Diesel (HTCD) Program (Heavy Duty HCCI Development Activities). 11th Diesel Engine-Efficiency and Emissions Research (DEER) Conference. Coronado.

Chao, M. and T. Lin, Chao, HR, Chang FH, Chen CB (2001). "Effects of methanol-containing additive on emission characteristics from a heavy-duty diesel engine." Sci Total Environ **279**: 167–179.

Chen, J. and X. G. D. Gussert, C. Gupta and D. Foster (1981). "Ethanol fumigation of a turbocharged diesel engine." Society of Automotive Engineers **SAE Paper 810680**.

Cheng, A. and R. Dibble, Buchholz, BA (2002). "The Effect of Oxygenates on Diesel Engine Particulate Matter." SAE Technical Paper Series, no.2002-01-2002.

Cheng, C. H. and C. S. Cheung, Chan,T.L., Lee,S.C., Yao, C.D., Tsang, K.S. (2008). "Comparison of emissions of a direct injection diesel engine operating on bio-diesel with emulsified and fumigated methanol." Fuel **87**: 1870–1879.

Cheng, C. H. and C. S. C. Cheung, T. L. Lee, S. C. Yao, C. D. (2008). "Experimental investigation on the performance, gaseous and particulate emissions of a methanol fumigated diesel engine." Sci Total Environ **389**(1): 115-124.

Cheung C S and C. T. L. Zhang Z H, Yao Chunde (2009). "Investigation on the Effect of Port-Injected Methanol on the Perfromance and Emissions of a Diesel Engine at Different Engine Speeds." Energy & Fuels **23**(11).

Cheung, C. S. and C. Chuanhui Cheng, T. L., Chunde, Lee, Yao, S. C., Tsang, K. S. (2008). "Emissions Characteristics of a Diesel Engine Fueled with Bio-diesel and Fumigation Methanol." Energy & Fuels **22**: 906–914.

Christensen, M. and A. a. J. Hultqvist, B. (1999). "Demonstrating the Multi-Fuel Capability of a Homogeneous Charge Compression Ignition Engine with Variable Compression Ratio " Society of Automotive Engineers **SAE Paper 1999-01-3679**.

Dec, J. (1997). "A Conceptual Model of DI Diesel Combustion Based on Laser-Sheet Imaging." Society of Automotive Engineers **SAE Trans. Paper 970873 106 (3)**: 1319-1348.

Demirbas, A. (2007). "Progress and recent trends in biofuels." Progress in Energy and Combustion Science **33**: 1–18.

DieselNet. (2013). from https://www.dieselnet.com/standards/in/.

Edling, C. and O. Axelson (1984). "Risk factors of coronary heart disease among personnel in a bus company " Int Arch Occup Environ Health **54**: 181-183

EPA. (2012). "US Environmental Protection Agency ".

EPA. (2013). "US Environmental Protection Agency ".

Fahd, M. and Y. L. Ebna Alam Wenming, P. S. Chou, S. K. Yap, Christopher R (2013). "Experimental investigation of the performance and emission characteristics of direct injection diesel engine by water emulsion diesel under varying engine load condition." Applied Energy **102(0)**: 1042-1049.

Flowers, D. L. and S. M. Aceves, Westbrook, C. K., Smith, J. R., and Dibble, R. W. (1999). Sensitivity of Natural Gas HCCI Combustion to Fuel and Operating Parameters Using Detailed Kinetic Modeling Proceedings of the ASME Advanced Energy Systems Division AES.

Flynn, P., et al (1999). Premixed Charge Compression Ignition Engine with Optimal Combustion Control. W. I. P. Organization.

Gamble, J. and W. Jones, Minshall, S (1987). "Epidemiological-environmental study of diesel workers: chronic effects of diesel exhaust on the respiratory system." Environ Res **44**: 6-17

Gerdes, K. R., Suppes, G.J. (2001). "Miscibility of ethanol in diesel fuels." Ind. Eng. Chem. Res. **40 (3)**: 949-956.

Goering, C. and D. Wood (1981). "Overfueling a diesel engine with carburated ethanol." American Society of Agricultural Engineers (ASAE) **ASAE Transactions Paper 81-1048**

Gray III, A. and T. Ryan III (1997). "Homogeneous charge compression ignition (HCCI) of diesel fuel." Society of Automotive Engineers **SAE Paper 971676**.

Gray, K. and L. Zhao, Emptage, M. (2006). "Bioethanol." Current Opinion in Chemical Biology **10**: 141-146.

Greeves, G. and I. Khan, Wang, CHT, and Fenne, I (1977). "Origins of Hydrocarbon Emissions from Diesel Engines " Society of Automotive Engineers **SAE Paper 770259**.

Griffith, D. and L. Savage, and Goering, CE (1988). "Evaluation of an Ethanol-Fumigated Diesel Tractor." American Society of Agricultural Engineers (ASAE) **ASAE Paper 88-1558**.

Gunston, B. and L. Bridgman (1994). German Power-Boosting Systems. In Jane's fighting aircraft of world war II, Avenel, NJ: Crescent Books.

Handbook, I. C. E. (2004). SAE.

Harada, A. and N. Shimazaki, Sator, S, Miyamoto T, Akagawa H, Tsujimura K (1998). "The effects of mixture formation on premixed lean diesel combustion." Society of Automotive Engineers **SAE Paper 980533**.

Harrington, J. and R. Shishu (1973). "A Single-Cylinder Engine Study of the Effects of Fuel Type, Fuel Stoichiometry, and Hydrogen-to-Carbon Ratio and CO, NO, and HC Exhaust Emissions." Society of Automotive Engineers **SAE Paper 730476**.

Hashizume, T. and T. Miyamato, Akagawa, H, Tsujimura, K (1998). "Combustion and emission characteristics of multiple stage diesel combustion." Society of Automotive Engineers **SAE Paper 980505**.

Havemann, H. and M. Rao, Natarajan, A., Narasimhan, TL (1954). "Alcohol in Diesel Engines." Automobile Engineer **44(6)**: 256 -262.

Hayes, T. and L. Savage, White RA., and Sorenson SC (1988). "The effect of fumigation of different ethanol proofs on a turbocharged diesel engine." Society of Automotive Engineers **SAE Paper 880497**.

Hebbar, G. S. and K. B. Anantha (2012). "Control of NO_x from a DI dieselengine with hot EGR and ethanol fumigation:an experimental investigation." IOSR Journal of Engineering **2**: 45–53.

Heisey, J. and S. Lestz (1981). "Aqueous Alcohol Fumigation of a Single-Cylinder DI Diesel Engine." Society of Automotive Engineers **SAE Paper 811208**.

Heywood, J. (1988). Internal Combustion Engines Fundamentals, McGraw-Hill Co, New York.

Hilden, D. and J. Eckstrom, and Wolf, LR (2001). "The Emissions Performance of Oxygenated Diesel Fuels in a Prototype DI Diesel Engine." Society of Automotive Engineers **SAE Paper 2001-01-0650**.

Hilliard, J. and R. Wheeler (1979). "Nitrogen Dioxide in Engine Exhaust." Society of Automotive Engineers **SAE Paper 790691**.

Hochgreb, S. (1998). Handbook of Air Pollution from Internal combustion engines Pollutant formation and control, Academic Press.

Horng-Wen Wu and D.-J. O. Ren-Hung Wang, Ying-Chuan Chen, Teng-yu Chen (2011). "Reduction of smoke and nitrogen oxides of a partial HCCI engine using premixed gasoline and ethanol with air." Applied Energy **88**(11): 3882-3890.

Hosoya, M. and M. Shimoda (1996). "The application of diesel oxidation catalysts to heavy duty diesel engines in Japan." Appl Catal B **10**: 83-97.

Huang, Z. and H. Lu, Jiang, DM, Zeng K, Liu B, Zhang JQ, et al (2004). "Engine performance and emissions of a compression-ignition engine operating on the diesel–methanol blends." Proc Inst Mech Eng D J Automob **218**: 435-447.

Hwanam, K. and C. Byungchul (2008). "Effect of ethanol–diesel blend fuels on emission and particle size distribution in a common-rail direct injection diesel engine with warm-up catalytic converter." Renewable Energy **33**: 2222–2228.

Imran, A. and M. M. Varman, H. H. Kalam, M. A. (2013). "Review on alcohol fumigation on diesel engine: A viable alternative dual fuel technology for satisfactory engine performance and reduction of environment concerning emission." Renewable and Sustainable Energy Reviews **26**: 739-751.

Ishibashi, Y. and M. Asai (1996). "Improving the Exhaust Emissions of Two Stroke Engines by Applying the Activated Radical Combustion." Society of Automotive Engineers **SAE Paper 960742**.

Ishida, M. Y. and H. S. Shohei Ueki, Daisaku (2010). "Remarkable improvement of NO_x–PM trade-off in a diesel engine by means of bioethanol and EGR." Energy **35**(12): 4572-4581.

Iwashiro, Y. and T. N. Tsurushim, Y. Asaumi, Y. Aoyagi, Y (2002). "Fuel consumption improvement and operation range expansion in HCCI by direct water injection." Society of Automotive Engineers **SAE Paper 2002-01-0105**.

Jacobs, T. J. and S. V. Bohac, Assanis, D. N., Szymkowicz, P. G. (2005). "Lean and Rich Premixed Compression Ignition Combustion in a Light-Duty Diesel Engine." Society of Automotive Engineers **SAE Paper 2005-01-0166**.

Jiang, Q. and P. Ottikkutti, VanGerpen, J., VanMeter, D. (1990). "The effect of alcohol fumigation on diesel flame temperature and emissions." Society of Automotive Engineers **SAE Paper 900386**.

Jiménez-Espadafor and M. V. Francisco J. Torres, José A. Carvajal, Elisa Becerra, José A (2012). "Experimental analysis of low temperature combustion mode with diesel and bio-diesel fuels: A method for reducing NO_x and soot emissions." Fuel Processing Technology **103**: 57-63.

Joel Martinez-Frias and D. F. Salvador M. Aceves, J. Ray Smith, and Robert Dibble (2000). "HCCI Engine Control by Thermal Management." Society of Automotive Engineers **SAE Paper 2000-01-2869**.

Juttu, S. and S. S. Thipse, Marathe, N., and Gajendra Babu, M. (2008). "Diesel HCCI Combustion Control Parameters Study using n-Heptane Reduced Chemical Kinetic Mechanism." Society of Automotive Engineers **SAE Paper 2008-28-0036**.

Kaahaaina, N. B. and A. J. Simon, Caton, P. A., and Edwards, C. F. (2001). "Use of Dynamic Valving to Achieve Residual-Affected Combustion." Society of Automotive Engineers **SAE Paper 2001-01-0549**.

Kahn, G. O., P; and J. Weeks (1988). "Acute overexposure to diesel exhaust:report of 13 cases." Am J Ind Med **13**: 405-406

Kaneko, N. and H. Ando, Ogawa, H., Miyamoto, N. (2002). "Expansion of the operating range with in-cylinder water injection in a premixed charge compression ignition engine." Society of Automotive Engineers **SAE Paper 2002-01-1743**.

Kern, D. Q. (1997). Process Heat Transfer, Tata McGraw-Hill Edition.

Kim, D. and C. Lee (2006). "Improved emission characteristics of HCCI engine by various premixed fuels and cooled EGR." Fuel **85(5-6)**: 695-704.

Kimura, S. and O. Aoki, Ogawa, H, Muranaka, S, Enomoto, Y (1999). "New combustion concept for ultra-clean and high-efficiency small DI diesel engines." Society of Automotive Engineers **SAE Paper 1999-01-3681**.

Kittelson, D. (1998). "Engines and nanoparticles: a review. Journal of Aerosol Science." **29(5/6)**: 575-588.

Kowalewicz, A. (2006). Eco-diesel engine fueled with rapeseed oil methyl ester and ethanol. Part 2: comparison of emissions and efficiency for two base fuels: diesel fuel and ester", Proceedings of the Institution of Mechanical Engineers, Part D. Journal of Automobile Engineering.

Kumar MS and N. B. Ramesh A (2003). "Use of hydrogen to enhance the performance of a vegetable oil fueled compression ignition engine." Int J Hydrogen Energy **28**: 1143-1154.

Ladommatos N and Z. H. Abdelhalim SM, Hu Z (1998). "Effects of EGR on heat release in diesel combustion." Society of Automotive Engineers **SAE Paper 980184, Society of automotive Engineers Inc., Warrendale, PA**.

Ladommatos N and Z. H. Abdelhalim SM, Hu Z (1997). "The dilution,chemical, and thermal effects of exhaust gas re-circulation on diesel engine emissions—part 4: effects of carbon dioxide and water vapor." Society of Automotive Engineers **SAE Paper 971660, Society of automotive Engineers Inc,Warrendale, PA**.

Ladommatos, N. and S. Abdelhalim, Zhao, H., and Hu., Z (1998). "Effects of EGR on Heat Release in Diesel Combustion." Society of Automotive Engineers **SAE Paper 980184**.

Ladommatos, N. and S. M. Abdelhalim, Zhao, H., Hu, Z. (1996b). "The dilution, chemical, and thermal effects of exhaust gas re-circulation on diesel engine emissions - Part2: Effects of carbon dioxide." Society of Automotive Engineers **SAE Paper 961167, International Spring Fuels and Lubricants Meeting, Dearborn, Michigan**

Ladommatos, N. and S. M. Abdelhalim, Zhao, H., Hu, Z. (1997). "The Dilution, Chemical, and Thermal Effects of Exhaust Gas Re-circulation on Diesel Engine Emissions – Part 3: Effects of Water Vapor." Society of Automotive Engineers **SAE Paper 971659, International Spring Fuels and Lubricants Meeting, Dearborn, Michigan**.

Ladommatos, N. and S. M. Abdelhalim, Zhao, H., Hu, Z., (1996a). "The dilution, chemical, and thermal effects of exhaust gas re-circulation on diesel engine emissions – Part1: Effect of reducing inlet charge oxygen." Society of Automotive Engineers **SAE Paper 961165, International Spring Fuels and LubricantsMeeting, Dearborn, Michigan**.

Lapuerta, M. and J. Hernandez, Gimenez, F (2000). Evaluation of exhaust gas re-circulation as a technique for reducing diesel engine NO_x emissions. Proc Inst Mech Eng Part D J Autom Eng 2000.

Lavoie, G. and J. Heywood, Keck, JC. (1970). " Experimental and Theoretical Investigation of Nitric Oxide Formation in Internal Combustion Engines." Combustion Science and Technology **1**: 313-326.

Leahey, D. and B. Jones, Gilligan, J., Brown, L. et al. (2007). "Combustion of Bio-diesel- and Ethanol-Diesel Intake Injection Mixtures with intake injection." Society of Automotive Engineers **SAE Paper 2007-01-4011**.

Letcher, T. M. (1983). "Diesel blends for diesel engines." South African Journal of Science **79 (1)**: 4-7.

Lim, O. and N. Taeck Iida (2012). "The investigation about the effects of thermal stratification in combustion chamber on HCCI combustion fueled with DME/n-Butane using Rapid Compression Machine." Experimental Thermal and Fluid Science **39**(0): 123-133.

Lin CY and H. JC (2003). "An oxygenating additive for improving the performance and emission characteristics of marine diese lengines." Ocean Engineering **30**: 1699-1715.

Liotta, F. and D. Montalvio (1993). "The effect of oxygenated fuels on emissions from a modern heavy-duty diesel engine." Society of Automotive Engineers **SAE Paper 932734**.

Lu, X. and L. Z. Ji, L. Hou, Y. Huang, C. Huang, Z. (2007). "Experimental study and chemical analysis of n-heptane homogeneous charge compression ignition combustion with port injection of reaction inhibitors." Combustion and Flame **149**(3): 261-270.

Machrafi, H. C. and J. Simeon Amouroux (2008). "A parametric study on the emissions from an HCCI alternative combustion engine resulting from the auto-ignition of primary reference fuels." Applied Energy **85**(8): 755-764.

Magnus Sjo¨berg, J. E. D. (2010). "Effects of EGR and its constituents on HCCI auto-ignition of ethanol." Proceedings of the Combustion Institute.

Mahrous, A. F. M. and A. W. Potrzebowski, M. L. Xu, H. M. Tsolakis, A. Luszcz, P. (2009). "A modelling study into the effects of variable valve timing on the gas exchange process and performance of a 4-valve DI homogeneous charge compression ignition (HCCI) engine." Energy Conversion and Management **50**(2): 393-398.

Maurya, R. K. and A. K. Agarwal (2011). "Experimental study of combustion and emission characteristics of ethanol fueled port injected homogeneous charge compression ignition (HCCI) combustion engine." Applied Energy **88**(4): 1169-1180.

Megaritis, A. and D. W. Yap, M. L. (2007). "Effect of water blending on bioethanol HCCI combustion with forced induction and residual gas trapping." Energy **32**(12): 2396-2400.

Mellor, A. and J. Mello, Duffy, KP., Easley, WL., Faulkner, JC. (1998). "Skeletal Mechanism for NO_x Chemistry in Diesel Engines." Society of Automotive Engineers **SAE Paper 981450**.

Merryman, E. L. and A. Levy (1975). Nitrogen Oxide Formation in Flames: The Roles of NO, and Fuel Nitrogen. Proceedings of Fifteenth International Symposium on Combustion, The Combustion Institute.

Miguel Torres Garcı and T. s. n. L. Francisco Jose´ Jime´nez-Espadafor Aguilar (2009). "Experimental study of the performances of a modified diesel engine operating in homogeneous charge compression ignition (HCCI) combustion mode versus the original diesel combustion mode." Energy **34**: 159-171.

Miłosław Kozak and J. Merkisz (2009). "The Influence of Oxygenated Diesel Fuels on a Diesel Vehicle PM/NO$_x$ Emission Trade-Off." Society of Automotive Engineers **SAE Paper 2009-01-2696**.

Mingfa, Y. and Z. Z. Zheng Chen, Bo Zhang, Yuan Xing (2006). "Study on the controlling strategies of homogeneous charge compression ignition combustion with fuel of dimethyl ether and methanol." Fuel **85** 2046–2056.

MiŠosŠaw Kozak and J. Merkisz (2008). "The Influence of Synthetic Oxygenates on Euro IVDiesel Passenger Car Exhaust Emissions - Part 3." Society of Automotive Engineers **SAE Paper 2008-01-2387**.

Miyamoto, N. and H. Ogawa, Nurun, NM., Obata, K., and Arima,T. J. Jacobs (1998). "Smokeless, Low NO$_x$, High Thermal Efficiency, and Low Noise Diesel Combustion with Oxygenated Agents as Main Fuel." Society of Automotive Engineers **SAE Paper 980506**.

Murugesan A and S. R. Umarani C, Nedunchezhian N (2009). "Bio-diesel as an alternate fuel for diesel engines – a review." Renew Sustain Energy Rev **3(3)**: 653-662.

Nabi, N. and M. Minami, Ogawa, H., Miyamoto, N. (2000). "Ultra Low Emission and High Performance Diesel Combustion with Highly Oxygenated Fuel " Society of Automotive Engineers **SAE Paper 2000-01-0231**.

Najt PM and F. DE (1983). "Compression ignited homogeneous charge combustion." Society of Automotive Engineers **SAE Paper 830264**.

Nakagome K and M. K. Shimazaki N, Kobayashi S (1997). "Combustion and emissions characteristics of premixed lean diesel combustion engine." Society of Automotive Engineers **SAE Paper 970898**.

Naoyam, K. and A. Hirokazu, Hideyuki, Ogawa, Noboru Miyamoto (2002). "Expansion of the Operating Range with In-Cylinder Water Injection in a Premixed Charge Compression Ignition Engine." Society of Automotive Engineers **SAE Paper 2002-01-1743**.

Nightingale, J. A., Maggs, R and P. D. Cullinan, LE; Rogers, DF; Kinnersley, R; Chung, KF; Barnes, PJ; Ashmore, M; Newman-Taylor, A. (2000). "Airway inflammation after controlled exposure to diesel exhaust particulates." Am. J. Respir. Crit. Care Med(162): 161-166

Noguchi, M. and T. Tanaka, Takeuchi, Y (1979). "A study on gasoline engine combustion by observation of intermediate reactive products during combustion." Society of Automotive Engineers **SAE Paper 790840**.

Olsson, J.-O. and P. Tunestal, Johansson Bengt (2001). "Closed-loop control of an HCCI engine." Society of Automotive Engineers **SAE papers 2001-01-1031**.

Onishi, S. and S. Jo, Shoda, K, Jo, PD, Kato, S (1979). "Active thermo-atmosphere combustion (ATAC) – a new combustion process for internal combustion engines." Society of Automotive Engineers **SAE Paper 790501**.

Osses, M. and G. E. Andrews, Greenhough, J. (1998). "Diesel fumigation partial premixing for reduced particulate soot fraction emissions." Society of Automotive Engineers **SAE Paper 980532**.

Padala, S. and C. Woo, Kook, S.,Hawkes, E.R. (2012). Performance Improvement of a Compression Ignition Engine by Ethanol and Diesel Dual-Fuelling. 18th Australasian Fluid Mechanics Conference Launceston, Australia.

Pandey Shyam and S. P. K. Sharma Amit (2012). "Experimental investigation on the performance and emission characteristics of a diesel engine fueled with ethanol, diesel and Jatropha based bio-diesel blends." International Journal of Advances in Engineering & Technology **4**(2).

Pickett, L. M. and D. L. Siebers, Cherian, I. (2005). "Relationship Between Ignition Processes and the Lift-Off Length of Diesel Fuel Jets." Society of Automotive Engineers **SAE Paper 2005-01-3843**.

Praj Industries, P. "Praj breaks ground on cellulosic demonstration plant in India." from http://ethanolproducer.com/articles/10119/praj-breaks-ground-on-cellulosic-demonstration-plant-in-india.

Praj Industries, P. "Sweet Sorghum to Ethanol ", from http://praj.net/featured-sections.html.

Pundir, B. P. (2007). I C engines Combustion and emissions, Narosa Publishing House Pvt. Ltd., New Delhi.

Qiang, F. and F. Junhua, Jian Zhuang, Zhen Huang (2012). "Influences of pilot injection and exhaust gas re-circulation (EGR) on combustion and emissions in a HCCI-DI combustion engine." Applied thermal engineering **48**: 97-104.

Rassweiler, G. M. and L. Withrow (1980). "Motion pictures of engine Flames Correlated with Pressure Cards." Society of Automotive Engineers **SAE Trans. Paper 83-1938, Reissued as SAE Paper 800131**: 185-204.

Robert Kavet and K. M. Nauss (1990). "The Toxicity of Inhaled Methanol Vapors." Critical Reviews in Toxicology **Vol. 21, No. 1** Pages 21-50

Ryan III, T. and T. Callahan (1996). "Homogeneous charge compression ignition of diesel fuel." Society of Automotive Engineers **SAE Paper 961160**.

S Juttu, P. M. and N. V. M. S S Thipse, M K Gajendra Babu (2011). "Combined PCCI-DI Combustion to Meet EURO-IV Norms on LCV Engine - Experimental and Visulisation Study." SAE International **26-0031**.

Satgé de Caro and P. Mouloungui, Vaitilingom, Z., Berge, G., Ch, J. (2001). "Interest of combining an additive with diesel–ethanol blends for use in diesel engines." Fuel **80**(4): 565-574.

Savage, L. and R. White, Cole, S., and Pritchett, G. (1986). "Extended Performance of Alcohol Fumigation in Diesel Engines through Different Multipoint Alcohol Injection Timing Cycles." Society of Automotive Engineers **SAE Paper 861580**.

Shahid, E. and Y. Jamal (2008). "A review of bio-diesel as vehicular fuel." Renewable and Sustainable Energy Reviews **12(9)**: 2484-2494.

Sharke and Paul (2000). "Otto or Not, Here it Comes,." Mechanical Engineering **122**(6): 62-66.

Sharp, C. A. and S. A. Howell, Jobe, J. (2000). "The effect of bio-diesel fuels on transient emissions from modern diesel engines, part 1 regulated emissions and performance." Society of Automotive Engineers **SAE Paper 2000-01-1967**.

Shi, L. and Y. D. Cui, K. Peng, H. Chen, Y. (2006). "Study of low emission homogeneous charge compression ignition (HCCI) engine using combined internal and external exhaust gas re-circulation (EGR)." Energy **31**(14): 2665-2676.

Shropshire, G. and C. Goering (1982). "Ethanol Injection into a Diesel Engine." Transactions of the ASAE **25(3)**: 570-575.

Shropshire, G. J. and L. L. Bashford (1984). "A comparison of ethanol fumigation systems for a diesel engine." Agricultural Engineering **65(5)**: 17-23.

Simescu, S. and T. W. Ryan, Neely,G.D., Matheaus, A.C., Surampudi B. (2002). "Partial pre-mixed combustion with cooled and uncooled EGR in a heavy-duty diesel engine." Society of Automotive Engineers **SAE Paper 2002-01-0963**.

Sjöberg, M. and J. E. Dec (2011). "Effects of EGR and its constituents on HCCI auto-ignition of ethanol." Proceedings of the Combustion Institute **33**(2): 3031-3038.

Stanglmaier, R. and L. J. Matthews, RD (1999). "The effect of in cylinder wall wetting location on the HC emissions from SI engines." Society of Automotive Engineers **SAE Paper 1999-01-0502**.

Stockinger, V. and H. Schapertons, Kuhlmann, U. (1992). "Investigations on a gasoline engine working with self-ignition by compression " MTZ **53**: 80-85.

Stone, R. (1999). "Intorduction to internal combustion engines." Society of Automotive Engineers Inc; 3 edition.

Stone, R. (1999). Introduction to Internal Combustion Engines. MacMillan Press LTD, Society of Automotive Engineers.

Sullivan, N. and L. Bashford (1981). "Pre-turbocharger alcohol fumigation in a diesel tractor." Transactions of the ASAE **81**: 3581-3588.

Surawski, N. C. and B. R. Miljevic, B. A. Modini, R. L. Situ, R. Brown, R. J. Bottle, S. E. Ristovski, Z. D. (2010). "Particle emissions, volatility, and toxicity from an ethanol fumigated compression ignition engine." Environ Sci Technol **44**(1): 229-235.

Tayor, C. (1985). The Internal-Combustion Engine in Theory and Practice, Cambridge MIT Press.

Theobald, M. A. and R. Henry (1994). "Control of Engine Load Via Electromagnetic Valve Actuators." Society of Automotive Engineers **SAE Paper 940816**.

Thring, R. (1989). "Homogeneous – charge compression ignition (HCCI) engines." society of Automotive Engineers **SAE Paper 892068**.

Tomazic, D. and A. Pfeifer (2002). "Cooled EGR—a must or an option for 2002/04." Society of Automotive Engineers **SAE Paper 2002-01-0962**.

Tsang, K. S. and Z. H. C. Zhang, C. S. Chan, T. L. (2010). "Reducing Emissions of a Diesel Engine Using Fumigation Ethanol and a Diesel Oxidation Catalyst." Energy & Fuels **24**(11): 6156-6165.

Wagner, G. and B. Zarah, Kozinski, AA (1979). "Practicality of alcohols as motor fuel." Society of Automotive Engineers **SAE Paper 790429**.

Wang, W. and N. Clark, Lyons DW, Yang RM, Gautam M, Bata RM, et al. (1997). "Emissions comparisons from alternative fuel buses and diesel buses with a chassis dynamometer testing facility." Environ Sci Technol(31): 3132-3137.

Wrage, K. E. and C. E. Goering (1980). "Technical feasibility of diesohol." Transactions of the ASAE **23 (6)** 1338–1343

Xing-Cai Chen and W. H. Lü, Zhen (2005). "A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR. Part 1. The basic characteristics of HCCI combustion." Fuel **84**(9): 1074-1083.

Xing-Cai Chen and W. H. Lü, Zhen (2005). "A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR. Part 2. Effect of operating conditions and EGR on HCCI combustion." Fuel **84**(9): 1084-1092.

Yao, C. and C. Cheung, Cheng,CH, Wang, YS (2007). "Reduction of smoke and NO_x from diesel engines using a diesel/methanol compound combustion system." Energy & Fuels **21**: 689-691.

Yao CD and C. C. Cheung CS, Wang YS, Chan TL, Lee SC (2008). "Effect of Diesel/methanol compound combustion on diesel engine combustion and emissions." Energy Conversion and Management **49**(6): 696-704.

Yao, M. Z. and H. Zhaolei Liu (2009). "Progress and recent trends in homogeneous charge compression ignition (HCCI) engines." Progress in Energy and Combustion Science **35**(5): 398-437.

Yap, D. and A. W. Megaritis, M. L. (2004). "An Investigation into Bioethanol Homogeneous Charge Compression Ignition (HCCI) Engine Operation with Residual Gas Trapping." Energy & Fuels **18**(5): 1315-1323.

Yeh, L. and D. Rickeard, Duff, JLC., Bateman, JR., Schlosberg, RH., Caers, RF (2001). "Oxygenates: An Evaluation of their Effects on Diesel Emissions." Society of Automotive Engineers **SAE Paper 2001-01-2019**.

Yoshinaka, T. and K. Nakagone (1996). "Emission characteristics of premixed lean diesel combustion with extremely early staged fuel injection." Society of Automotive Engineers **AE Paper 961163**.

Youn, I. M. P. and H. G. L. Su Han Roh, Chang Sik (2011). "Investigation on the fuel spray and emission reduction characteristics for dimethyl ether (DME) fueled multi-cylinder diesel engine with common-rail injection system." Fuel Processing Technology **92**(7): 1280-1287.

Yu, R. and S. Shahed (1981). "Effects of Injection Timing and Exhaust Gas Re-circulation on Emissions from a D.I. Diesel Engine." Society of Automotive Engineers **SAE Paper 811234**.

Zhang Z.H. and K. S. Tsang, Cheunga,C.S., Chana, T.L., Yao (2011). "Effect of fumigation methanol and ethanol on the gaseous and particulate emissions of a direct-injection diesel engine." Atmospheric Environment **45** 2001-2008.

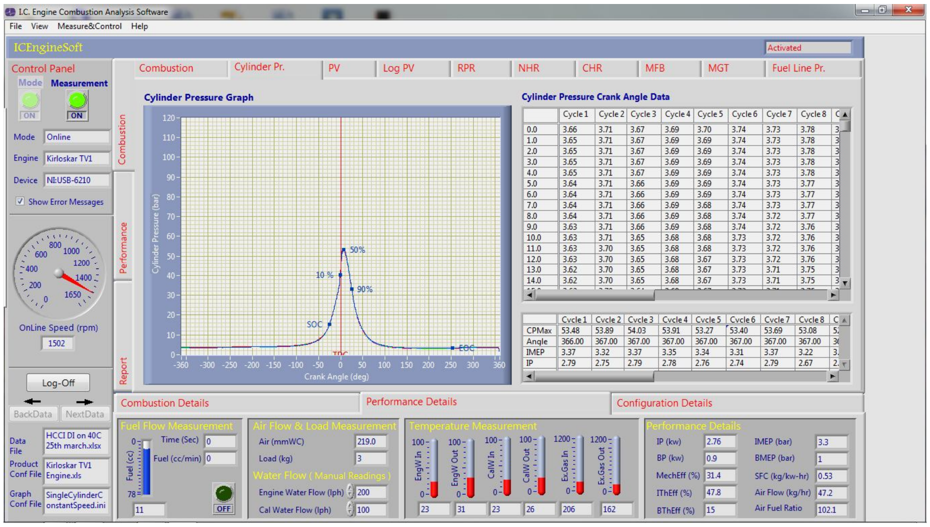
Zhang, Z. H. and C. S. C. Cheung, T. L. Yao, C. D. (2009). "Emission reduction from diesel engine using fumigation methanol and diesel oxidation catalyst." Sci Total Environ **407(15)**: 4497-4505.

Zhang, Z. H. and C. S. Cheungb, Chan, T.L., Yao, C.D. (2010). "Experimental investigation of regulated and unregulated emissions from a diesel engine fueled with Euro V diesel fuel and fumigation methanol." Atmospheric Environment **44**: 1054-1061.

Zhao F and A. D. Asmus TW, Dec JE, Eng JA, Najt (2003). Homogeneous charge compression ignition (HCCI) engines: key research and development issues, Society of Automotive Engineers.

Zhao, H. and Z. Peng, Ladommatos, N (2001). "Understanding of controlled auto-ignition combustion in a four-stroke gasoline engine." Proc. Inst Mech Eng **215**: 1297-1310.

Zuohua Huang and D. J. Hongbing Lu, Ke Zeng, Bing Liu, Junqiang Zhang, Xibin Wang (2004). "Combustion behaviors of a compression-ignition engine fueled with diesel/methanol blends under various fuel delivery advance angles." Bioresource Technology **95**.



Mode: Online
Engine: Kirloskar TV1
Device: NI:USB-6210
☑ Show Error Messages
OnLine Speed (rpm): 1502
Data File: HCCI DI on 40C 25th march.xlsx
Product Conf File: Kirloskar TV1 Engine.xls
Graph Conf File: SingleCylinderConstantSpeed.ini

Cylinder Pressure Crank Angle Data

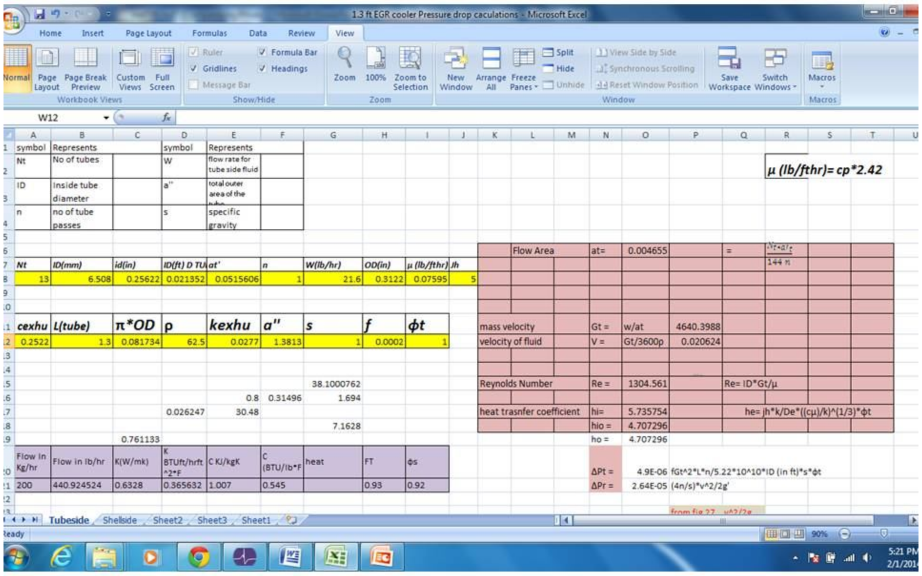
	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Cycle 5	Cycle 6	Cycle 7	Cycle 8
0.0	3.66	3.71	3.67	3.69	3.70	3.74	3.73	3.78
1.0	3.65	3.71	3.67	3.69	3.69	3.74	3.73	3.78
2.0	3.65	3.71	3.67	3.69	3.69	3.74	3.73	3.78
3.0	3.65	3.71	3.67	3.69	3.69	3.74	3.73	3.78
4.0	3.65	3.71	3.67	3.69	3.69	3.74	3.73	3.78
5.0	3.64	3.71	3.66	3.69	3.69	3.74	3.73	3.77
6.0	3.64	3.71	3.66	3.69	3.69	3.74	3.73	3.77
7.0	3.64	3.71	3.66	3.69	3.68	3.74	3.73	3.77
8.0	3.64	3.71	3.66	3.69	3.68	3.74	3.72	3.77
9.0	3.63	3.71	3.66	3.69	3.68	3.74	3.72	3.76
10.0	3.63	3.71	3.65	3.68	3.68	3.73	3.72	3.76
11.0	3.63	3.70	3.65	3.68	3.68	3.73	3.72	3.76
12.0	3.63	3.70	3.65	3.68	3.67	3.73	3.72	3.76
13.0	3.62	3.70	3.65	3.68	3.67	3.73	3.71	3.75
14.0	3.62	3.70	3.65	3.68	3.67	3.73	3.71	3.75

	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Cycle 5	Cycle 6	Cycle 7	Cycle 8
CPMax	53.48	53.89	54.03	53.91	53.27	53.40	53.69	53.08
Angle	366.00	367.00	367.00	367.00	367.00	367.00	367.00	367.00
IMEP	3.37	3.32	3.37	3.35	3.34	3.31	3.37	3.22
IP	2.79	2.75	2.79	2.78	2.76	2.74	2.79	2.67

Fuel Flow Measurement: Time (Sec) 0; Fuel (cc/min) 0; Fuel (cc) 11
Air Flow & Load Measurement: Air (mmWC) 219.0; Load (kg) 3
Water Flow (Manual Readings): Engine Water Flow (lph) 200; Cal Water Flow (lph) 100
Temperature Measurement: EngW In 23; EngW Out 31; CalW In 23; CalW Out 26; Ex.Gas In 206; Ex.Gas Out 162

IP (kw)	2.76	IMEP (bar)	3.3
BP (kw)	0.9	BMEP (bar)	1
MechEff (%)	31.4	SFC (kg/kw-hr)	0.53
IThEff (%)	47.8	Air Flow (kg/hr)	47.2
BThEff (%)	15	Air Fuel Ratio	102.1

Plate A1: User interface of EnginesoftV sofatwere.



symbol	Represents	symbol	Represents
Nt	No of tubes	W	flow rate for tube side fluid
ID	Inside tube diameter	a''	total outer area of the tube
n	no of tube passes	s	specific gravity

μ (lb/fthr)= cp*2.42

Nt	ID(mm)	id(in)	ID(ft) D TU	at'	n	W(lb/hr)	OD(in)	μ (lb/fthr)	Jh
13	6.508	0.25622	0.021352	0.0515606	1	21.6	0.3122	0.07595	5

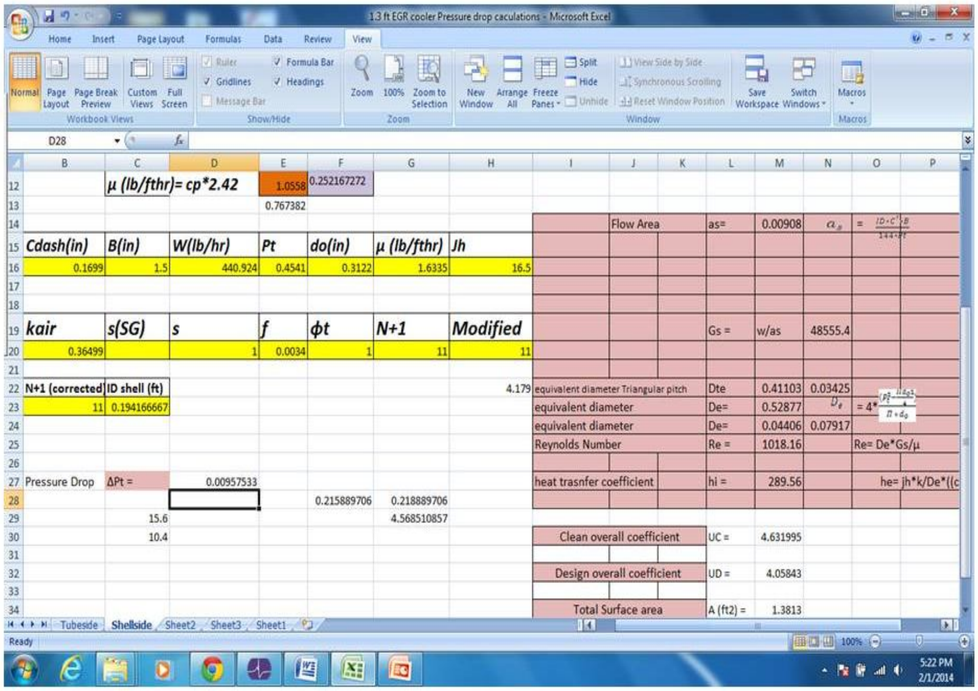
cexhu	L(tube)	π*OD	ρ	kexhu	a''	s	f	ϕt
0.2522	1.3	0.081734	62.5	0.0277	1.3813	1	0.0002	1

38.1000762
0.8 0.31496 1.694
0.026247 30.48
7.1628
0.761133

Flow In Kg/hr	Flow in lb/hr	K(W/mk)	K BTUft/hrft^2*F	C KJ/kgK	C (BTU/lb*F	heat	FT	ϕs
200	440.924524	0.6328	0.365632	1.007	0.545		0.93	0.92

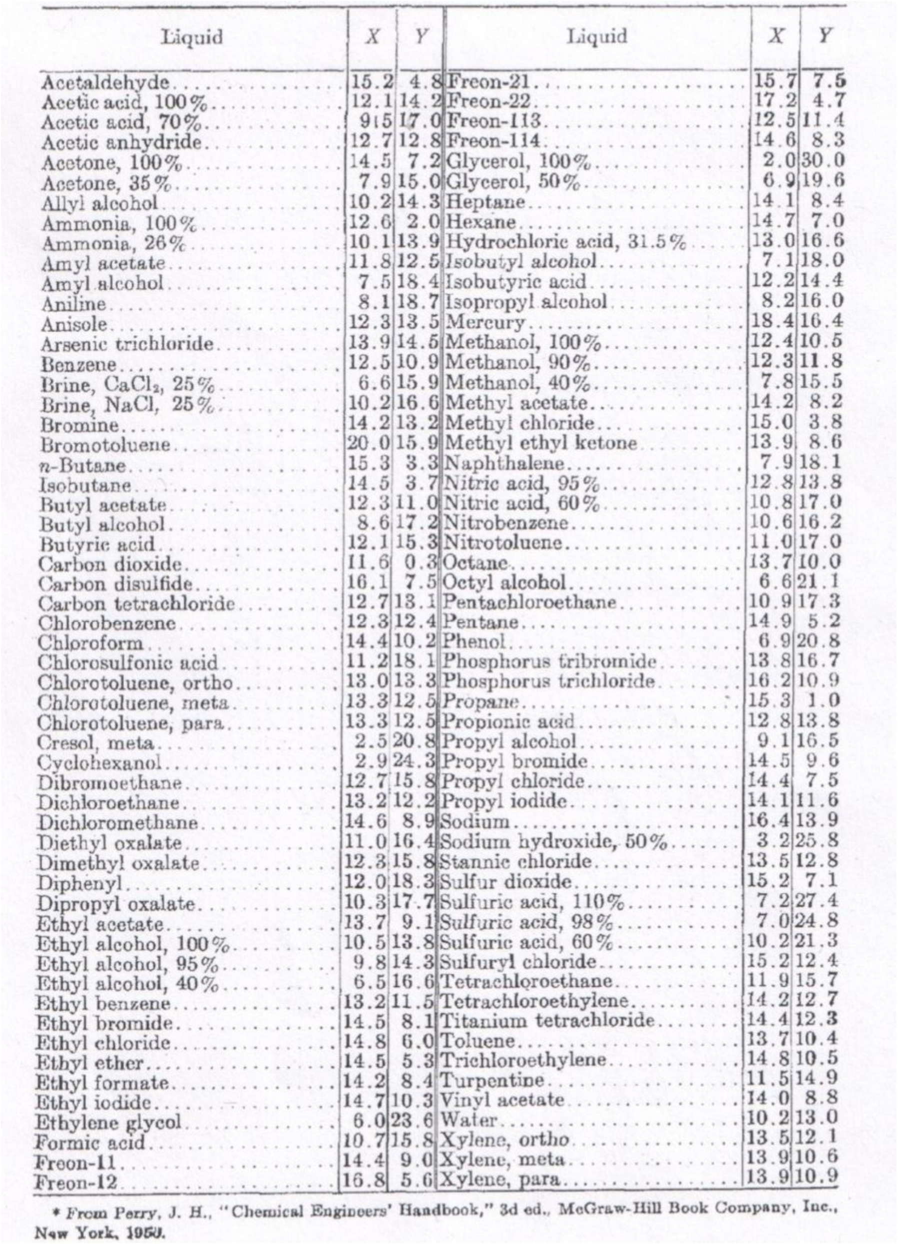
Flow Area at= 0.004655 = $\frac{N_t * a/t}{144\ n}$
mass velocity Gt = w/at 4640.3988
velocity of fluid V = Gt/3600ρ 0.020624
Reynolds Number Re = 1304.561 Re= ID*Gt/μ
heat trasnfer coefficient hi= 5.735754 he= jh*k/De*((cμ)/k)^(1/3)*ϕt
hio = 4.707296
ho = 4.707296
ΔPt = 4.9E-06 fGt^2*L*n/5.22*10^10*ID (in ft)*s*ϕt
ΔPr = 2.64E-05 (4n/s)*v^2/2g'
from fig 27 v^2/2g

Plate A2: Excel calculation sheet to determine heat transfer co-efficient (tube side)



	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
12		μ (lb/fthr)= cp*2.42		1.0558	0.252167272										
13				0.767382											
14									Flow Area		as=	0.00908	a_s	=	
15	Cdash(in)	B(in)	W(lb/hr)	Pt	do(in)	μ (lb/fthr)	Jh								
16	0.1699	1.5	440.924	0.4541	0.3122	1.6335	16.5								
17															
18															
19	kair	s(SG)	s	f	ϕt	N+1	Modified				Gs =	w/as	48555.4		
20	0.36499		1	0.0034	1	11	11								
21															
22	N+1 (corrected)	ID shell (ft)					4.179	equivalent diameter Triangular pitch			Dte	0.41103	0.03425		
23	11	0.194166667						equivalent diameter			De=	0.52877	D_e	= 4*	
24								equivalent diameter			De=	0.04406	0.07917		
25								Reynolds Number			Re =	1018.16		Re= De*Gs/μ	
26															
27	Pressure Drop	ΔPt =	0.00957533					heat trasnfer coefficient			hi =	289.56			he= jh*k/De*((c
28					0.215889706	0.218889706									
29		15.6				4.568510857									
30		10.4						Clean overall coefficient			UC =	4.631995			
31															
32								Design overall coefficient			UD =	4.05843			
33															
34								Total Surface area			A (ft2) =	1.3813			

Plate A3: Excel calculation sheet to determine heat transfer co-efficient(shell side)



Liquid	X	Y	Liquid	X	Y
Acetaldehyde	15.2	4.8	Freon-21	15.7	7.5
Acetic acid, 100%	12.1	14.2	Freon-22	17.2	4.7
Acetic acid, 70%	9.5	17.0	Freon-113	12.5	11.4
Acetic anhydride	12.7	12.8	Freon-114	14.6	8.3
Acetone, 100%	14.5	7.2	Glycerol, 100%	2.0	30.0
Acetone, 35%	7.9	15.0	Glycerol, 50%	6.9	19.6
Allyl alcohol	10.2	14.3	Heptane	14.1	8.4
Ammonia, 100%	12.6	2.0	Hexane	14.7	7.0
Ammonia, 26%	10.1	13.9	Hydrochloric acid, 31.5%	13.0	16.6
Amyl acetate	11.8	12.5	Isobutyl alcohol	7.1	18.0
Amyl alcohol	7.5	18.4	Isobutyric acid	12.2	14.4
Aniline	8.1	18.7	Isopropyl alcohol	8.2	16.0
Anisole	12.3	13.5	Mercury	18.4	16.4
Arsenic trichloride	13.9	14.5	Methanol, 100%	12.4	10.5
Benzene	12.5	10.9	Methanol, 90%	12.3	11.8
Brine, $CaCl_2$, 25%	6.6	15.9	Methanol, 40%	7.8	15.5
Brine, NaCl, 25%	10.2	16.6	Methyl acetate	14.2	8.2
Bromine	14.2	13.2	Methyl chloride	15.0	3.8
Bromotoluene	20.0	15.9	Methyl ethyl ketone	13.9	8.6
n-Butane	15.3	3.3	Naphthalene	7.9	18.1
Isobutane	14.5	3.7	Nitric acid, 95%	12.8	13.8
Butyl acetate	12.3	11.0	Nitric acid, 60%	10.8	17.0
Butyl alcohol	8.6	17.2	Nitrobenzene	10.6	16.2
Butyric acid	12.1	15.3	Nitrotoluene	11.0	17.0
Carbon dioxide	11.6	0.3	Octane	13.7	10.0
Carbon disulfide	16.1	7.5	Octyl alcohol	6.6	21.1
Carbon tetrachloride	12.7	13.1	Pentachloroethane	10.9	17.3
Chlorobenzene	12.3	12.4	Pentane	14.9	5.2
Chloroform	14.4	10.2	Phenol	6.9	20.8
Chlorosulfonic acid	11.2	18.1	Phosphorus tribromide	13.8	16.7
Chlorotoluene, ortho	13.0	13.3	Phosphorus trichloride	16.2	10.9
Chlorotoluene, meta	13.3	12.5	Propane	15.3	1.0
Chlorotoluene, para	13.3	12.5	Propionic acid	12.8	13.8
Cresol, meta	2.5	20.8	Propyl alcohol	9.1	16.5
Cyclohexanol	2.9	24.3	Propyl bromide	14.5	9.6
Dibromoethane	12.7	15.8	Propyl chloride	14.4	7.5
Dichloroethane	13.2	12.2	Propyl iodide	14.1	11.6
Dichloromethane	14.6	8.9	Sodium	16.4	13.9
Diethyl oxalate	11.0	16.4	Sodium hydroxide, 50%	3.2	25.8
Dimethyl oxalate	12.3	15.8	Stannic chloride	13.5	12.8
Diphenyl	12.0	18.3	Sulfur dioxide	15.2	7.1
Dipropyl oxalate	10.3	17.7	Sulfuric acid, 110%	7.2	27.4
Ethyl acetate	13.7	9.1	Sulfuric acid, 98%	7.0	24.8
Ethyl alcohol, 100%	10.5	13.8	Sulfuric acid, 60%	10.2	21.3
Ethyl alcohol, 95%	9.8	14.3	Sulfuryl chloride	15.2	12.4
Ethyl alcohol, 40%	6.5	16.6	Tetrachloroethane	11.9	15.7
Ethyl benzene	13.2	11.5	Tetrachloroethylene	14.2	12.7
Ethyl bromide	14.5	8.1	Titanium tetrachloride	14.4	12.3
Ethyl chloride	14.8	6.0	Toluene	13.7	10.4
Ethyl ether	14.5	5.3	Trichloroethylene	14.8	10.5
Ethyl formate	14.2	8.4	Turpentine	11.5	14.9
Ethyl iodide	14.7	10.3	Vinyl acetate	14.0	8.8
Ethylene glycol	6.0	23.6	Water	10.2	13.0
Formic acid	10.7	15.8	Xylene, ortho	13.5	12.1
Freon-11	14.4	9.0	Xylene, meta	13.9	10.6
Freon-12	16.8	5.6	Xylene, para	13.9	10.9

* From Perry, J. H., "Chemical Engineers' Handbook," 3d ed., McGraw-Hill Book Company, Inc., New York, 1950.

Plate A4: Viscosities of liquids –cordinate to be used

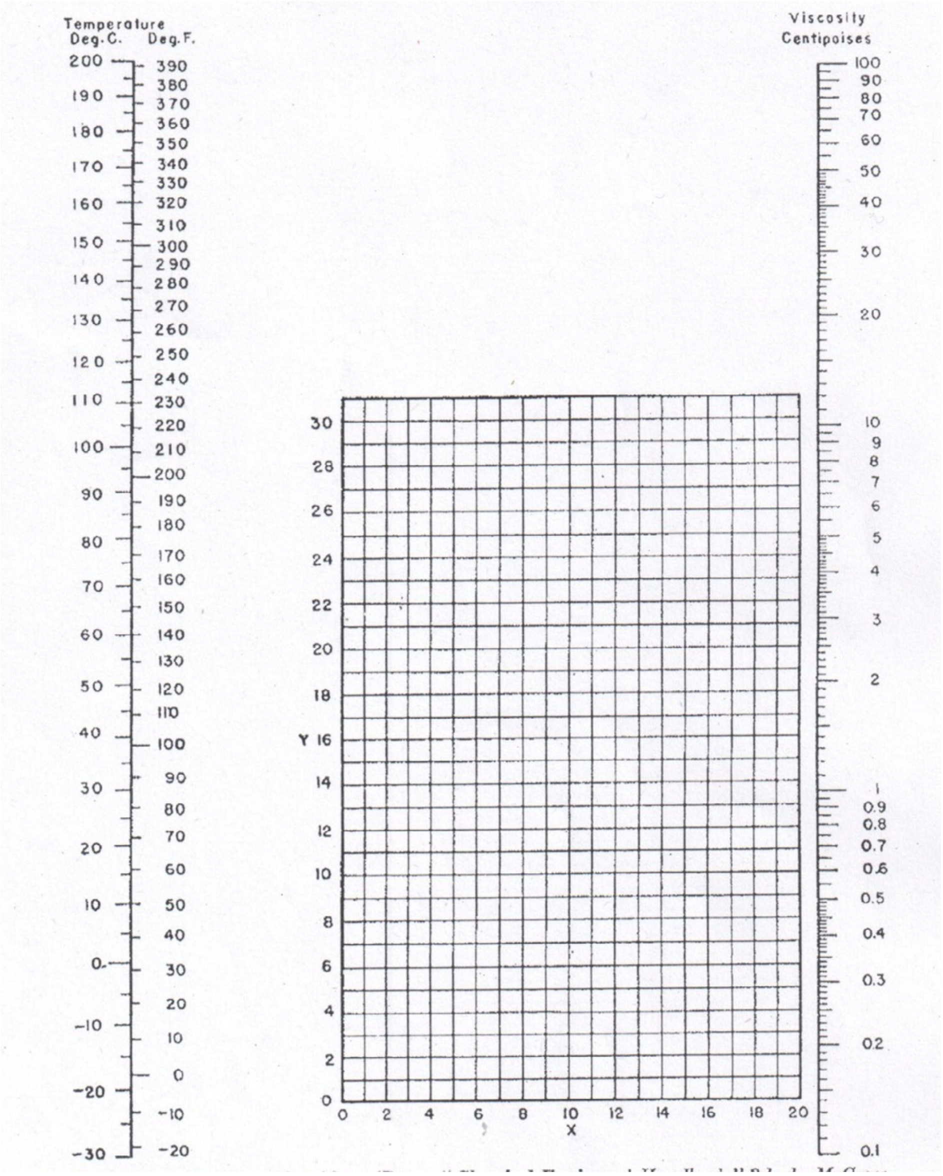
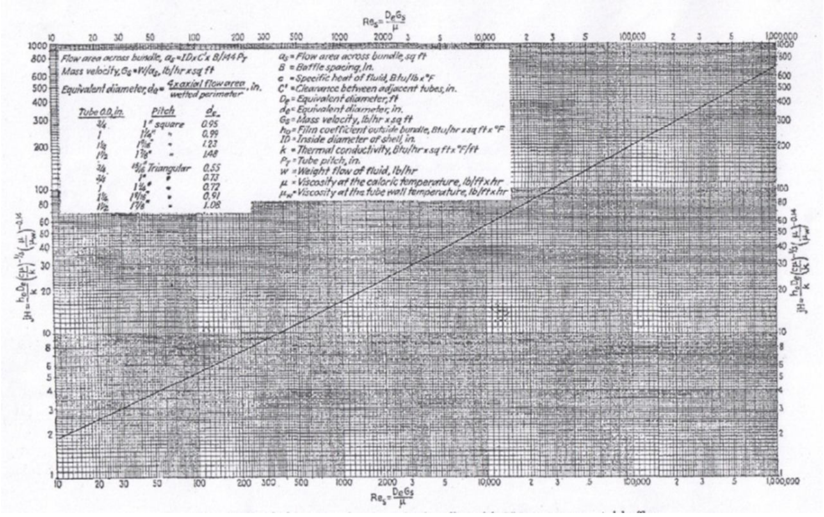


Plate A5: Viscosities of liquids- exhaust gases (based on co-ordinates)



Flow area across bundle, $a_s = ID \times C' \times B/144 P_T$

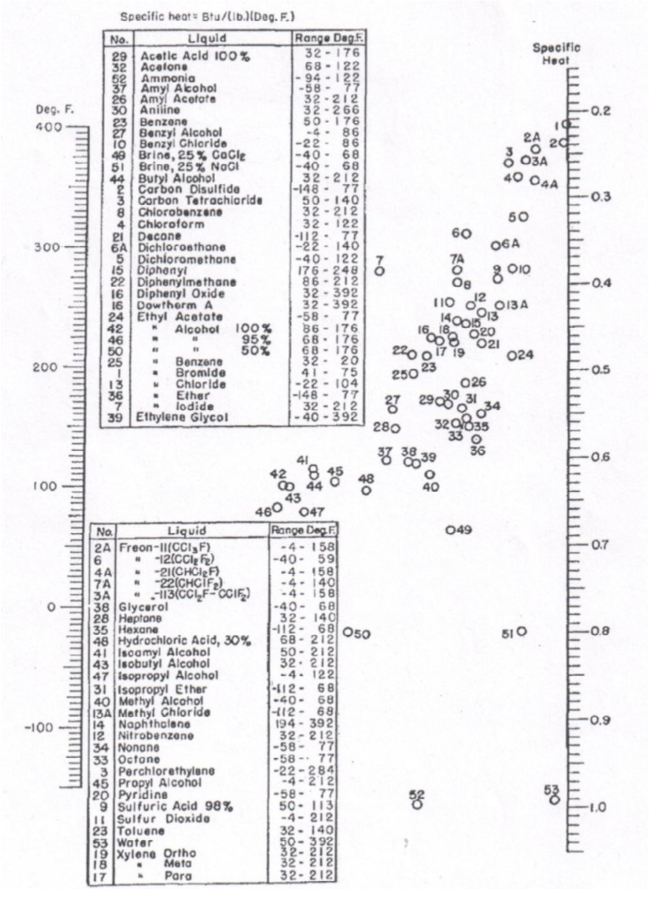
Mass velocity, $G_s = W/a_s$, lb/hr x sq ft

Equivalent diameter, $d_e = \frac{4 \times \text{axial flow area}}{\text{wetted perimeter}}$, in.

Tube O.D., in.	Pitch	d_e
3/4	1" square	0.95
1	1¼" "	0.99
1¼	1 9/16" "	1.23
1½	1 7/8" "	1.48
3/4	15/16" Triangular	0.55
3/4	1" "	0.73
1	1¼" "	0.72
1¼	1 9/16" "	0.91
1½	1 7/8" "	1.08

a_s = Flow area across bundle, sq ft
B = Baffle spacing, in.
c = Specific heat of fluid, Btu/lb x °F
C' = Clearance between adjacent tubes, in.
D_e = Equivalent diameter, ft
d_e = Equivalent diameter, in.
G_s = Mass velocity, lb/hr x sq ft
h_o = Film coefficient outside bundle, Btu/hr x sq ft x °F
ID = Inside diameter of shell, in.
k = Thermal conductivity, Btu/hr x sq ft x °F/ft
P_T = Tube pitch, in.
W = Weight flow of fluid, lb/hr
μ = Viscosity at the caloric temperature, lb/ft x hr
μ_w = Viscosity at the tube wall temperature, lb/ft x hr

Plate A6: Shell side heat transfer curve.



Specific heat = Btu/(lb.)(Deg. F.)

No.	Liquid	Range Deg.F.
29	Acetic Acid 100%	32 - 176
32	Acetone	68 - 122
52	Ammonia	- 94 - 122
37	Amyl Alcohol	-58 - 77
26	Amyl Acetate	32 - 212
30	Aniline	32 - 266
23	Benzene	50 - 176
27	Benzyl Alcohol	-4 - 86
10	Benzyl Chloride	-22 - 86
49	Brine, 25% CaCl$_2$	-40 - 68
51	Brine, 25% NaCl	-40 - 68
44	Butyl Alcohol	32 - 212
2	Carbon Disulfide	-148 - 77
3	Carbon Tetrachloride	50 - 140
8	Chlorobenzene	32 - 212
4	Chloroform	32 - 122
21	Decane	-112 - 77
6A	Dichloroethane	-22 - 140
5	Dichloromethane	-40 - 122
15	Diphenyl	176 - 248
22	Diphenylmethane	86 - 212
16	Diphenyl Oxide	32 - 392
16	Dowtherm A	32 - 392
24	Ethyl Acetate	-58 - 77
42	" Alcohol 100%	86 - 176
46	" " 95%	68 - 176
50	" " 50%	68 - 176
25	" Benzene	32 - 20
1	" Bromide	41 - 75
13	" Chloride	-22 - 104
36	" Ether	-148 - 77
7	" Iodide	32 - 212
39	Ethylene Glycol	-40 - 392

No.	Liquid	Range Deg.F.
2A	Freon-11(CCl$_3$F)	-4 - 158
6	" -12(CCl$_2$F$_2$)	-40 - 59
4A	" -21(CHCl$_2$F)	-4 - 158
7A	" -22(CHClF$_2$)	-4 - 140
3A	" -113(CCl$_2$F-CClF$_2$)	-4 - 158
38	Glycerol	-40 - 68
28	Heptane	32 - 140
35	Hexane	-112 - 68
48	Hydrochloric Acid, 30%	68 - 212
41	Isoamyl Alcohol	50 - 212
43	Isobutyl Alcohol	32 - 212
47	Isopropyl Alcohol	-4 - 122
31	Isopropyl Ether	-112 - 68
40	Methyl Alcohol	-40 - 68
13A	Methyl Chloride	-112 - 68
14	Naphthalene	194 - 392
12	Nitrobenzene	32 - 212
34	Nonane	-58 - 77
33	Octane	-58 - 77
3	Perchlorethylene	-22 - 284
45	Propyl Alcohol	-4 - 212
20	Pyridine	-58 - 77
9	Sulfuric Acid 98%	50 - 113
11	Sulfur Dioxide	-4 - 212
23	Toluene	32 - 140
53	Water	50 - 392
19	Xylene Ortho	32 - 212
18	" Meta	32 - 212
17	" Para	32 - 212

Plate A7: Specific heats of liquids (water).

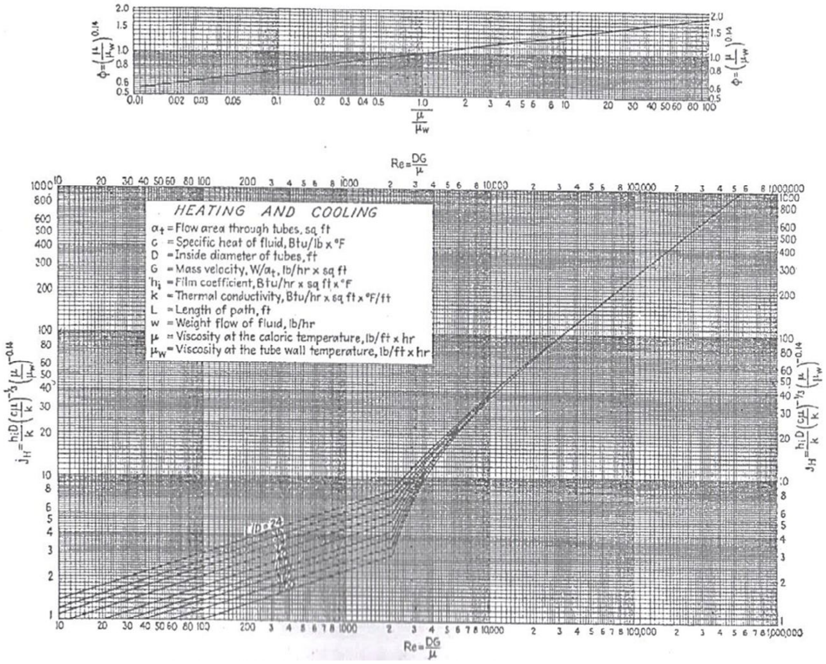


Plate A8: Tube-side heat-tranfser curve.

CURRICULUM-VITAE

SHYAM PANDEY
Assistant Professor –
Mechanical Engineering
University of Petroleum and
Energy Studies, Dehradun
(U.K.)

Educational Qualification: M. Tech. (Industrial Metallurgy)
- 2001-2002, Indian Institute of Technology Roorkee,8.3 CGPA.

B.E. (Mechanical Engg.)
- 1997-2001, Hitkarini College of Engineering and Technology Jabalpur (M.P.), RGPV University, 71%.

Professional Experience:

Total Experience: 12 Years, (Teaching : 10½ Years &Industrial : 1 ½ Years)
Details are given in Annexure –I.

Paper Published: International Journal -02 Nos
National Conferences Papers- 01 Nos
Details are given in Annexure –II.

Patent filed: Indian Design patent -01 Nos
Fuel Injection System -01
Details are given in Annexure –II.

Sponsored Project: Uttarakhand Council of Science and Technology, Dehradun -01
Details are given in Annexure –II.

FDP Organized 01

Seminars/ FDP Attended : 03 Nos, Details are given in Annexure –III.

Teaching Experience:

Aug 2007– till date UNIVERSITY OF PETROLEUM & ENERGY STUDIES, DEHRADUN.
Asst. Professor (SG)- Mechanical Engineering Department

Jan 2006-July 2007	ENERCON (INDIA) LTD. DAMAN Senior Engineer – Enercon Training Academy.
May 2003 – Dec 2005	Lecturer - Mechanical Engg., Hitkarini College of Engineering & Technology, Jabalpur (M.P.)
Other Activities :	▪ Team member of SAE NIS Collegiate Club Convention at UPES in 2013. ▪ Team member of organizing Uaarja ▪ Worked as a Proctor CoES
Technical Skills :	▪ Knowledge of Fluent, Auto-CAD Packages.
Personal Information :	Father's Name: Late Shri Purshottam Pandey Date of Birth: 05.07.1977 Address: Lane No. -6, Adarsh Vihar, Shyampur, Near Rana Chowk, Via Prem Nagar Indore (M. P.) Pin 248007, Mob- 9897912708 Email: shyam@ddn.upes.ac.in

(Shyam Pandey)

Annexure -I

Professional Experience

Teaching & Professional Experience:

Aug 2007– till date	**UNIVERSITY OF PETROLEUM & ENERGY STUDIES, DEHRADUN.**

Asst. Professor (Mechanical Engineering)

- Teaching Fluid Mechanics, Theory of Machines, Thermodynamics, Thermodynamics and Heat Transfer, Gas turbines power plants to the under-graduate and post graduate students.
- Developed testing rigs in the laboratories such as – computerized VCR Engine Test Rig in the alternate fuel research Lab.
- Arranging internship for the Automotive Design Engineering students.
- Working as an Alumni co-ordinator for CoES.
- Guided more than 15 UG and PG Project to the students.

Jan. 2006 – July 2007	ENERCON (INDIA) LTD. DAMAN

Senior Engineer – Enercon Training Academy.

- Formulation of Process and Work Instruction for site personnel for Service and for Installation activities.
- Working at sites with Service & Installation team on special activities like bearing replacement, gearbox replacement, Gear rim shifting, Shimming in between tower flanges Turbine re-erection, blade extender activity, rotor conversion etc. & developing guidelines for same.
- Root cause analysis for Wind turbine failure on technical front.

- Compiling technical instructions on various issues as per Indian operation conditions and updating it as per requirement.
- Training of Service, Installation and Commissioning team at ENERCON TRAINING ACADEMY for function of Wind Machines.
- Training Blade Production employees at technical or moulder level as well as supervisor level.
- Administration of Enercon (I) Ltd e –learning (Moodle) portal, content creation and management.
- Data extraction from SCADA Data Base Management System (S.D.B.M.S.) for

May 2003–Dec 2005 HITKARINI COLLEGE OF ENGINEERING & TECHNOLOGY, JABALPUR (MP)

Lecturer - Mechanical Engg.

- Taught Thermal Engg., Material Science, I.C. Engine etc to the undergraduate students.

....

Annexure- II

List of Paper Published

Paper Published in International Journal -02 Nos

Papers Presented in National Conferences - 01 Nos

INTERNATIONAL JOURNALS:

1. Experimental investigation on the performance and emission characteristics of a diesel engine fueled with ethanol, diesel and jatropha based bio-diesel blends International Journal of Advances in Engineering & Technology, Sept 2012. ISSN: 2231-1963
2. Effect of Compression Ratio on Diesel Engine Performance and Emission with Diesel- Ethanol Blends. International Journal of Scientific & Engineering Research, Volume 4, Issue 10, October-2013 ISSN 2229-5518

NATIONAL CONFERENCES:

1. Experimental Investigation on the performance and emissions of a diesel engine fueled with butanol-bio-diesel-diesel blends. - 7th Uttarakhand State Science and Technology Congress-2012

SPONSORED R-&-D PROJECT

1. Endurance analysis and tribological studies of diesel engines fueled with Bio-butanol, diesel and their blends as alternate fuels.

   Funding Agency: Uttarakhand State Council for Science &Technology(govt. of uttarakhand)dehradun- 248001

   Project Duration : 03 Year

   Total approved cost of the project (in Rs.): 15 Lakhs

Patent filed:

1. Fuel Injection system for compression ignition engines of stationary applications, particularly for diesel engines.(Application no. -2792/DEL/2013)
2. Shell and tube type EGR cooler for small off-road diesel engine. (Application no.- 262875)

………………………

Annexure -III

Faculty Development Program Coordinated/Attended

- Organized a two weeks main workshop on “Fluid Mechanics” duringMay 20th -30th, 2014 at University of Petroleum and Energy Studies, Dehradun (U.K.).
- Attended “Coordinators workshops on fluid mechanics” at IIT KharagpurduringMarch 11-15, 2014.
- Short term Training Programme on “ Diesel Particulate and NO_x control” at IIT Kanpur (QIP)
- Attended workshop on “Harnessing Intellectual Property & its management for growth and prosperity” at Graphic Era University, Dehradun on 26th April 2014.

……………