PERFORMANCE AND EMISSION OPTIMIZATION OF AN ADVANCED DIESEL ENGINE FOR DUAL FUEL APPLICATION

A Thesis Submitted to the
University of Petroleum & Energy Studies, Dehradun

For the Award of
Doctor of Philosophy
in
Engineering (Power)

BY
M. Muralidharan

August 2022

Internal Supervisor
Dr. Ajay Kumar

External Supervisor
Dr. M. Subramanian,

Department of Mechanical Engineering
School of Engineering
University of Petroleum & Energy Studies
Dehradun – 248007: Uttarakhand

PERFORMANCE AND EMISSION OPTIMIZATION OF AN ADVANCED DIESEL ENGINE FOR DUAL FUEL APPLICATION

A Thesis Submitted to the
University of Petroleum & Energy Studies, Dehradun

For the Award of
Doctor of Philosophy
in
Engineering (Power)

BY
M. Muralidharan
SAP ID : 500046020

August 2022

Internal Supervisor
Dr. Ajay Kumar
Professor
Department of Mechanical Engineering
University of Petroleum & Energy Studies

External Supervisor
Dr. M. Subramanian,
General Manager –Retd.
Automotive Research, Indian Oil Corporation Ltd
R&D Centre, Faridabad

Department of Mechanical Engineering
School of Engineering
University of Petroleum & Energy Studies
Dehradun – 248007: Uttarakhand

August 2022

DECLARATION

I declare that the thesis entitled "Performance And Emission Optimization Of An Advanced Diesel Engine For Dual Fuel Application**"** has been prepared by me under the guidance of Dr. Ajay Kumar, Professor &Head, Department of Mechanical Engineering, University of Petroleum & Energy Studies, Dehradun and Dr. M. Subramanian, General Manager –Retd., Automotive Research, Indian Oil Corporation Ltd, R&D Centre, Faridabad. No part of this thesis has formed the basis for the award of any degree or fellowship previously.

M. Muralidharan
Department of Mechanical Engineering,
University of Petroleum & Energy Studies, Dehradun
Date : 18.08.2022

CERTIFICATE

I certify that M. Muralidharan has prepared his thesis entitled "Performance And Emission Optimization Of An Advanced Diesel Engine For Dual Fuel Application**"**, for the award of PhD degree from the University of Petroleum & Energy Studies, under my guidance. He has carried out work at the Department of Mechanical Engineering, University of Petroleum & Energy Studies.

Internal Supervisor
(Dr. Ajay Kumar)
Department of Mechanical Engineering,
University of Petroleum & Energy Studies, Dehradun
Date : 18.08.2022

CERTIFICATE

I certify that M. Muralidharan has prepared his thesis entitled "Performance And Emission Optimization Of An Advanced Diesel Engine For Dual Fuel Application", for the award of PhD degree from the University of Petroleum & Energy Studies, under my guidance. He has carried out work at the Department of Mechanical Engineering, University of Petroleum & Energy Studies.

External Supervisor
(Dr. M. Subramanian)
General Manager –Retd., Automotive Research,
Indian Oil Corporation Ltd, R&D Centre, Faridabad
Date : 18.08.2022

ABSTRACT

Researchers are captivated by dual-fuel technology as a potential solution to pollution and energy depletion, especially with the increasing fuel availability such as compressed natural gas (CNG). It is a technology that is environmentally sustainable since it emits less particulate matter (PM) and smoke while maintaining diesel combustion efficiency. Large engines, such as marine, locomotive, and stationary engines, have typically used dual fuel technology. However, its usage in automotive engines has previously been limited due to the scarcity of alternate fuels. Under dual-fuel mode operation, CNG is a suitable fuel with different degrees of achievement. The induction method reduces volumetric efficiency, eliminates a pre-mixed natural gas-air mixture, and causes power loss at higher speeds. While nitrogen oxide (NOx) emissions are decreased at low-intermediate loads and lower engine operating temperatures, carbon monoxide (CO) and total hydrocarbon (THC) emissions are significantly increased under these conditions.

The utilization of compressed natural gas (CNG) in dual-fuel mode for compression ignition (CI) engines remains largely unexplored in countries such as India due to commercial considerations, primarily due to the intricate process of modifying and adjusting the engine control unit settings. To investigate its possible applications and analyse Selective Catalytic Reduction (SCR) performance, a commercial heavy duty diesel engine with six cylinders, turbocharging, intercooling, and common rail direct injection was modified for dual fuel operation that can run on both diesel and CNG fuels. An eddy current engine dynamometer was utilized to optimize the CNG flow rate for different load and speed conditions, taking into account combustion characteristics, engine performance, and emission standards such as Euro 4 and Bharat Stage (BS) –IV. The engine's electronic control unit (ECU) regulates the amount of diesel fuel used, while CNG is initially introduced through the manifold (intake) at a variable rate of energy substitution and a mass flow rate ranging from 0.65 to 4.0 kg/hr. The investigation was carried out under four steps namely (i) Full

Throttle Condition, (ii) Part Throttle/Varying Load Condition, (iii) European Stationary cycle testing and (iv) Engine Control Unit (ECU).

(i) A maximum energy substitution rate of 10.2% utilising CNG was noticed at full throttle conditions, resulting in an 8.9% increase in power and a 5.8% improvement in torque. While brake thermal efficiency increased, volumetric efficiency declined slightly, necessitating the use of a dedicated turbocharging system with a dual fuel (DF) engine to achieve higher brake power. The ideal engine speed for improved fuel economy was between 1250 and 2250 rpm with a 1% reduction in brake-specific energy and fuel consumption. With CNG substitution, brake-specific CO_2 and CO emissions were significantly reduced, while non-methane hydrocarbon, NO_x, and methane emissions were marginally higher; however, all emissions were under Euro 4 emission standards. Acetylene, Formaldehyde, ethylene and formic acid have marginally increased while unregulated emissions including Propane, NH_3 and SO_2 have reduced. SCR has proven to be effective at lowering mass emissions from diesel engines and increasing the conversion efficiency.

(ii) The impact of in-cylinder combustion on emissions were studied under different CNG energy fractions of 1.9-60.2% at different load and speed conditions. Under full throttle load conditions, an increase in CNG energy fraction resulted in an increase in in-cylinder pressure, pressure rise rate and heat release rate. Under full load conditions, an increase in the flow rate of CNG leads to a prolonged combustion duration and a decrease in ignition delay. With a longer combustion duration, higher fuel burn rate and longer diesel injection duration, CO & THC emissions are reduced, however at higher IMEP it was lower. In the case of NO_x and CO_2 emission, it is reduced with shorter combustion parameters.

(iii) Diesel dual fuel (DDF) has demonstrated improved performance in respect of volumetric efficiency, power, exhaust temperature and brake thermal efficiency under various speed and load conditions, making it appropriate for commercial use. Under specific CNG flow rates, DDF showed higher emissions than regular diesel for regulated mass emissions such as THC, CO and NOx, although these emissions were still under the Euro 4 emission standard. An array of CNG flow rates between 0.65 and 3.0 kg/hr at various load and speed situations was established based on several criteria, including performance of the engine and emissions, and the diesel engine was optimised for use in CNG-diesel dual fuel mode.

(iv) Similar to performance evaluation at different load & speed conditions, at various mass flow rates between 0.65 and 4.0 kg/h, CNG was inducted for the test condition conducted as per Euro 4/BS -IV emission norms, i.e. under European Stationary Cycle (ESC) engine test cycle. DDF's engine performance has been on par with neat diesel in respect of brake-specific energy consumption, brake thermal efficiency, brake-specific fuel consumption, volumetric efficiency and power. However, in the case of emission performance, with an increased flow rate of CNG, while emissions of NOx increased up to 1.42 kg/h and then decreased, THC and CO emissions increased significantly. While emissions like CO & THC were higher than Euro 4 emission norms, NOx emission complies with the Euro 4 emission standard. As expected methane emission increased with an increase of flow rate of CNG and CO_2 reduced. Based on the above performance, a normalised matrix was prepared to determine the optimum flow rate of CNG for the different ESC modes.

(v) A DDF induction kit was developed wherein four CNG injectors were connected to the intake manifold and the CNG flow rates were controlled through ECU based on the optimal CNG flow rates for different speeds and loads including ESC modes that was generated during the engine and emission performance. The DDF engine was thereafter run for 100 hours of operation at different engine speeds/loads and engine oil analysis was carried out at initial and final in addition to the emission testing as per Euro 4 emission norms. With the DDF engine, marginal reduction in fuel consumption and brake-specific energy was observed and comparable performance in respect of brake, volumetric and thermal efficiency. Under the regulated emissions, CO and THC increased under DDF though CO emission was marginally higher than Euro 4 emission limit, THC emission was much higher. NO_x emission though increased under DDF, it was under 3.5 g/kWh as per Euro 4 limit. There was a marginal increase in methane emission while CO_2 was reduced under DDF. The used engine oil of neat diesel and DDF engine revealed that the lubricant performance is comparable in terms of kinematic viscosity at 40°C & 100 °C and total base number (TBN), while there has been marginal increase in fuel dilution, oxidation and nitration though within the limit.

(vi) With the development of the DDF kit, it has been demonstrated that a commercial engine can be run on diesel and CNG dual fuel without affecting engine performance and meeting emission standards if the CNG flow rate is optimised based on various parameters, including engine performance and emissions. To further reduce CO and THC emission, diesel oxidation catalyst (DOC) can be employed along with exhaust gas recirculation for reduction of NO_x emission.

(vii) While most of the previous research on DDF has been conducted on single-cylinder engines or older low-power engines, the DDF kit developed through this research is a novel work as a commercial heavy-duty engine is adapted to operate on CNG-diesel dual fuel technology that meets the BS-IV emission standards implemented during 2017.

This study makes a contribution to the field of DDF kit development for a commercial six-cylinder heavy-duty diesel engine that meets BS-IV emission standards and can run on CNG and diesel fuel without affecting engine parameters and complying with regulatory emission standards. This is regarded as the innovative aspect of this research since, despite what researchers have reported, there are currently no commercial kits for retrofitting automobiles, with the exception of a few initiatives for diesel generator sets.

Keywords: Diesel Engine, Compressed Natural Gas, Dual Fuel, Optimization, Performance, Power, Emission, Combustion, etc.

ACKNOWLEDGEMENT

I would like to dedicate this thesis to my beloved parents, Mr. I. Murugesu and Mrs. M. Tamilvani. I am grateful to them for raising me and providing me with mental and financial assistance throughout my life. Without their noble devotion, I would not be where I am now.

First and foremost I would like to thank my wife; Dr. Jothilakshmi M. (PT) for always being by my side, pushing me to overcome challenges, sharing my joy, and keeping me active. No words can adequately explain my feelings for her. I also thank my children Master M. Thirucharan and Master M. Nikhil who has consistently offered the necessary respite from philosophy as well as the motivation to complete my degree as quickly as possible. I also thank my brother Mr. M. Rajashekar and sister Ms. M. Karthika for their inspiration.

I thank my supervisors, Professor Ajay Kumar and Dr. M. Subramanian for their support, encouragement, guidance and understanding during this entire journey of research project.

I gratefully acknowledge the support provided by my ex-employer, Indian Oil Corporation Ltd., R&D Centre, Faridabad for allowing me to pursue Ph.D and offering the laboratory facilities for carrying out the research project. I sincerely thank the management, officers, staff and contractual employees of Indian Oil Corporation, especially Mr. Mukul Maheshwari, Mr. M.Sithananthan, Dr. A.S.Ramadhas, Mr. Neeraj Kumar, Mr. Chhavideep Meena, Mr. Navarun Saikia, Mr. Padamveer Chouhan, Mr. Begraj, Mr. Sunil Kumar and Mr. Harender for their invaluable help in all things.

Finally, I want to express my gratitude to all the friends and colleagues I have made over this study endeavour for the insightful conversations. Special thanks to my current organisation Hindustan Petroleum Green R&D Centre, Bengaluru for showing compassion to complete the thesis.

List of Symbol

ATDC	after top dead center
BAP	Boost air pressure
BAT	Boost air temperature
BP	Brake power
BSEC	Brake specific energy consumption
BSFC	Brake specific fuel consumption
BS-VI	bharat stage VI
BTE	brake thermal efficiency
CA	Crank angle
CEF	CNG Energy Fraction
CH_4	methane
CHV	combined heating value
CI	compression ignition
CLD	chemiluminescent detector
CMFR	CNG Mass Flow Rate
CNG	compressed natural gas
CO	carbon monoxide
CO_2	carbon di oxide
COV	Coefficient of variation
DAS	Data Acquisition System
DAU	Data Acquisition Unit
DDF	diesel-CNG dual fuel
DEF	diesel exhaust fluid
ECU	engine control unit
EGR	Exaust Gas Recirculation
ELR	European Load Response
ESC	Europen stationary cycle
ESR	Energy substitution rate
ETC	European Transient Cycle
EV	Electric Vehicle
FBR	Fuel burn rate
FID	flame ionization detector
FS	Full scale

FTIR	fourier transform infrared
GISFC	gross indicated specific fuel consumption
HC	hydrocarbon
HD	heavy duty
HRR	Heat release rate
HPDI	High Pressure Direction Injection
ID	Ignition delay
IHR	initial heat release
IMEP	Indicated mean effective pressure
LHV	Low heating value
LNG	Liquid natural gas
LNT	Lean NOx trap
MFSR	Mass fuel substitution rate
NDIR	non destruction infrared
NGV	Natural gas vehicle
NMHC	Non methane hydrocarbon
NOx	nitric oxides
OEM	Original equipment manufacturer
PM	Particulate matter
PRR	Pressure release rate
R/P	Reserve to production
RAFR	Relative air fuel ratio
RCCI	Reactivity controlled compression ignition
RH	Relative humidity
RON	Research Octane Number
SCR	selective catalytic reduction
SO_2	sulphur di oxide
TDC	Top dead centre
THC	Total hydro carbon
UHC	Unburned hydro carbon
VE	Volumetric efficiency
VOC	Volatile organic compound
WHSC	World harmonised stationary cycle

Table of Contents

List of Figures

List of Tables

CHAPTER

1. INTRODUCTION

Energy is widely regarded as one of the key contributors to economic development and personal comfort. Economic expansion and energy use are closely related to one another. The availability of affordable, environmentally acceptable energy sources is necessary for the growth of an economy and for it to remain competitive on a global scale. Conversely, the level of economic development is considered to be affected by energy consumption. Due to rising population and economic development, India's consumption has recently increased at one of the quickest rates in the world. India is fortunate to have both moderate and renewable energy resources, yet it's per capita energy consumption is still modest when compared to other emerging nations. India lacks resources with a high energy density, such as nuclear power plants and fossil fuels which is the cause of this.

Fig 1.1 shows that India's energy requirement is primarily met by petroleum. Worldwide, the third-largest importer of crude oil is India, and petroleum energy also serves as the foundation of its expanding economy. As a result, efficient management of petroleum energy use is required. In India, diesel usage is almost 3.5 times that of gasoline [1]. Diesel should be used in a regulated and efficient manner in order to sustain economic growth.

Diesel engines have slowly become more popular over the past century as a fuel-efficient and dependable form of transportation for commodities and the general public, as well as for other essential social needs, including small-scale power generation and other similar activities. They often have advantages over spark-ignition engines due to their reduced regulated emissions of CO, unburned HC & carbon dioxide (CO_2) and better thermal efficiency [2] [3].

Diesel engines also have the advantage of using low-energy alternative fuels like biogas since they can run at a higher compression ratio. On the other hand, diesel engines generate dangerous pollutants, including PM and NOx. Because of their potential health hazards and effect on visibility, these emissions are a hazard. When exposed, particulate emissions have the potential to cause occupational cancer and have a number of other negative health effects [4]. Diesel engines, when used as part of a transportation system, are widely acknowledged as a significant source of ambient particulate matter [5].

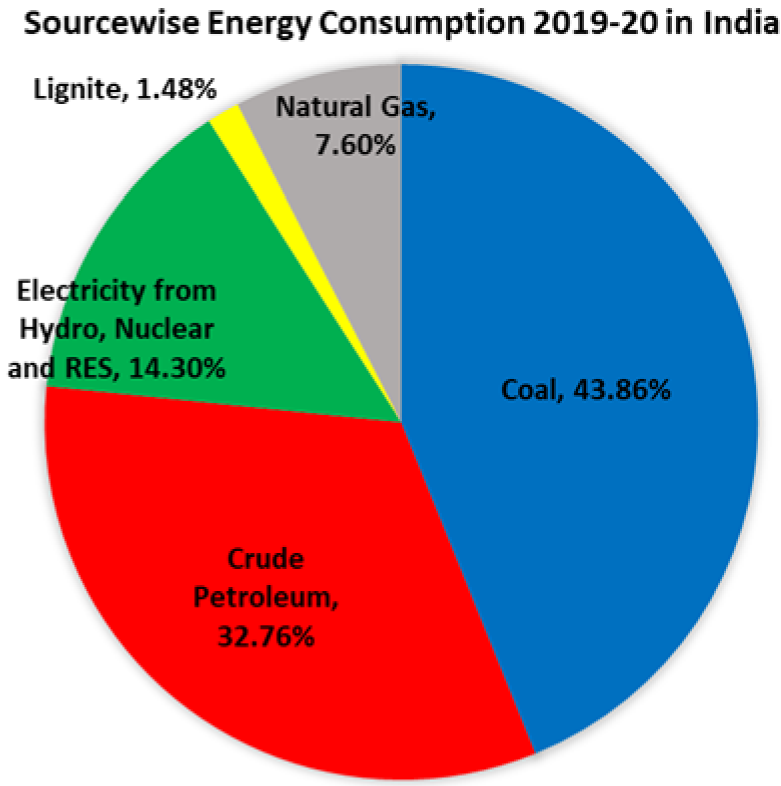


Fig 1.1: Sourcewise Energy Consumption during 2019-20 [6]

As public awareness about pollution increases, more stringent emission regulations are being enacted. Oil companies and engine manufacturers have made considerable improvements in diesel innovation as a result of the stringent regulations, including the use of enhanced, ultra-low sulphur fuels and improved engines; exhaust after-treatment, etc. Newer engines have lower PM levels than older ones. Stricter regulations have resulted in significant improvements in lowering the emissions of diesel exhaust. However, aged diesel engines without after-treatment devices will continue to predominate, particularly in developing nations, preventing future advancements. As a result, several strategies for reducing PM and NOx emissions from CI engines have been developed. Alternate fuel is required for the limited usage of diesel in CI engines. Utilizing gaseous fuels in existing CI engines emerges as a highly practical approach to address these challenges [7].

Natural gas, a widely recognized gaseous alternative fuel, consists of various gas species and is derived from fossil sources. *Table 1.1* presents a representative spectrum of chemical composition for natural gas [8]**.** It is possible to find fossil natural gas either by itself or in conjunction with other fossil fuels (eg, coal in coal beds and crude oil in oil fields). Natural gas's properties are fundamentally identical to those of methane (CH_4) which is its principal composition.

Table 1.1***: Natural Gas Chemical composition.***

Gas Species	Range in percentage volume
Methane	87.0–96.0
Ethane	1.8–5.1
Nitrogen	1.3–5.6
Propane	0.1–1.5
Carbon dioxide	0.1–1.0
n-Butane	0.01–0.3
Isobutane	0.01–0.3
Oxygen	0.01–0.1
n-Pentane	Traces to 0.14
Isopentane	Traces to 0.14
Hexane	Traces to 0.06
Hydrogen	Traces to 0.02

There is a global emphasis on prioritizing natural gas due to its larger reserves compared to crude oil. The existing reserve-to-production (R/P) ratio for crude oil is estimated to be around 40 years, whereas natural gas boasts a ratio of approximately 60 years [9]. *Fig 1.2* illustrates the changes in the R/P ratios throughout the past 25 years. Natural gas and crude oil R/P ratios have been mostly stable over the previous 20 years. This is because, notwithstanding the discovery of new reserves, formerly inaccessible natural gas supplies can now be accessed because of modern technology that makes practical and affordable recovery possible.

Additionally, compared to gasoline or diesel, natural gas combustion enthalpy per unit mass is higher. that is demonstrated in *Fig 1.3* by comparing the energy densities [7]. The figure depicts the overall mass of various fuels stored in various medium, which also includes the mass of the fuel and storage container. The penalty for the container's weight is represented by the grey areas in the figure. Real values that are lower than the bars themselves are represented by the smaller columns (in some cases, to make it noticeable, smaller bars have been overstated in the chart). Because of its higher enthalpy than conventional fuels, natural gas may be the best alternative.

Natural gas cannot be used with conventional liquid fueling methods since it is gaseous. This fuel presents a challenge for on-board storage of fuel and fuel carburation and injection systems in particular. To replace a typical vehicle's liquid fuel tank, high-pressure CNG tanks and fuel lines are required. Customized fuel injectors or a fuel inducement system are needed since the low density of natural gas necessitates a higher volume and mass flow rate. Despite these advancements, natural gas-powered vehicles (NGVs) cannot travel as long distances as conventional automobiles. This is illustrated in *Fig 1.3*, whcih depicts the energy densities of various fuels per unit volume under different storage media. Invariably, a normal pressurised natural gas tank carried by an NGV carries less fuel energy (measured in MJ) than a diesel or gasoline tank of the same volume. Preserving liquid hydrogen within the temperature range of approximately -260 °C (at approximately 1 bar) to -245 °C (at pressures ranging from 20 to 100) presents notable obstacles in terms of insulation and preventing excessive evaporation.

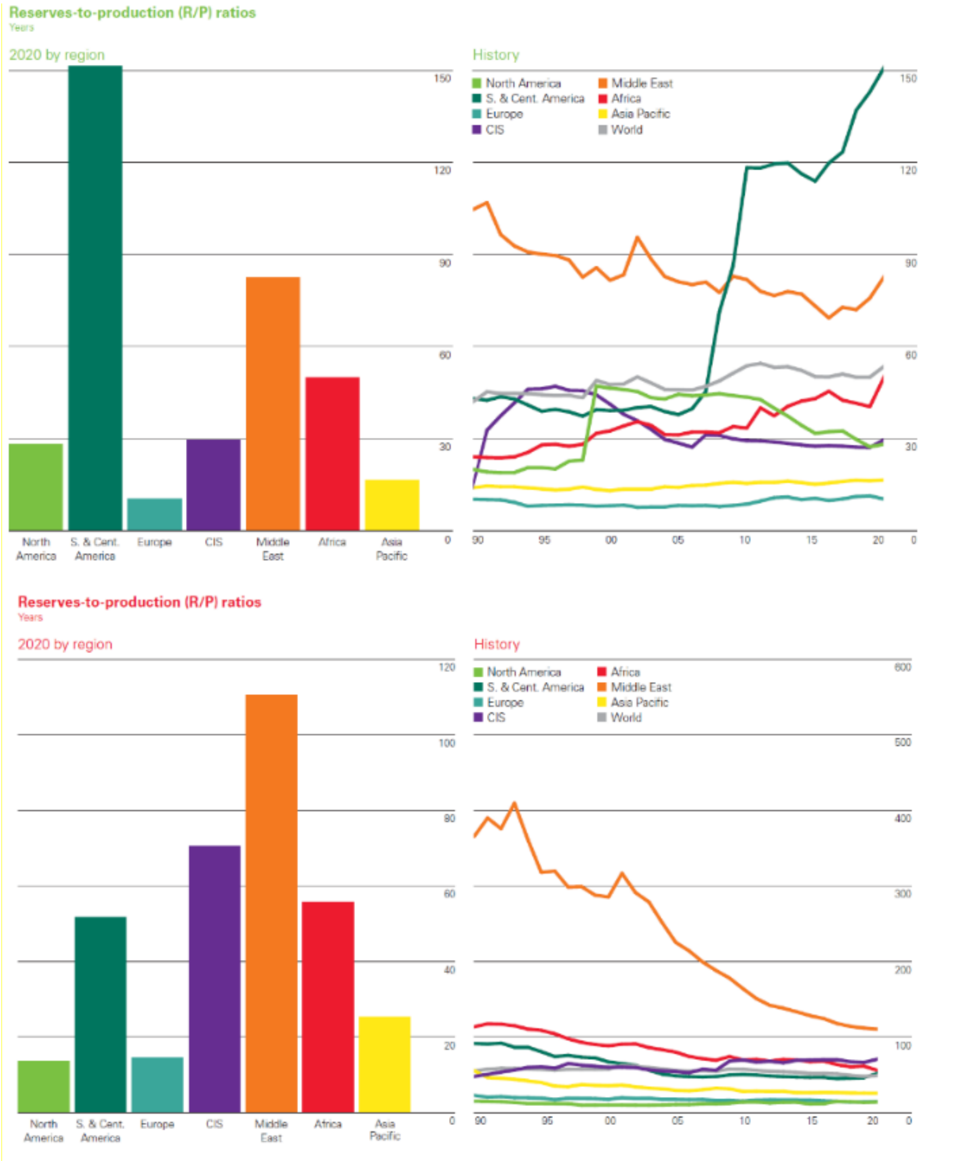


Fig 1.2: Reserves-to-production ratios of natural gas (bottom) and crude oil (top) from 1990 to 2020 [9]

However, there is no specific natural gas refuelling guideline for NGVs, particularly in developing countries. This is despite the fact that the infrastructure and setup for providing and treating natural gas are infinite (power plant, warming of building). Outside of South America, infrastructure for refuelling with natural gas is not as widespread as infrastructure for refuelling with gasoline. In this sense, it is difficult to see a gaseous fuel completely replacing cars that run on gasoline or diesel in the near future. However, because of the fairly short travel lengths and reduced levels of non-methane HC emissions and CO, both of which are hazardous to humans, and HC is a proven carcinogen, NGV's use is rising in urban areas [10] [11].

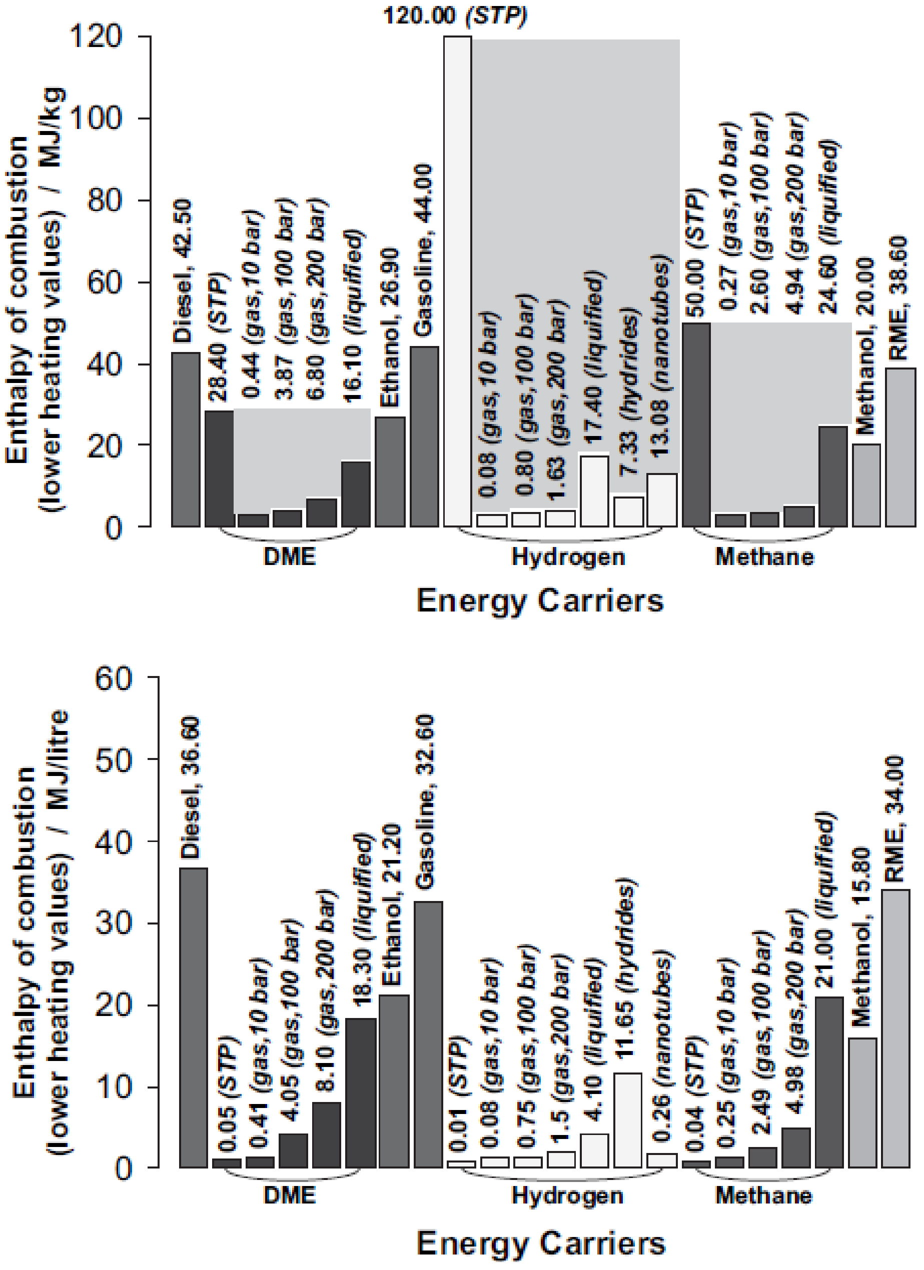


Fig 1.3: Enthalpy of combustion per unit volume and mass - Fuel energy densities [7]

Methane is the main hydrocarbon emission in natural gas engines, which is another characteristic of these engines. Although methane has low reactivity during the photochemical smog cycle, which is the basis for HC emission limitations, methane's impact on greenhouse gases has been contributing to global warming for a century that is over 25 times greater than that of CO_2 [12]. In densely populated areas, there is also an issue with photochemical smog, in which volatile organic compounds (VOCs) and ground-level ozone and particles are produced when NOx react in the presence of sunlight. Hence, natural gas-powered engines produce relatively little particulate matter and smoke compared to gasoline and diesel-powered engines, so their impact on smog formation is minimal. As a result, NGVs are gaining popularity in densely populated metropolitan areas where community transportation and trucks have undergone natural gas conversions [11]. Public transportation networks frequently include central refuelling stations, which makes setting up a natural gas refuelling infrastructure simpler. Furthermore, buying natural gas is less expensive than buying traditional fuels.

Natural gas is not renewable because it is a fossil fuel similar to that of gasoline and diesel. However, the primary gas in natural gas, methane, may be produced through sustainable means [13] [14]. The direct utilization of methane from landfills and biomass decomposition is notably distinct from the Fischer-Tropsch technique employed in biomass-to-fluid and gas-to-liquid technologies. Unlike the elaborate procedure of gathering, purifying, and employing methane from these sources, the process is simplified and straightforward.

Natural gas has garnered attention as a potential alternative for automobiles due to its reduced greenhouse gas emissions and enhanced safety characteristics when compared to traditional fuels. One of the several successful strategies for lowering emissions while conserving fossil diesel fuel is the DDF operation, which uses both diesel and natural gas. A dual-fuel engine is commonly known as an engine that utilizes both diesel and gaseous fuel for combustion. Straight diesel operation is defined as using diesel only fuel; while dual fuel operation is defined as using two fuels simultaneously. In the context of dual fuel operation, a unique approach is employed where a precise mixture of air and gaseous fuel is carefully incorporated at minimal gas-to-air ratios. Following this, the compressed mixture undergoes the compression stroke and towards the end, the engine is injected with diesel fuel. Following a short delay in ignition, the diesel fuel ignites initially, setting off the combustion process. Following this, the ignition process is initiated for the natural gas, indicating the commencement of the flame propagation stage. DDF combustion displays distinct emission and performance attributes due to the influence of natural gas on the chemical and thermodynamic properties of the mixture within the cylinder. The diesel fuel used as an igniting source is called pilot diesel. A dual fuel engine's efficiency and emission performance are influenced by the concentration of natural gas and the amount of pilot diesel in the combustion chamber.

Due to the restricted supply of alternative fuels, dual fuel technology for transportation is still in its nascent stages [15]. Furthermore, dual fuel technology is still being researched in India for automotive applications [16]. This thesis focuses on evaluating the operational characteristics and emissions of a commercially available, heavy duty diesel engine that utilizes a dual-fuel configuration consisting of diesel and CNG. The objective is to examine the engine's performance and emission levels under this specific setup. The optimization of the dual-fuel engine has been achieved by incorporating the insights gained from these research findings. As a result, a CNG kit has been developed to operate at the optimum CNG flow rate for 100 hours at varying engine loads and speeds, with performance monitored in terms of emission and engine oil.

1.1. NATURAL GAS AS AN ALTERNATIVE FUEL

Natural gas, a viable gaseous fuel, can be effectively employed in both SI and CI engines. CI engines, known for their higher compression ratios, demonstrate superior thermal efficiency compared to SI engines. According to *Table 1.2*, natural gas has a 19.0 % lower stoichiometric fuel-air ratio and a 5.9 % higher LHV_f than diesel [17]. In the majority of applications, natural gas is presently introduced into the intake manifold either through injection or induction methods, which decreases volumetric efficiency and, consequently, power. Natural gas is also directly injected into the cylinder in some applications.

Table 1.2*: Common fuels' chemical composition and physical characteristics under typical atmospheric conditions.*

Fuels	**Diesel**	**Gasoline**	**Methane**	**Hydrogen**
Chemical formula	$C_nH_{1.8n}$	$C_nH_{1.87n}$	CH_4	H_2
Density/kg/m^3	827-840	750	0.725	0.09
Lower heating value (LHV_f)/MJ/kg	42.5	44	45	120
Resarch Octane Number (RON)	-	95	120	>120
Cetane number	52	-	-	-
Auto-ignition temperature/°C	250	280	650	585
Stoichiometric fuel-air ratio ($(F/A)_{st}$)	0.069	0.068	0.058	0.029

The octane number, which gauges the knocking tendencies in fuel used in SI engines, and the cetane number, which evaluates the ignition delay characteristics in fuel used in CI engines, are indicators of a fuel's ignition properties. Although octane and cetane values have an inverse relationship for a particular fuel, it is not a proportional one. It is noteworthy that methane, the dominant constituent of natural gas, possesses a higher octane rating compared to gasoline, yet it exhibits a lower cetane rating than diesel. Nonetheless, the cetane rating of methane is typically negligible and infrequently documented. The cetane and octane ratings in *Table 1.2* demonstrate how substantially methane's ignition quality differs from that of normal gasoline and diesel fuel. Because natural gas, like diesel, cannot ignite spontaneously at conventional CI compression ratios (and associated temperatures), both direct-injected as well as port-injected natural-gas CI engines need a source of governed ignition. As a result, the utilization of different high-cetane fuels, including diesel, renewable diesel, and biodiesel, can introduce the potential for multiple ignition sources by utilizing the conventional fuel injector in CI engines. This process, commonly referred to as dual-fueling in CI engines, serves as a pilot fuel for natural gas combustion [7].

1.2. COMPRESSED NATURAL GAS

Natural gas primarily consists of methane, along with other hydrocarbons including propane, butane, and ethane. Methane's composition is supplemented with other gases such as carbon dioxide, nitrogen, hydrogen sulphide, water vapour, and helium. Natural gas is extracted from reservoirs deep underground, sandwiched between rock and sand layers, just as other fossil fuels. Due to its lower density than other confined substances like water and crude oil, natural gas will drift over them. To use natural gas as an automotive fuel, either it is compressed into CNG or used as liquefied natural gas (LNG).

1.2.1. Specificities of CNG

CNG is a type of fuel that primarily consists of methane (CH4) gas compressed to a pressure of around 200 bar. The specificities of CNG compared with methane is listed below :

- Composition: CNG is predominantly composed of methane gas, typically containing 95% to 98% methane with smaller amounts of other hydrocarbons such as ethane (C_2H_6). On the other hand, methane is a hydrocarbon compound that consists of a single carbon atom bonded to four hydrogen atoms. Methane itself can exist as a gas, primarily known as natural gas when found in large quantities underground.

- Storage: CNG is stored under high pressure in specially designed cylinders or tanks to maintain its gaseous state. The pressure helps to maximise the amount of gas that can be stored within a given volume. Methane, in its natural state, can also be stored as a gas but at a lower pressure compared to CNG, such as in underground storage facilities.

- Energy Content: Both CNG and methane have similar energy contents per unit volume. Methane, as a gas, also has a high energy content.

- Applications: CNG is primarily used as a transportation fuel, especially in vehicles with specially designed CNG engines or converted gasoline or diesel engines. It is widely used in fleet cars, taxis, and buses for public transit. On the other hand, methane has uses outside of transportation. It is a raw ingredient for the manufacture of chemicals and fertilisers and is used as a fuel for heating, cooking, and the creation of electricity.

- Combustion Characteristics: CNG has favorable combustion characteristics due to its high methane content. It has a high octane rating, meaning it has good resistance to knocking and allows for higher compression ratios. This can result in improved engine efficiency and power output. Methane also exhibits similar favorable combustion characteristics as CNG. It offers good combustion efficiency, resulting in improved thermal efficiency and power output.

- Compression Ratio: CNG requires high-pressure storage to maintain its gaseous state, typically ranging from 200 to 250 bar. Therefore, vehicles running on CNG require specially designed engines with higher compression ratios to maximize the energy extracted from the fuel. Methane can be stored and used at lower pressures compared to CNG. It can be liquefied under moderate pressure (around 40 bar) at low temperatures (-162°C) to form LNG. LNG provides a higher energy density compared to compressed methane, allowing for longer driving ranges.

- Environmental Impact: CNG is considered a relatively clean-burning fuel compared to gasoline and diesel. It produces fewer emissions of pollutants such as CO, NOx, and PM. Methane, being the primary component of CNG, is also a potent greenhouse gas. However, when used as a fuel, its combustion produces fewer CO_2 emissions compared to other fossil fuels.

- Infrastructure: CNG requires a specific refuelling infrastructure, including compressors and storage facilities, to supply vehicles with the gas. CNG refuelling facilities are often located in regions where the demand for CNG cars is higher. Methane, on the other hand, is delivered through the current natural gas distribution and pipeline systems, making it broadly accessible for a variety of uses.

CNG is a particular type of methane gas that is compressed under high pressure for use as an alternative fuel. Despite their similarities, CNG stands out because of its unique infrastructure and storage needs, which are predominantly centred on automotive use. Methane, on the other hand, is frequently used for heating, the production of electricity, and as a raw ingredient in a variety of industries. Its uses go beyond transportation.

1.2.2. Properties of CNG

Table 1.3 compares the physical characteristics of CNG with conventional diesel fuel [7].

Table 1.3: The Properties of CNG and Normal Diesel [7]

Property	Compressed Natural Gas (CNG)	Conventional Diesel
Chemical Formula	CH_4	C_3 to C_{25}
Molecular Weight	16.04	~200
Specific Gravity	0.424	0.81-0.89
Composition by weight,%		
Carbon	75	84-87
Hydrogen	25	13-16
Density, kg/m3	128	802-886
Boiling temperature, °C	-31.7	188-343
Flashpoint, °C	-184	73
Freezing point, °C	-182	-74.4
Autoignition temperature, °C	540	316
Specific Heat, J/kg K	-	1800
Flammability limits, % volume		
Lower	5.3	1
Higher	15	6

CNG has been regarded as a favourable alternative fuel due to its clean combustion characteristics and minimal exhaust emissions. Installing a Bi-Fuel Conversion kit enables the utilization of CNG in SI engines, which faciliatates the converted engine to run on either CNG or gasoline. Diesel engines have the potential to be converted into CNG usage through two methods: using a dual fuel conversion kit or making alterations to the current diesel engine in order to function as a SI engine. The majority of modern CNG vehicles still retain bi-fuel capabilities and have aftermarket retrofit conversions to their petrol engines. The inclusion of an aftermarket conversion kit in bi-fueled modified engines frequently results in power loss and drivability issues, attributed to the design modifications. On the other hand, single-fuel engines that have been CNG-optimized are likely to be far more desirable in relation to emissions and performance. By retaining the thermal efficiency of a typical diesel engine, dual fuel mode can be used with CNG as a fuel in diesel engines that can reduce NOx, particulate matter, and CO_2 emissions [18]. CNG vehicle owners have concerns about the safety implications of switching to CNG-powered engines. As opposed to gasoline, diesel, or LPG, CNG offers four important safety aspects that make it a safer fuel. CNG cylinders are safer to store than diesel or gasoline fuel tanks since they are constructed of special materials and are manufactured to the highest safety requirements.

By combining CNG in the correct proportions, it exhibits superior mixing capabilities with air, resulting in efficient and swift mixing. This ultimately enhances engine's combustion efficiency. Due to CNG's higher RON of about 130 than that of gasoline (87), it may be compressed at a higher ratio (15.6:1), which results in more effective fuel use. Because diesel engines have a higher compression ratio, CNG may also be used in them. However, due to CNG's low cetane rating, it cannot totally replace diesel, unlike gasoline [19]. As a result, using CNG in diesel engines seems to be a very effective substitute. Different natural gas concentrations have a substantial effect on engine and emissions performance, according to recent studies on CNG vehicles. The origin of natural gas plays a significant role in determining the levels of unburned hydrocarbons and other emissions, in addition to their concentration, efficiency, and heating value. Additionally, it has been stated that due to higher engine knock, this influence is prominent in heavy-duty engines with high compression ratios.

1.3. CONVERSION OF DIESEL ENGINE

Diesel engines are extensively employed in heavy-duty applications and fleet transportation due to their enhanced engine efficiency resulting from higher compression ratios (CR). Diesel engines can have compression ratios of up to 20:1, whereas spark ignition engines have compression ratios of about 8:1.

1.3.1. Bi-fuel engine

An engine that runs on two different fuel systems typically CNG and gasoline is referred to as a bi-fuel engine. One way to transform SI engines into bi-fuel engines is by incorporating a CNG kit into their existing engine setup. The user may select which fuel to use via a fuel selector. It's been observed that when CNG is operated at wide open throttle, vehicles with bi-fuel engines lose around 10-15% of their power.

1.3.2. Dedicated CNG Engine

Dedicated CNG engines need more adaptations than a vehicle that uses a bi-fuel system. CNG vehicles, which utilize SI engines to combust fuel, necessitate the replacement of numerous components in a diesel engine. For example, for a dedicated CNG engine, a fresh gas supply and ignition systems are essential. As these engines are specifically engineered to run on natural gas, they exhibit enhanced efficiency and emit reduced pollutants throughout their entire operating system.

1.3.3. Dual Fuel Engine

Natural gas and diesel fuel are both simultaneously ignited in the combustion chamber of dual-fuel engines to generate power. Natural gas is utilised for around 80% of the energy required, with diesel providing the remaining 20%. In order to optimize performance, the ignition of methane within the combustion chamber is facilitated by utilizing diesel as a pilot fuel. To enable the smooth transition of conventional diesel engines to dual fuel systems, the incorporation of an electronic control unit (ECU) becomes necessary. Using both electrical and mechanical sensors, the ECU controls engine speeds simultaneously maintaining a focus on engine pressure, temperature, and oxygen concentration in the exhaust. This aids in ensuring the dual-fuel system's optimal operation. The diesel and CNG injectors are connected with sensors and solenoids so that the ECU may operate them with a pulse width modulated signal, resulting in maximum efficiency. The advantage of dual-fuel engines is that they generate less NOx and particulate matter while still delivering the same amount of power as a typical diesel engine.

1.4. MERITS AND DEMERITS OF USING CNG-DIESEL ENGINE

1.4.1. Merits

- The dual-fuel CNG-diesel engine offers a notable benefit in terms of its ability to seamlessly transition from operating on diesel fuel to solely relying on conventional diesel.
- The initiation of the combustion process does not require an electrical system or a spark ignition. Instead of the premixed combustion process used by spark-ignition engines, the dual-fuel engine uses a high-pressure diffusion combustion technique.
- It offers fuel adaptability. If natural gas is not available, it may be run in full diesel mode.
- Dual fuel engines could provide higher power densities, more efficiency, and the capacity to burn fuel leanly, which reduces misfire NOx emission.
- The usage of diesel pilot fuel leads to longer maintenance intervals and reduced maintenance expenses due to its ability to lubricate valves and rings.
- Dual-fuel CNG engines provide a number of advantages over spark-ignition engines, including the ability to efficiently burn mixtures at a relatively low air/fuel ratio [20].

- The normal diesel engine only needs minor modifications to run on CNG because dual-fuel engines operate at as low as a 16:1 compression ratio.
- Another benefit of employing a CNG-Diesel engine is that it produces more power than a normal diesel engine.
- The DDF engine is less likely to knock than a typical diesel engine because of its high ignition temperature and high octane rating.
- To reduce emissions, the DDF engine can be operated at a significantly reduced replacement level.
- It provides improved energy security and is more environmentally friendly because it emits fewer harmful pollutants like CO, NOx, SO_2, and particulate matter.
- Moreover, it derives advantages from and exhibits similar characteristics to methane, the principal constituent of natural gas, thereby reducing the potential for accidents in case of a leakage.

1.4.2. Demerits

- The CNG-Diesel engine is still restricted in some ways. The appropriate calibration of the CNG-Diesel mixing ratio and achieving control over the injector pump proves to be a difficult challenge when it comes to ensuring homogenous combination.
- The supply system is another big drawback of CNG-diesel engines that has led to a large drop in vehicle sales.
- For dual-fuel vehicles, natural gas must be stored in separate, pressurised tanks. Due to the increased weight of the vehicle, it will have less power, less storage room, and other drivability problems. Because natural gas has a lower energy density, it has a shorter driving range and a longer refuelling time—roughly twice as long as regular gasoline vehicles [4].
- Finally, it performs poorly when running at part or no load condition, although it's not a big disadvantage because the vehicle would be idling.

1.5. OPERATION OF DUAL FUEL ENGINE

During the final stage of the compression stroke, a considerable number of DDF engine employ a direct injection technique to introduce a minimal quantity of diesel fuel into the cylinder. This serves as an ignition source and undergoes compression to initiate combustion [21]. Natural gas is often used as the primary fuel in these engines. It is generally known that when using two fuels simultaneously, NOx and PM emissions may be significantly decreased, which causes THC emissions to rise [22]. Apart from the economic and environmental benefits, the DDF engine offers the advantage of easy engine conversion and the ability to retain complete diesel functionality in situations where gaseous fuel is unavailable. These factors play a vital role in facilitating the commercialization of this technology [23]. Furthermore, numerous recent studies have investigated the potential of Reactivity Controlled Compression Ignition (RCCI) combustion technique to function in a dual-fuel operation, leading to reduced emissions and improved performance [24]. These studies have also explored the applicability of RCCI in heavy-duty diesel engines that utilize both compressed natural gas (CNG) and diesel fuel [25]. The findings indicate that maintaining precise and consistent injection control enables the RCCI combustion technique for increased effectiveness and reduced emissions.

IC engine emissions and efficiency could be improved utilizing either RCCI or diesel-CNG dual fuel technology. While they share the same goal of reducing dependency on neat diesel fuel, their operational strategies and fuel injection techniques are different. Diesel-CNG dual fuel technology involves using a combination of CNG and diesel fuel to power the engine, as explained earlier. The RCCI combustion approach, on the other hand, combines the concepts of both spark ignition and compression ignition. The system uses a high-reactivity fuel like diesel in combination with a low-reactivity fuel like petrol or ethanol, both of which have unique reactivity properties. The low-reactivity fuel is combined with air and introduced into the intake manifold, whereas the high-reactivity fuel is directly injected into the combustion chamber. RCCI results in controlled auto-ignition, where the low-reactivity fuel ignites under compression and by adjusting the ratio of these fuels, the high-reactivity fuel combustion is initiated. By streamlining the combustion process and generating a more uniform fuel-air mixture, RCCI can offer high efficiency and low emissions. Both technologies hold the possibility of decreasing emissions and enhancing fuel efficiency, but they use distinct strategies to do so.

Depending on what way natural gas is introduced to diesel engines, three different types of dual-fuel engines are available. It includes [26]:

a) Intake premixed

b) Sequentially manifold injection

c) High-pressure direct injection (HPDI)

Due to reduced modification costs, the premixed type of dual-fuel engine is currently pervasive. One of the primary drawbacks associated with the dual-fuel engine is the diminished level of dynamic responsiveness and increased methane emissions. Fortunately, these drawbacks may be mitigated by using the intake manifold to individually inject natural gas into each cylinder in a sequential per cycle, allowing the quantity and timing of injection to be modified on the move. Furthermore, when compared to premixed dual-fuel engines, methane emissions may be drastically reduced by preventing the injection of natural gas before the closure of the exhaust valve. But a sequentially injected dual-fuel engine's primary problem is that compared to a premixed engine, it is more challenging. The primary concern lies with the engine control unit, that plays a crucial role in managing natural gas injection and pilot diesel injection configurations. Just like the eGD Flex technology, effective communication between the ECU of dual fuel and the existing diesel ECU is crucial [27].

The Westport HPDI injector, which is seen in *Fig 1.4*, is an illustration of a modern dual-fuel engine that uses an integrated gas/diesel injector. The injector is capable of precise injection of both diesel and CNG. Initially, diesel fuel is injected to act as a pilot ignition source for the diffusion flame of natural gas.

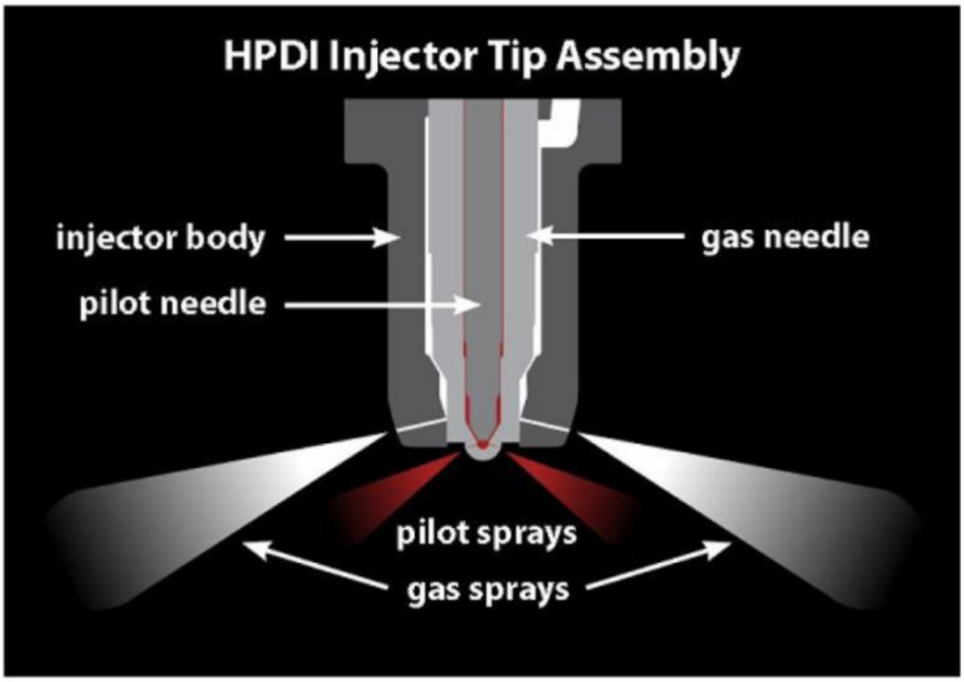


Fig 1.4:Westport Innovation HPDI Injector [28]

The HPDI dual-fuel engine offers the advantages of potentially reduced THC emissions and a higher replacement rate for diesel fuel. However, in the absence of natural gas, the engine can only operate in a limited "limp home" mode, thus preventing it from running solely on diesel fuel [26] [28]. As depicted in *Fig 1.4*, the system also requires a completely new diesel and natural gas fuel supply system, which includes high-pressure diesel and natural gas fuel pumps, an engine controller, and specially made fuel injectors [28]. As a result, the system often costs more than other dual-fuel system types.

CHAPTER

2. LITERATURE REVIEW

There is a significant amount of literature in relation to dual fuel conversion of existing engines. A gas mixer or port injectors are typically used in experiments to quickly convert a diesel engine to operate on dual fuel. The effect on the engine is evaluated by controlling a few parameters on emissions and performance, including pilot timing, pilot quantity, load, intake temperature, and speed [29] [30] [31] [32] [33]. While these publications support common characteristics in dual fuel operation, they usually lack depth and don't enable us to comprehend the combustion process any better. The findings of these tests are also highly dependent on the configuration of the engine chosen. More research is needed on the influence of air movement and the gas delivery technique using optical observation using an endoscope [34].

2.1. PERFORMANCE

Several studies have looked into the specific dual fuel engine performance. To gain a better understanding and identify areas that require more research, some of the key factors and patterns attributed to dual fuel operation are studied.

2.1.1. Combustion Characteristics

With standard CI-engine compression ratios, under compression alone, no natural gas auto-ignites. The utilization of CI engines necessitates the use of a unique operation mode called dual-fueling operation. A spontaneously igniting "pilot" fuel provides this ignition source. In the combustion chamber, a limited amount of high-cetane fuel is injected directly, known as a "pilot" injection.

This fuel mixes with an already pre-mixed natural gas-air charge. After a short period of ignition delay, the pilot fuel ignites and initiates combustion. Depending on how much the natural gas and air have mixed, there is a second delay before the ignition of natural gas and the majority of the energy generated during combustion is released. The objective of dual-fueling was to reduce the smoke and NOx emissions that typical CI engines produce. While maintaining appropriate thermal efficiency across the different operating ranges, dual-fuel engines are reasonably effective at the above [21] [35] [36]. In CI engines, natural gas operation's slow flame speed is a less significant issue than it is in SI engines. There are several reasons for this. There can only be one ignition source in a typical spark-ignition engine (i.e. spark plug). After the ignition from spark plug, the flame expands in all directions, spreading throughout the fuel mixture. In a standard operation of a CI engine, the cylinder receives fuel injection, which then blends with the charge. Under favorable charging conditions (such as appropriate fuel-air mixing levels, temperature, and pressure), the fuel droplets disperse within the air and spontaneously ignite. The chamber is filled with several ignition points, which leads to various flame fronts and a faster burn rate. This vast distribution of locations compensates for natural gas's slower flame speed and enables dual-fuel engines to burn fuel more quickly (where the cylinder is directly injected with the pilot fuel).

Since CI engines have no throttle valves across their intake system, they have better volumetric efficiency than throttled SI engines. Changing the fuel delivery rate to the cylinder impacts load control in CI engines. Pumping losses, which frequently cause SI engines to lose efficiency, are reduced as a result of this in CI engines. Previous research [32] indicated that dual-fueled engines have an average of 1-4 % lower volumetric efficiency than conventional CI engines. The effect is minimal because of the way the system is set up, where natural gas travels through a tube made of steel to the intake manifold adjacent to the intake valve. At higher engine speeds, under these dual fuel studies, the maximum natural gas flow rate is also restricted by this technique leading to a loss in power output. Relatively, there is a fuel shortage in terms of the supply of natural gas at a given flow rate of natural gas when supplied through the intake tube at lower speeds. Certain studies [35], have shown that this effect is not present when air mixture and natural gas are combined in a chamber before it enters the intake manifold.

The cycle-to-cycle variations of dual fuel engines exhibit similar levels of fluctuation as conventional CI engines, with both experiencing approximately 1% COV in IMEP values [30]. There are several contributing factors caused by minor discrepancies that are present both in SI and CI engines. Natural gas-air mixtures has specific heat capacity comparatively higher, because of which the pilot fuel igniting delay was significantly increased by around 0.08 ms during dual-fuel operation [37].

Dual-fueled engines exhibit increased noise levels when operating within their standard performance parameters, in contrast to conventional CI engines [37] [38] and at extremely high loads knocking is common [38]. Compared to diesel combustion, natural gas combustion has a relatively shorter duration, which further contributes to the occurrence of increased knocking [37]. In terms of net energy conversion rate throughout combustion, *Fig 2.1* [39] compares the performance of a dual-fuel engine (intake manifold injection of natural gas) with that of a standard CI engine ("heat-release" rate). The graph shows that, although the energy conversion rate during combustion falls to 0 at 5^{o} ATDC under dual-fuel operation, the energy conversion rate falls to 0 at around 30^{o} ATDC during conventional engine operation. However, in other publications, for all circumstances, the shortened combustion period observed in *Fig 2.1* is not sustained.

This variance may be the result of the various mass fractions of natural gas used to maintain a certain workload in a given engine. As a result, the mass fraction of natural gas may range between 70 and 86 % [37]. This discrepancy is seen in *Fig 2.2*, where the working fluid's peak dual-fuel energy change rate is shown to be less than that of conventional fueling, although *Fig 2.1* shows the reverse. The "Z" term in *Fig 2.2* stands for the amount of enthalpy substitution of natural gas [40].

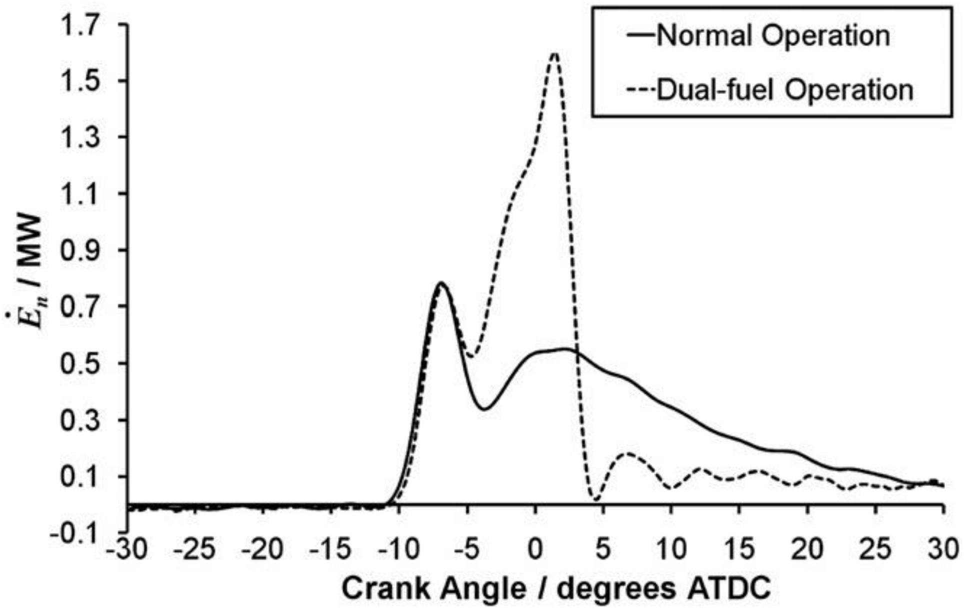


Fig 2.1: Comparison of the working fluid's net rate of energy change for normal engine and dual-fuel operation (dEn/dt plots) [39]

While other experimental investigations [37] indicate substantially lower efficiency, some of them indicate that dual-fuel thermal efficiencies are equivalent to those of traditional CI engines (at maximum BMEP values). The successful and efficient operation of a dual-fuel engine relies greatly on the induction system of natural gas and the specific engine utilized. This is due to the fact that the combustion process in dual-fuel engines is greatly influenced by the original design characteristics of the engine.

In dual-fuel engines, EGR without cooling improves low- to medium-load combustion while also minimising knocking tendencies. Normal dual-fuel operation at *Fig 2.3* is compared to a 2% mass EGR replacement, which results in a higher calculated energy change's peak rate and shorter duration of combustion of the working fluid [41]. The figure illustrates the effects of chemical (Ch), thermal (Th), and radical (Ra) factors on the working fluid's rate of energy change when EGR is implemented. Other studies [42] indicate that for low loads, low EGR levels (on mass substitution of about 5%) leads to increased rates of pressure rise. This data suggests that the working fluid transforms energy at higher rates. By diluting the mixture of intake, reducing the oxygen level, and delaying combustion, EGR reduces knocking at higher loads. However, if the charge is diluted with exhaust gas, more EGR will be required to reduce NOx emissions, which will result in slower combustion. In these situations, the injection of hydrogen gas improves combustion stability and lowers ignition delay [43]. By substituting natural gas with 23 % hydrogen by volume and intake oxygen with 40 % EGR by volume, around 1% less COV of IMEP was observed.

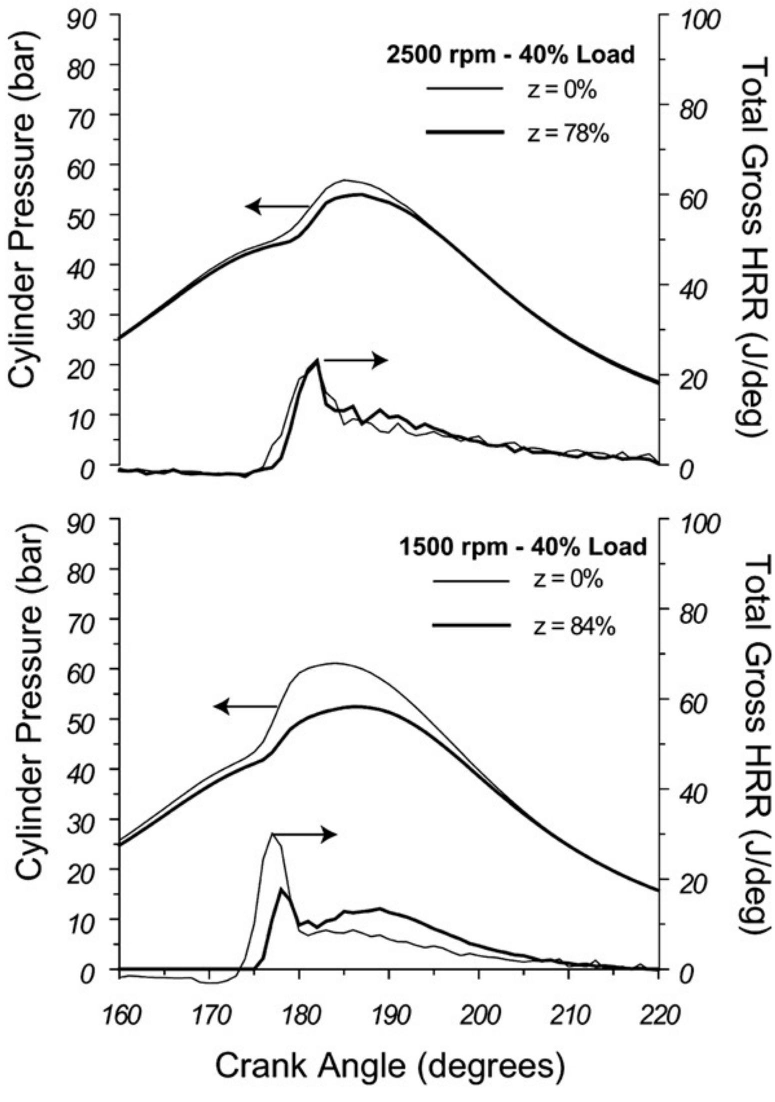


Fig 2.2: Comparison of dual-fueling and conventional fueling operations with respect to the rate of energy change and pressure of the operating fluid (HRR) [40]

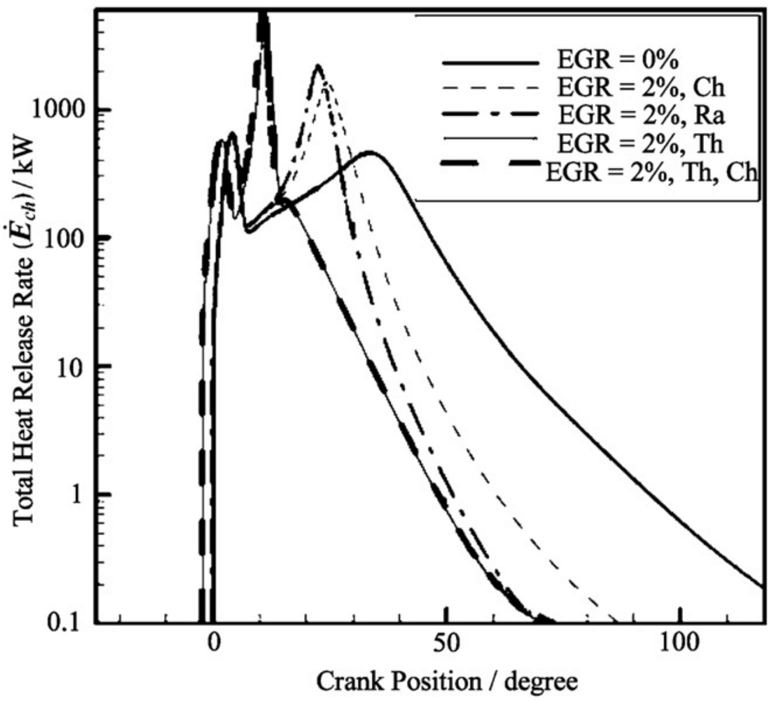


Fig 2.3: Comparison of the graphs of the working fluid's estimated rate of energy (heat-release rates) under dual-fuel operation for the various EGR components [41]

With increased hydrogen addition, the working fluid’s peak energy change rate (against injection timing of gaseous fuel for this case) increases. In *Fig 2.4*, the combustion time is defined by a 50 % "IHR". The figure also demonstrates that when the timing of hydrogen fuel injection is advanced compared to the combustion TDC, there is a noticeable decrease in the COV of IMEP.

2.1.2. Exhaust Emissions

The utilization of dual fueling in CI engines leads to considerably reduced NOx emissions in comparison to conventional diesel engines [35] [36]. The reduction is around 50% for low and moderate loads, this is dependent upon the engine's operating condition and type. The average NOx emissions in terms of BMEP from a dual-fuel engine running at different loads and speeds are shown in *Fig 2.5*.

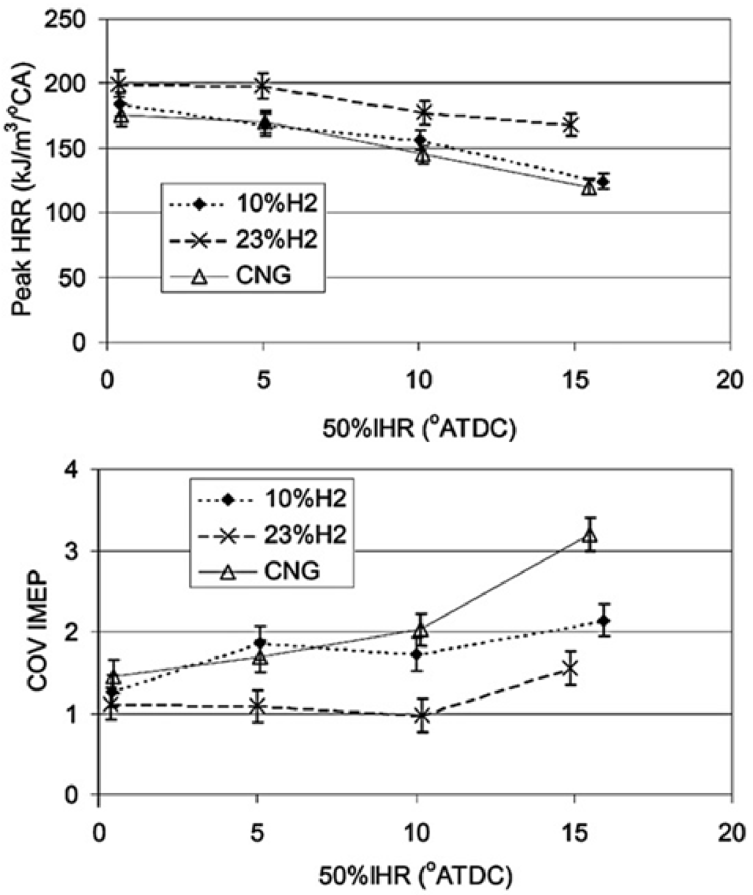


Fig 2.4: Peak energy change rate of COV variations and the operating fluid (HRR) for dual-fuel operation with 40% EGR [43]

The lower combustion temperatures caused by natural gas's slower flame speed are primarily responsible for the reduction in NOx. Based on additional studies, it has been found that most of the NOx emissions produced from dual-fuel combustion can be attributed to the combustion of the pilot fuel. Consequently, the type and amount of pilot fuel used have a significant impact on the overall NOx emissions released during dual-fuel operation. Researchers have additionally conducted trials to enhance the timing of pilot fuel injection in order to achieve a higher reduction in NOx emissions. Dual-fuel engines emit extremely little smoke, soot, and particle emissions, which are sometimes undetectable [21] [35] [36]. Natural gas combustion creates less smoke than diesel because natural gas has fewer carbon atoms. Methane has a low propensity for soot formation because it has a high hydrogen-to-carbon ratio and lacks carbon-carbon bonds [43].

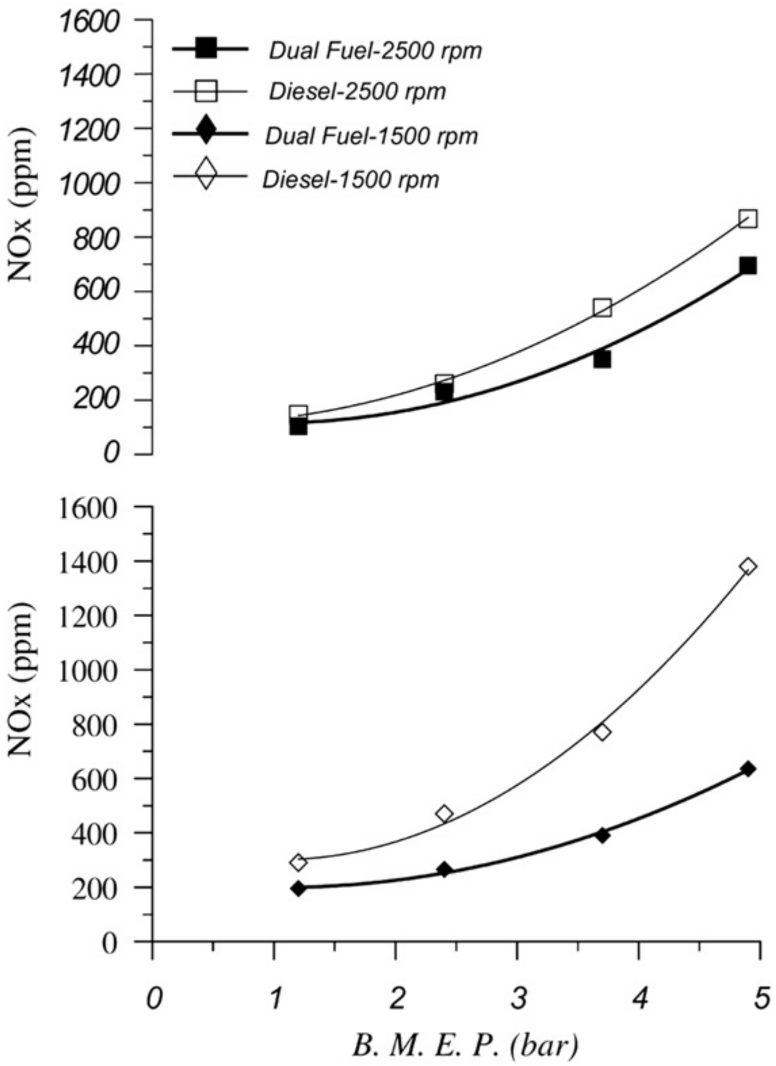


Fig 2.5: NOx emissions comparison under dual-fuel operation based on BMEP [37]

Dual-fuel engines emit considerably more part-load unburned HC than typical diesel engines do at low to moderate loads [21] [35] [36]. As opposed to less than 100 ppm during regular diesel operation, the HC concentrations are in the range of 6000 ppm as illustrated in *Fig 2.6* for dual fuel operation. Unburned natural gas, usually methane, is still present in the exhaust, which is the source of this. The unburned HC emission is comparable for both dual-fuel CI engines as well as conventional diesel-fuel operation at higher engine load conditions due to higher combustion temperature caused by a sub-stoichiometric fuel-rich mixture that oxidises most of the fuel [37].

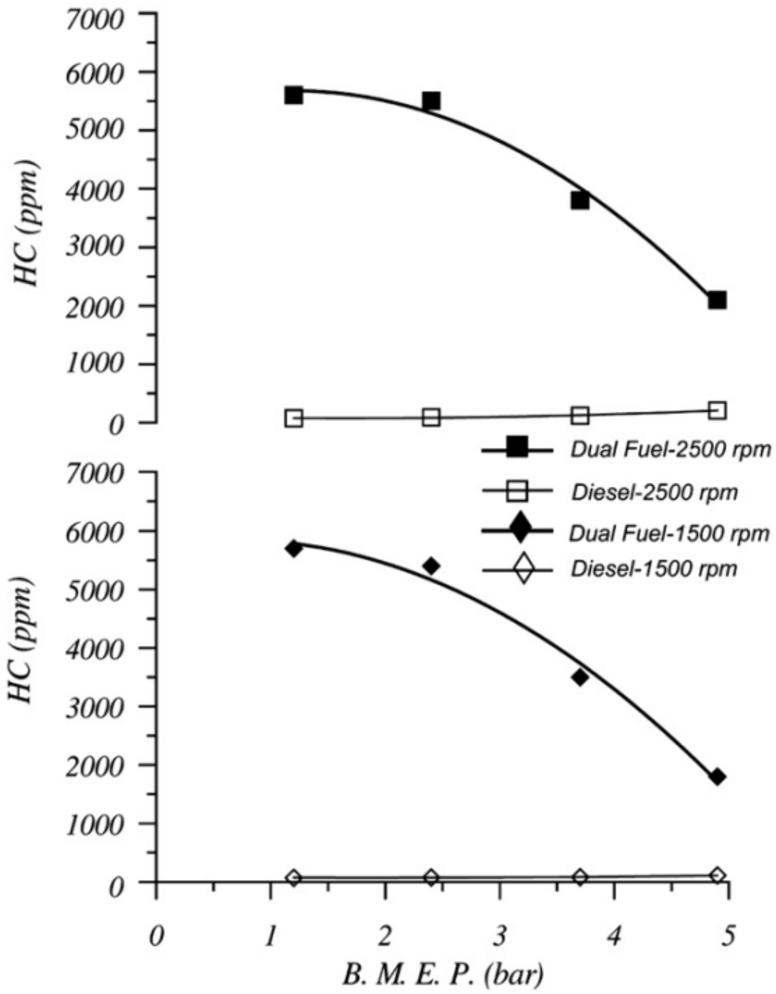


Fig 2.6: HC emissions comparison under dual-fuel operation based on BMEP [37]

Fig 2.7 illustrates the relationship between CO emissions and BMEP. Considering the load range at the speeds, the CO emissions are much higher than those of a conventional CI engine. These results are supported by some studies, which also show an even higher increase during part load conditions when operating in dual-fuel mode [37]. Several techniques have been suggested to improve the dual-fuel engine's lower-load emission characteristics. An effective strategy involves introducing supplementary pilot fuel into the overall combustion energy, leading to a rise in both the ignition points and the effectiveness of the combustion process for the natural gas-air mixture [44] [45]. This technique, while partially effective, falls short of the objective of lowering the amount of diesel fuel used in CI engines.

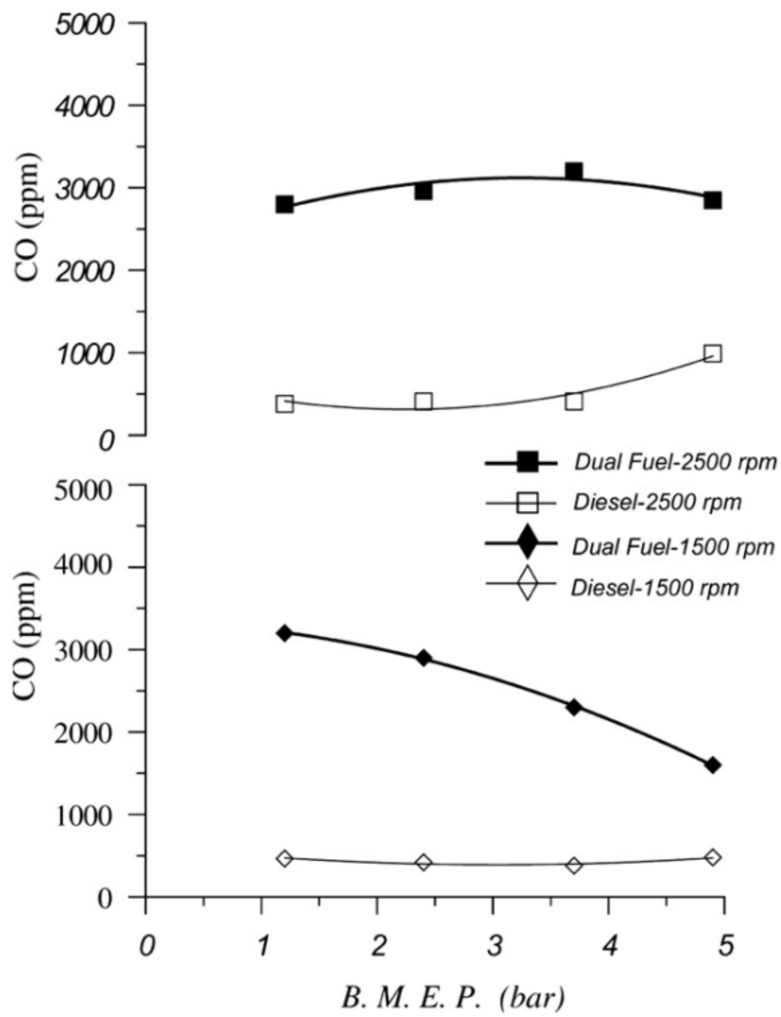


Fig 2.7: CO emissions comparison under dual-fuel operation based on BMEP [37]

Another option is to utilise small volumes of uncooled EGR (less than 5 % volume) as a low-load emission reduction technique [41] [42]. The elevated temperature of the combustion process is facilitated by the heat generated by the exhaust gas, which lowers emissions of unburned HC and CO [42]. This is caused by a hotter charge and a higher equivalency ratio (due to the use of EGR in place of air). The EGR system also exerts a chemical influence by introducing active radicals into the exhaust. These radicals, which are combustion byproducts in a partially oxidized state, play a crucial role in maintaining the chemical reactions involved in combustion. Because EGR has a dilution effect on the charge, NOx (*Fig 2.8*) and HC emissions (*Fig 2.9*) are reduced [46].

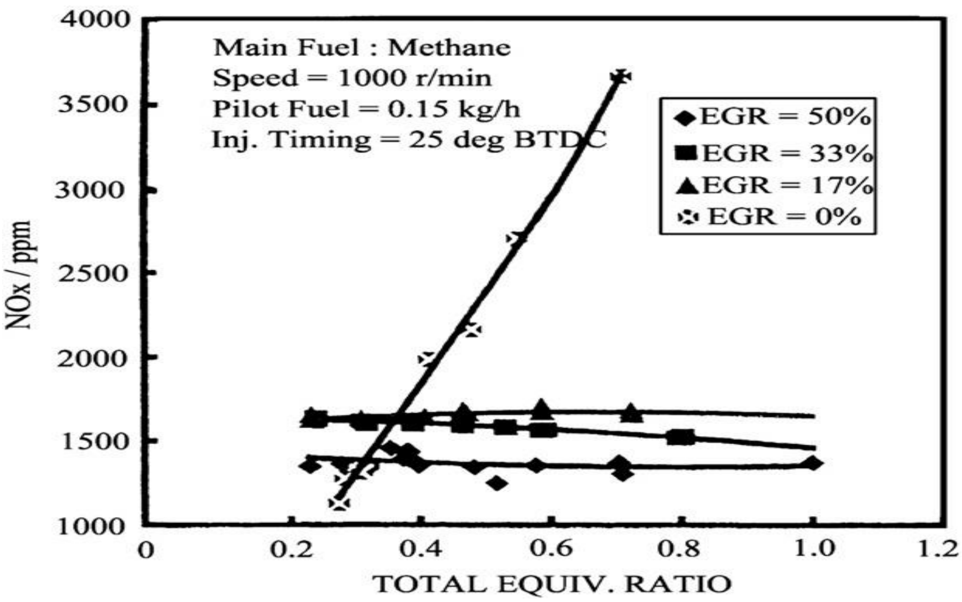


Fig 2.8: Variation of NOx emissions under dual-fuel operation with equivalence ratio at various EGR rates [46]

At low loads, the rise in charge temperature caused by EGR addition negates the diluting impact of EGR, however opposite result occurs at high load conditions. Extremely high amounts of CO and unburned HC are produced together with relatively low NOx emissions when EGR substitution rates of the intake air are more than 5% [42]. The favourable benefits of the hot EGR gases are outweighed by the dilution of the intake charge at high EGR rates. In addition, it was reported that an attempt was made to increase the maximum EGR rates by incorporating hydrogen into the charge [43]. The addition of 23 vol.% of hydrogen and a 40 vol.% EGR rate marginally increased NOx emissions but there was a significant decrease of 60% in CO emissions and a 40% decrease in unburned HC emissions [43]. When comparing dual-fuel engines to standard diesel engines, they generate around 30% less CO_2 [37]. This is because diesel has a lower hydrogen-to-carbon ratio than natural gas. When methane is combusted in the atmosphere, it produces approximately 2.8 grams of CO_2 through a stoichiometric reaction. On the other hand, diesel combustion under stoichiometric conditions results in the generation of approximately 3.2 grams of CO_2.

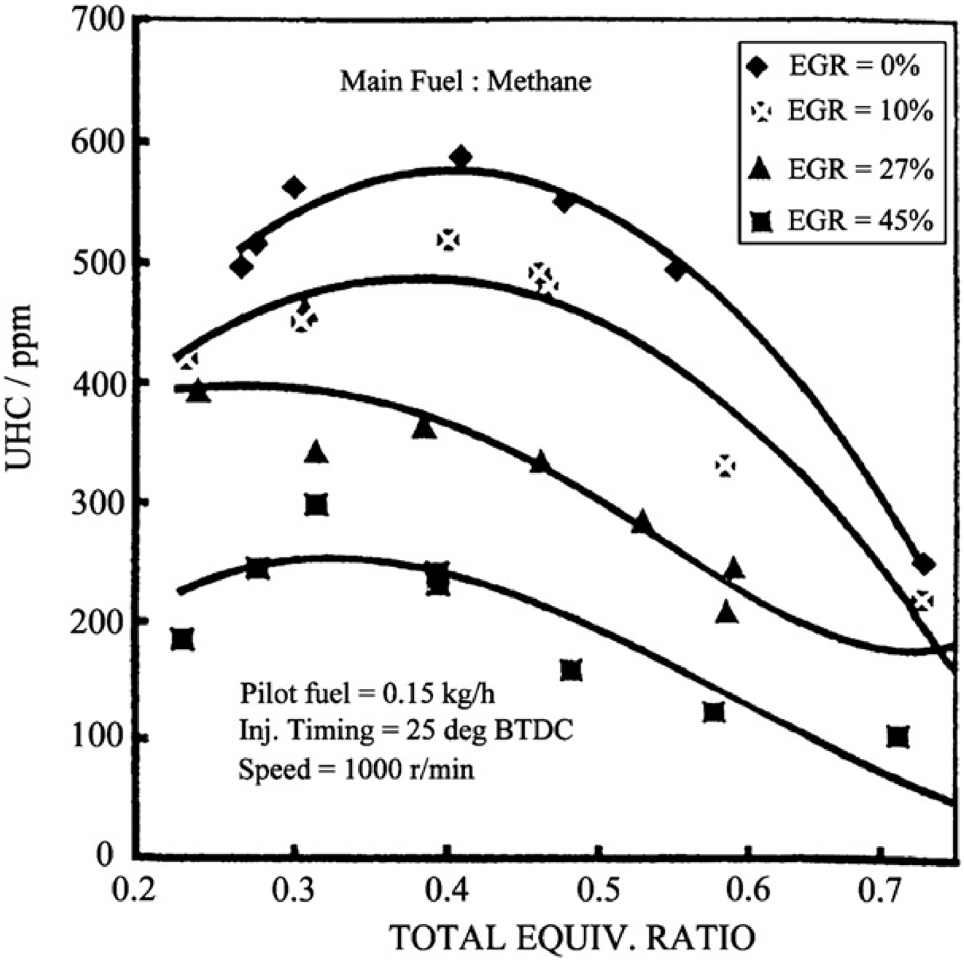


Fig 2.9: Variation of HC emissions under dual-fuel operation with equivalence ratio at various EGR rates [46]

2.2. NATURAL GAS DIRECT INJECTION IN CI ENGINES

The majority of previous studies conducted on dual-fuel engines have primarily concentrated on the modification of conventional diesel engines. These engines use the intake manifold to introduce natural gas into the combustion chamber while keeping the original in-cylinder injectors for pilot fuel injection. The addition of natural gas to the intake manifold has an impact on both the potential power output and volumetric efficiency, much as port-injected SI engines. Based on a specific study [43], it has been found that in dual-fuel engines, a single injector is employed to directly introduce both natural gas and pilot fuel into the cylinder. This configuration allows the dual-fuel engines to maintain their thermal efficiency and power output, distinguishing them from conventional diesel engines that are not equipped for dual-fuel operation. Additionally, it was reported that NOx and particle emissions were lower than anticipated [43].

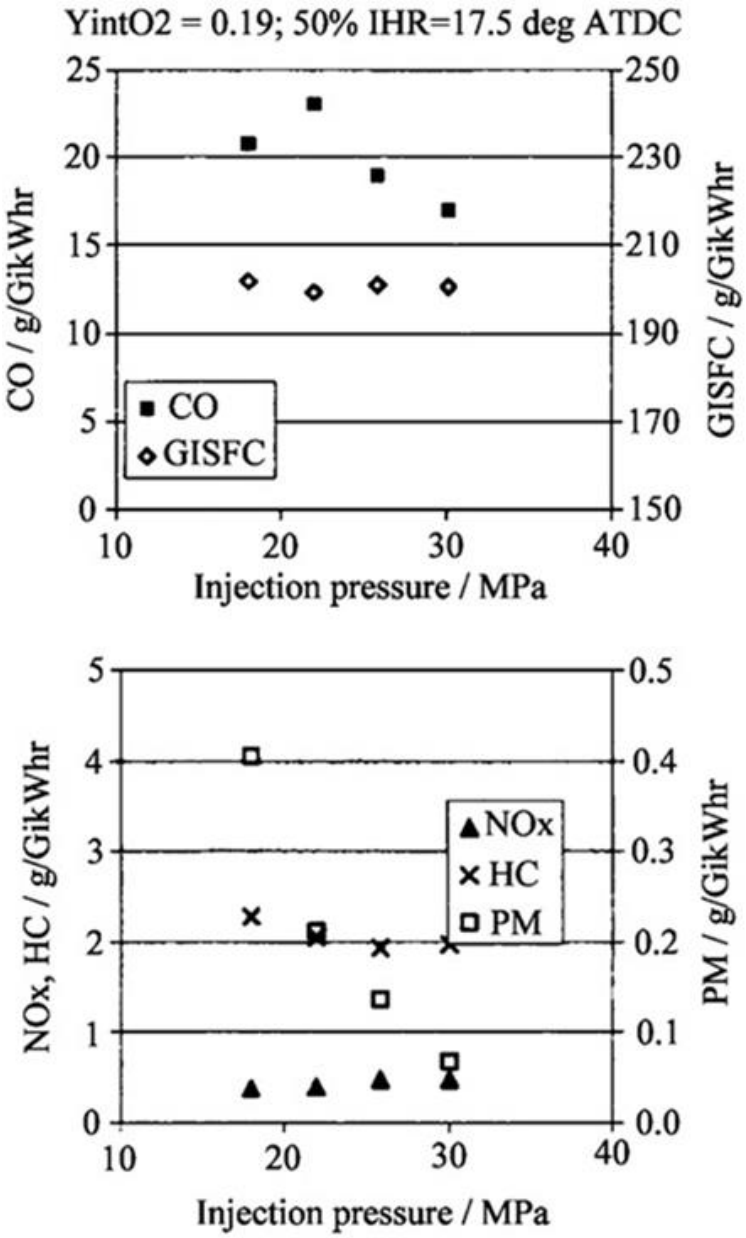


Fig 2.10: Specific fuel consumption and emissions trends with different injection pressure at 1200 rpm [47]

By fine-tuning the injection pressures of diesel and natural gas, direct-injected dual-fueled CI engines can achieve enhanced performance. The augmentation of injection pressures (from 21 MPa to 30 MPa) for both natural gas and pilot fuel leads to a decrease in the ignition delay of the diesel pilot fuel. This modification promotes a swifter and more effective blending of pilot fuel with air, specifically during the ignition delay phase. A shorter total combustion duration is the outcome of the increased rates of combustion progress [47].

Contrasted lower injection pressure settings, CO, HC, and smoke emissions are much lower, while NOx emissions are slightly higher. Improved levels of natural gas, air, and pilot fuel mixing as well as higher combustion-progress rates are the main contributors to such emission trends. The thermal efficiency levels were unaffected by changing the injection pressure, and *Fig 2.10* shows these emission patterns [47]. The graph also depicts the inverse relationship between gross indicated specific fuel consumption (GISFC) and thermal efficiency. Depending on the operating settings, especially the engine speed, these patterns vary significantly.

2.3. CONCLUSIONS BASED ON LITERATURE REVIEW

The major conclusions based on the literature survey are summarised below :-

- Dual-fueling is a process employed in CI engines where natural gas serves as a feasible alternative fuel with varying degrees of efficacy. Within CI engines, the combustion of natural gas is initiated through the utilization of pilot fuel. Due to the functioning of the induction mechanism, it is not possible for the intake manifold to create a mixture of natural gas and air.

- The reduction in the number of natural gas cycles being introduced leads to a decrease in power output at higher speeds. However, the impact of adding natural gas to the intake manifold on volumetric efficiency is reduced..

- Due to the difficulty to completely ignitie the natural gas-air mixture by the pilot diesel fuel, the thermal efficiency is decreased at low and moderate loads. The HC, NOx, and CO emissions are still comparable to those seen in normal CI engines. However, there is a noticeable increase in HC and CO emissions as the load increases.

- To minimize the emissions of CO and HC while achieving dual fuel operation, smaller quantities of natural gas can be effectively used at lower loads compared to pilot diesel fuel. This approach allows for the dispersion of ignition sites throughout the natural gas-air mixture, resulting in improved atomization and mixing of the pilot fuel with the natural gas. This is achieved through the incorporation of multiple smaller diesel injector holes, facilitating a more even distribution of the two fuels.

- At low to moderate loads, uncooled EGR reduces CO and unburned HC emissions by increasing combustion efficiency and accelerating combustion. In addition to these changes, the intake charge could include an additional fuel that improves natural gas's burning characteristics.

- Hydrogen in CI natural gas dual fuel engines enhances combustion flame speed, reduces COV, and improves engine stability. Furthermore, a reduced volume of hydrogen induction enhances EGR tolerance and boosts thermal efficiency.

- Many fuels, besides natural gas, have laminar flame speeds that have been extensively studied. It is necessary to conduct studies on natural gas combustion that replicate normal operating conditions for reciprocating piston engines. For instance, the properties of pure natural gas's flame propagation, turbulent flame speed, and emissions formation at these engine operating conditions are unknown. Even less is established about how other inputs like EGR, water, or hydrogen will influence performance.

- While there are a few computational assessments in the public literature, there aren't yet enough experimental data to support the numerical work or provide fundamental knowledge. Further investigation is essential to grasp and evaluate emission patterns linked to different engine designs, choices of engine variables, and regimes of combustion, including NOx-HC tradeoff. This requires a combination of computational and experimental research efforts.

- The impact of various performance characteristics, including fuel-injection timing, equivalence ratio, and compression ratio, on exhaust emissions from natural gas engines remains uncertain. A combination of experimental and computational investigations is necessary to develop better models for predicting the environmental impacts of all engine fuels.

- Further investigation is required to enhance the performance of the dual-fuel compression ignition engine utilizing natural gas as a fuel. To optimize emission characteristics, it is imperative to conduct thorough studies on adjusting injection pressure and timing, in addition determining the appropriate quantities of pilot fuel for different engine loads or equivalency ratios.

- Direct in-cylinder fuel injection is essential for enhancing the power output and reducing the low volumetric efficiency of CI natural gas engines compared to intake manifold induction or injection. However, there is a need for advancements in the techniques used for direct gas injection. At present, the majority of direct natural gas injectors are either custom-built or in the prototype stage. Because of lubrication and wear characteristics that are unknown, mass production is hampered.

- Gaseous fuels can cause lubrication issues with various engine parts, such as the valve seat and intake valves. More investigation into various engine component materials and alternative or supplementary lubrication techniques is required for natural gas-powered engines.

The literature survey findings on natural gas as a dual-fuel alternative for CI engines indicate that it is a viable option, but its effectiveness varies. Higher speeds experience reduced power due to limitations in the induction mechanism. Using pilot diesel fuel to ignite the complete natural gas-air charge is difficult that results in lower thermal efficiency and increased HC and CO emissions at higher loads. Improving combustion through better pilot fuel atomization and mixing with natural gas, as well as implementing uncooled EGR and additional fuel in the intake charge, can reduce emissions and enhance combustion efficiency. The addition of hydrogen improves flame speed, engine stability, and thermal efficiency. Additional study is required to comprehend natural gas combustion, the influence of EGR, water, and hydrogen, and to optimize dual-fuel CI engines through computational and experimental investigations. Direct in-cylinder fuel injection is crucial for improved power output, but current techniques require improvement for mass production. Furthermore, investigating lubrication issues and alternative lubrication techniques is necessary for natural gas-powered engines.

2.4. NULL HYPOTHESIS

For a DDF engine, the null hypothesis can be expressed as follows:

Null Hypothesis (H_0): Similar to that of normal diesel engine, the diesel-CNG dual fuel engine exhibits comparable performance and emission characteristics.

In this hypothesis, it is assumed that a diesel engine's power and emissions are not significantly impacted by the addition of CNG. In statistical hypothesis testing, the null hypothesis is often used as a benchmark for comparison to see if there is sufficient evidence to reject it in favour of an alternative hypothesis. To conduct a comprehensive analysis, we will explore various aspects concerning the efficiency and environmental impact of DDF engines that utilize both diesel and CNG.

Performance: The null hypothesis states that the utilization of CNG as an substitute fuel has no impact on variables such as power output, torque, thermal efficiency, and specific fuel consumption. It implies that, as compared to a traditional diesel engine, the addition of CNG has no appreciable impact on the engine's overall performance.

Emissions: According to the null hypothesis, a diesel-CNG dual fuel engine's emission characteristics are not significantly different from those of a regular diesel engine. It is based on the assumption that the combustion of CNG alongside diesel fuel won't have a major impact on the quantities of pollutants like NOx, PM, CO, and HC that are emitted.

Combustion: According to the null hypothesis, combustion in diesel-CNG dual-fuel engine functions similarly to that of a traditional diesel engine. The premise underlying this assumption is that the inclusion of CNG will not significantly impact the ignition delay, duration of combustion, or overall combustion stability.

Durability and Reliability: The null hypothesis also takes into account that the durability and reliability of the engine are unaffected by the use of CNG as a supplemental fuel. It is based on the idea that CNG combustion won't cause injectors, valves, and pistons to experience additional wear and tear. It implies that the engine can maintain levels of dependability and longevity comparable to those of a normal diesel engine.

Extensive study and experimentation are required to evaluate the null hypothesis. If there is sufficient data to reject the null hypothesis in favour of the alternative hypothesis, statistical analyses can be employed to assess the data collected, establishing compelling evidence of the notable disparities between the operational and emission characteristics of DDF engines and conventional diesel engines.

2.5. STATEMENT OF PROBLEM

Researchers are becoming more interested in dual fuel technology as a viable solution in the context of energy depletion and pollution as a result of the rising access to gaseous fuels like CNG and renewable methane. Large engines, such as marine and stationary engines used for power generation, have traditionally utilised dual fuel technology. However, because of the restricted availability of CNG and the lack of a network, it was limited for usage in vehicle engines in the past. When dual-fueling of CNG in diesel engines involves injecting or inducting CNG into the intake manifold without modification, there is a volumetric efficiency penalty. Furthermore, since the volume of CNG inducted every cycle decreases at higher speeds, there is a power reduction. At lower to moderate load conditions, the natural gas-air charge fails to get ignited through pilot fuel, resulting in decreased thermal efficiency. Although NO_x emissions are reduced, HC and CO emissions increase significantly. To increase the efficiency and performance of CI engines running on natural gas, extensive performance and emissions tuning are necessary. It's expected that a variety of engine operating modes will be required. *As a result, performance and emissions must be optimised when utilising CNG in a dual-fuel diesel engine.*

2.6. MOTIVATION FOR RESEARCH

The challenges associated with the performance of CNG diesel dual fuel operation concerning speed and load [23] are illustrated in *Fig 2.11*. Poor flame propagation and incomplete combustion result from unthrottled operation under low load and high-speed conditions. High loads cause pre-ignitions, knocking, and diesel injector tip overheating, indicating heat and cooling issues. Despite the load, crevice slip generation of NOx and methane must be minimised. One of the objectives of the present research is to address these challenges and enhance the operational capabilities of the dual-fuel engine, thereby expanding its operating range. The current status of each of these occurences as well as their contributions must be looked at with CNG-dual fuel combustion at lower and higher loads.

Furthermore, it is evident from the aforementioned literature study that research on ECU-based diesel-CNG dual fuel engine systems has been limited. Also, there is no commercial dual fuel diesel vehicle available in India. The area of research that remains to be addressed is the advancement of dual fuel operation for new generation diesel engines using ECU.

In light of the aforementioned, the diesel engine's performance can be enhanced in the following ways:

(i) Similar to Westport HPDI technology, which uses diesel as a pilot ignition source or "liquid spark plug" and natural gas as the main fuel the design of a high-pressure direct injection injector.

Or

(ii) Undertake hardware modifications in the engine side which may include the installation of a CNG fuel supply system, manifold gas injection system, injection timings, Dual-Fuel ECU, etc. for better performance

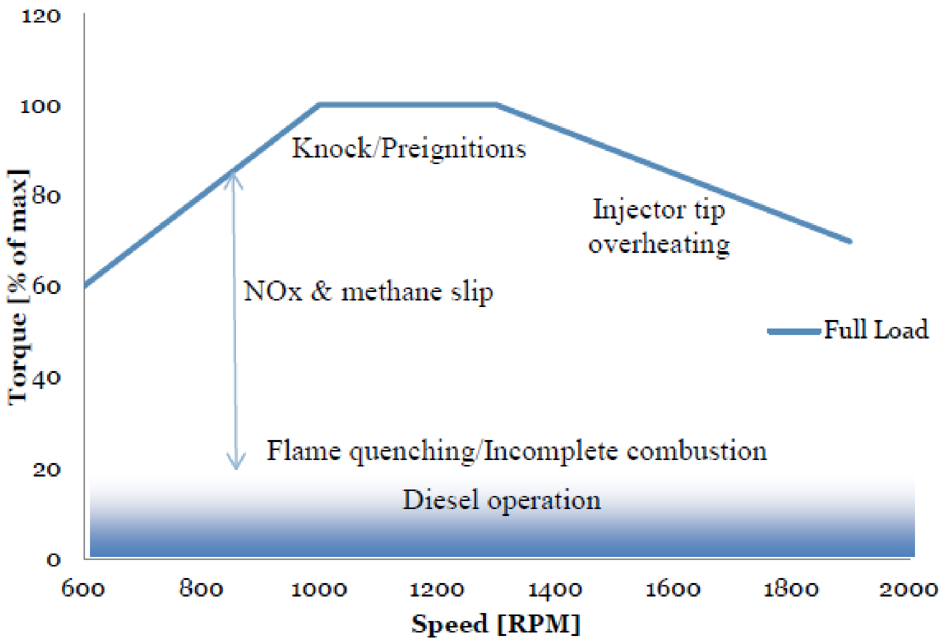


Fig 2.11: Challenges of CNG-Diesel Dual Fuel Engine operation.

2.7. OBJECTIVES OF RESEARCH

Considering the technical constraints in designing High-Pressure Direction injection due to infringement of Patent, the following objectives are proposed for the current study, which would enable the utilization of CNG in diesel engines without compromising their performance.

- To identify and select suitable CNG injection system for a commercial BS-IV diesel engine, based on multiport gas injection dual-fuel ECU.
- To identify the correct CNG – Diesel ratio for optimum engine and emission performance.

- Performance evaluation of the modified dual-fuel engine in comparison to the original diesel engine.
- Performance assessment and analysis of the effect on engine oil of a modified dual-fuel engine.

2.8. POTENTIAL NOVELTY OF THE WORK

The potential of diesel-CNG dual fuel technology can be examined from various angles, particularly in relation to the growing electric vehicle (EV) industry. The EV market is clearly growing quickly, but diesel-CNG technology can still be advantageous in specific circumstances and applications. The following are some potential applications for this technology:

- Transitional Solution: For businesses and sectors that rely significantly on diesel engines, such as commercial trucks, heavy-duty vehicles, and marine vessels, diesel-CNG dual fuel technology can act as a transitional solution. These industries frequently need long-range capabilities and better energy density, which the present EV infrastructure or technology may not be able to fully support. In comparison to traditional diesel engines, diesel-CNG engines can offer a cleaner and more environmentally friendly alternative while lowering emissions and reliance on fossil fuels.

- Fuel Cost Reduction: CNG is typically less expensive than diesel fuel, providing fleet operators and other sectors with the possibility of cost savings. The entire fuel usage can be optimised by combining CNG and diesel, which lowers operating expenses. This financial benefit might be especially alluring for long-distance transportation, when fuel costs account for a sizable portion of the operating budget.

- Emissions Reduction: When compared to conventional diesel engines, there is a high potential for diesel-CNG dual fuel technology to significantly reduce emissions. Reduction in particulate matter (PM) and sulphur emissions from CNG combustion result in better air quality and a lower environmental impact. Even though EVs produce no emissions when in use, the overall emissions from energy production and battery manufacturing must be taken into account. Diesel-CNG technology can offer a cleaner option in some areas where the electricity mix is still dependent on fossil fuels.

- Infrastructure Availability: CNG already has a widespread distribution network, unlike EVs, which need a strong charging infrastructure. The use of diesel-CNG dual fuel engines can benefit from this infrastructure, making it simpler to install this technology in some places. This benefit is especially important in areas with sparse charging infrastructure or where it's possible that the power grid can't handle a significant increase in EV deployment.

- Operational Flexibility: Diesel-CNG dual fuel engines offer the flexibility for vehicles to operate on either diesel fuel, CNG, or a combination of both, providing a range of fueling options. Fleet operators can respond to shifting conditions, such as differences in fuel supply or governmental regulations, owing to this flexibility. Additionally, it has a backup option that enables vehicles to switch to diesel operation if the supply of CNG is disrupted.

It is significant to highlight that the development of infrastructure, regulatory frameworks, and advances in battery technology all play an important role in shaping the future of diesel-CNG dual fuel technology. The need for diesel-CNG technology may be impacted as the EV market develops and picks up steam. However, diesel-CNG dual fuel technology can provide an alternative and more environmentally conscious option for particular applications and industries where EVs may not be immediately practicable.

CHAPTER

3. RESEARCH METHODOLOGY

3.1. THEORETICAL FRAMEWORK

The suggested study topic was thoroughly surveyed in several peer-reviewed journals, books, patents, and other research resources for information on various facets of the issue. Based on the literature review conducted so far, the following methodology was proposed.

- Selection of engine and preparation of experimental set-up.
- Emission and performance with engine mapping.
- Generation of base data with diesel fuel.
- Develop a dual fuel conversion kit.
- Optimize the design through laboratory testing by varying the CNG flow rate, EGR, CO_2 and hydrogen etc.
- Generate performance, lubricants and emission data under different operating conditions.
- Development of multiport gas injection dual-fuel ECU.
- Development of unique control strategy of CNG fuel under Dual-Fuel operation.
- Generate and analyze performance and emission data across various operational scenarios.
- Assess the impact of dual fuel operation on emission control devices.
- Analysing of lube oil properties after a longer period of operation.
- Analysis and inference of the test data.
- Documentation, publications and thesis writing.

3.2. TEST MATRIX

The study utilized the test matrix presented in *Table 3.1* as a reference.

Table 3.1: Test Matrix

Engine Speed, rpm	Neat Diesel	CNG Flow Rate, kg/hr								Load
		0.65	0.93	1.41	2.1	2.5	3.0	3.5	4.0	
1000	✓	✓	✓	✓	✓	✓	✓	✓	✓	Idle
1250	✓	✓	✓	✓	✓	✓	✓	✓	✓	25%
1500	✓	✓	✓	✓	✓	✓	✓	✓	✓	
1750	✓	✓	✓	✓	✓	✓	✓	✓	✓	50%
2000	✓	✓	✓	✓	✓	✓	✓	✓	✓	
2250	✓	✓	✓	✓	✓	✓	✓	✓	✓	75%
2500	✓	✓	✓	✓	✓	✓	✓	✓	✓	100%

The experimention initialy followed the test matrix as per Table 3.1 under different modes of engine operation viz. throttle-speed, speed-torque and full throttle condition. The parameters monitored during the experimentation includes :-

- Engine : Engine Speed, Air Mass flow rate, Throttle, Blow-by, Combustion Air Pressure, Combustion Air Temperature, Combustion Air Humidity, Intake Air Temperature, Intake Air Pressure, Boost Air Pressure, Boost Air Temperature, Fuel Pressure, Fuel Temperature, Oil Sump Temperature, Fuel Flow Rate, Coolant Temperature, Power, Torque, Exhaust Manifold Temperature, Exhaust Manifold Pressure, Pre-SCR Temperature, Pre-SCR Pressure, Post-SCR Temperature, Post-SCR Pressure, Exhaust Back Pressure.
- Emission: Pre and Post SCR Raw Emission – CO, THC, CH_4, NOx, O_2 and Post SCR Unregulated Emission - C_2H_2, C_2H_4, C_3H_6, C_3H_8, CH_4, CO_2, H_2O, HCOOH, HCHO, N_2O, NH_3, NO, NO_2, SO_2.

- Combustion : Crank angle, Volume, Energy, Pressure, Pressure per angle, Burn, Speed, Current

3.3. PARAMETERS STUDIED

Engine Performance

- Power, Torque, Energy Substitution Rate, Fuel Mass Flow Substitution Rate, Relative Air Fuel Ratio, Brake Thermal Efficiency, Volumetric Efficiency, Brake Specific Energy Consumption, Brake Specific Fuel Consumption, Engine Exhaust Temperature, Boost Air Temperature and Boost Air Pressure

Emission Performance

- Regulated emissions (Pre-SCR & Post-SCR) : THC, NOx, CO, CO_2 & CH_4
- Unregulated Emissions : Acetylene, Ethylene, Propene, Propane, Formic Acid, Formaldehyde, Ammonia & Sulphur Dioxide

Combustion Performance

- Heat Release Rate, In-cylinder Pressure, Pressure Rise Rate, Combustion Duration, Fuel Burn Rate, Ignition Delay, Indicated Mean Effective Pressure and Diesel Injection Duration.

3.4. WORK PLAN

The activity schedule and the timeline of achieving the same are indicated in *Fig 4.1*.

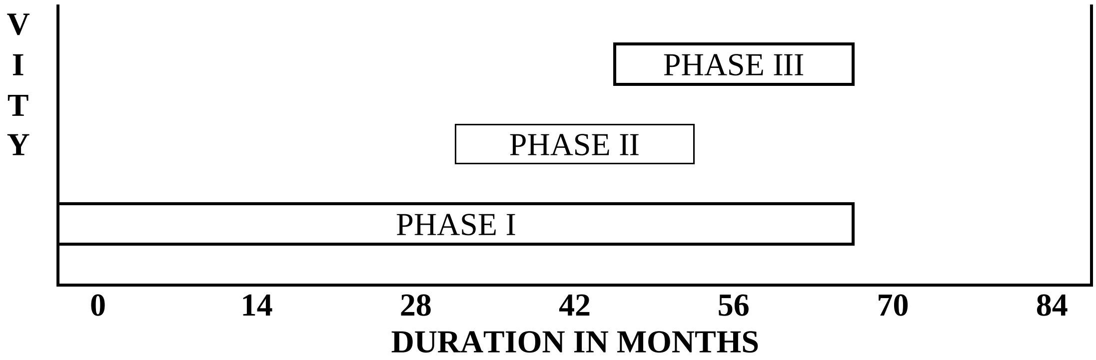


Fig 3.1: Work Plan

Phase I : Course work completion and extensive literature survey

Phase II : Selection of engine, preparation of experimental set-up, generation of base data.

Phase III : Develop a dual fuel conversion kit, and optimise the dual fuel engine's performance in terms of power, fuel consumption, emissions, engine oil, etc.

Phase IV : Development of Dual Fuel Kit, control strategy, generate performance, emission data under different operating conditions, assess the impact.

Phase V : Analysis and inference of the test data, conclusions and/or recommendations

Phase VI : Documentation, publications, submission of thesis and desertion.

3.5. EXPERIMENTAL SET-UP

3.5.1. Commercial Heavy Duty Diesel Engine

A commercial in-line six-cylinder turbocharged-inter-cooled diesel engine compliant with BS-IV emission regulations and fitted with a Selective Catalytic

Reduction (SCR) system that runs on a typical diesel pilot fuel injection system was used for the investigation. By implementing CNG injection before the intake manifold, it became possible to convert the engine for DDF operation. In this mode, the diesel engine's original ECU was utilized, and its engine map played a crucial role in conducting performance tests. For reference, *Table 3.2* provides the specifications of the engine.

Table 3.2: Detailed Engine Specification.

Parameter	Specification
Engine Type	Six-cylinder, twelve valves, water cooled, in-line Turbo-charged
Emission Norms	BS-IV with SCR
Capacity	5756 cc
Bore/Stroke	104 mm x 113 mm
Maximum Torque	800Nm @ 1400-1900 rpm
Maximum Power	225 HP (165 kw) @ 2500 rpm
Idle engine speed	600 rpm
Maximum engine speed	2750 rpm
Timing and arrangement	ECU controlled
Fuel injection system	Common-rail high-pressure direct

3.5.2. Engine dynamometer Engine Test Bench

For this study, an AVL Alpha 350 eddy current dynamometer with a PUMA control system was utilized. The dynamometer can produce a torque of 1500 Nm and 350 kW of power and can operate at 8000 rpm at its top speed. The engine test bench is fully automated, with multiple test variables and measurements controlled and monitored. It is capable of performing emission legislation test cycles in accordance with both Euro IV and BS IV criteria. The AVL engine test bench also has other parts, including a combustion air handling system, an airflow metre, a fuel measurement system, a fuel conditioning system, a gas flow metre, a coolant conditioning system, a boost air conditioning system, and an oil conditioning system in addition to the dynamometer automation that is controlled by computer. The engine testing facilities used for this study are depicted in *Fig 3.2*.

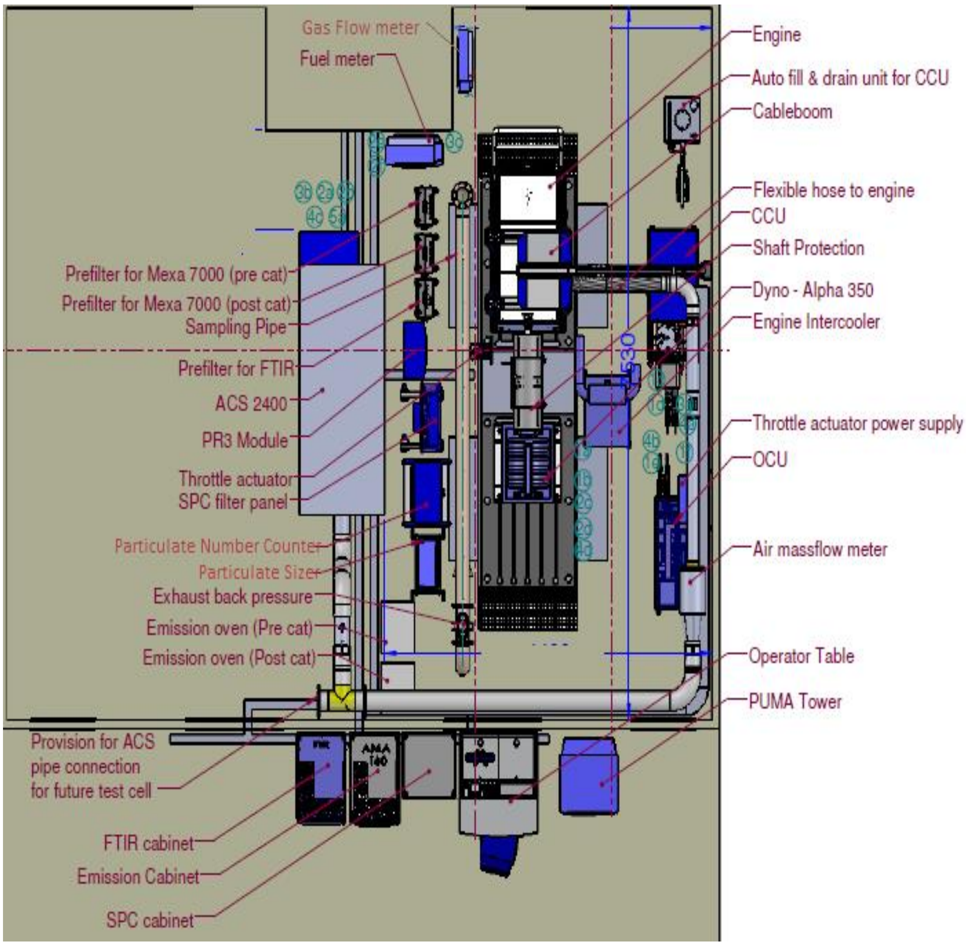


Fig 3.2: Engine Test Facility commissioned for the Research Study.

3.5.3. Engine Intake Air Measurement System

During the testing, the direct mass flow of the engine's intake air was measured using a dynamic air mass flow metre based on the hot-film anemometer principle.

3.5.4. Fuel Measurement System for Diesel and CNG

Separately, a fuel monitoring system based on the Coriolis flow principle was utilised to measure diesel and CNG use. Fuel consumption was measured during the research using a diesel fuel mass flow meter. To maintain a consistent fuel temperature of 25°C, the meter was linked to a fuel conditioning system.

Additionally, great attention was given to maintaining the specified return and inlet pressures of the diesel fuel line, ensuring compliance with the engine test requirements.

3.5.5. Engine Coolant Conditioning System

The engine coolant conditioning system used in the study was compatible with steady-state test standards for set points ranging from 70 to 110°C. Throughout the DDF mode of operation, the engine's coolant temperature was kept at 85°C.

3.5.6. Engine Oil Conditioning System

Engine oil's temperature and pressure were controlled at a predetermined point value in the range of 70-140°C using an engine oil conditioning system. The engine's oil temperature was also kept at 90°C throughout the DDF mode of operation testing.

3.5.7. Boost Air Conditioning System

While conducting the test, the intercooler of the engine was disconnected. As a result, the turbocharger directed the high-pressure, high-temperature air into a water-cooled boost air conditioning system. This system effectively maintained the air temperature to the engine at a constant 30°C.

3.5.8. Mass Emission Measurement System

The Horiba MEXA 7500 DEGR, which meets Euro IV requirements, was used to monitor the raw emissions from the CAT before and after, as well as the EGR CO_2 sample and regulated mass emissions like THC, CO_2, NMHC, CO, O_2, NOx and CH_4. For certification testing, catalyst efficiency evaluation, EGR testing, air/fuel calculation, and other applications, the emission analyser provides exhaust emission measurement capability. The system includes a remote heated analyzer that is equipped with sample handling capabilities, as well as a Flame Ionization Detector (FID) for total hydrocarbon measurement, a second FID with Methane cutter, and a Chemiluminescent analyser for nitrogen oxide detection. A non-dispersive infrared (NDIR) detector located in the analyzer rack measures the CO_2 and CO emissions. A chamber that is illuminated by infrared light is pumped with the sample gas. The wavelength of the light is chosen particularly for each component to be detected. The component absorbs light if it is in the sample, which reduces the amount of light that exits the sample chamber. The concentration of the specific component is determined by comparing the intensity of the light emitted by the sample with that of the reference light source. By passing a sample through an alternative magnetic field while it is in a nitrogen stream, the Magneto-pneumatic analyzer utilizes the magnetic characteristics of O_2 to assess the oxygen levels present in a given sample. The pressure variation observed in the detection plates is directly influenced by the presence or absence of oxygen in the sample. It is possible to calculate the total amount of hydrocarbons in a sample by passing it through a hydrogen flame. The amount of carbon atoms in the hydrocarbons in the sample determines how much ionisation current is generated. A collector electrode that is placed around the flame is used to measure the ionisation current. The amount of hydrocarbons in a sample is measured in parts per million (ppm-C). The underlying assumption made in this statement is that the response generated by a propane molecule (C_3H_8) is threefold higher compared to that of a methane molecule (CH_4), which consists of only one carbon atom. Flame ionisation (FID) is used to determine the levels of THC and CH_4 in the engine exhaust gas. THC is detected using the hydrogen flame ionisation method, while CH_4 is detected using the non-methane (non CH_4, NMHC) cutter method and hydrogen flame ionisation. In the chemiluminescent method for detecting nitric oxide, the photo-diode is utilised to analyse the interaction between nitric oxide (NO) and ozone (O_3). The process results in the production of a light photon. The quantity of NO present is correlated with the number of photons emitted. A sample containing NOx must be run through a converter that dissociates NOx in order to detect NOx.

3.5.9. FTIR Motor Exhaust Gas Analyser

The MEXA 6000 FT Series developed by Horiba combines the Fourier Transform Infrared (FTIR) technique with an advanced multivariate analytical algorithm. This advanced technology enables the simultaneous detection of various exhaust gas components, such as NH_3, N_2O, CH_4, CO, CO_2, Aldehyde, Ketones, SO_2, and more. This system can accurately identify undiluted gases from engines even without the need for dehumidification. It achieves this by utilizing a sample device and a heated gas cell within the unit. This system complies with the heavy-duty emission testing criterion for NH_3 under the Euro IV norms.

3.5.10. Combustion Analyzer

The investigation of combustion parameters involved the utilization of Kistler's KiBOX. This analyzer served the purpose of collecting and analyzing data, offering valuable insights into combustion quality, injection timing, and other relevant parameters. In order to accurately gauge the in-cylinder pressure, a meticulous approach was employed to insert a piezoelectric pressure sensor through a specialized sleeve designed for pressure sensors, specifically into the head of the sixth cylinder. To establish the crank angle, an adapter was seamlessly connected to the flywheel pulse sensor of the engine. Measurement of injection timing was accomplished by employing a current clamp, which was attached to the injector cable. The combustion analysis encompassed several aspects, including the pressure rise rate (PRR), indicative mean effective pressure (IMEP) and heat release rate (HRR). These values were obtained by calculating the average in-cylinder pressure across a continuous sequence of 300 cycles.

3.5.11. Data Acquisition System

For multiple parameter measurements, AVL PUMA 1.5 engine test bed, a sophisticated system for testing and evaluating the engine was used. It incorporates a comprehensive Data Acquisition System (DAS) to collect and analyze various engine parameters during the testing process. The components of DAS and their functions is given below:

- Sensors: The DAS is equipped with a wide range of sensors that are strategically placed on the engine and its peripherals. These sensors capture data related to engine speed, temperature, pressure, torque, fuel consumption, emissions, and other relevant parameters. They convert physical measurements into electrical signals for further processing.

- Signal Conditioning: The electrical signals from the sensors often require conditioning to ensure accuracy and compatibility with the DAS. Signal conditioning modules are used to amplify, filter, and convert these signals to appropriate levels for digitization and processing.
- Data Acquisition Unit: The core of the DAS is the Data Acquisition Unit (DAU). It receives the conditioned signals from the sensors and performs analog-to-digital conversion. The DAU typically consists of multiple high-speed analog-to-digital converters that sample the incoming signals at a high rate, typically in the kilohertz range. It also synchronizes the data acquisition process across all channels to ensure accurate and time-aligned measurements.
- Data Storage: The acquired data is stored in a dedicated storage medium, viz. high-capacity hard drive. The storage system have sufficient capacity to accommodate the large volume of data generated during extended engine tests.
- Real-time Control and Monitoring: The DAS provides real-time control and monitoring capabilities that is achieved through a user interface, which displays live sensor data, engine parameters, and other relevant information. The operator can set test parameters, configure the data acquisition process, and monitor the engine's performance during the test.
- Data Analysis and Post-Processing: After the test is completed, the acquired data is transferred to a separate analysis workstation / software for further processing and interpretation.
- Integration with Test Automation Systems: The DAS of AVL PUMA 1.5 engine test bed is integrated with test automation systems to enable automated test sequences and control.

The Data Acquisition System consists of sensors, signal conditioning modules, a DAU, data storage, real-time monitoring and control interfaces, data analysis tools, and integration with test automation system. It enabled accurate and comprehensive data collection during engine tests, facilitating performance evaluation, optimization, and analysis.

3.5.12. Test Fuels

For DDF operation, typical high-speed diesel fuel that complies with the fuel requirements of IS:1460 BS-VI specification and CNG in accordance with IS:15958 fuel regulations were used. The CNG was inducted into the intake manifold along with diesel acting as the pilot fuel, with the attributes stated in *Table 3.3*.

Table 3.3: Specification of Test Fuel.

Fuel properties	BS VI Diesel	CNG [48]
Auto-ignition temperature (°C)	316	650
Cetane number	52.5	–
Boiling point (°C)	180–360	−162
Carbon content (%)	87	75
Evaporation heat (kJ/kg)	250	–
Octane number	–	130
Low heating value (MJ/kg)	42.91	48.6
Stoichiometric air-fuel ratio (kg/kg)	14.69	17.2
Specific Gravity	0.820	0.466
Lean combustion limits in air's volume ratio (%)	1.4	5.0
Lean combustion limits in air's volume ratio (%)	7.6	13.9

According to the design specifications, at a pressure of 220 bar, CNG was kept in a cascade system. Then, using a two-stage regulator, it was decompressed to a pressure of 6 bar. After the intercooler, a gas induction system was devised and put into operation to directly inject CNG into the intake manifold. Prior to introducing CNG into the intake manifold, the gas flow rate was controlled manually using a control valve. To determine the required mass flow rate of CNG, a digital gas flow meter was utilized.

CHAPTER

4. TEST METHODOLOGY

4.1.TEST PROCEDURE

4.1.1. Test Set-up

Fig 4.1 depicts the schematic for the CNG induction system and the position of the gas fuel induction system.

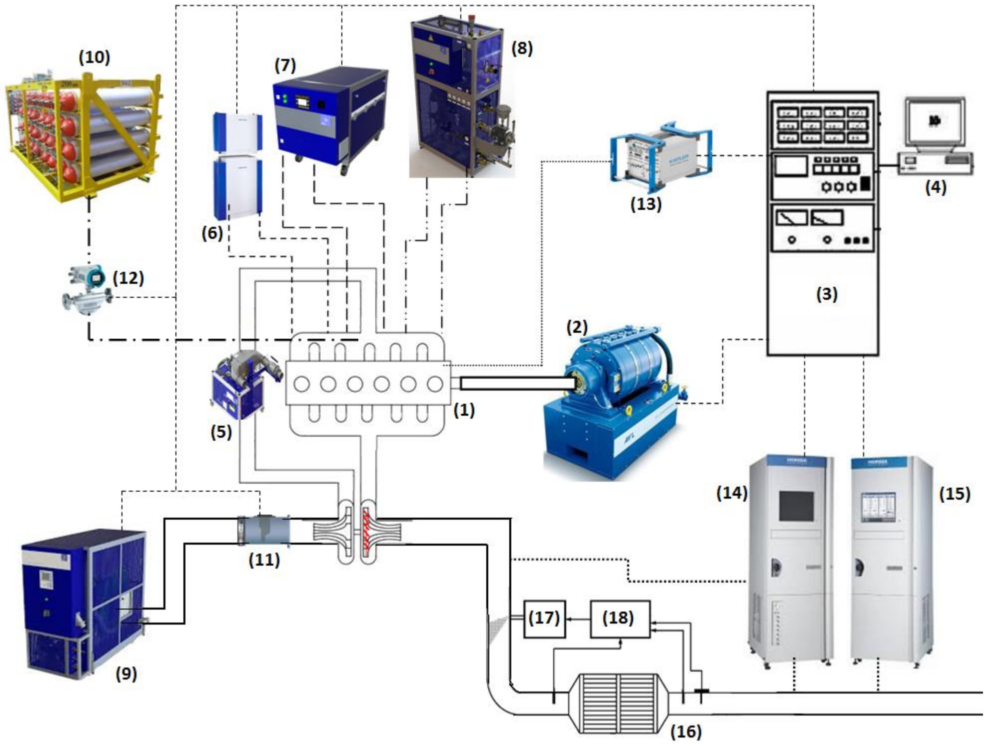


Fig 4.1: Schematic of Experimental Test Configuration.

(1) Commercial Heavy Duty Six Cylinder Turbocharged Intercooled Direct Injection Diesel Engine, (2) Eddy Current Dynamometer, (3) Control Unit of Dynamometer, (4) Data Acquisition and Dynamometer Control Computer, (5) Boost Air Conditioning Unit (Intercooler), (6) Diesel Mass Flow Measurement system and Fuel Conditioning Setup, (7) Unit for Conditioning Oil, (8) Unit for Conditioning Coolant, (9) Combustion air conditioning unit, (10) CNG Cascade, (11) Air Mass Flow Meter, (12) CNG Gas Flow Meter, (13) Combustion Analyser, (14) Mass Emission Analyser, (15) FTIR Motor Exhaust Gas Analyser, (16) Selective Catalyst Reduction, (17) Diesel Exhaust Fluid Injection Unit, and (18) Engine Control Unit.

Fig 4.2: Experimental Test Configuration

The engine test stand's experimental test arrangement is shown in *Fig 4.2.* Throughout the research project, the original ECU that was already installed in the diesel commercial engine was utilized due to the limitations imposed by the engine's design characteristics, which restricted the extent to which CNG substitution could be implemented. Using CNG in diesel engines intends to promote the use of less expensive CNG fuel in India while decreasing crude oil imports. Software for the AVL PUMA 1.5 engine test bench was used to run the engine at various speeds that ranged between 1000 and 2725 rpm at an interval of about 250 rpm and under various load conditions of 25, 50, 75, and 100%. While the CNG port fuel induction was adjusted throughout a variety of flow rates, especially between 0.67 and 4.0 kg/h, the diesel fuel flow remained consistent in accordance with the engine's design. As a result, data collected with various CNG flow rates were compared to data generated with neat diesel fuel. The torque that was developed exceeded the engine's design parameters, preventing the constant flow rate of CNG from being raised further. As previously stated, the operational conditions for each experiment were maintained throughout the research.

CNG was considered as 99 % methane (CH_4) gas in this study. For CNG substitution rates and diesel fuel, the tests were conducted using the equivalent total energy approach. The percentage of energy substitution for CNG was determined by assessing the percentage of CNG energy within the overall energy composition of both CNG and diesel fuel. The Appendix contains a list of relevant mathematical equations.

4.2.TEST UNCERTAINTIES/ACCURACY

In scientific research and data analysis, calculations of uncertainty and the use of accuracy play a key role. The measurement error or the range of values that the true value of a quantity is anticipated to fall under are both examples of uncertainty. To ensure the dependability and repeatability of research findings, it is crucial to measure and disseminate uncertainties. Several statistical and mathematical methods, including error propagation, confidence intervals, and standard deviation, are used to calculate uncertainty. Researchers can evaluate the degree of confidence in their measurements and experimental findings using these computations. On the other hand, accuracy measures how closely a measured value matches the genuine value. It serves as a gauge of the measurement method's dependability and the lack of systematic mistakes. Calibration, reference standards, and repeated measurements can all be used to determine accuracy. Since they serve as the foundation for deducing meaningful conclusions and making legitimate decisions, accurate measurements are essential for achieving meaningful and reliable outcomes.

The uncertainties of the tested data are impacted by a number of errors, including test methodology, calibration of the test bench, examined correctness, and arbitrary fluctuation of the utilised equipment. In this thesis, the accuracy of the experimental measurements has been assessed using strict uncertainty analysis methodologies. It aims to give a thorough assessment of the research's constraints and potential sources of inaccuracy by quantifying uncertainty. Furthermore, efforts have been made to ensure accuracy through the application of suitable calibration processes, the use of trustworthy reference standards, and repeated measurements. These efforts have improved the findings' reliability and validity, adding to the thesis' overall credibility and expanding the extent of knowledge in the area.

Before each test, all of the instruments used in this research were calibrated to verify that the measurements were accurate. In terms of emissions and performance, data was collected for one minute under each operating condition, while the combustion study utilised an average of 200 consecutive cycles. To ensure the accuracy of the measured readings before moving on to the next test, the emission analyzers underwent purging and calibration after each test. The results presented in this paper represent the average values obtained for each test or operating condition. To determine the overall uncertainty of the tested values, a combination of systematic and random uncertainties was integrated. In the study, crucial experimental equipment was utilized to gather data, and *Table 4.1* presents comprehensive details concerning the precision and measuring range of these instruments.

Table 4.1: Precision of the primary testing equipment employed

Measured Parameter	Measuring Range	Accuracy/Sensibility
Dynamometer	0-350 kW	
Engine Torque	0-1500 Nm	0.2% FS
Engine Speed	0-8000 rpm	1 rpm
Intake Air Temperature	10-35°C	+/- 0.5°C
Intake Air Humidity	4 to 30 g/kg	+/- 3% RH
Intake Air Pressure	920 to 1020 mbar	1 mbar
Intake Air Mass Flow	0-2400 kg/h	< 1% MV
Boost Air Temperature	15 to 70°C	+/- 1 K
Diesel Fuel Mass flow	0-125 kg/h	Us <= 0.12%
Diesel Fuel Temperature	10 to 80°C	+/- 0.02 °C
Coolant Temperature	70 -110 °C	<= +/- 2 °C
Engine Oil Temperature	70 – 140 °C	<= +/- 2 °C
CNG Mass Flow	0-150 kg/h	+/- 0.25% of rate
CO (NDIR)	0-5000 ppm	<+/- 0.5 % FS

CO_2 (NDIR)	0-20%	<+/- 0.5 % FS
THC (H.FID)	0-50000 ppmC	<+/- 0.5 % FS
NOx (H.CLD)	0-10000 ppm	<+/- 0.5 % FS
CH_4 (NMC+ H.FID)	0-25000 ppmC	<+/- 0.5 % FS
O_2 (Magnetic)	0-25%	<+/- 0.5 % FS
NH_3 (FTIR)	0-1000 ppm	<+/- 1% FS
Aldehyde (FTIR)	0-500 ppm	<+/- 1% FS
SO_2 (FTIR)	0-200 ppm	<+/- 1% FS
Ketone (FTIR)	0-500 ppm	<+/- 1% FS
In Cylinder Pressure (Piezoelectric)	0-250 bar	- 37pC/bar

The resulting measurement values were deemed suitable for the investigation based on the findings of the uncertainty analysis.

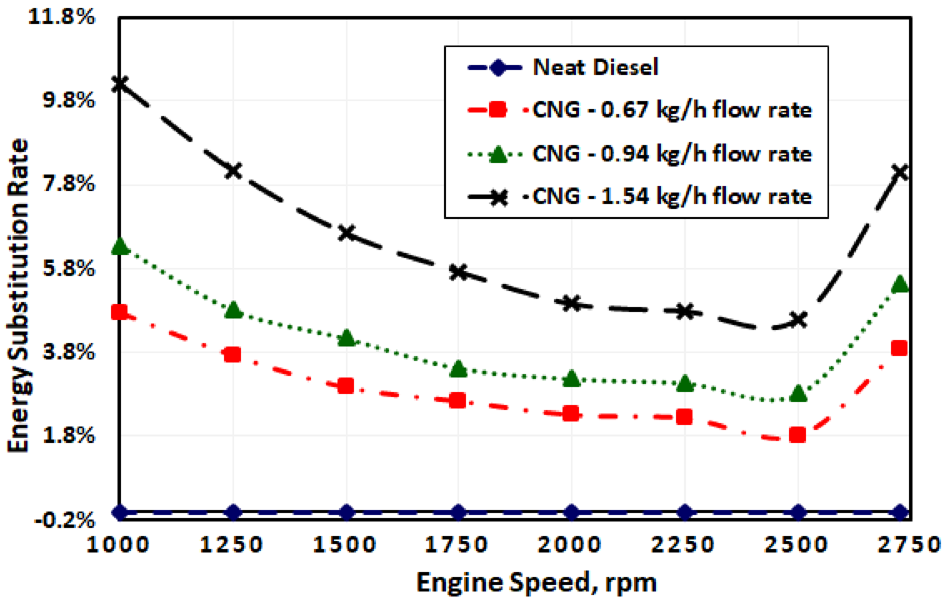
CHAPTER

5. RESEARCH FINDINGS

A commercial heavy-duty diesel engine was modified to run on a combination of diesel and CNG. The engine's performance and emissions were analyzed under different speed and load conditions, as well as Euro 4/ Bharat Stage (BS) -IV emission regulations. The CNG flow rate was optimized using an eddy current engine dynamometer, ranging from 0.65 to 4.0 kg/hr. The engine's ECU regulated the amount of diesel fuel used, while the intake manifold was utilised as the introduction location for CNG at a variable energy substitution rates. The investigation was carried out under four steps namely (i) Full Throttle/Load Condition, (ii) Part Throttle/Varying Load Condition, (iii) European Stationary cycle testing and (iv) Development of DDF Engine.

5.1. PERFORMANCE UNDER FULL LOAD CONDITION

The combustion process of CNG when ignited by diesel fuel is noticeably influenced by the fuel flow. In the course of this investigtion, CNG was introduced at a consistent flow rate. Consequently, the Mass Flow Substitution Rate (MFSR) of CNG, which represents the proportion of mass flow of CNG in relation to diesel and CNG's combined mass flow, was determined based on the stated experimental parameter. Also the MFSR in a DDF engine can be referred as the proportion of CNG that replaces diesel fuel in the combustion process. It indicates the amount of CNG injected into the engine relative to the amount of diesel fuel used. The mass flow substitution rate is typically expressed as a percentage or a ratio. The energy substitution rate (ESR) under a DDF mode of operation refers to the percentage of energy supplied by CNG in relation to the total energy input to the engine. It indicates how much of the engine's power is derived from CNG compared to diesel fuel. The ESR can vary depending on factors such as the engine design, operating conditions, and the specific dual fuel system used. The ESR is typically expressed as a percentage and can be altered in accordance with the needs of the engine and CNG's availability. By adjusting the CNG flow rate, energy substitution rate can be modified to optimize performance and efficiency according to the specific operating conditions. The corresponding equations of ESR and MFSR is given in Appendix. *Fig 5.1* shows the tested fuels' ESR and MFSR values



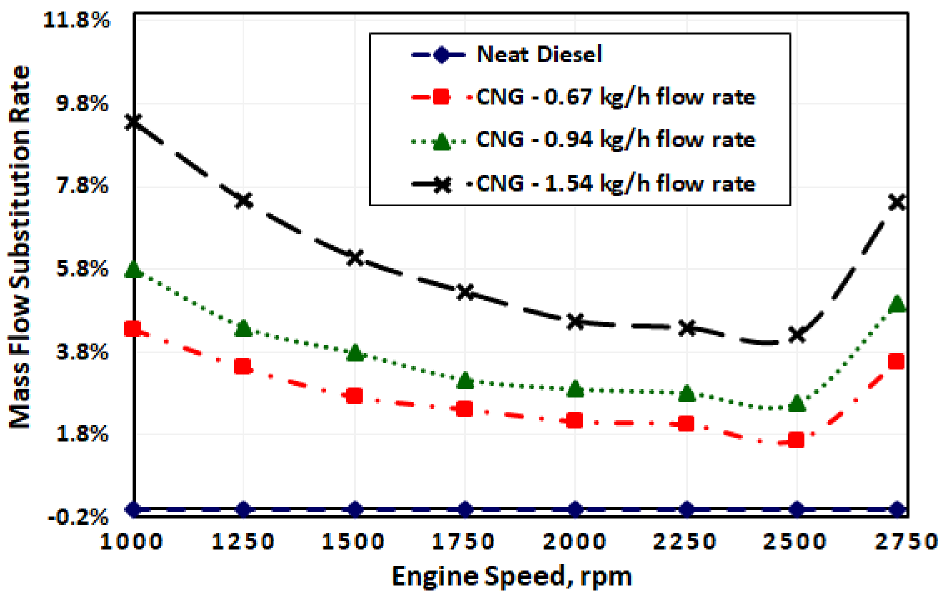


Fig 5.1: ESR and MSR of CNG

ESR and MFSR both decrease as engine speed rises, up to the engine's rated speed of 2500 rpm, and then rise to the engine's maximum speed. The minimal ESR for a steady flow of CNG at 0.67 kg/h is 1.8% when the engine operates at 2500 rpm, while it increases to 4.8% at 1000 rpm. The minimum MFSR is 1.8% for a flow of 0.94 kg/h at 2500 rpm and 4.3% at 1000 rpm. The minimum MFSR is 4.2 % and the highest is 9.4% for a flow rate of 1.54 kg/h, that corresponds to 4.6 and 10.2 % for ESR. Due to the enhanced brake power's ability to exceed engine design restrictions, which resulted in exhaust emissions that were black smoke-colored and engine stalling, the maximum increase in ESR was constrained to 10.2%.

The subsequent sections will present a comprehensive examination of the findings, discussing diverse attributes linked to engine speed and ESR concerning performance and emissions.

5.1.1. Engine Performance Analysis

5.1.1.1. Brake Power and Engine Torque

According to the results of the trial, DDF has a higher engine power than normal diesel fuel. Specifically, when considering the whole range of engine speeds, the highest engine power was observed at 1.54 kg/h CNG flow rate, compared to other DDF operations. The power improved together with the engine speed up to 2725 rpm until it reached 2500 rpm, after which it began to decline. This decrease can be attributed by the rise in friction losses, which become more prominent during exceptionally high speeds , which can be used to explain this drop [49]. *Fig 5.2* provides a visual representation of the experimental findings. With a CNG mass flow rate of 1.54 kg/h, there was an average increase of around 5.2% in the maximum power. Additionally, the lowest ESR was recorded at 1.8% at 2500 rpm, resulting in a 2.9% increase in power. The highest ESR was observed at 10.2% at 1000 rpm, leading to an 8.9% increase in power. Moreover, it is important to mention that CNG has a higher lower heating value (LHV) compared to diesel fuel. Additionally, the combined heating value (CHV) for DDF surpasses that of diesel fuel when the mass flow rate of CNG rises. Consequently, the power output of DDF improves as the combined heating value increases. In other words, substituting more CNG for diesel fuel results in higher power output.

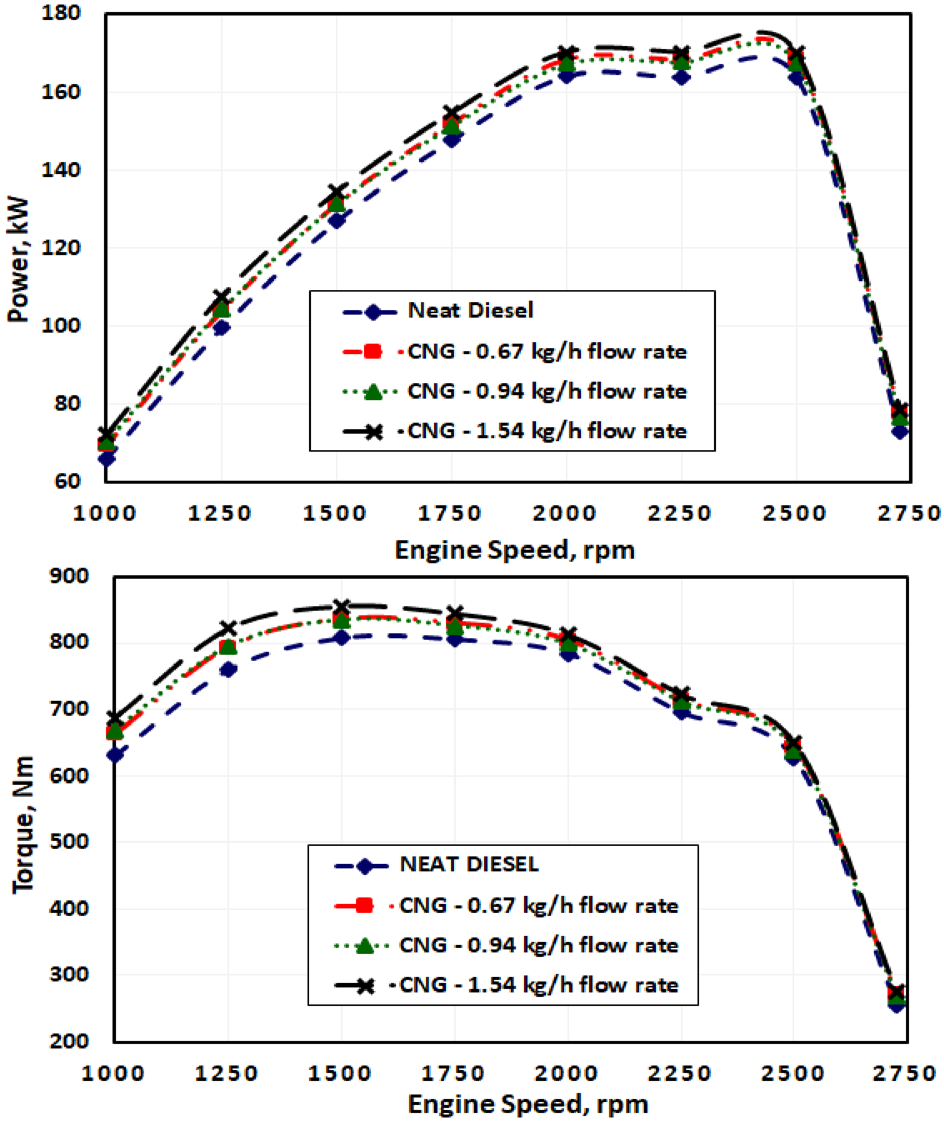


Fig 5.2: Engine Torque and Power under different DDF modes under full load conditions.

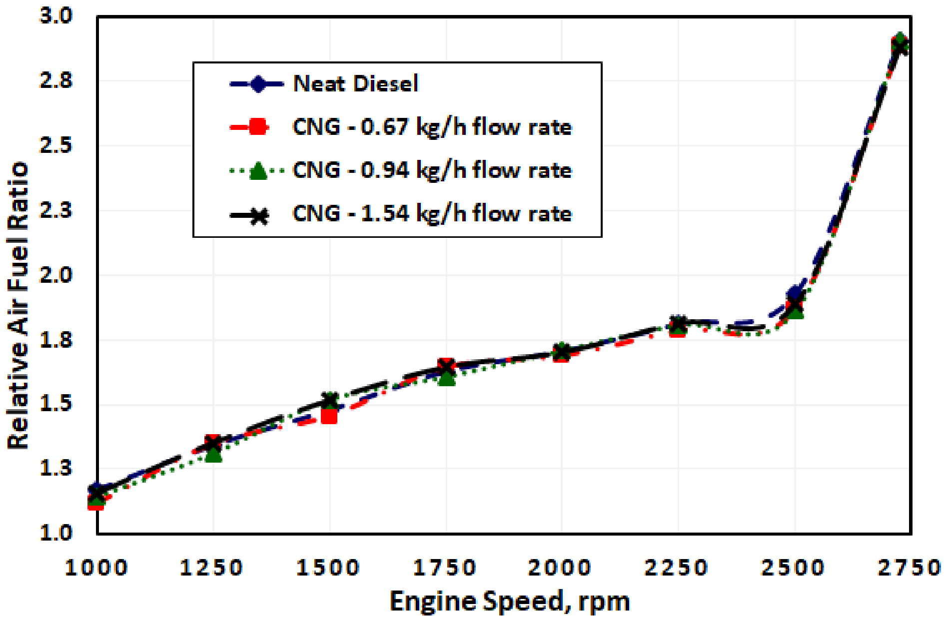
Engine torque at various engine speeds is illustrated in *Fig 5.2*. The DDF operation demonstrates a notable advantage in terms of torque output compared to diesel operation, exhibiting approximately 5.8% more torque when utilizing a 1.54 kg/h flow rate for CNG. Interestingly, torque increases steadily with engine speed until reaching a peak at 1500 RPM, after which it gradually declines. Both diesel and DDF operations exhibit this behaviour and can be attributed to the significant influx of fresh charge at higher speeds, resulting in a larger displacement effect.

In terms of ESR, the highest torque was attained when around 6.5% of the energy was consumed within the 1500–2000 RPM range. This finding establishes a direct correlation between engine speed and the proportion of energy allocation required for optimal performance of engine. Moreover, it was noted that a significant amount of approximately 10% of the energy could be substituted at lower RPMs. However, as the load and RPM increase, this energy share diminishes.

Surprisingly, an increase of 7.5% in torque was observed at 2725 RPM, contrary to the typical preference for lower torque levels. This unexpected result could be attributed to the heightened air-fuel mixture's presence at higher speeds, facilitated by the turbocharger, intercooler, and the engine components' favourable thermal conditions. These factors enhance the combustion process, leading to greater fuel consumption and consequently an increase in torque output.

5.1.1.2. Relative Air-Fuel Ratio (RAFR)

The RAFR, an abbreviation used in the context of a DDF engine, represents the proportion of air in relation to the combined mass of diesel fuel and CNG injected into the engine for the purpose of combustion. It is an important parameter that affects the combustion process and performance of the engine. For effective combustion and top performance while using two fuels, it's critical to maintain the proper relative air-fuel ratio. It has an impact on a number of parameters, including combustion stability, power output, emissions, and fuel efficiency. The specific air-fuel ratio requirements may vary depending on the engine design, operating conditions, and desired performance characteristics. During DDF operation, it is essential to maintain a constant quantity of the air-fuel mixture that is being introduced. The quantity of air being substituted corresponds precisely to the quantity of CNG being used to replace the intake air. While the quantity of diesel is sufficient to handle the load, additional CNG replacement is used to meet the full load requirements at the specified speed. This, in turn, reduces the proportion of intake air [35]. Due to variations in the intake air volume at a particular speed, the ratio of diesel is adjusted accordingly, leading to fluctuations in the RAFR. In this investigation, the maximum MSFR recorded is 9.4%, whereas the lowest RAFR is 1.1%. These values are illustrated in *Fig 5.3* at engine speed of 1000 rpm and 0.67 kg/h CNG flow rate.



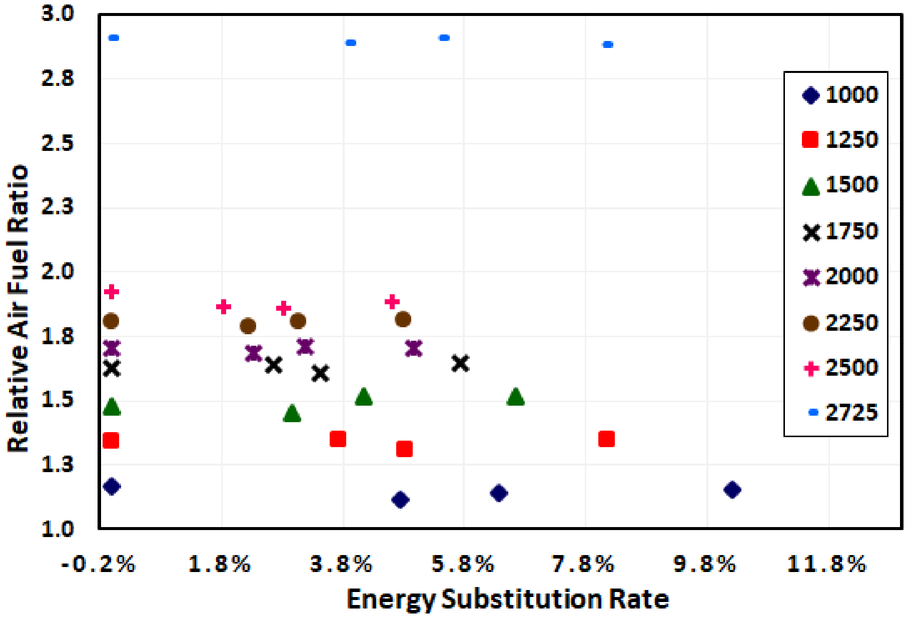


Fig 5.3: Relative Air Fuel Ratio Effect on Torque and Power for different DDF modes at Full Load.

The selected engine features lean combustion and an RAFR of 1-3. Typically, as the speed of engine increases, RAFR increases as well; however, under full load, as the CNG substitution rates increase, RAFR decreases in all speed ranges compared to neat diesel, as observed in *Fig 5.3.* It was to be expected since the quantity of diesel required to produce the specified load at a particular speed is constant; when additional CNG is replaced, the RAFR drops even while the power output increases. Furthermore, for all energy shares, RAFR increases with increasing speed, and when compared to RAFR, the DDF power curve improves with the percentage of energy substitution. The utilization of CNG as a substitute in the lower RAFR has resulted in a substantial boost in torque and power of up to 1.85. However, when the RAFR is further increased, the power and torque experience a decrease. Additionally, the data presented in the figures indicate that at low RAFR values, the DDF engine generates more power in comparison to diesel engine. Conversely, at high RAFR values, the DDF engine produces less power and torque than the pure diesel engine.

5.1.1.3. Exhaust Temperature

The quality of the combustion is gauged by the exhaust temperature and is critical for after-treatment device performance, efficiency, and durability. A variable geometry turbocharger is integrated into the engine, delivering varied delta temperatures across the turbocharger. Part of the heat energy in the exhaust gas is dispersed by the compressor wheel (of the turbocharger), which is driven by the pressure and heat energy in the exhaust gas. As a result, it is commonly observed that the turbine outlet experiences a decline in maximum temperature of the exhaust gas of approximately 300-400 degrees Celsius compared to the turbine inlet [50]. *Fig 5.4* shows the impact of exhaust temperature at different speeds and CNG substitution rates. It was observed that, regardless of the fuel used, the exhaust temperature declines continuously as engine speed increases. This tendency of lower exhaust gas temperatures with higher speed is interesting as temperature measurements are made after the turbocharger. In comparison to diesel, the temperature of exhaust gas has increased slightly as CNG replacement has increased in DDF mode. This phenomenon occurs due to the higher proportion of CNG inside the cylinder as opposed to using diesel. As the amount of CNG increases, the amount of air inside the combustion chamber is decreasing. This, in turn, enhances the quality of CNG combustion, leading to a decreased combustion duration [35]. The continuous combusting of the mixture throughout the expansion and exhaust strokes may also be the cause of increased exhaust temperatures in diesel CNG dual fuel engines operating at lower speeds [51]. Efficient after-treatment devices necessitate a high exhaust gas temperature. Diesel and DDF engines have exhaust temperatures that are more than 500°C up to 2500 rpm, after which they drop to 375°C.

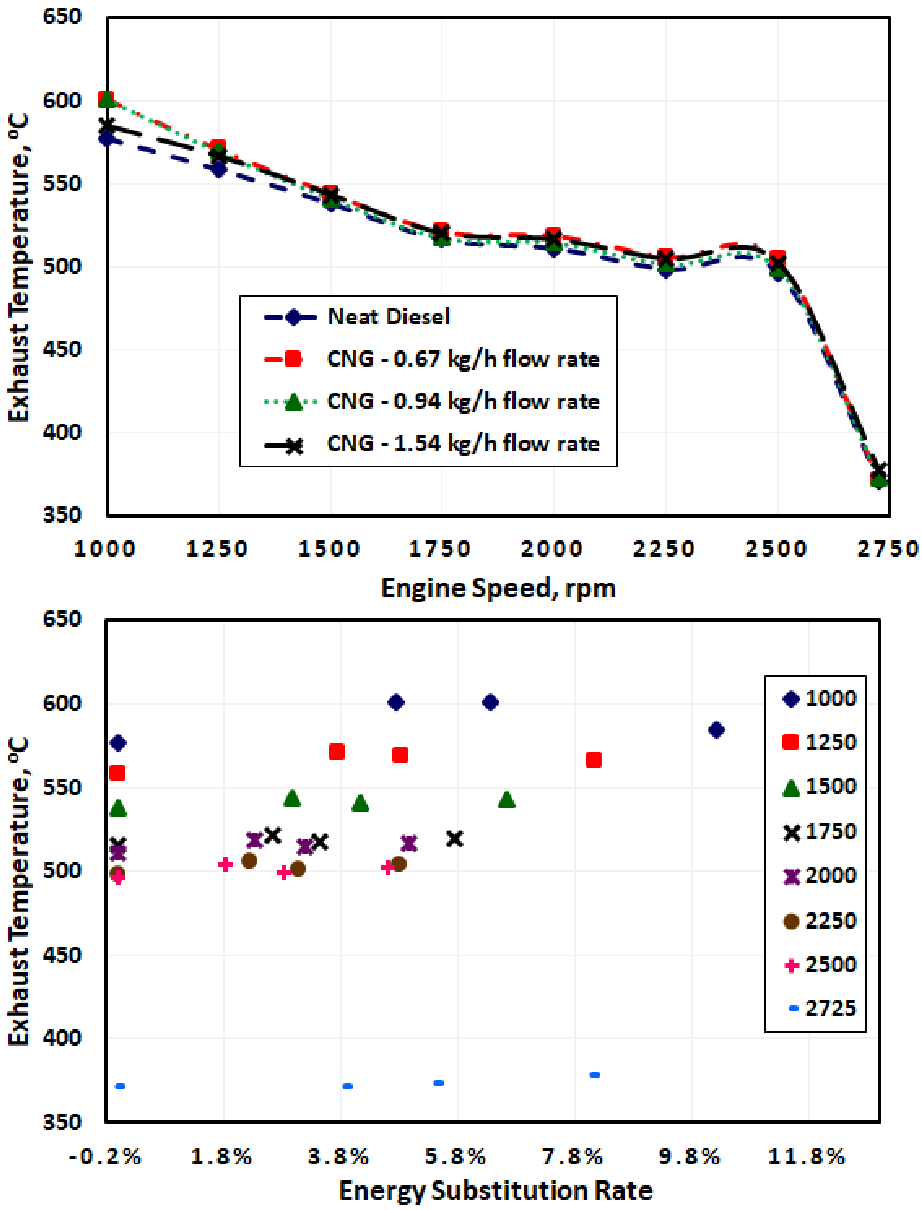


Fig 5.4: Effect of Exhaust Temperature under different DDF modes under full load conditions.

5.1.1.4. Brake Thermal Efficiency

Based on engine speed and the CNG energy substitution rate, the fluctuation of brake thermal efficiency (BTE) is shown in *Fig 5.5*. The conversion efficiency of input energy to useable output energy is measured by BTE [52]. It is evident that BTE decreases with speed, regardless of fuel and energy sharing, reaching its maximum at 1500–1750 rpm. In the rated torque speed range, there is a higher BTE. Compared to the fuel's (burned or unburned) chemical potential energy, thermal efficiency measures how effectively the engine uses the thermal energy that is actually contributed to the cylinder [53]. It can also be seen that when the ESR rises, the BTE rises as well. However, this is only true for the specified torque speed range; BTE is almost constant at all other speeds. The increased BTE plays a crucial role in enabling the engine to effectively convert the potential energy derived from improved energy substitution [44]. Premixed combustion occurs more frequently when energy substitution increases, hence BTE conversion effectiveness is also improving. A flow rate of 0.94 kg/h resulted in a maximum BTE of 43% in the energy share of 6-7%.

5.1.1.5. Volumetric Efficiency

When CNG is introduced into a diesel engine's intake manifold, the volumetric efficiency (VE), which compares the actual volume of air + CNG to the geometric volume, is used to determine how well the engine can operate. The VE of an engine can be influenced by various factors, including the pressure and temperature of the air entering the engine, the engine speed, the fuel-to-air ratio, the composition of the fuel, the humidity of the surroundings, and additional characteristics such as the temperature of the oil, the rate of coolant flow, and more [54]. The VE of different DDFs in relation to engine speed and ESR is depicted in *Fig 5.6*. In comparison to diesel and CNG flow rates, an inverse relationship between engine speed and VE was observed. In DDF mode, the air intake is partially replaced by a specific amount of CNG, which depends on the ESR, leading to a decrease in VE.

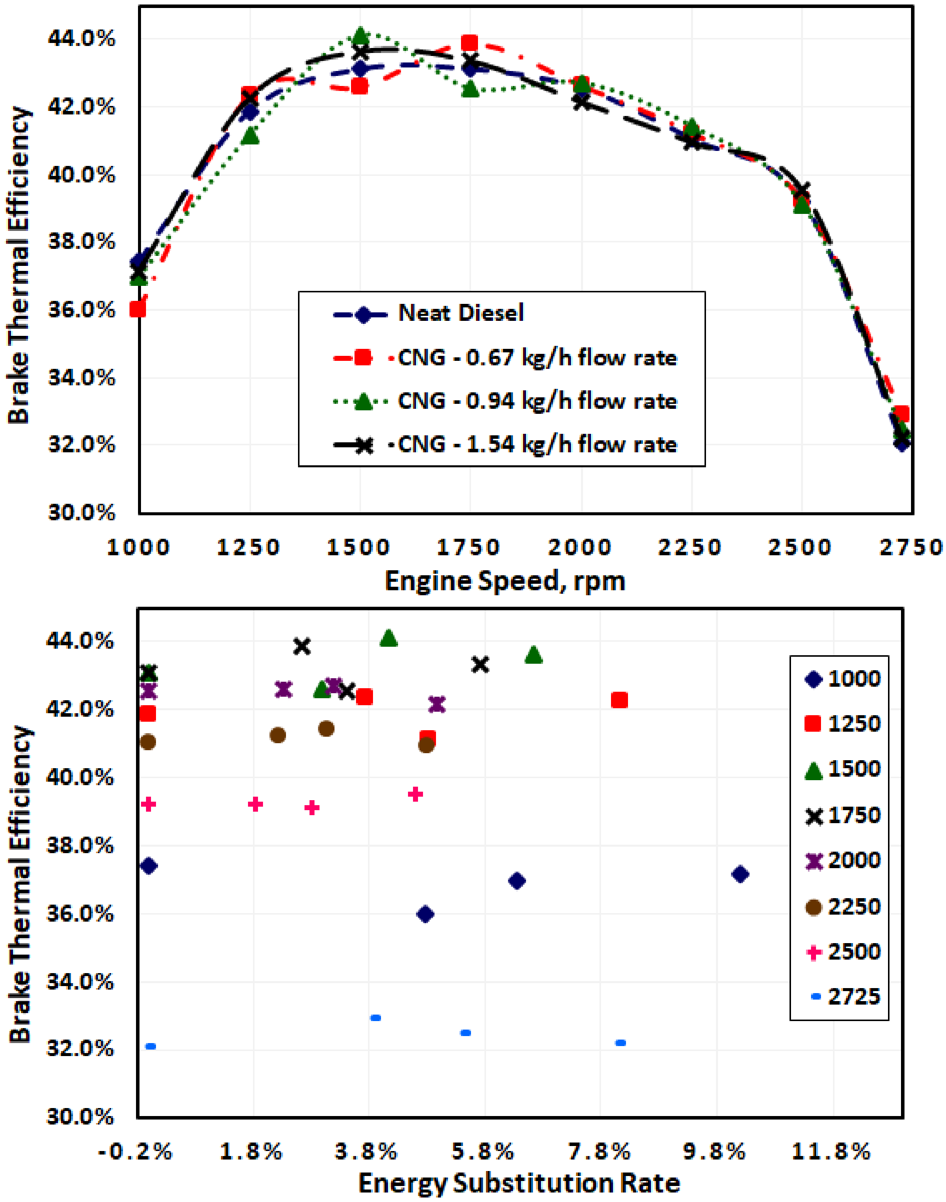
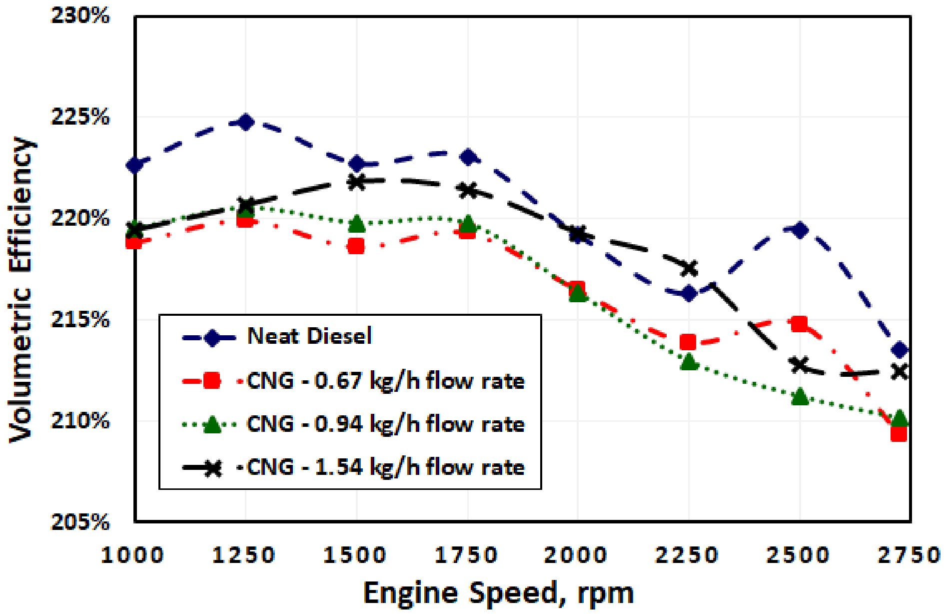


Fig 5.5: Brake Thermal Efficiency of different DDF modes under full load conditions.

Diesel operation resulted in a higher VE, followed by a 1.54 kg/h CNG flow rate, while DDF operation showed a lower VE. The relationship between higher VE and higher ESR in different DDF modes was not well understood. When evaluating various engine speeds, it was found that the highest VE was attained at 1250 rpm across all test fuels. Conversely, the lowest VE was registered at 2725 rpm when comparing the VE of different DDF modes. Generally, the VE decreases at lower speeds and increases as the engine speed rises, reaching its peak at 1250 rpm before declining. Since CNG has a lower density and specific gravity in its gaseous state compared to diesel, for the same amount of engine power, an extensive amount of CNG is required. When CNG is introduced into the intake manifold during DDF, both air and CNG must be compressed by the engine, resulting in reduced volumetric efficiency and increased compression work. Air displacement and the compression effect are accountable for the reduced volumetric efficiency.



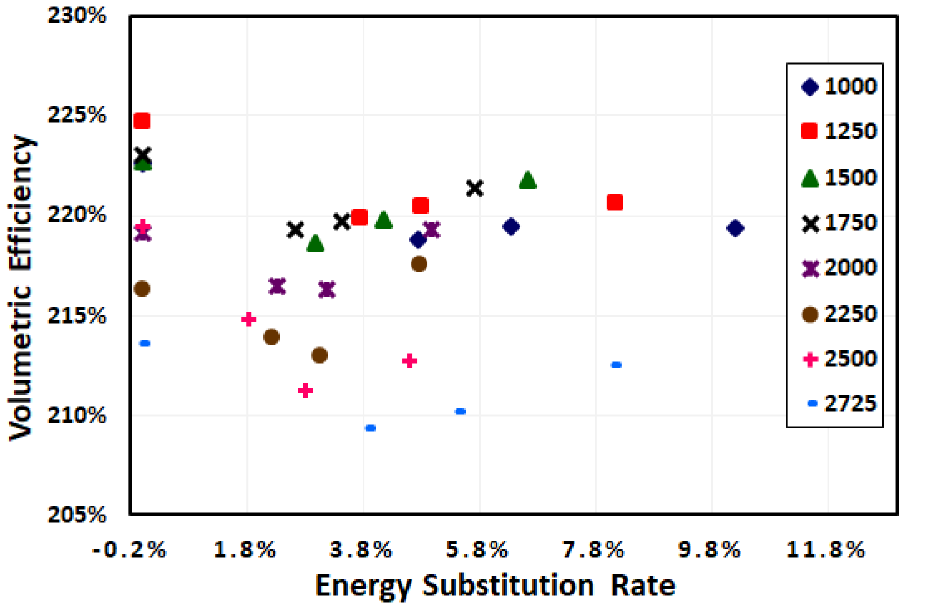


Fig 5.6: Volumetric Efficiency at different speeds under full load conditions.

5.1.1.6. Brake Specific Energy/Fuel Consumption

The measurement of an engine's fuel efficiency is determined by its brake-specific fuel consumption (BSFC), which quantifies the amount of fuel consumed to generate power. The effectiveness in terms of fuel efficiency is determined by the calorific value of individual fuel when an engine runs on a dual fuel system or combines various fuels. It is more advantageous to represent efficiency using brake-specific energy consumption (BSEC) rather than BSFC if there are changes in the density and calorific value of the fuels.

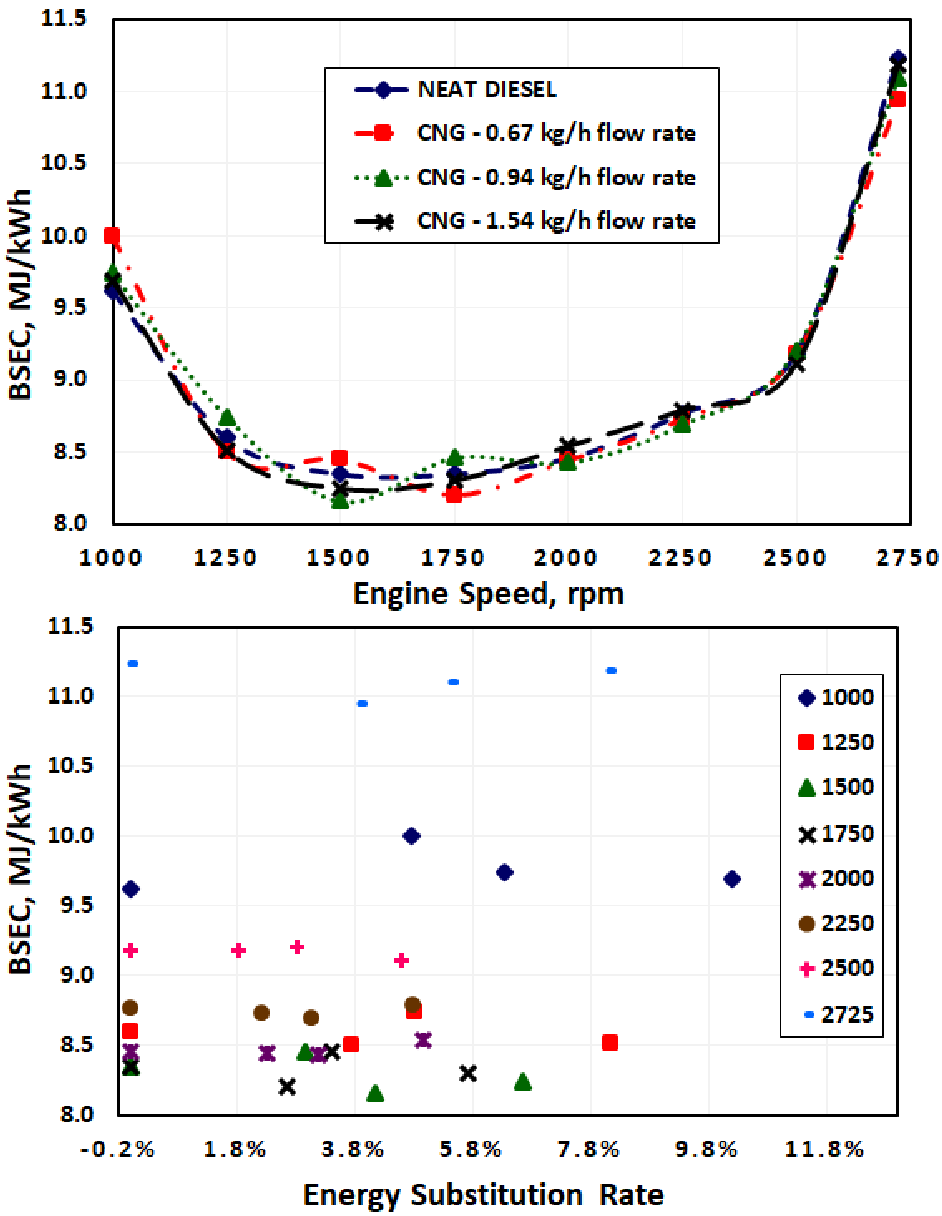


Fig 5.7: Effect of BSEC for varying DDF modes under full load conditions.

The BSEC and BSFC at various speeds as a function of various modes of DDF operation is displayed in *Fig 5.7* & *Fig 5.8*. At lower speeds, both BSEC and BSFC exhibit higher values. However, as the engine speed increases, these values decrease until reaching a range of 1500-1750 rpm. Beyond this range, they start increasing with speed, and this trend remains consistent for all fuels. Moreover, when the ESR is increased, there is a notable reduction of up to 1% in BSFC for a 1.54 kg/hr CNG flow rate. Conversely, the decrease in BSEC is marginal, amounting to 0.2%. The lower BSEC/BSFC can be attributed to the higher LHV of CNG in comparison to diesel. Consequently, the engine requires a smaller amount of fuel to generate an equivalent output when operating in DDF mode. Moreover, research has indicated that the most favorable engine speed for dual fuel operation, leading to enhanced BSEC/BSFC, lies within the RPM range of 1250 to 2250, with an energy distribution ranging from 4 to 8 percent. Therefore, when designing an engine for enhanced fuel efficiency during DDF driving mode, the engine designer should take into consideration the BSFC values within these speed ranges.

5.1.2. Mass Emission Performance

Before and after the SCR, emissions were monitored for all regulated and unregulated emissions. Selective catalytic reduction, sometimes known as SCR, is a well-known technique for lowering NOx emissions. NOx emissions from diesel engines are lowered by injecting a reducant, such as ammonia solution into the exhaust stream, commonly referred to as DEF (diesel exhaust fluid). The ECU regulates the amount of diesel exhaust fluid injected. A NOx sensor, which is mounted in the exhaust pipe after the SCR unit, controls how much DEF is injected. According to the emission study, to meet Euro IV emission norms, DEF is only injected when the NOx value is higher than 900 ppm. The engine's emission performance before and after SCR under different DDF modes is discussed in the next section.

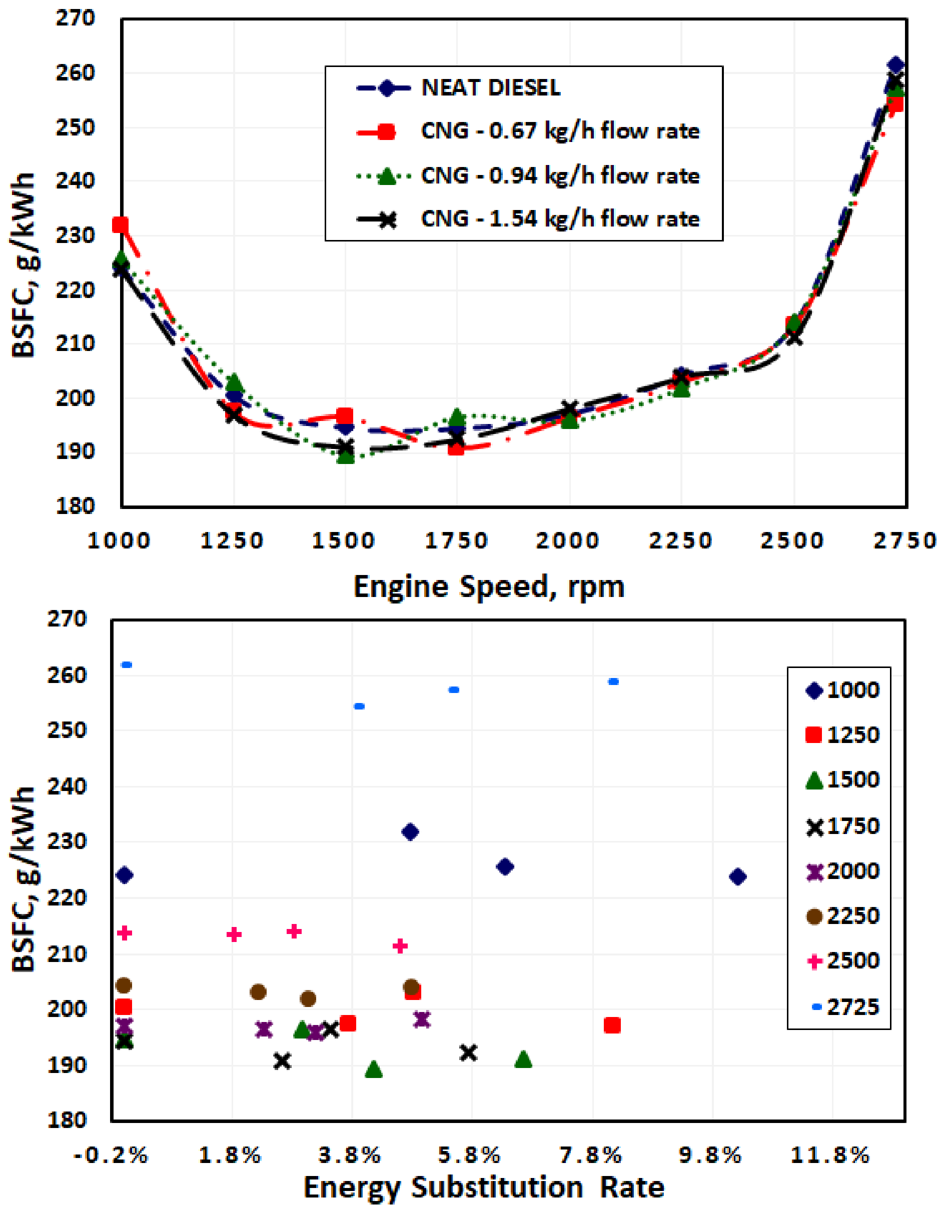


Fig 5.8: Effect of BSFC for varying DDF modes under full load conditions.

5.1.2.1. CO Emission

The engine's fuel-to-air equivalency ratio controls CO emissions in substantial part and is the result of incomplete combustion, which occurs when the oxidation process is not completed [10] [55]. The oxidation of CO emissions to CO_2 has only been documented in a few cases for diesel engines with exhaust gas after-treatment devices, and it is determined to be less than 5% [56]. The emission levels of the study were assessed in ppm (parts per million), and the empirical equation below was used to create mass emission values from the raw emission data:

$$\dot{m}_{e,i} = \frac{x_i \times MW_i \times \dot{m}_{ex} \times CF \times 3600}{MW_{ex} \times BP} \quad \text{Eqn. (6.1)}$$

Where, $\dot{m}_{e,i}$ (kg/kWh) = specific mass emission of species i (= CO, HC, NOx, CO_2, CH_4, N_2O, etc.), MW_i = molecular weight of emission i, x_i= volume fraction of emission i (i = %/100 (or) ppm X 10-6), MW_{ex} = exhaust gas' molecular weight, $\dot{m}_{ex}$ = mass flow rate of exhaust gas, and CF = correction factor that is affected by the temperature, pressure, and relative humidity of the ambient air.

During low-speed operation, the concentration of CO is significantly elevated. However, as the engine speed gradually rises, the CO level gradually decreases until it reaches 1500 rpm. Beyond this point, the CO concentration remains relatively constant until reaching 2500 rpm. Subsequently, there is a slight incremental increase in CO levels, as illustrated in *Fig 5.9* & *Fig 5.10*. Measurements of pre- and post-SCR CO demonstrate the same pattern. While the CO limit under Euro 4/6 is 1.5 g/kWh, this does not apply to full load operation. At all engine speeds, the CO with DDF is less than the legislation limit and less than neat diesel which is due to CNG fuel's lower carbon content.

Except for low and maximum speeds, the CO emissions have reduced at all engine speeds compared to neat diesel [57]. This is supported by BSFC and air/fuel ratio data, which show that for both low and high speed DDF modes, the values are higher, signifying incomplete combustion and higher CO. Prior to selective SCR treatment, during moderate engine speeds, it was observed that the levels of CO were comparatively lower as opposed to conventional diesel fuel. The minimum recorded value was 0.62 g/kWh, which was achieved when utilizing a 0.94 kg/h CNG flow rate at 2000 rpm [58].

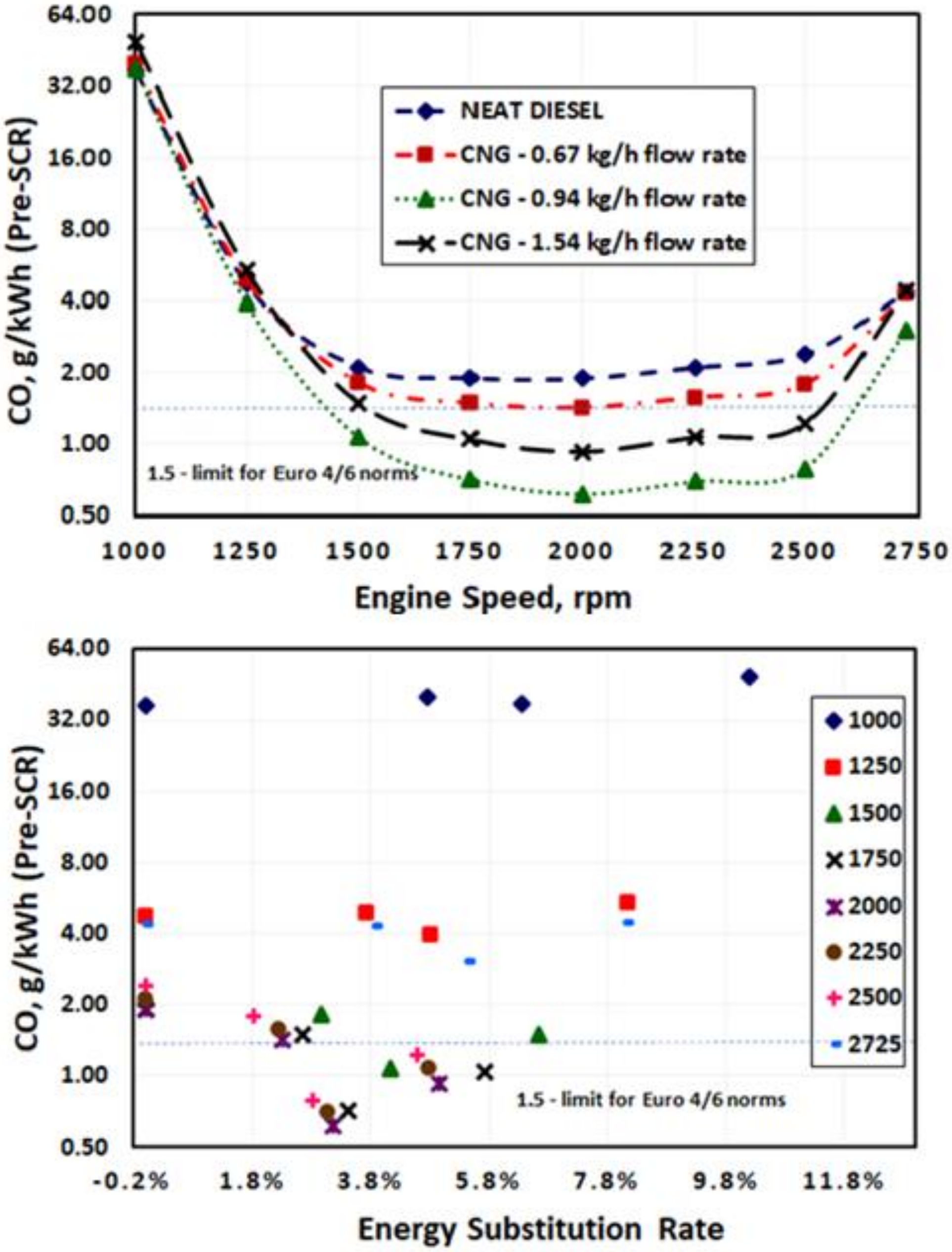


Fig 5.9: CO Emission at Pre-SCR under full load conditions.

As mentioned earlier, the SCR system has the potential to achieve a maximum reduction of 5% in CO emissions. However, in the present study, the average CO reduction achieved for a 0.94 kg/h CNG flow rate was 2.5%. Interestingly, when looking at individual CO results at various engine speeds, it was observed at intermediate speeds that the SCR system led to an increase in CO values. The rise in CO could be caused by the soluble organic component and hydrocarbon partially oxidising across the SCR catalyst [59]. While the introduction of DDF resulted in a decrease in the average CO value, it also resulted in a decline in SCR conversion efficiency because of an increase in ESR [60]. Although the SCR conversion efficiency was lower, the CO levels at intermediate speed range of 1500-2500 rpm remained below Euro 4/6 norms for CNG flow rates of 0.94 and 1.54 kg/h. For commercial applications, the ideal CNG flow rate is determined by the ESR, which falls within the range of 2.6% to 6.8% at these high speeds. This specific range, 2.6-6.8%, proves to be the most suitable while developing DDF.

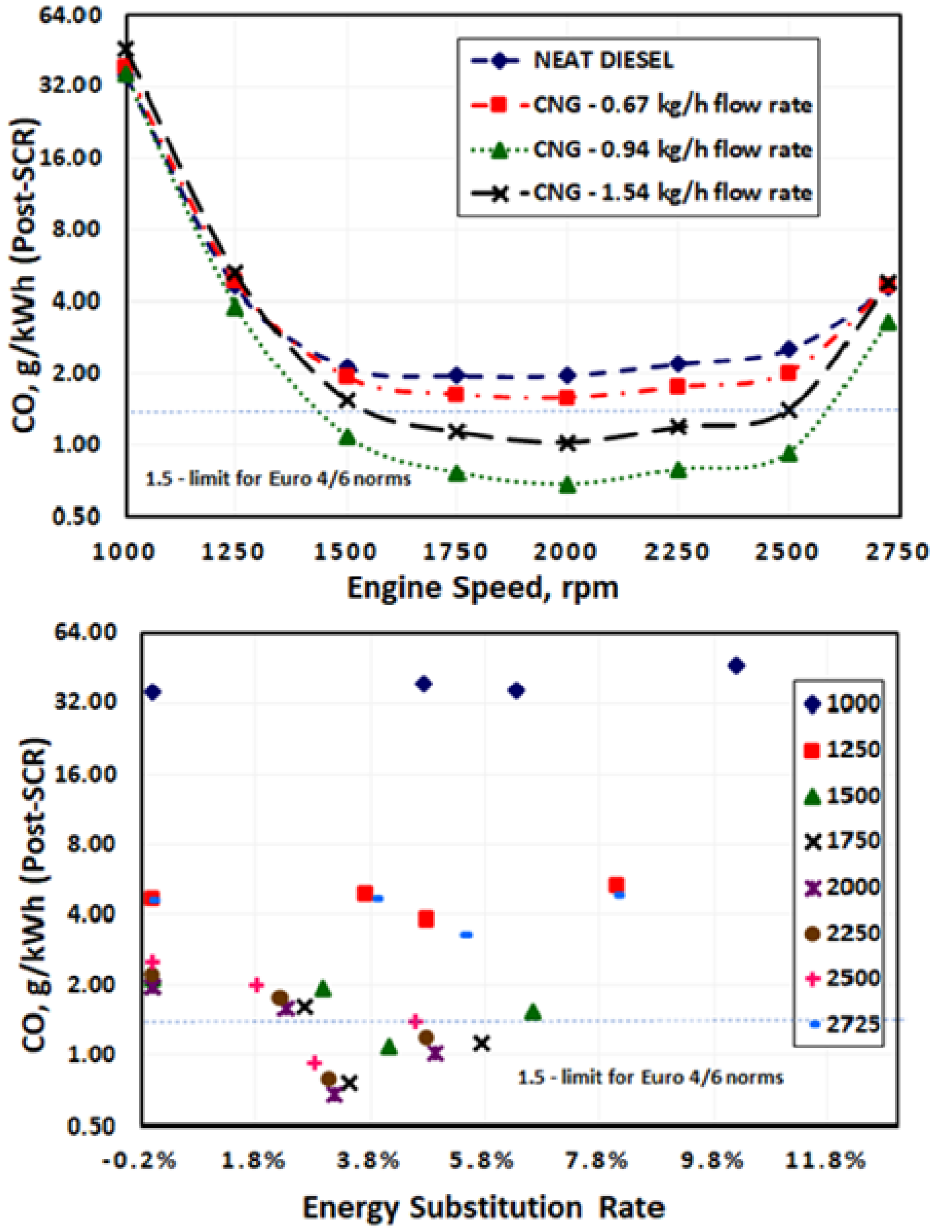


Fig 5.10: CO Emission at Post-SCR under full load conditions.

5.1.2.2. THC Emission

Diesel fuel's incomplete combustion produces hydrocarbon emissions, which are referred to as Total Hydrocarbon (THC) and contain both non-methane HC and methane. In a diesel engine, hydrocarbon emissions are largely produced by two sources [61]. The extended delay period results in an increased production of HC, as the composition within the reaction zone is excessively lean to ignite in the initial source. Some of the fuel is trapped between the nozzle holes and seat, which is the second source and also produces HC. Furthermore, the relevance and different compositions of HC are influenced by the fuel composition [10]. If the olefin and aromatic concentration is higher, the amount of reactive hydrocarbons in the exhaust is likely to be higher.

Methane emissions outnumber THC emissions by a factor of 52-87, but because it is non-toxic, for diesel engines, legislation controls it as THC or non-methane hydrocarbon (NMHC) [61]. The THC emissions were expected to comply with Euro 4 standards when utilizing regular diesel during the European Stationary Cycle (ESC). However, our examination revealed that even before passing through the SCR system (as shown in *Fig 5.11* & *Fig 5.12*), the THC levels were already below the Euro 4 limits. The THC levels were found to be below the Euro 6 standard at specific engine speeds. Furthermore, when assessed after the SCR system, the measurements consistently demonstrated a remarkable decrease in emissions compared to the Euro 6 standards, indicating the SCR system's higher efficiency in converting diesel fuel.

THC levels have been found to increase with a relatively higher proportion of CNG in both pre and post-SCR measurements [62]. Furthermore, when the engine operates at full load condition, THC levels for all tested fuels rise with increasing engine speeds. Three primary processes explain the increased THC under DDF modes: (i) quenching of the flame, (ii) the fuel mixture may get trapped in the cylinder's crevices during the compression stroke, and (iii) increased air-to-fuel ratio [63]. In DDF modes, the CNG-air mixture is already present during the compression stroke. Because of this, the CNG that has been trapped in the combustion chamber's crevices participates in later phases of combustion, which raises the THC levels. Although it was anticipated that THC would be lowered in DDF modes, similar to how it is reduced in SCR compared to normal diesel, the conversion effectiveness of SCR declines as the percentage of CNG replacement increases.

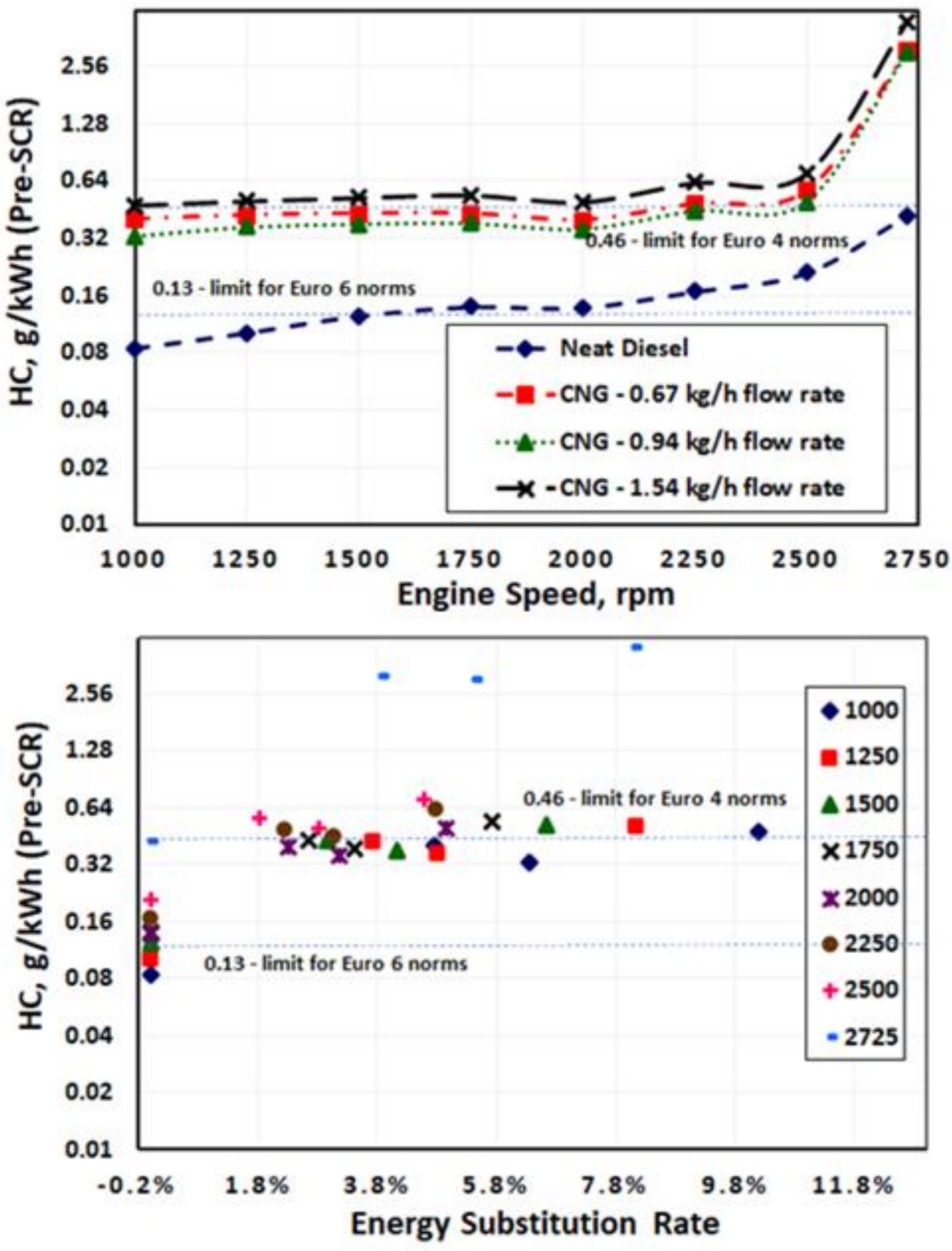


Fig 5.11: Hydrocarbon Emission at Pre-SCR under full load conditions.

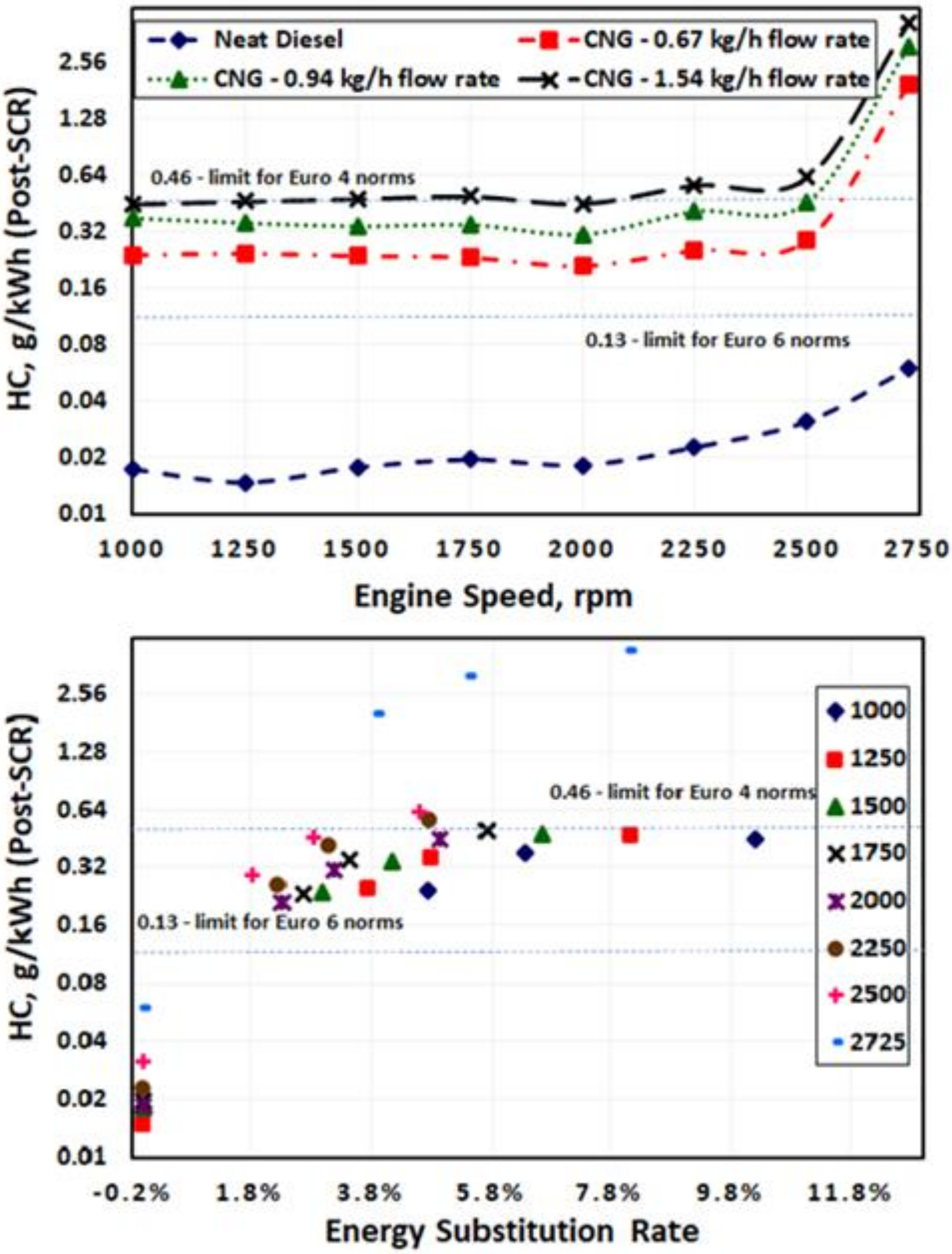


Fig 5.12: Hydrocarbon Emission at Post-SCR under full load conditions.

THC conversion with conventional diesel was 86%, while conversion with 0.67, 0.94, and 1.54 kg/h CNG flow rates was 48%, 12%, and 10%, respectively.As previously noted, the production of unburned hydrocarbon (UBHC) is affected due to the composition of the fuel, which in turn leads to a prolonged ignition delay. The SCR catalyst would have been primarily intended for diesel operation, however under DDF mode, it is a mixutre of both CNG and diesel [64]. In diesel engines, THC levels are generally low compared to CO.

However, as the ESR increases, THC levels can exceed those of diesel engines. Even so, these elevated THC levels remain within the limits set by Euro 4 regulations, ranging from 0.17 to 0.64 g/kWh [65]. In addition to the reduction process that SCR is designed for, an appropriate modification in the catalyst coating maybe beneficial in oxidising unburned hydrocarbons [66].

5.1.2.3. NOx Emission

One of the principal pollutants responsible for climate change and air pollution, NOx emissions constitute a significant factor. Rich fuel-air ratios in diesel engines produce CO2, soot, smoke, and unburned hydrocarbon emissions, whereas at high temperatures, low air fuel ratios emit higher NOx [67]. In the combustion chamber, NOx is primarily created in three ways: thermal NOx, fuel NOx, and prompt NOx. These mechanisms rely on factors such as residence time, temperature, turbulence and excess oxygen [10]. Controlling the aforementioned parameters in the in-cylinder combustion chamber possibly affects particles, HC, CO, and other emission elements like CO_2. Among the numerous control techniques available, after-treatment devices such as lean NOx trap, SCR, and various others have demonstrated their effectiveness in reducing nitrogen oxide (NOx) emissions. As a result, these devices have gained significant popularity and are extensively utilized in light-duty vehicles, heavy-duty vehicles, as well as passenger vehicles. DEF is an aqueous solution made up of 67.5% purified water and 32.5 urea, serves as a highly effective reducing agent in SCR systems, especially at exhaust temperatures between 350 and 450^{o}C. Below 200°C, SCR conversion efficiency is diminished, leading to ammonia deposition on the SCR system. Conversely, at temperatures exceeding 600°C, before the ammonia reaches the catalyst-coated SCR system, it is burned. The reduction of NOx emissions is effectively achieved through the occurrence of the following reaction within the specified operating temperature range:

$$(NH_2)_2CO \rightarrow NH_3 + HNCO \text{ (thermolysis) Eqn. (6.2)}$$

$$HNCO + H_2O \rightarrow NH_3 + CO_2 \text{ (hydrolysis) Eqn. (6.3)}$$

After the above process, the main reaction is

$$2NO + 2NO_2 + 4NH_3 \rightarrow 4N_2 + 6H_2O \quad \text{Eqn. (6.4)}$$

$$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O \quad \text{Eqn. (6.5)}$$

$$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O \quad \text{Eqn. (6.6)}$$

Despite equation (4) exhibiting the highest conversion efficiency and being ideally combined with DOC, this research exclusively focuses on the implementation of the SCR method. The effects of NOx emissions under full load condition is shown in *Fig 5.13* & *Fig 5.14*.

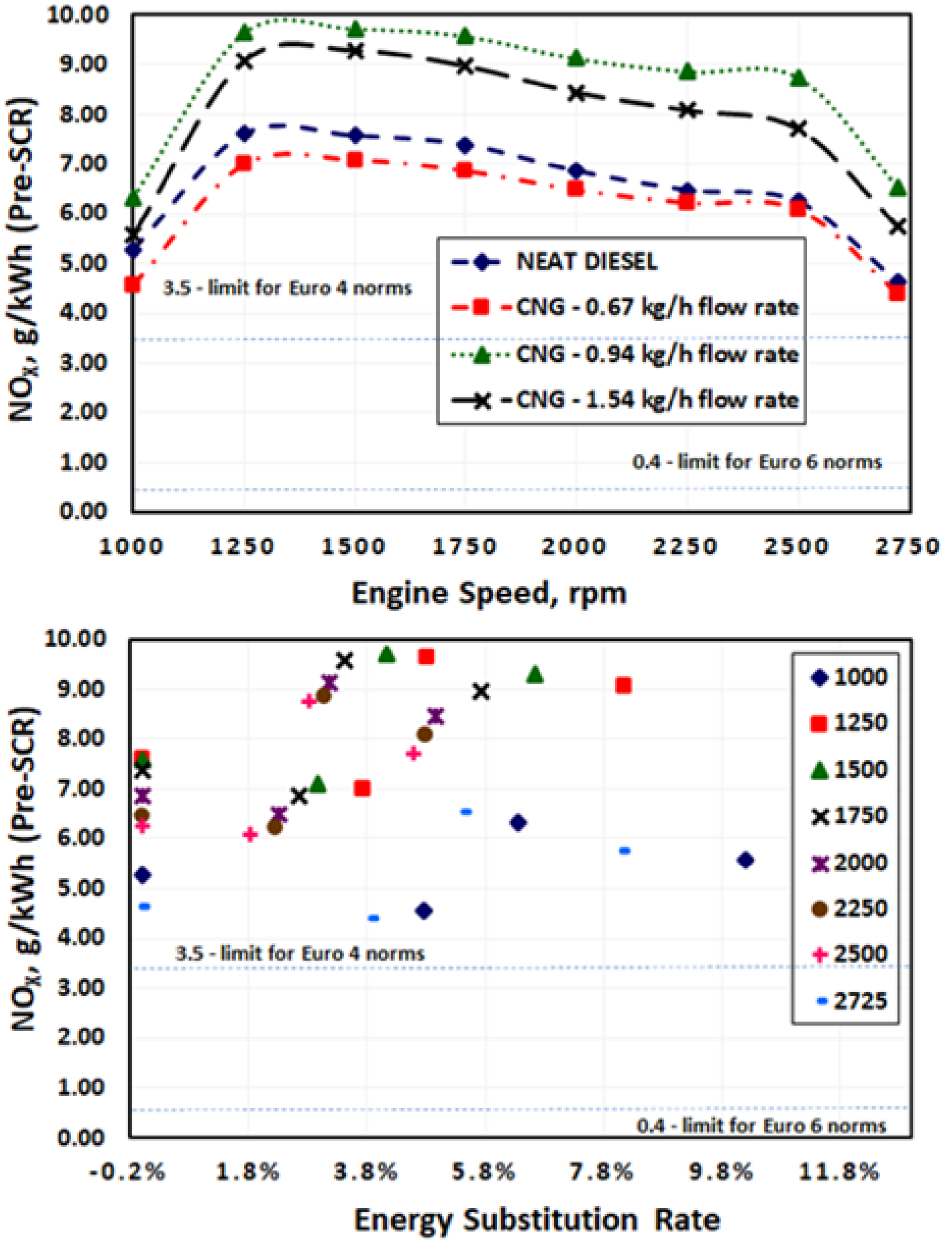


Fig 5.13: NOx Emission at Pre-SCR under full load conditions.

Contrary to previous research, it was observed that the NOx emissions increased even when operating under DDF mode at a lower CNG flow rate of 0.67 kg/h [40] [68] [63] [69]. While evaluating the impact of different CNG flow rates on NOx levels upstream of the SCR system, it was observed that the NOx levels were higher when a CNG flow rate of 0.94 kg/h was utilized. However, the NOx levels decreased significantly when a higher CNG flow rate of 1.54 kg/h was employed. However, even with higher flow rate, NOx levels remained higher compared to those of conventional diesel.

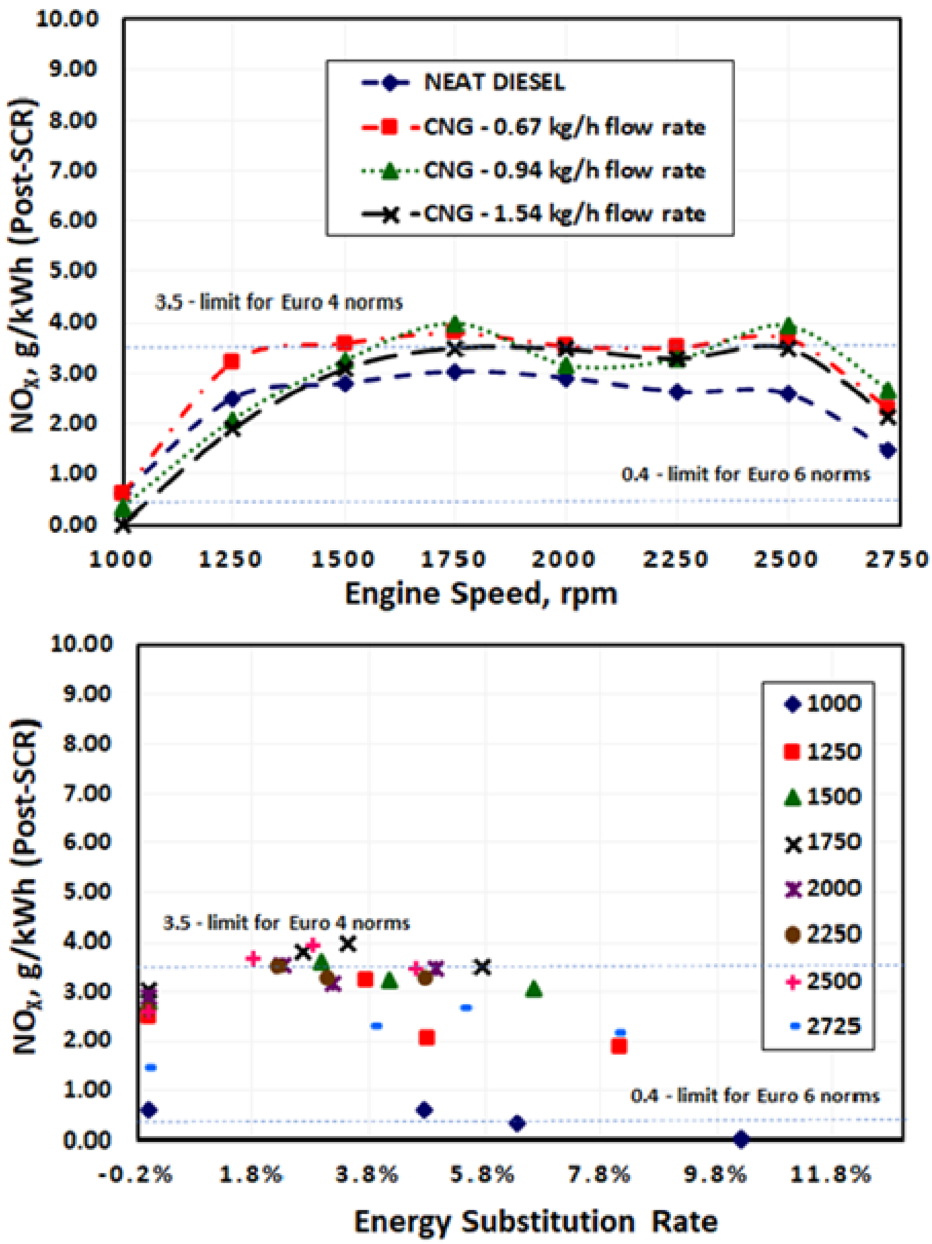


Fig 5.14: NOx Emission at Post SCR under full load conditions.

This study could not establish a definitive trend of NOx decrease or increase in the DDF mode. Increased heat release from CNG combustion results in higher combustion temperatures, which raises NOx levels, this is evident from the pattern in exhaust gas temperature. [70] [71]. While NOx levels have been found to be lower at moderate speeds, such as 1000 rpm, they rapidly increased at 1250 rpm and then began to decline. Before SCR, all of the fuel mixtures that were evaluated exhibited this tendency of lowering NOx emission at higher engine speeds [72]. The shorter working cycle has led to this phenomenon, as it results in a decreased combustion process.

As a result, this trend is influenced by a decrease in the charge mixture's residence duration at high temperatures in the combustion chamber. The residency time of the charge mixture at elevated combustion temperatures, one of the mechanisms for the production of NOx, is shortened as engine speed increases, resulting in a decrease in emissions [73]. The full load conditions as shown in *Fig 5.14* illustrates that all of the fuels tested had NOx levels within the Euro 4 standard of 3.5 g/kWh.

The mean NOx levels ranged from 2.31 to 3.04 g/kWh. Specifically, the normal diesel fuel yielded a NOx level of 2.31 g/kWh, while a NOx level of 3.04 g/kWh was produced when 0.67 kg/h of CNG flow was used under DDF mode. Despite conducting SCR measurements, the NOx levels in the DDF mode remained higher than those observed in regular diesel. The conversion efficiency of NOx SCR ranged from 50% to 67%, and it was observed to be influenced by the fuel type. The optimal efficiency was attained when the flow rate of CNG was adjusted to 0.94 kg/h. However, it should be noted that there was no consistent pattern, as the DEF flow rate was adjusted to regulate NOx emissions under the SCR system.

5.1.2.4. CH_4 Emission

Although the research on THC and NMHC has been extensively documented, the effects of methane emissions, commonly referred to as methane slip, remain poorly understood within the context of DDF mode. This section, which is represented in *Fig 5.15* & *Fig 5.16*, covers investigations into methane emissions (CH_4), which is a highly prospective GHG that may have a detrimental influence on after-treatment devices. As measured before the SCR system, CH_4 levels in DDF modes are higher than in conventional diesel, which is essential given the importance of this topic [74].

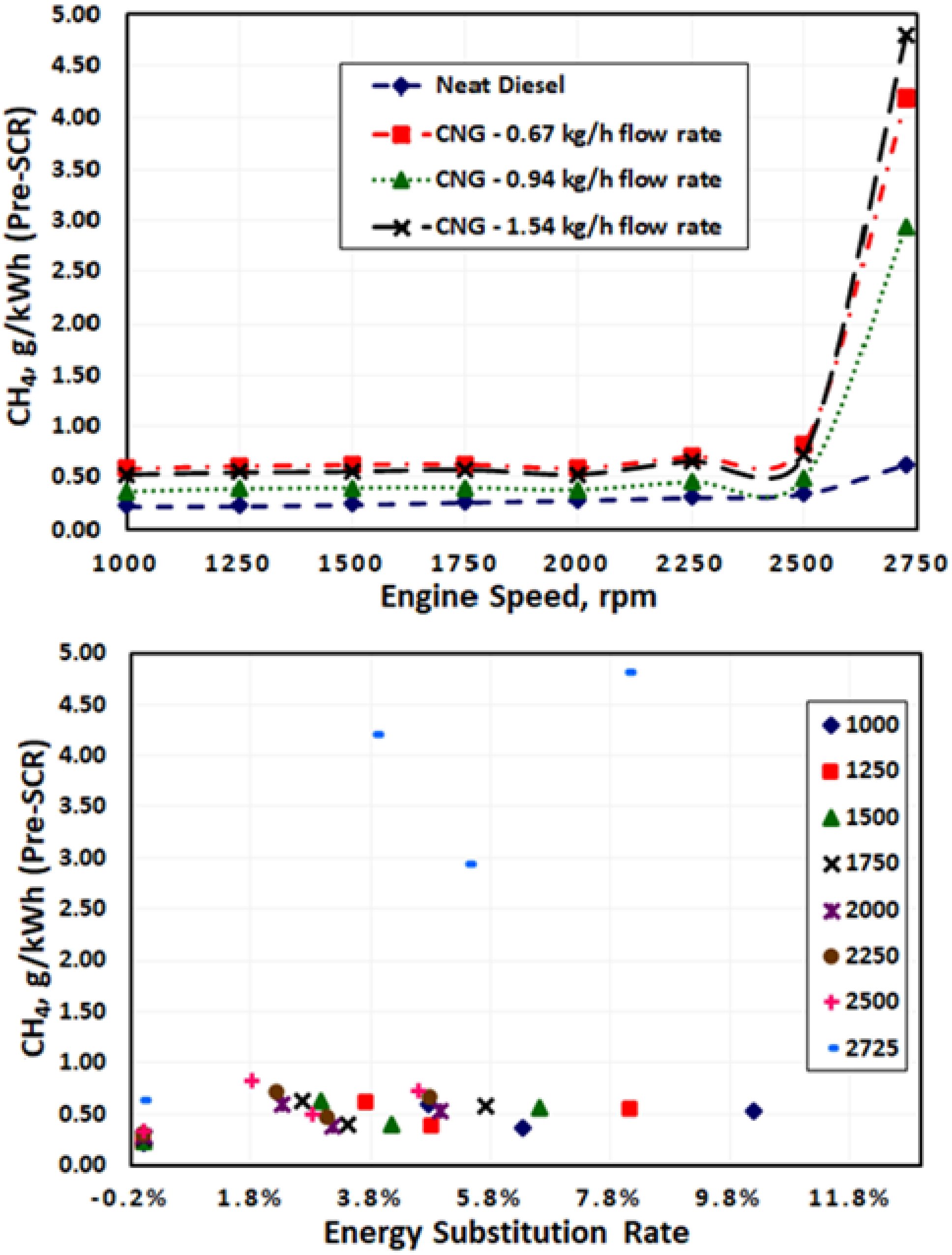


Fig 5.15: Methane Emissions at Pre SCR under full load conditions.

Despite the fact that there is no correlation between increased CNG substitution rates and increased CH_4, all energy substitution using CNG within the DDF system result in higher levels of CH_4. At all engine speeds, CH_4 levels remained relatively consistent across all fuels. However, when the engine speed reached 2725 rpm which is its maximum, higher CH_4 emissions were observed. Further, the highest CH_4 emission was recorded at 1.54 kg/h CNG flow rate, reaching 4.80 g/kWh. The average CH_4 emissions for neat diesel are recorded at 0.31 g/kWh. Nevertheless, under the conditions of DDF operation and utilizing diverse CNG flow rates (0.67, 0.94, and 1.54 kg/h), the mean CH_4 emissions experience an elevation, reaching 1.10, 0.72, and 1.12 g/kWh, correspondingly. The usage of DDF technology is expected to result in increased levels of CH_4. This is primarily due to the introduction of CNG into the engine via the intake manifold, where it mixes with air before combustion. In all DDF modes, increased CH_4 emissions results from the exhaust and inlet valves being open simultaneously during the exhaust and intake strokes of the engine cycle. During the overlap phase, a mixture of CNG and air ultimately leaves directly to the exhaust through the exhaust valve. Elevated levels of CH_4 result from a decreased combustion temperature, a more lean mixture near the cylinder wall, and the confinement of CNG fuel within the crevice zone [75] [76]. Furthermore, 49–72 % account for methane emissions of the total THC emissions under DDF.

The potential global warming of methane is influenced by the quantity of CNG replaced in DDF mode. To ensure high conversion efficiency and address the issue of increased methane slip, it is crucial to design after-treatment devices like SCR and DOC accordingly [77]. The efficiency of SCR has been at its maximum when using diesel fuel, with an average CH_4 conversion of 0.31-0.03 g/kWh. With DDF mode, things were different. The highest achieved conversion rate was 27.77%, accompanied by a 0.67 kg/hr CNG flow rate, whereas the SCR conversion for other DDF modes ranged from 12.12 to 25%. Despite the absence of a clear pattern indicating an elevated methane slip when CNG was added to diesel, the results showed higher levels, comparable to the measurements taken prior to the SCR [78]. The effectiveness of methane emission conversion in SCR is solely determined by the methane concentration;

the lower the amount, the higher the conversion efficiency. In comparison to THC, methane has a significantly better conversion efficiency under DDF mode.

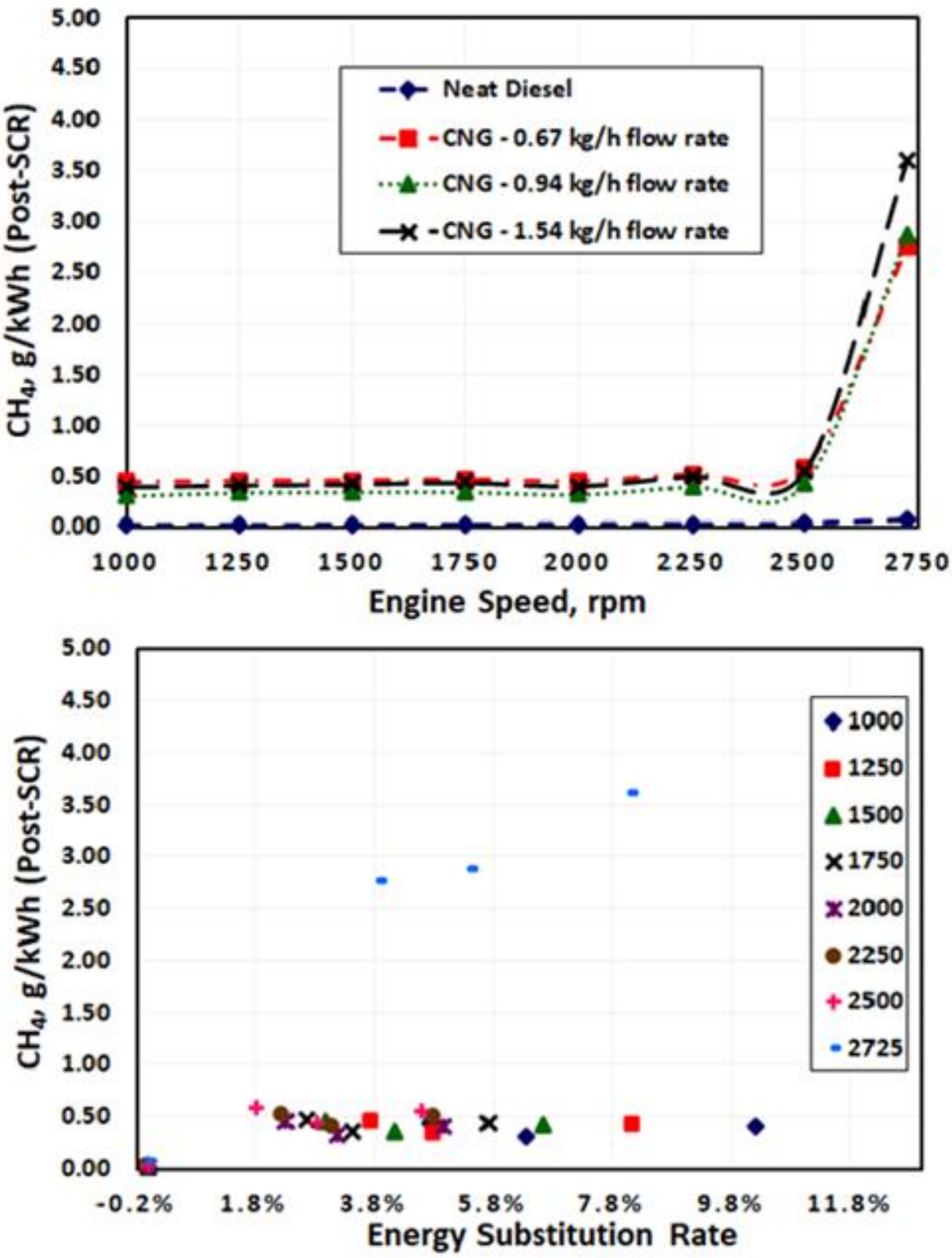


Fig 5.16: Methane Emissions at Post SCR under full load conditions.

5.1.2.5. CO$_2$ Emission

CO_2 and water vapour are the major products of hydrocarbon combustion in an engine. However, in most engines, combustion is incomplete, resulting in NOx emissions as well as intermediate products such CO, UBHC, CH_4, particulate matter, and soot. Moreover, the volume of CO_2 released into the atmosphere is influenced by several factors, including the specific fuel employed, the quantity of fuel utilized, the presence of oxygen, and the fuel's chemical composition. The CO_2 emissions are depicted in *Fig 5.17* & *Fig 5.18.*

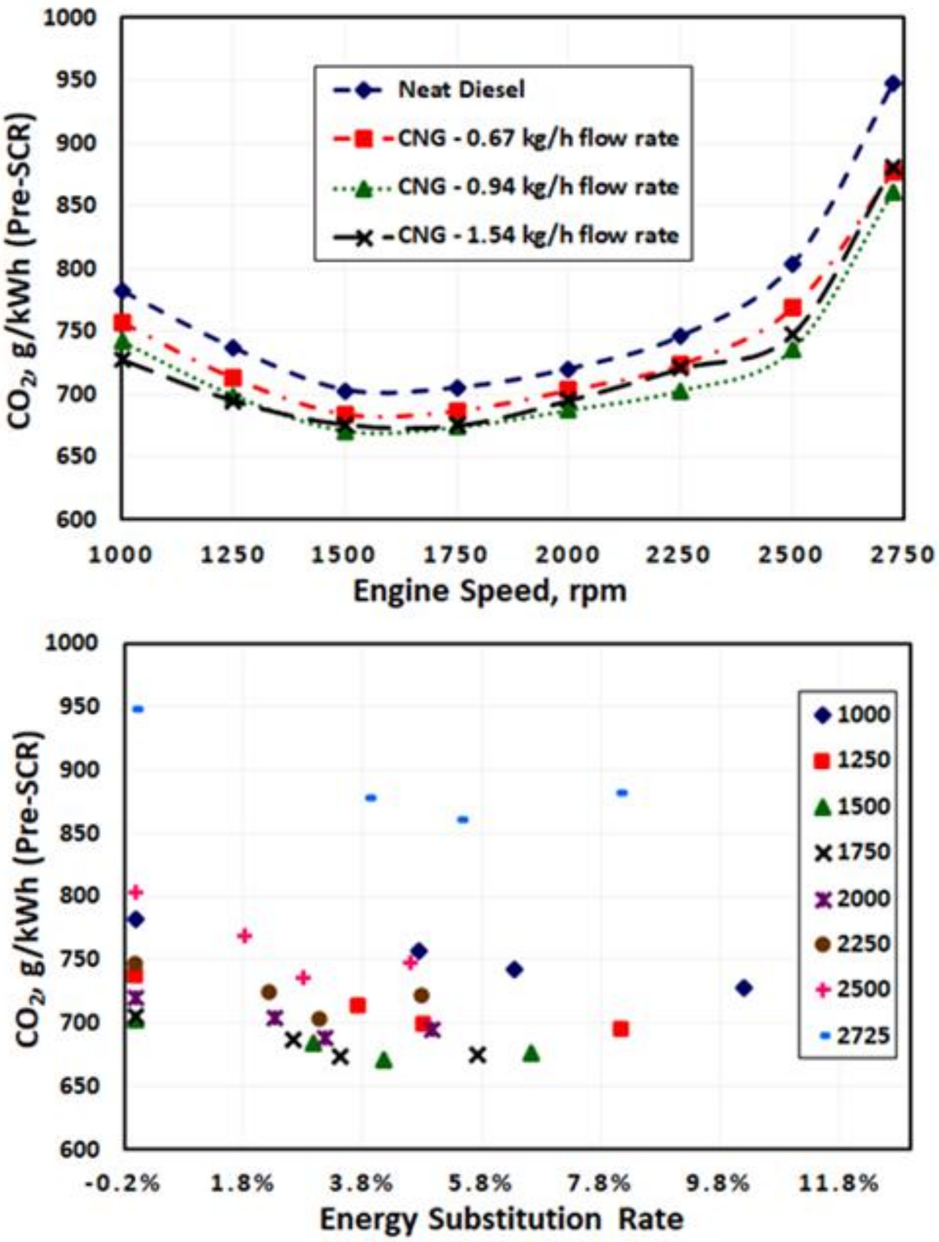


Fig 5.17: CarbonDiOxide at Pre-SCR under full load conditions.

When measured before SCR, the average CO_2 with neat diesel was 768 g/kWh. Furthermore, CO_2 dropped under DDF as ESR increased [79] [80]. Regarding 0.94 kg/h and 1.54 kg/h CNG flow rates, the corresponding lowest values were 722 g/kWh and 727 g/kWh, respectively. Because CNG fuel has a low carbon content, DDF mode is expected to reduce CO_2 [81]. CO_2 levels were marginally higher for all fuels at lower engine speeds; however, as engine speeds increased, CO_2 levels decreased until 1500 rpm, then increased [58].

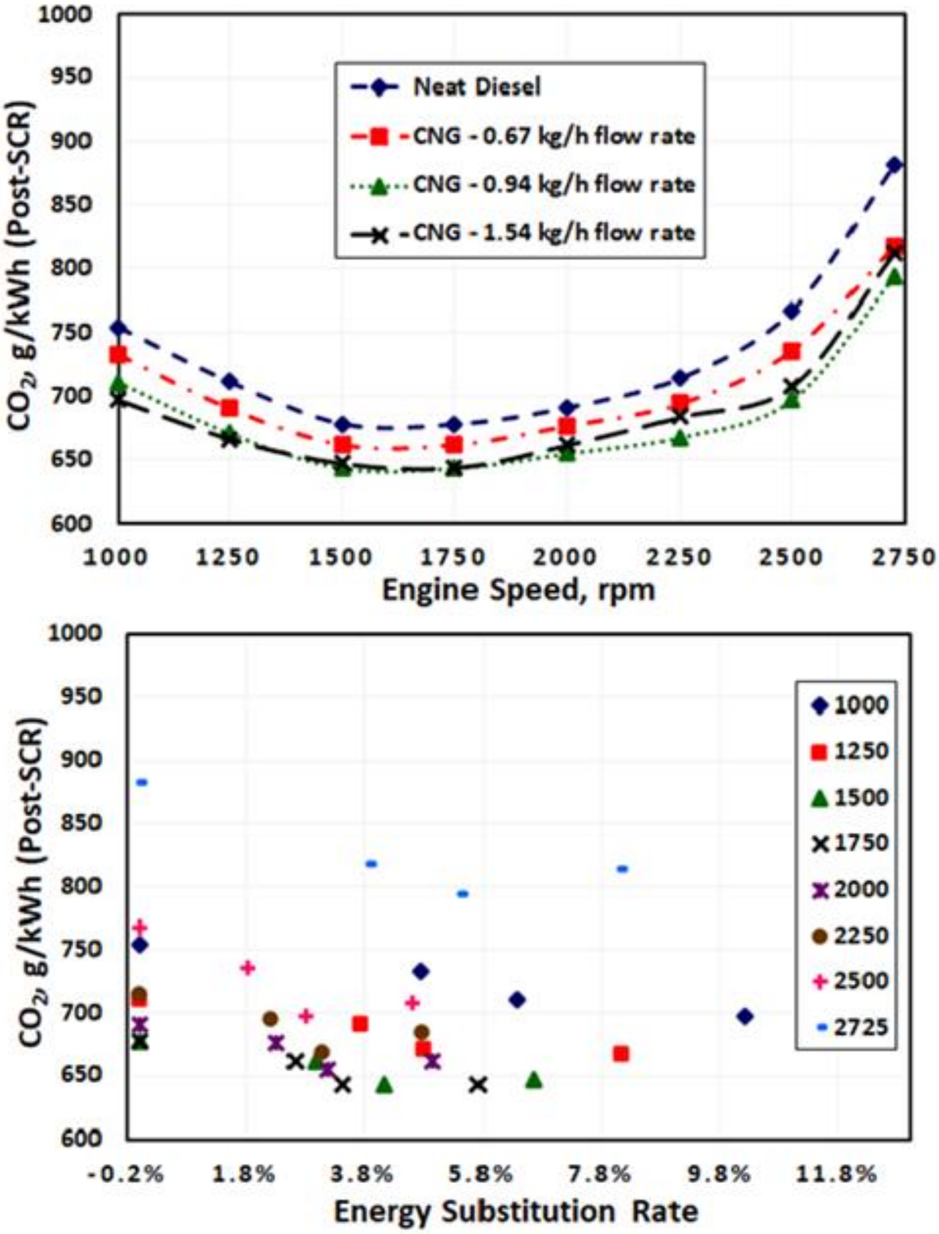


Fig 5.18: CarbonDiOxide at Post-SCR under full load conditions.

In the speed range of 1500-1750 rpm, it was observed that the CO_2 emissions were at their lowest levels. This was specifically observed when the CNG flow rate was either 0.97 or 1.54 kg/h, where the recorded value was approximately 674 g/kWh. Notably, when the DDF mode was employed, a substantial decrease in CO_2 emissions was achieved due to a reduction of around 20% in carbon content present in the air-fuel mixture. This utilization of DDF mode proved to be effective in mitigating CO_2 emissions. [58] [82]. CO_2 conversion rates in SCR systems are seldom documented, but this latest study reveals a surprising consistency, with CO_2 conversion occurring at a fixed rate of 4.5%, regardless of the type of fuel used. CO_2 emissions have decreased from 768 to 734 g/kWh in neat diesel operation, and from 730 to 695 g/kWh in DDF mode. While CO_2 levels were already low in DDF mode, they were far lower with the SCR system.

In *Fig 5.19*, the graph illustrates the mean conversion rate of regulated emissions using SCR technology under DDF mode. Most emissions are decreased by a percentage when using regular diesel, however, the percentage reduction is significantly reduced when using different CNG substitution rates.

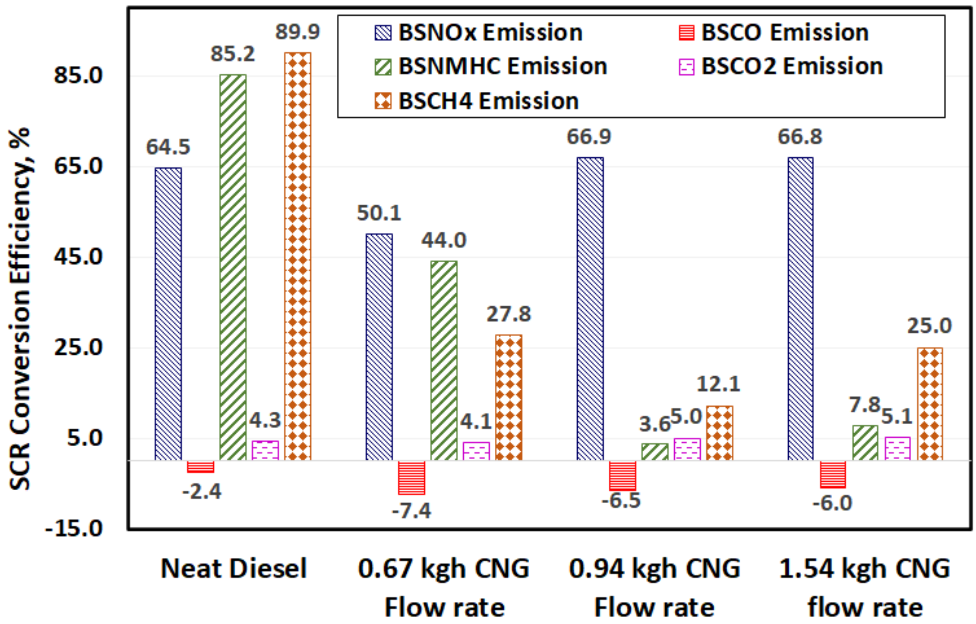


Fig 5.19: SCR conversion efficiency in various DDF modes at full conditions.

5.1.3. Unregulated Emission

While some of the toxic pollutants in diesel engine exhaust are regulated by strict legislation as previously described, others are not and are referred to as unregulated emissions. When exposed for longer periods of time, these unregulated emissions may have negative environmental and health consequences [83].

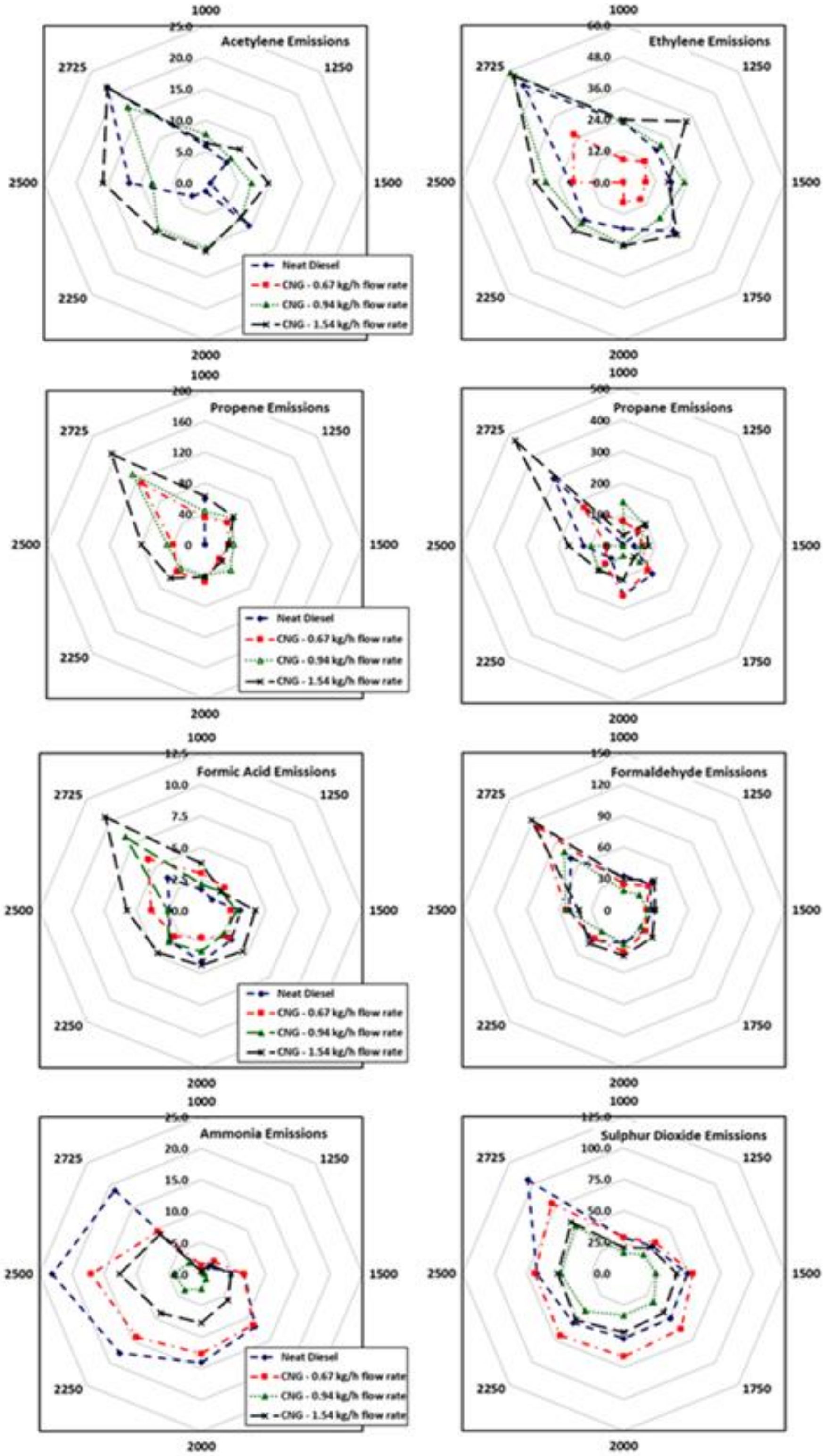


Fig 5.20: For different DDF modes, Unregulated Emission under full load conditions.

As previously stated, these unregulated emissions were assessed using FTIR equipment after the SCR system, and the results are depicted in *Fig 5.20.* The instrument is capable of measuring approximately 18 different components, encompassing acetylene, ethylene, propane, propene, formaldehyde, formic acid, ammonia, and sulphur dioxide emissions. All of these components fall within the instrument's measurement range. The concentrations of the unregulated emissions stated have been adjusted to a mass basis (mg/kWh) for clarity. Propane, SO_2, and formaldehyde were the most prevalent unregulated emissions tested. SO_2 emissions decreased under DDF, owing to diesel fuel's low sulphur content (below 10 parts per million) and the absence of sulphur in CNG.

Byproducts of combustion like formaldehyde have become more prevalent in the DDF mode. Lower combustion temperatures are anticipated as CNG's percentage is increased in the air-fuel mixture. As a result, as the ESR rises, there is an increase in formaldehyde emissions [84] [85]. As the ESR increased, the concentration of propane species reduced. These species fall under the category of saturated emissions and are generated during fuel combustion. In contrast, there was an observed rise in the levels of propene, a different type of emission [86]. With an increase in ESR, ethylene and acetylene both slightly increased. There were traces of formic acid found in this investigation, but no significant differences were found with partial substitution of CNG. Though there was evidence of ammonia slip during the DDF mode of operation, there was a slight reduction in the same.

5.2.COMBUSTION STUDIES UNDER FULL LOAD AND PARTIAL LOAD

When operating a diesel engine, the timing for injecting liquid fuel is typically set for the latter stage of the compression stroke. Within the combustion chamber, the fuel vaporizes as a result of the elevated temperatures, and it combines with the surrounding air before auto-igniting. Nevertheless, the introduction of premixed CNG and air into the combustion chamber brings about changes in the combustion process. Furthermore, the process of combustion has an impact on engine performance, emission production, and component durability [87]. In this section, we will explore the engine and combustion characteristics, as well as their influence on emissions in the context of DDF technology. Specifically, we will examine the gaseous fuel induction system and the engine setup, which includes an in-cylinder pressure sensor, as depicted in *Fig 5.21.*

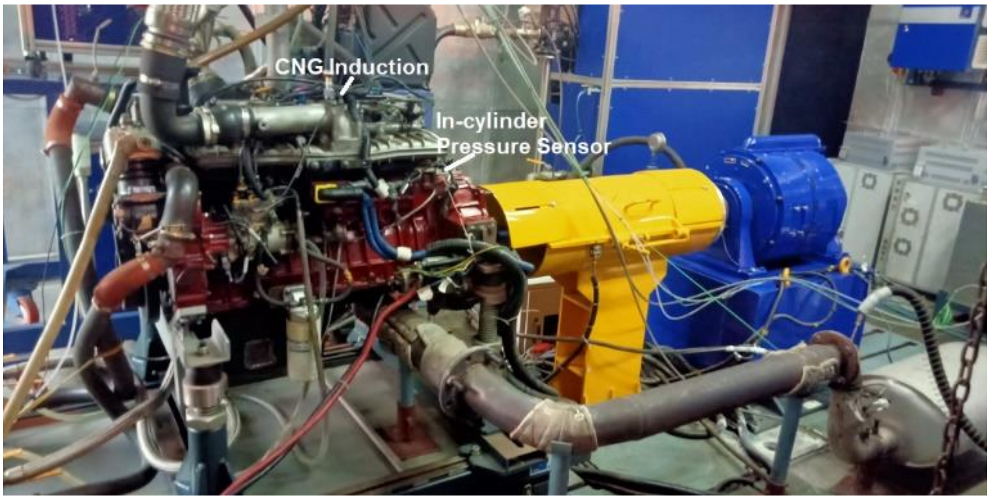


Fig 5.21: Test engine's CNG fuel induction and mounting position of In-Cylinder Pressure Sensor.

5.2.1. Engine characteristics

5.2.1.1. <u>CNG Energy Substitution Rate / CNG Energy Fraction</u>

Under this study, CNG used was considered as 99% methane, and the experiment took into account the total equivalent energy of DDF. When compared to the total energy of the DDF system, the CNG Energy Substitution Rate/CNG Energy Fraction (CEF) shows how much energy is obtained from CNG (corresponding equation given in Appendix). The relationship of CEF under DDF operation is shown in *Fig 5.22* for various loads and speeds. As expected, the CEF also increases as CNG mass flow rate (CMFR) under all the specified load and speed conditions, increases. CEF was reduced as the speed and load increased and as the CMFR increased. Due to the limitations of the pilot fuel's combustion capabilities, the engine was unable to operate with CNG flow rates exceeding 2.1 kg/hr under lower load and speed conditions. In a specific scenario, where the engine was running at 1000 rpm and operating at a 25% load, the ignition of the CNG-air mixture proved challenging for the pilot fuel. As a result, the engine was unable to sustain higher CNG flow rates and function properly. The CEF ranged from 1.9 to 60.2 % for the various CMFR, with lower CEF at lower speed and load and; higher CEF at higher speed and load. The pilot diesel fuel injection in the combustion chamber varies based on the quantity of CNG needed to generate the desired torque, while keeping the CMFR consistent across different DDF operations. Furthermore, as the speed or load increases, the CEF decreases. This paper's findings and discussion include combustion research based on CMFR/CEF at various speeds/loads, as well as the emission performance at 1750 rpm and full load is determined by analyzing several combustion parameters.

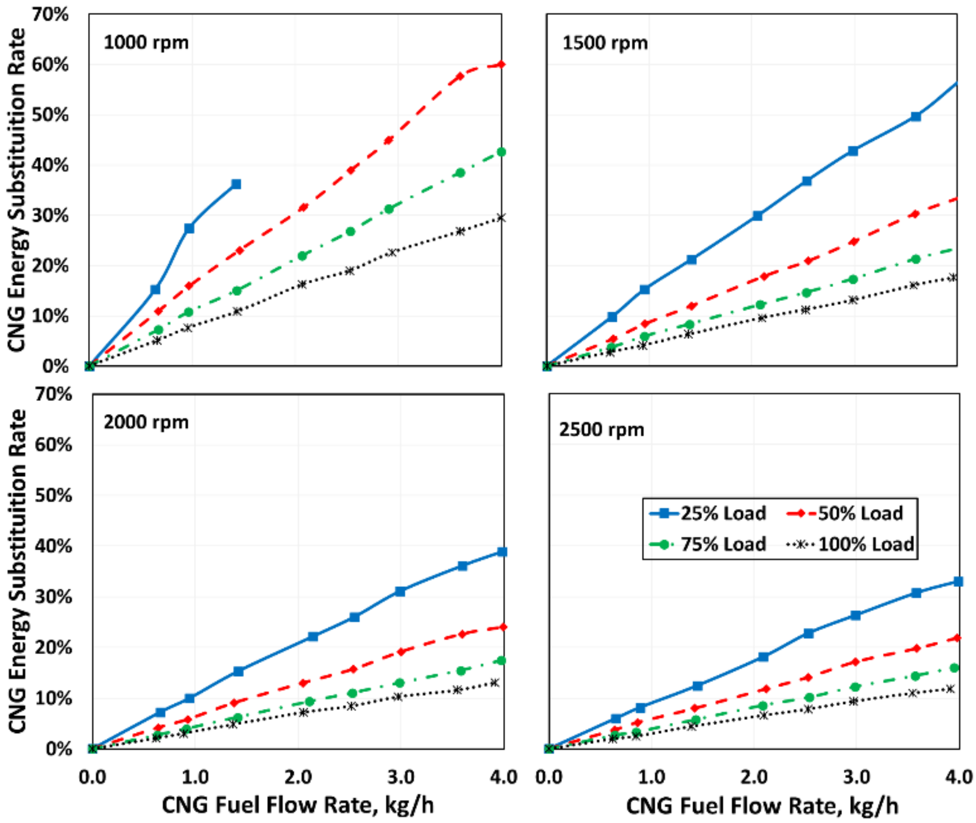


Fig 5.22: CNG Energy Fraction comparing CNG mass flow rates at different speeds/loads.

5.2.1.2. Brake Specific Energy Consumption (BSEC)

Fig 5.23 depicts the impact of BSEC on CEF under various load and speed conditions. BSEC, or Combustion Efficiency, is a metric utilized to assess the power output obtained from a specific fuel quantity. It quantifies the energy efficiency when diverse fuels, characterized by varying calorific values, are employed. The equation corresponding to the relationship between BSEC and CEF can be found in the Appendix.

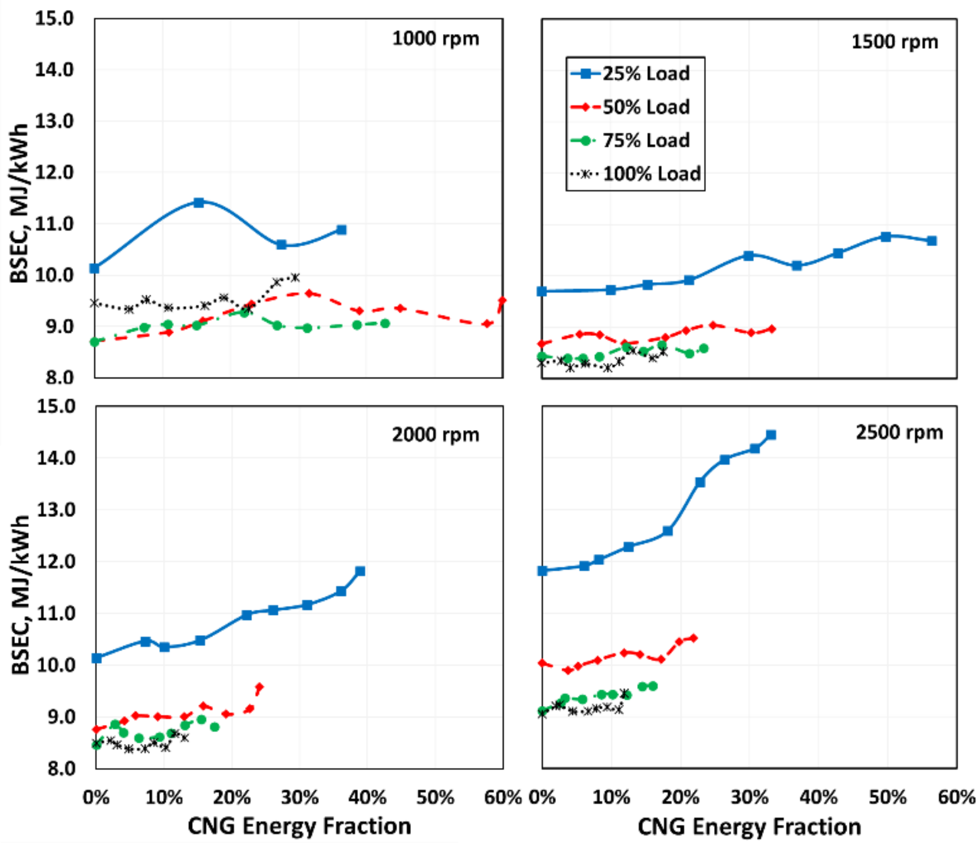


Fig 5.23: Brake Specific Energy Consumption comparing CEF at different speeds/loads.

In general, the behavior of a diesel engine's efficiency varies with its speed. Initially, the efficiency curve decreases until a certain point, and then it starts to climb. This pattern was also observed during the DDF operation. The BSEC decreases as the engine speed increases up to 1500 rpm, but it increases for every CEF value beyond that threshold. However, when the energy fraction is increased, the lowest BSEC is reported with pure diesel fuel, while the BSEC for DDF remains relatively constant or slightly increases. Under DDF operation, regardless of the energy substitution rate, the BSEC increases for all engine speeds when the load is at a lower level of 25%. However, for other load conditions, the BSEC remains nearly constant or shows a slight increase. Moreover, the BSEC decreases with increasing CEF for all speeds except at 1000 rpm as the load condition increases. At this particular speed, the BSEC decreases up to 50% load and then starts to increase again. The BSEC values for the 4% CEF at 1500 rpm and 100% load indicated the lowest value, whereas the BSEC values for the 33% CEF at 2500 rpm and 25% load indicated the highest recorded value.

5.2.1.3. Brake Thermal Efficiency (BTE)

At maximum throttle, the BTE of a diesel engine is typically around 40%, which is a function of energy of the fuel, with the remaining energy dispersed through heat losses, cooling medium, and exhaust gas. Three factors namely compression ratio, cut-off ratio and specific heat ratio determine theoretical BTE [88]. The trends of BTE with DDF at various loads and speeds are depicted in *Fig 5.24*.

While BTE improves with increasing load, it was higher at 75% load at lower speeds. Furthermore, at all CNG energy fractions, BTE was higher at higher loads. Nevertheless, as the proportion of CNG energy increased, the BTE decreased, especially when under lesser loads. But at higher loads, BTE was slightly reduced. When operating at lower speeds and higher loads (up to 1500 rpm), a marginal decrease was observed. However, at higher speeds, a substantial reduction in BTE was seen with lower loads (25%). Despite the fact that the flame was premixed, the incorporation of higher CNG energy content is anticipated to result in reduced thermal efficiency due to slower flame propagation [89].

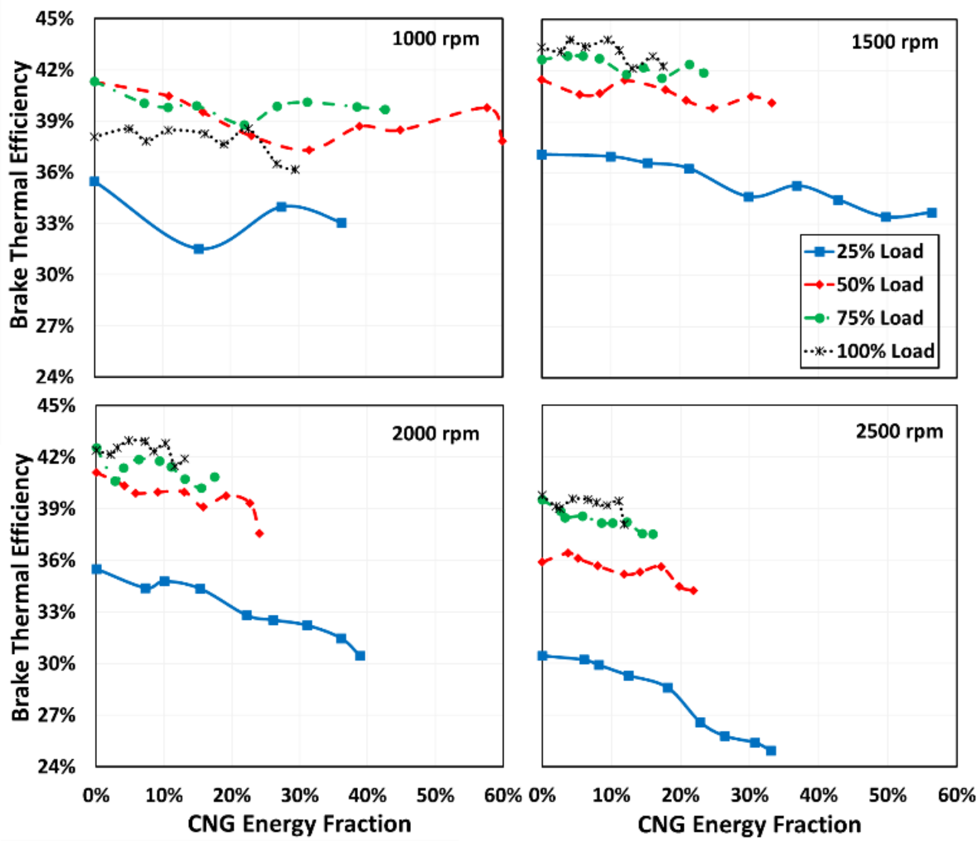


Fig 5.24: Brake Thermal Efficiency comparing CEF at different speeds/loads.

5.2.2. Combustion characteristics

High-frequency pressure measurement is a technique used in combustion analysis in an IC engine. It is possible to detect significant thermodynamic processes by visualising the pressure against crankshaft position. This information offers insightful information on the effectiveness and performance of the engine. Analysing combustion and gas exchange processes through cylinder pressure inquiry enables more accurate calibration and development to reach certain objectives. In comparison to other methods, cylinder pressure indication and combustion analysis offer exceptional information. Under this sub-section, from the pressure traces vs crank angle captured from Kistler KiBox combustion analyser, functions like heat release rate, in-cylinder pressure, pressure rise rate, combustion duration, ignition delay, etc. are calculated through the combustion analysis software and presented below.

5.2.2.1. In-cylinder Pressure

DDF combustion differs from conventional diesel combustion in several aspects [90]. The process involves short-term diesel pilot injection into a pre-mixed CNG-air mixture, followed by autoignition of diesel, a shift in combustion scenario, and premixed flame propagation. The primary means of monitoring combustion parameters relies on the utilization of a piezoelectric transducer to measure the in-cylinder pressure [91]. By analyzing pressure values, other essential combustion parameters can be derived. The measurement of in-cylinder pressure plays a crucial role in diagnosis of the engine and control as it offers valuable information about the combustion processes happening within the cylinder. To capture this data, a Kistler pressure sensor, specifically a water-cooled 6061C model, was utilized. It was appropriately installed within the cylinder head of engine cylinder number 6.

A comparison of CMFR at various engine speeds using the in-cylinder pressure curve while the engine is running at full load (100% of the intended engine torque) is shown in *Fig 5.25*. As the speed increases, the in-cylinder pressure rises until it reaches a peak between 1500 and 2000 rpm, then drops, with peak pressure occurring beyond TDC. By increasing the CMFR, the maximum pressure increases and the speed at which it occurs shifts from 2000 rpm to 1500 rpm. As an illustration, at 2000 rpm, the maximum pressure for diesel fuel was recorded, measuring 145.17 bar, while the maximum pressure for a CMFR of 4.0 kg/h occurred at 1500 rpm, measuring 152.89 bar.

Additionally, the study found that at lower engine speeds, the maximum pressure occurrence moved closer to TDC as the CMFR increased, ranging from 10.36 to 8.47 °CA. However, at 1500 rpm, this position remained relatively constant. Alternatively, when operating at higher speeds ranging from 2000 to 2500 rpm, there was a notable displacement of the maximum pressure which occurred beyond the level recorded for neat diesel fuel, ranging from 12.04 to 12.34 °CA, and away from TDC. The in-cylinder pressure trend implies that DDF burns early and has a shorter ignition delay at lower speeds, and burns later and has a little longer ignition delay at higher speeds, while having a higher maximum pressure [92] [93] [94] [95]. *Table 5.1* presents the cyclic variations expressed as the coefficient of variation (COV) for various CNG flow rates under full load conditions and at different engine speeds. Particularaly at rated speed, the impact of increasing flow rate of CNG on IMEP COV is evident. This effect is pronounced, with cyclic variations increasing by up to 4 kg/h when the flow rate of CNG reaches its maximum of 4 kg/h during DDF operation.

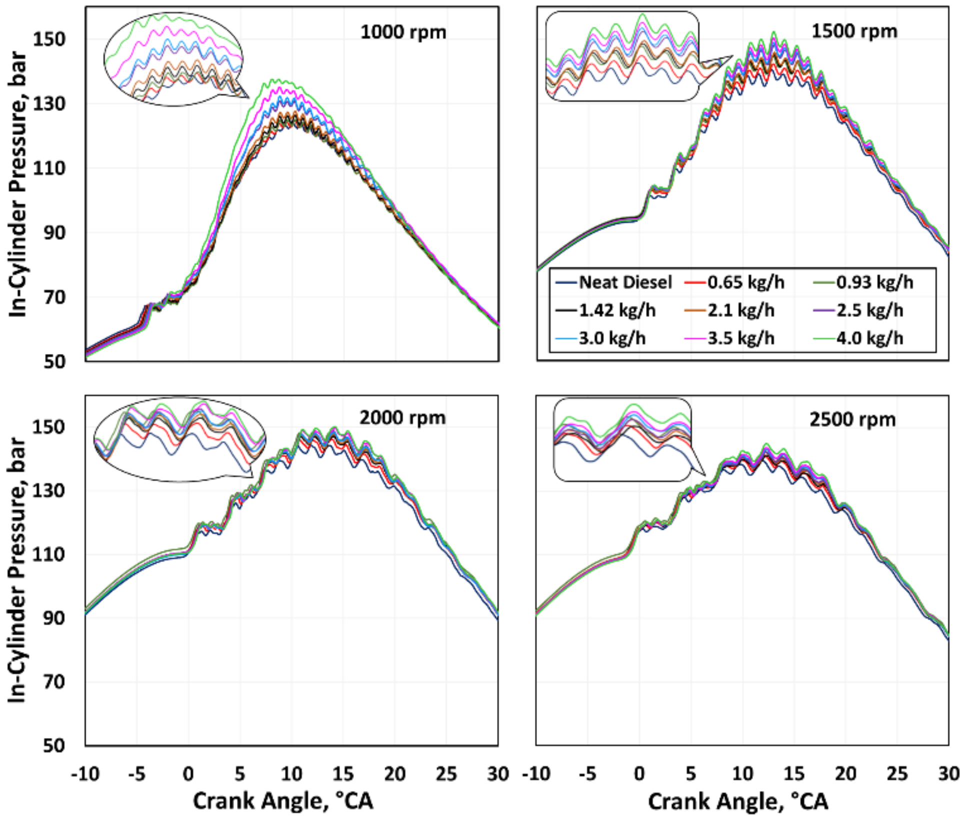


Fig 5.25: In-cylinder pressure profiles that compare CMFR at various speeds.

Table 5.1: IMEP COV (%) of DDF at various engine speeds while under full load (with varying CNG flow rates)

DDF /Engine Speed	Neat Diesel	CNG flow rate in kg/h								
		0.65	0.93	1.42	2.1	2.5	3.0	3.5	4.0
1000	0.69	0.85	0.78	-	0.54	0.6	0.61	0.46	0.54
1500	0.6	0.6	0.64	0.73	0.83	0.76	0.87	0.84	0.92

2000	0.39	0.54	0.47	0.87	0.63	0.7	0.7	0.99	0.9
2500	0.56	0.64	0.78	0.78	0.78	0.9	0.96	1.01	1.18

5.2.2.2. Heat Release Rate

Identification of combustion parameters can be facilitated by thermodynamic analysis of data from in-cylinder pressure measurements. "Burn rate analysis" and "heat release analysis" are the two primary approaches utilised in the analysis of thermodynamics [96]. In petrol engines, the useful technique of "burn rate analysis" is used to estimate burn angles and figure out how much mass has been consumed. On the other hand, in diesel engines, heat release analysis serves as a tool to ascertain the absolute energy levels. The heat release rate (HRR) is a significant factor that affects the combustion noise, rate of pressure rise, and NOx emissions. The HRR curves that compares CMFR at various speeds is depicted in *Fig 5.26*. HRR trends for each CMFR follow a similar pattern, and diffusion and pre-mixed combustion phases could be seen for all CMFRs. Peak HRR for all fuels decreases as engine speed increases, although peak HRR increases as CMFR increases (i.e. as pilot diesel fuel quantity decreases) [97].

This study discovered that maximum HRR increased at full load across all engine speeds, contrary to earlier research that claimed HRR reduced with DDF. For all fuels and speeds, the initial HRR peak emerged shortly before or just before the top dead centre (TDC). It's interesting to note that the second peak, which appeared further from TDC, had values that were far higher than the first peak. Additionally, at higher speeds several peaks were seen for all fuels between 2000 and 2500 rpm.

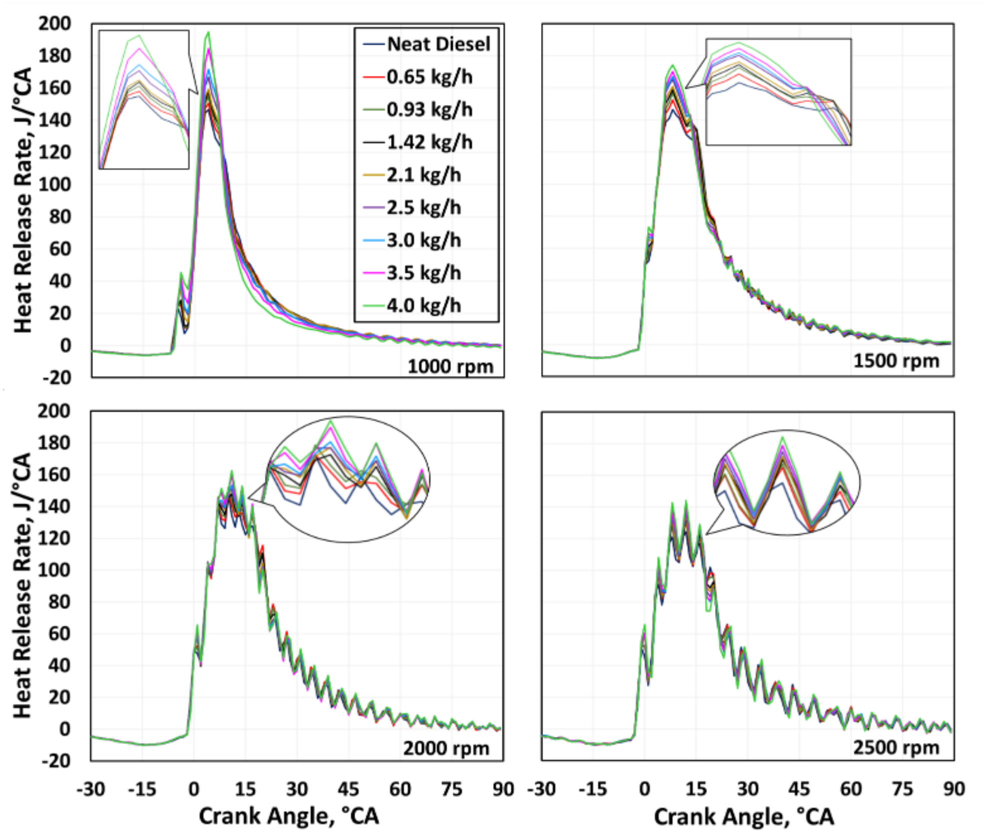


Fig 5.26: Heat Release Rate curves comparing CMFR at different speeds.

The peak HRR exhibited variations between 5 and 9 degrees crank angle (°CA) for engine speeds ranging from 1000 to 1500 rpm. When the engine was running at 2000 rpm, an elevation in CMFR caused the peak HRR to escalate from 10.76 °CA to 11.96 °CA. In contrast, when the engine was operating at 2500 rpm, a rise in CMFR caused the peak HRR to decrease from 12.32 °CA to 11.72 °CA. *Table 5.2* presents the 50% HRR in °CA for various CNG flow rates at varying engine rpm while the engine is fully loaded. The higher HRR observed during DDF can be attributed to the delayed combustion of CNG fuel [92] [94] [98] [99] [21] [90] [35].

Table 5.2: Crank Angle (°CA) corresponding to 50% Heat Release Rate (CA50) of various CNG flow rates under full load condition at different engine speeds

DDF / Engine rpm	Neat Diesel	CNG flow rate in kg/h							
		0.65	0.93	1.42	2.1	2.5	3.0	3.5	4.0

1000	9.1	8.9	8.7	-	8.5	7.9	7.7	7	6.1
1500	14.2	14.3	14.1	14	13.9	13.7	13.6	13.5	13.4
2000	16	16.1	15.9	16.1	15.9	15.8	15.8	15.7	15.7
2500	15.9	15.9	15.8	15.8	15.7	15.7	15.6	15.5	15.3

5.2.2.3. Pressure Rise Rate

An engine cycle's pressure rise rate (PRR), which affects acoustic emissions, indicates how quickly initial energy is discharged [100]. During the combustion process, it also describes the rate of increase in combustion pressure. The PRR variance when CMFR is compared at different speeds is depicted in *Fig 5.27*. Under DDF, PRR decreases as engine speed increases and increases when CMFR increases, with 4.0 kg/h being the highest value [70]. The duration of the combustion process decreases as engine speed increases. This reduced combustion duration limits the overall pressure rise rate. Consequently, the pressure rise rate tends to decrease with increasing engine speed in a DDF engine. The results also point to the possibility that increasing the CMFR in the presence of DDF could result in enhanced combustion noise, particularly at lower RPMs, with a notable difference between normal diesel and a CMFR of 4.0 kg/h in maximum PPR of about 2.8 bar/°CA. The difference is only 0.5 bar/°CA at higher RPMs, though. CNG has a higher combustion speed compared to diesel fuel which implies that the CNG portion of the fuel mixture burns more rapidly during the combustion process, leading to a faster pressure rise rate in the cylinder. This faster pressure rise rate results in an increase in combustion noise which is primarily due to the rapid and intense combustion of the CNG portion, which creates higher peak pressures and temperature gradients within the cylinder. Additionally, it was discovered that as the CMFR augmented, the maximum PRR exhibited a displacement away from the TDC and extended beyond the realm of pure diesel combustion, encompassing the range of 2000 rpm. However, at 2500 rpm, the incidence diminished, reverting back to the vicinity of TDC. This is because CNG and diesel fuel have distinct combustion characteristics. With a higher octane rating and a faster burning rate than diesel, CNG has different ignition and combustion characteristics. With an increase in CNG's mass flow rate, the combustion process becomes more rapid, resulting in a shorter ignition delay and faster flame propagation. This faster combustion rate can cause the peak pressure to occur earlier in the engine cycle, shifting the maximum pressure rise rate away from TDC. Additionally, depending on the demands, the injection time can be adjusted to minimise the maximum PRR by delivering pre-mixed CNG into the combustion chamber [46].

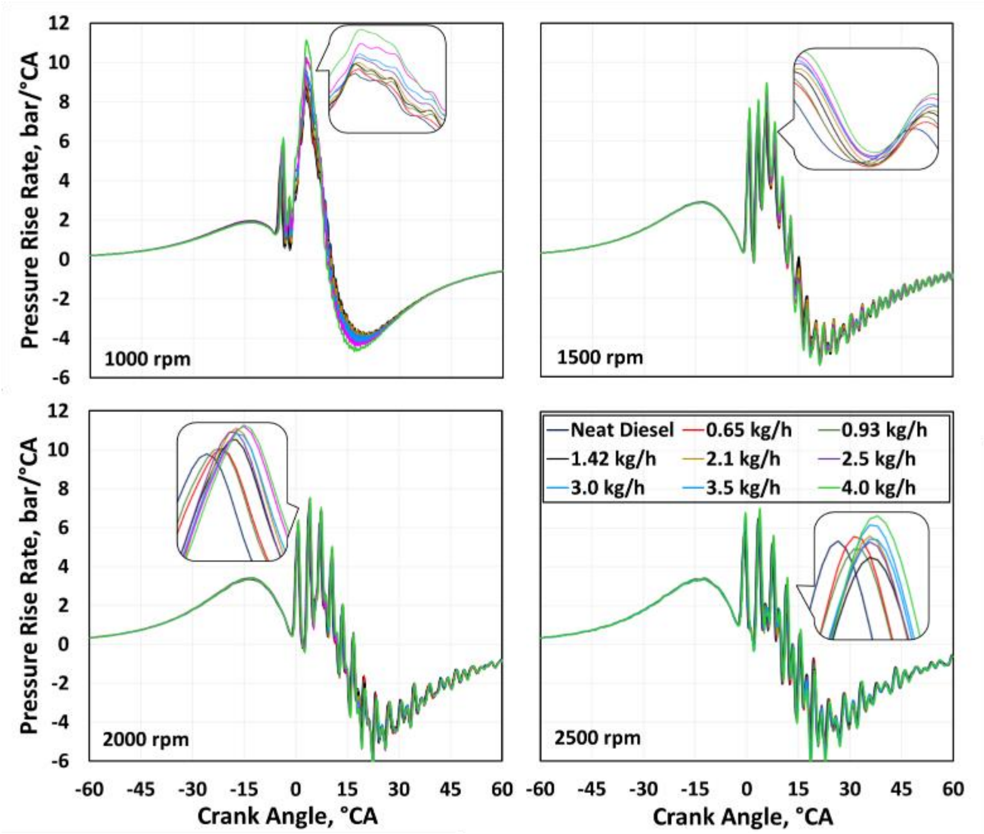


Fig 5.27: Curves of Pressure Release Rate comparing CMFR at various speeds.

5.2.2.4. <u>Combustion Duration</u>

Combustion duration is a critical parameter for understanding the combustion development process, as shown in *Fig 5.28*, which compares different CEFs under various load/speed conditions. The combustion duration is referred to in this study as the time between 10% and 90% of the mass fraction burned. In general, the combustion duration increases as the load increases however it rises up to 1500 rpm when the engine speed increases and then declines [72] [71].

Additionally, when the CEF rises, the combustion duration during DDF combustion experiences an increase across all load and speed conditions. However, there is an exception at lower speeds of 1000 rpm, where the combustion duration decreases at a CEF of 20-40%. This difference could be explained by the fact that the thorough blending of CNG and diesel occurs more rapidly at higher CEF values, leading to shorter combustion durations [101] [70]. On the other hand, when engine speed increases, less time is available to mix CNG and diesel, thereby increasing the combustion duration.

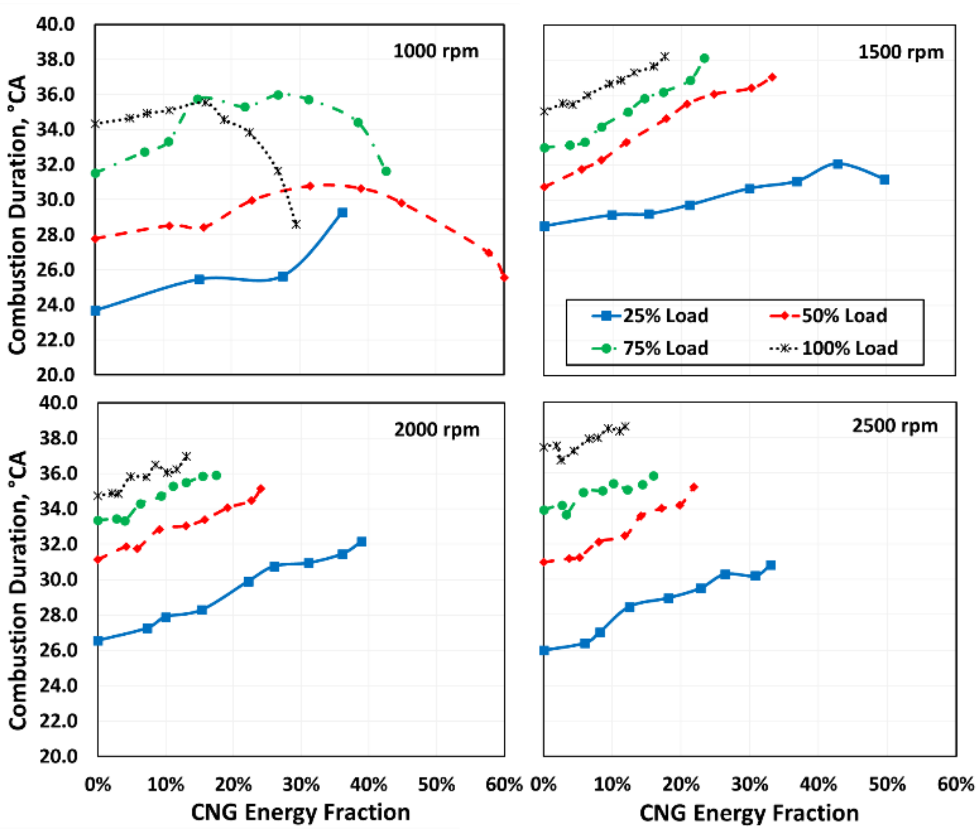


Fig 5.28: Comparing the CEF at various speeds and loads during combustion.

5.2.2.5. Ignition Delay

The time between the beginning of diesel fuel injection and the beginning of combustion is known as the ignition delay (ID). Under DDF operation, ID refers to the time delay between the start of injection of diesel fuel and the actual ignition of the diesel fuel-air mixture in the combustion chamber. When considering DDF systems, it is crucial to acknowledge that the existence of pre-blended CNG-air within the combustion chamber can result in distinct variations in both physical and chemical delays. These delays are influenced by the energy content, which is determined by factors such as viscosity and volatility. The variation in ignition delay at various speeds and loads comparing CEF is shown in *Fig 5.29*. ID decreases with increasing load (6.21-4.67 °CA) at lower engine speeds of 1000 rpm, but ID is nearly constant across different load conditions (between 9 and 11 °CA) during intermittent speeds of 1500-2000 rpm. However, when the engine speed is increased to 2500 RPM, the ID increases (13.34-14.24 °CA). Additionally, ID increases as engine speed increases ranging between 4.67 and 14.24 °CA.

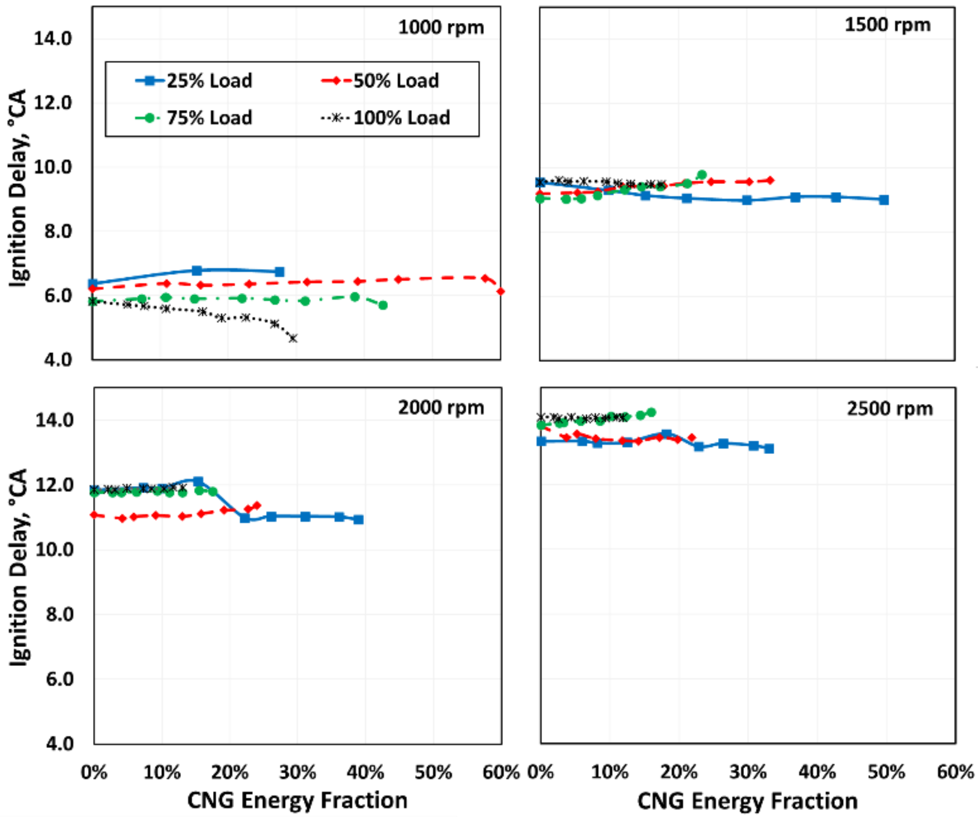


Fig 5.29: Ignition Delay comparing CEF at different loads and speeds.

At low speeds and low loads, the combustion chamber's temperature is inadequate to initiate auto-ignition of the fuel-air mixture, leading to a decrease in ID. On the other hand, even if the temperature is high at high loads and speeds, the available time for auto-ignition is shortened, resulting in an increase in ID. Furthermore, when CEF was increased, ID decreased at 100% load, while

ID increased marginally at most engine speeds for other load conditions [21] [102] [103] [104] [70]. The increased ID is because of the physical-chemical characteristics of CNG and diesel, which include lower thermal conductivity and specific heat [105]. Further under lower load conditions, the engine operates with a reduced fuel quantity that results in a lean mixture of air and fuel, where the ratio of air to fuel is higher than the stoichiometric ratio. Lean mixtures have a longer ID due to slower flame propagation and lower reactivity of the fuel; and also due to the reduced time available for pilot flame development when the pilot fuel quanity is reduced.

5.2.3. Emission Characteristics with Combustion Parameter

Raw emissions were evaluated using the same analyzer that was used for legislative purposes. Because the DDF engine's maximum torque is between 1400 and 1900 rpm, in this section, we analyze the variations in CEF at varying loads, while considering the diverse emission characteristics of the DDF engine operating at 1750 rpm. We examine various combustion parameters, including diesel injection duration (DID), fuel burn rate (FBR), IMEP and combustion duration. This analysis enables us to evaluate the effects of load and emission characteristics on the combustion performance of the engine.

5.2.3.1. CO Emission

The impact of various combustion parameters on CO emissions can be observed in *Fig 5.30*. In general, FBR increases with increasing load, and it can be derived from the graph that with longer FBR, CO emissions decrease. In terms of combustion duration, IMEP, and DID, a similar trend could be noticed. Under DDF, FBR reduces with increasing CEF at low loads while increasing at higher loads, resulting in increased CO emissions, though at lower levels. Increased CEF causes a rise in CO emission, whilst increased IMEP and combustion duration cause a decrease.

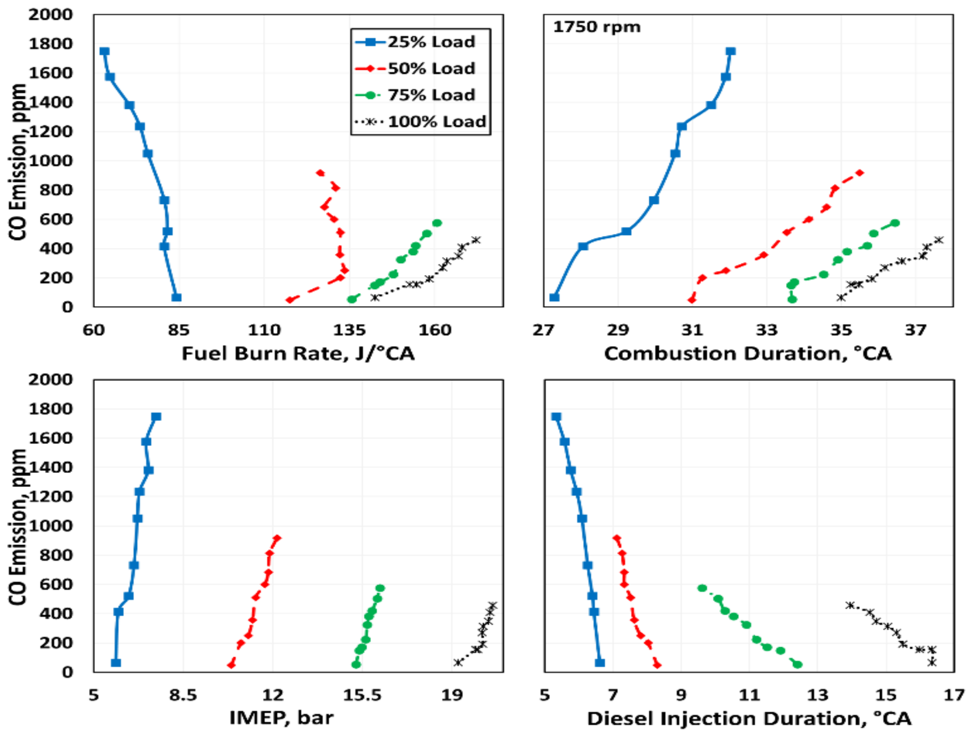


Fig 5.30: Trend of CO Emissions under Various Combustion Parameters.

When considering the concept of DDF, it becomes apparent that as CEF increases, the value of DID decreases. Furthermore, a rise in DID promotes a reduction in CO emissions, while a decrease in DID results in a rise in CO emissions. Therefore, it can be concluded that an increase in load causes a decrease in CO emissions regardless of the combustion parameters. Similarly, CO is decreased under increasing FBR, IMEP, Combustion duration and DID.

5.2.3.2. NOx Emission

The NOx emission pattern under varied combustion conditions is depicted in *Fig 5.31*; unlike CO emission, NOx emission does not show a trend. The emission of NOx increases up to a specific CEF for each load condition and subsequently decreases. However, when the load increases, the amount of NOx emitted increases. NOx levels were similar in neat diesel and increased DDF %ages. When the load is increased, every combustion parameter demonstrates an increase in NOx emission. It's also important to note that NOx emission increases when IMEP, FBR, DID, and combustion duration increase.

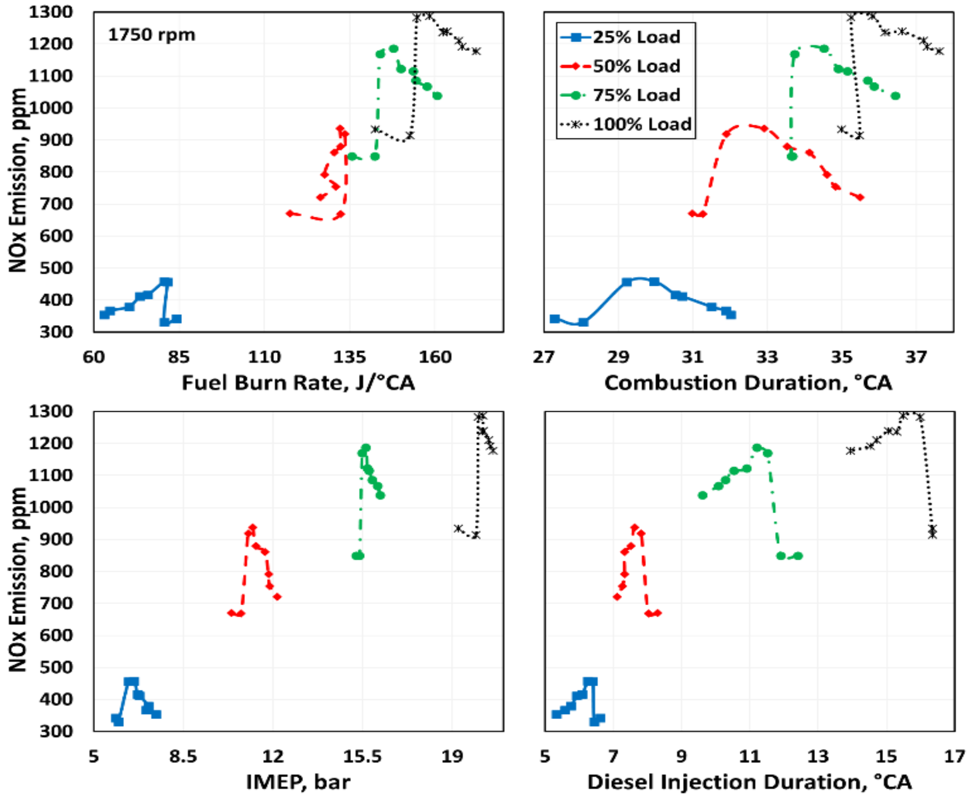


Fig 5.31: Trend in NOx emissions under various combustion conditions.

5.2.3.3. THC Emission

Fig 5.32 shows how THC emissions under DDF respond to different combustion parameters. THC emission follows the same pattern as CO emission, i.e., it reduces as load increases, while THC emission increases as CEF increases at different combustion parameters. Higher IMEP, FBR, DID and combustion duration values result in less THC emission, while lower values result in higher THC emission.

5.2.3.4. CO_2 Emission

The CO_2 emission trend for a DDF engine is depicted in *Fig 5.33* with various combustion parameters. CO_2 emissions have decreased with an increase in CEF and DDF, as shown in the graph. CO_2 emission increased with increasing load, although it was reduced with increasing FBR, combustion duration, and IMEP at each load condition, while CO_2 emission decreased with increasing DID, which decreased with increasing CEF.

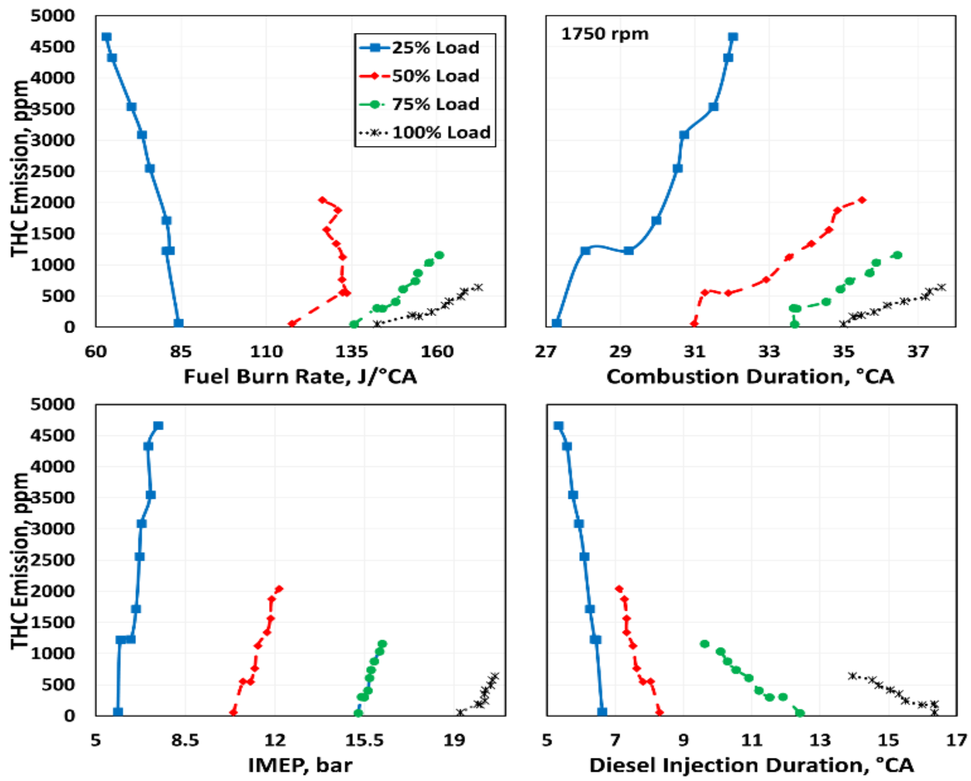


Fig 5.32: Trend in THC Emission under Various Combustion Parameters.

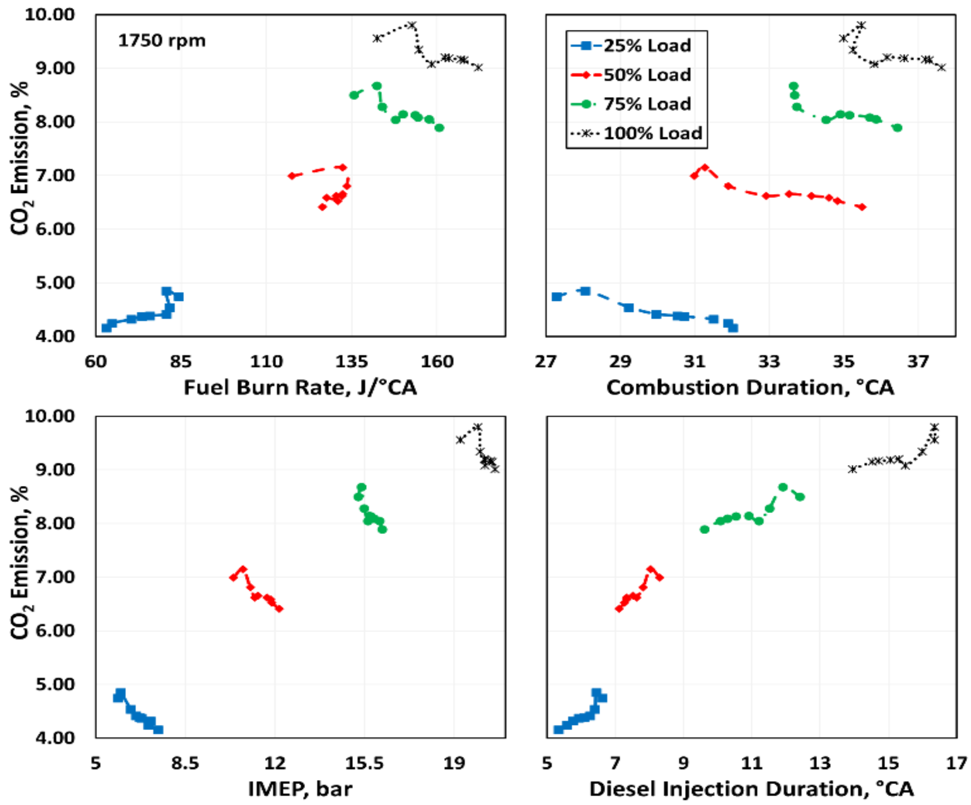


Fig 5.33: CO_2 Emission trend under different combustion parameters.

5.3. OPTIMIZATION OF DDF ENGINE BASED ON ENGINE PERFORAMANCE AND EMISSION UNDER DIFFERENT LOAD AND SPEED

In this section, we examine how varying CNG flow rates affect engine efficiency, emissions, and the SCR after-treatment device under various engine conditions. Additionally, we employ a statistical tool to determine an optimal CNG flow rate that matches or surpasses the performance of diesel fuel operation. Subsequently, we present a matrix that can be utilized to implement the recommended CNG flow rate measured by the sum of the performances at various engine speeds and loads. The automotive CNG specification (IS:15958) requires a methane concentration of at least 90% and CNG is assumed to be 99 % methane for the purposes of this study. Second-degree polynomial curves are employed to represent them when there isn't a trend in the graph.

5.3.1. CNG flow rate's effect on MFSR and ESR

In order to showcase the operation of DDF in a commercial heavy-duty multi-cylinder engine, several distinct CNG mass flow rates were employed. These rates were 0.65, 0.93, 1.41, 2.1, 2.5, 3.0, 3.5, and 4.0 kg/hr. It should be noted that the quantity of diesel fuel utilized as a substitute for each consistent CNG flow rate may differ. This discrepancy arises from the fact that CNG was manually introduced at a steady flow rate into the intake manifold for each individual trial, conducted at varying engine speeds & loads. The MFSR and ESR of various DDF operations under different load and speed is demonstrated in *Fig 5.34* & *Fig 5.35*. The CNG MFSR represents the proportion of CNG mass in relation to the total mass of DDF. The ratio of CNG energy to the overall energy of DDF is denoted by ESR. The Appendix contains the precise formulas for ESR and MFSR calculation..

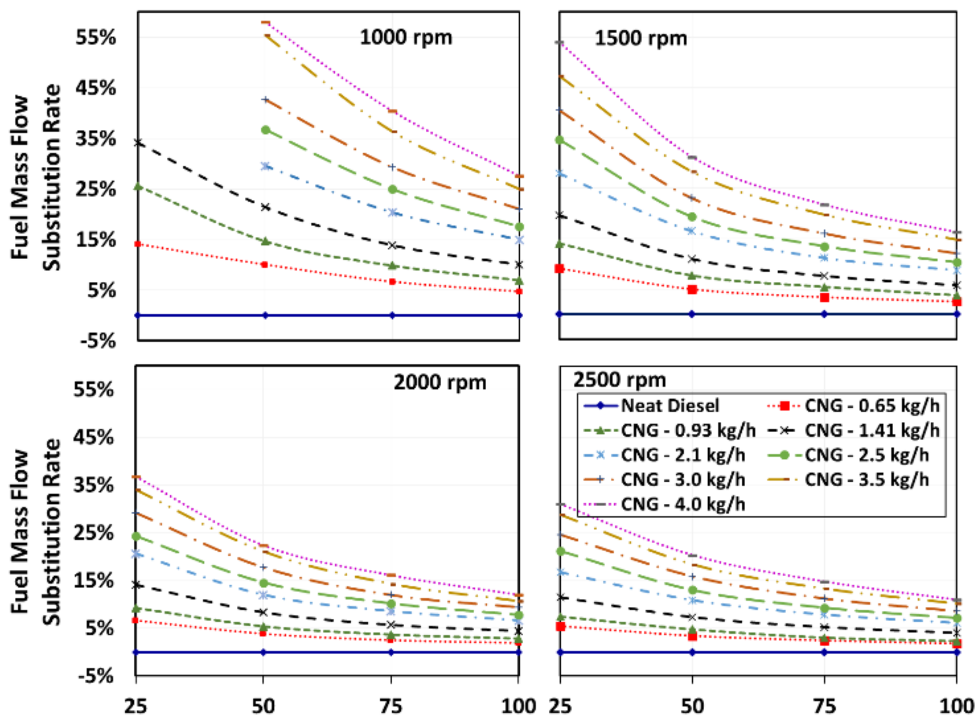


Fig 5.34: MFSR of CNG under different loads and speeds.

As seen in both figures, the maximum fuel mass and energy substitution can be observed at lower speeds and loads. As the flow rate of CNG increases, so does the rate of mass and energy replacement. However, as engine speed and load increase, the fuel mass and energy substitution rate drops, whereas as the CNG's mass flow rate increases, both mass and energy substitution rates increase. The lowest recorded MFSR value was 1.8%, observed when the engine operated at 2500 rpm and 25% load, with a 0.65 kg/hr. On the other hand, the highest MFSR value reached 58% when the engine ran at 50% load and 1000 rpm, with a CNG flow rate of 4.00 kg/hr CNG flow rate. However, it should be noted that at lower load conditions the engine could not function when operating at 1000 rpm and with CNG flow rates ranging from 2.1 to 4.00 kg/hr.

The combustion chamber experienced an increased concentration of CNG, resulting in incomplete burning of the air-fuel combination. As a result, the pilot diesel fuel was unable to fully combust, leading to engine stalling and the emission of black smoke. However, this issue was not encountered during higher speeds and load conditions, as it was more easily detectable. Specifically, when operating at 2500 and 1000 rpm, the ESR ranged from 1.9% to 60.2%.

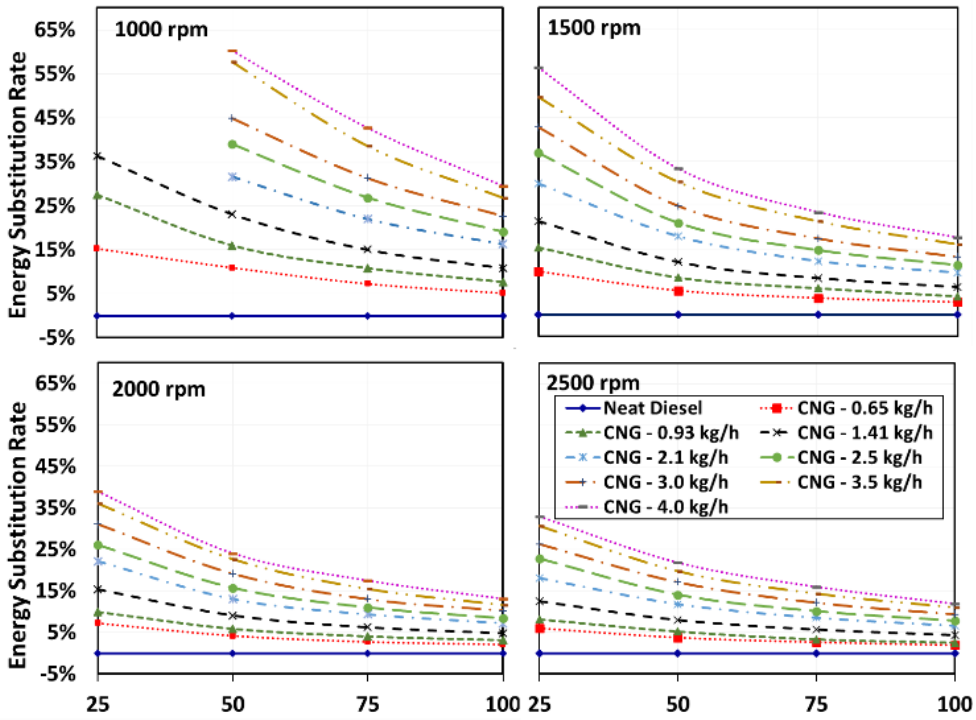


Fig 5.35: Energy Substitution rate of CNG under different loads and speeds.

The ESR was lower at a 0.65 kg/hr CNG flow rate, despite rising to higher levels at a 4.00 kg/hr CNG flow rate. Because of the proportional amount of pre-mixed versus diffusion burn combustion, the MFSR/ESR has decreased. When CNG is introduced into the intake and completely utilized for combustion, it predominantly undergoes pre-mixed combustion under low loads. However, under higher loads, diffusion combustion becomes necessary due to the increased fuel demand. Consequently, the utilization of CNG does not enhance the situation, and to achieve the required load, additional diesel fuel is required [106].

5.3.2. Engine Performance under different DDF operation

5.3.2.1. Relative Air-Fuel Ratio

The air-fuel ratio has a significant impact on an engine's output, fuel efficiency, and pollutants. However, a rich mixture is intended to produce more power, a lean mixture is predicted to consume less fuel and produce lower emissions like

HC and NOx, while NOx emissions will be higher [10]. For the DDF operation, the relative air-fuel ratio (RAFR) is displayed in *Fig 5.36*.

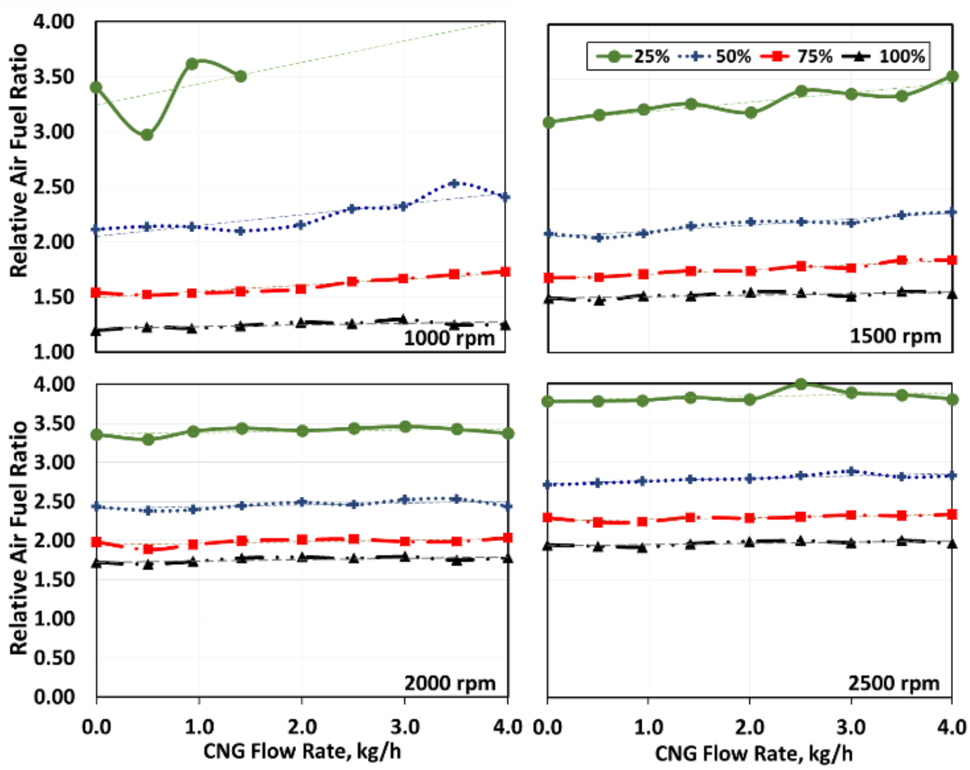


Fig 5.36: Relative Air-Fuel Ratio under different loads and speeds.

Typically, diesel engines operate on a leaner combination of fuel and air, characterized by a RAFR greater than 1. Throughout this study, a specific test engine was utilized, and its RAFR ranged from 1.2 to 3.78 when the speed was between 1000 and 2500 rpm. As depicted in the figure, as engine speed increases, RAFR also increases. However, it is worth noting that as the load percentage rises at all engine speeds, the RAFR gradually decreases and tends towards the stoichiometric ratio. At a load of 25% and an engine speed of 2500 rpm, all the fuels under test display a higher RAFR in the 3.78–4.00 range. A RAFR of 3.78 corresponds to regular diesel, while a RAFR of 4.00 corresponds to a 2.5 kg/hr CNG flow rate. Conversely, at 1000 rpm engine speed with a 100% load, a lower RAFR ranging from 1.20-1.30 was observed. The lowest RAFR was recorded when using regular diesel, whereas the highest RAFR was achieved with a 3.0 kg/hr CNG flow rate. It is significant to remember that a diesel engine's combustion chamber air volume fluctuates with engine speed and load. However, when it comes to DDF engines, along with air, CNG is added to the intake manifold thereby affecting the air-fuel mixture's composition in the combustion chamber. For the most part, RAFR is higher than normal diesel during DDF operation, which is to be expected given CNG's higher stoichiometric ratio. Additionally, as torque remains consistent across various CNG flow rates and operating conditions, to obtain the necessary rated torque, the amount of diesel injected is changed and reduced.. Consequently, this leads to an increased RAFR in the context of DDF.

5.3.2.2. Boost Air Pressure, Boost Air Temperature, and Exhaust Temperature

The impact of boost air pressure (BAP) and temperature (BAT) on engine performance and emissions is significant, as illustrated by the data presented in *Fig 5.37* & *Fig 5.38*. Various engine load and speed conditions were considered, and the measurements obtained for specific fuel consumption, power output, NOx emissions, particulate emissions, exhaust temperature, and CO emissions were analyzed. It is clear that the engine's overall performance and emission characteristics are greatly influenced by the boost air temperature and pressure [107] [108].

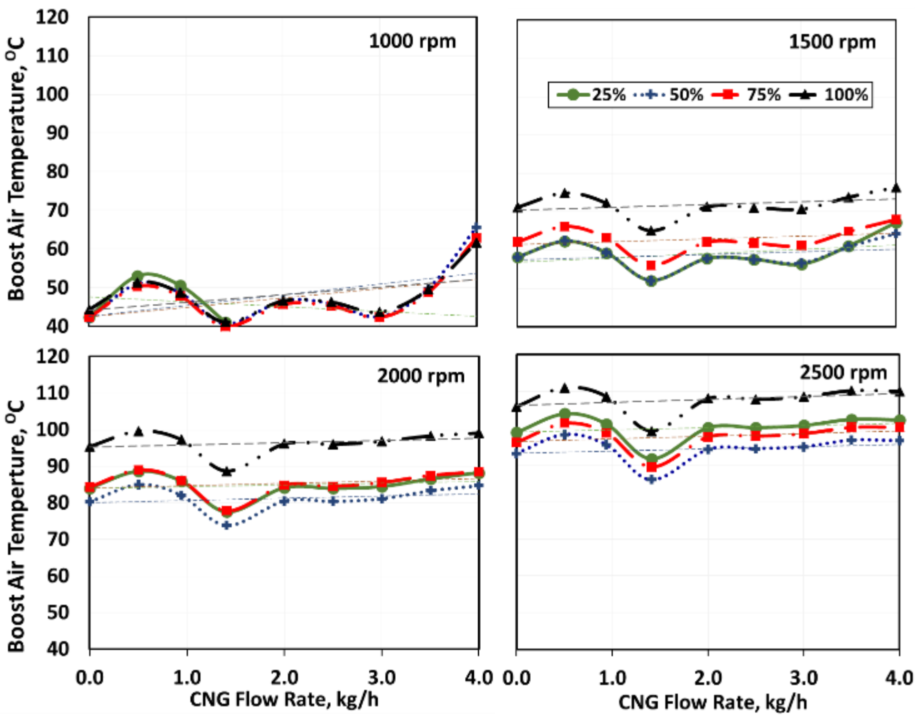


Fig 5.37: Variation in BAT under various loads and speeds.

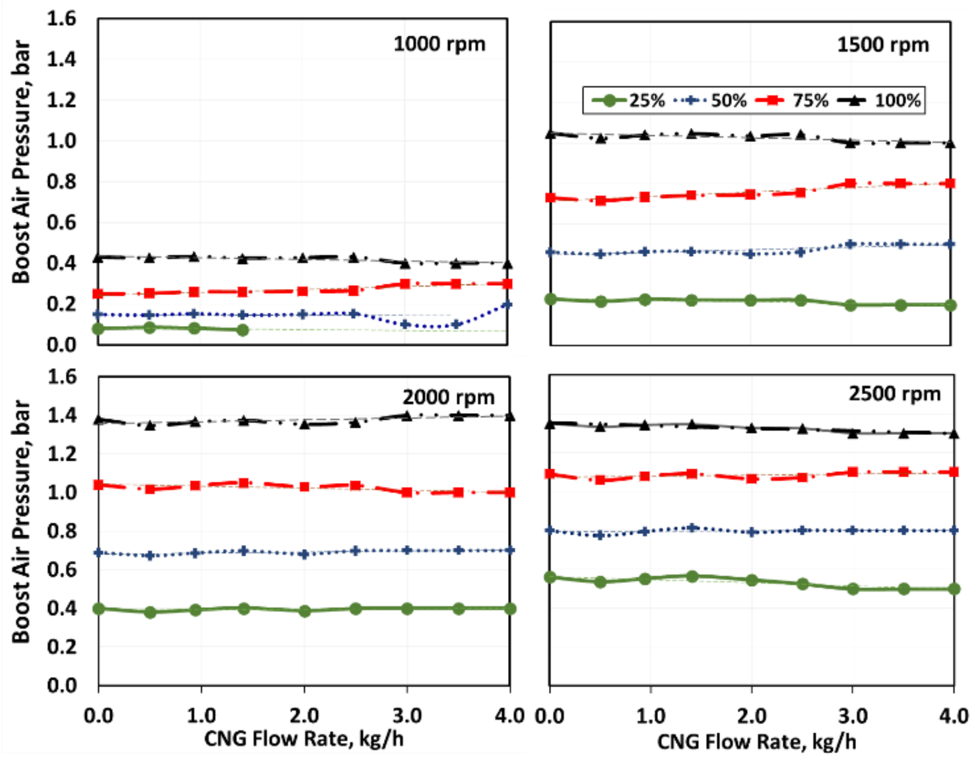


Fig 5.38: Variation in BAP under various loads and speeds.

Throughout the research, the intake air pressure and temperature were kept at 1000 mbar and 25 ^{o}C, respectively. The turbocharger on the selected heavy-duty engine is integrated into the exhaust manifold. The compressor of the turbocharger is driven by the pressure and heat from the exhaust gas, this, through the use of kinetic and pressure energy, results in an increase in BAP and BAT. Before to the intercooler and following the turbocharger, measurements were taken for the BAP and BAT. The behavior of BAT, as shown in *Fig 5.37*, indicates that it increases with the rise in engine speed but decreases as the load increases.

Based on the findings presented in *Fig 5.37*, it can be observed that the BAT demonstrates a positive correlation with engine speed, indicating that as the speed of the engine increases, BAT also increases. Conversely, BAT exhibits a negative relationship with increased load, suggesting that as the load on the engine rises, BAT decreases. Furthermore, under intermittent load conditions, BAT tends to decrease, whereas under full load conditions, it tends to increase. Nevertheless, during DDF operation there is no discernible correlation between the increase of BAT and the augmented CNG flow rate. Throughout various engine speeds and loads, the BAT consistently showed the lowest values when the flow rate was set at 1.41 kg/hr. It is worth mentioning that during the range of 1000-1500 rpm, an increased BAT was detected at a 4.0 kg/hr CNG flow rate. Conversely, for speeds ranging from 2000-2500 rpm, a higher BAT was recorded with a 0.65 kg/hr CNG flow rate.

Under various load-speed circumstances, the association between BAP and DDF shows an upward trend as the load conditions intensify, peaking at 100% load. As the speed increases, BAP also exhibits a corresponding rise. However, when the load is at its maximum, BAP is observed to be lower at higher speeds and higher at 2000 rpm. Interestingly, under DDF, the introduction of increased CNG flow rate results in a decrease in BAP at all speeds for both lower and higher load conditions. Nevertheless, at intermediate loads, there is a slight increase in BAP despite the implementation of DDF. At engine speeds between 1000 and 1500 rpm, an increase in BAP was noted when the CNG flow rate was set at 4.0 kg/hr, whereas using a flow rate of 1.41 kg/hr between 2000 and 2500 rpm led to a higher BAP. While assessing various fuels at higher engine speeds, it was observed that the BAP remained relatively consistent across all load conditions, despite some fluctuations in DDF at lower engine speeds.

As illustrated in *Fig 5.39*, an increase in load was accompanied by an increase in exhaust temperature generally, although it has increased with increased speed up to 1500 rpm and then reduced. Furthermore, during high-load high-speed and low-load low-speed operations, exhaust temperature decreased with DDF, whereas for all other loads and speeds, it increased proportionately to an increase in CNG flow rate. The temperature range achieved with DDF was approximately 236-536 ^{o}C, which falls within the optimal range for effective functioning of the exhaust after-treatment equipment. The utilization of CNG results in an elevation of the exhaust gas' exergy, potentially affecting the turbocharger owing to increased energy supply to the turbine. At a 4.0 kg/hr CNG flow rate and an engine speed of 1000 rpm, the highest exhaust temperature was recorded. Conversely, the lowest temperature of the exhaust was recorded at 25% load, also at 1000 rpm, with a 1.41 kg/hr CNG flow rate.

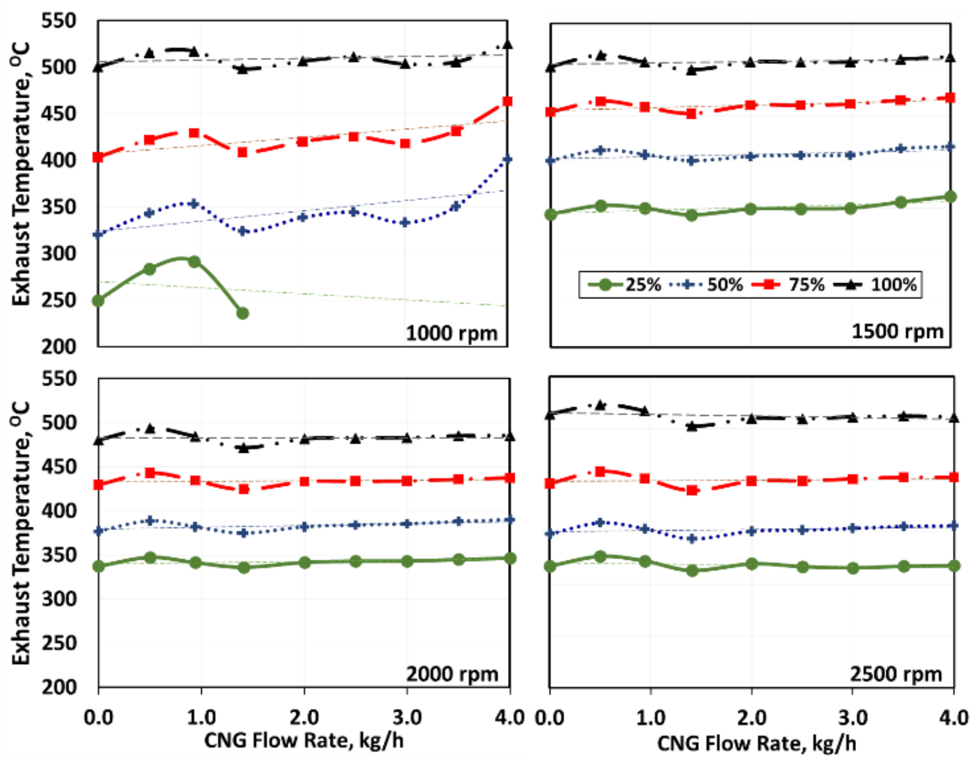


Fig 5.39: Exhaust Temperature Variation under different loads and speeds.

5.3.2.3. Brake Thermal Efficiency

Brake thermal efficiency (BTE) refers to the assessment of an engine's braking capability relative to the energy provided by the fuel, and it is based mostly on thermal energy rather than chemical energy [53]. The variation of BTE with DDF at various speed-loads is depicted in *Fig 5.40*.

As can be seen in the graph, BTE increased with increasing load, however at lower speeds, the highest observed BTE was noted when the load ranged from 50 to 75%. All fuels evaluated under DDF show a tendency of increasing BTE with increasing load. While there is an improvement with load, maximum BTE was noted at engine speed of 1500 rpm and thereafter reduced till 2500 rpm as engine speed increased. Furthermore, it was noted that augmenting the CNG flow rate during the DDF process resulted in a reduction of BTE at various speeds and loads.

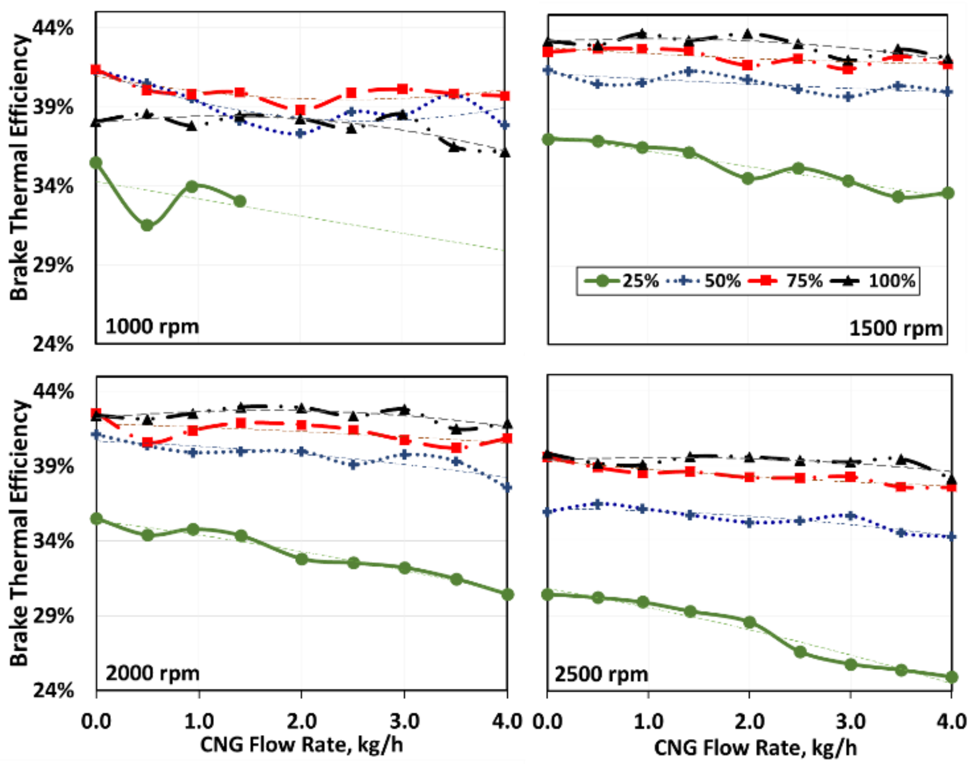


Fig 5.40: Brake Thermal Efficiency Variation under different loads and speeds.

Certain aspects were enhanced by replacing CNG with a higher proportion in premixed combustion, but it hindered the advancement of flame propagation, resulting in a decline in thermal efficiency [55]. In particular, the BTE achieved its highest value of 44% when operating at 100% load and 1500 rpm, accompanied by a 0.94 kg/hr CNG flow rate. Conversely, the lowest BTE of 42% was observed when the flow rate was set at 4.0 kg/hr. According to the results, when operating at a 25% load and an engine speed of 2500 rpm, the DDF mode demonstrated the lowest BTE with a 4.0 kg/hr CNG flow rate. Conversely, the maximum BTE was achieved with pure diesel under the same operating conditions but at a 30% load. In terms of BTE, diesel outperformed CNG flow rates of 1.41 and 0.93 kg/hr, respectively. This result suggests that the engine was specifically designed for diesel, and therefore, a higher BTE can be expected when using neat diesel.

The decline in BTE with increasing CNG flow rates can be attributed to the combustion process deviating from the optimum conditions. With increasing flow rate of CNG, there is a greater tendency for the combustion to occur in a premixed phase, leading to earlier combustion. As a consequence, deviating from the optimal diesel combustion point would lead to a decreased BTE, considering that the engine was initially intended for diesel fuel utilization. The impact on BTE is greatly influenced by the possible shift from Compression Ignition combustion to flame propagation, making it a critical element to consider.

5.3.2.4. Volumetric Efficiency

The air-fuel ratio supplied to the cylinder regulates the combustion process of an engine. Increasing the amount of air introduced results in improved torque and power as a result of enhanced fuel combustion. Volumetric efficiency (VE) refers to the ability of an engine to manage real air volume, rather than the volume it was originally designed for. The relationship between VE and DDF, as observed *Fig 5.41* , demonstrates a decline in VE as engine speed increases. However, there is a slight increase in VE at 1500 rpm under specific load conditions. The decreased duration needed for expelling exhaust gases and utilizing fresh fuel mixture at higher engine speed can be credited for this phenomenon. The trend of VE varies for different load conditions at different speeds. At an engine speed of 1000 rpm, there is a decline in VE as the load intensifies. Nevertheless, when the speed of the engine falls within the range of 1500 to 2000 rpm, VE initially rises until it reaches 50% load, after which it starts to decline. On the other hand, when the engine speed reaches 2500 rpm, there is an observed correlation between load increase and a corresponding increase in VE.

Furthermore, when considering DDF operation, the VE initially decreases for CNG flow rates up to 1.5 kg/hr. However, beyond this threshold, VE begins to increase across all speeds and engine load conditions. It is important to note that as the flow rate of CNG increases, a certain amount of it replaces the intake air. Surprisingly, despite the substitution of intake air with CNG, VE experiences a boost. This peculiar phenomenon can be attributed to the rise in BAP under DDF, which subsequently leads to elevated combustion and exhaust temperatures.

Increasing the BAP yields a decreased delay period, and when coupled with DDF, the intake air becomes a homogenous combination of air and CNG, ready for efficient combustion. This results in a rise in combustion temperature, which correlates with the noticeable enhancement of VE at higher flow rates of CNG. Specifically, when using neat diesel fuel and a 4.0 kg/hr CNG flow rate, the average VE reaches around 220%. Conversely, when the flow rate is decreased to 1.41 kg/hr, the VE shows a minor decline, stabilizing at 217%.

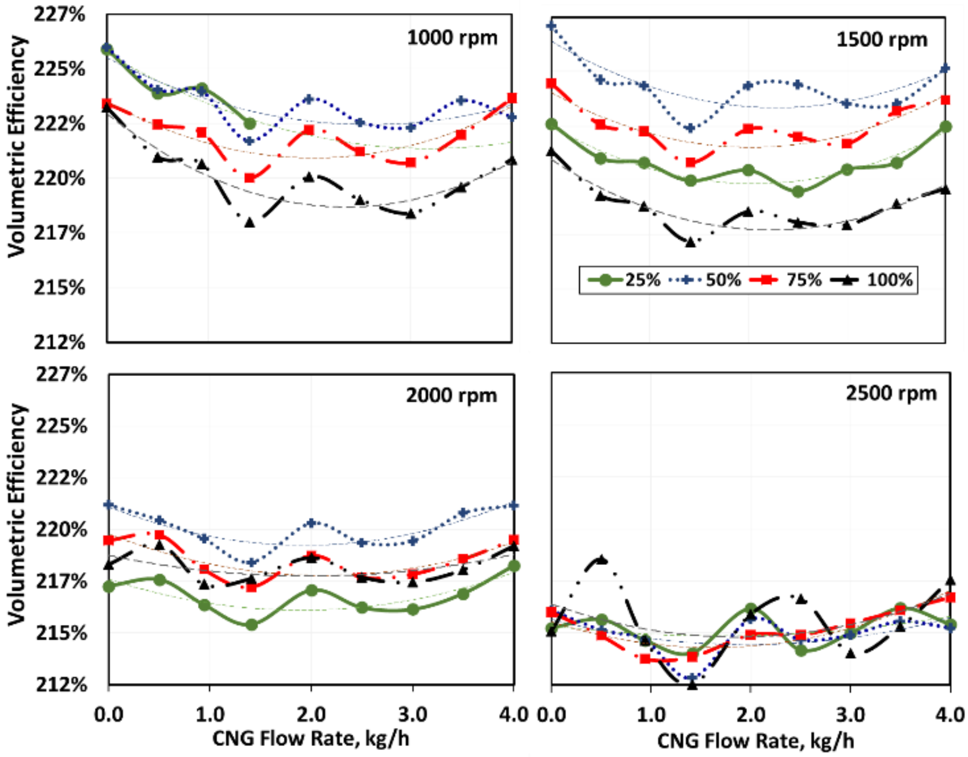


Fig 5.41: Volumetric Efficiency Variation under different loads and speeds.

5.3.2.5. Brake Specific Fuel/Energy Consumption

When multiple fuels with different calorific values are used, the brake specific fuel consumption (BSFC) is utilized as a metric to evaluate the fuel efficiency of an engine. Furthermore, the investigation delves into brake specific energy consumption (BSEC), offering a comprehensive analysis of its correlation with BSFC and CNG mass flow rate across various engine loads & speeds (refer to *Fig 5.42* & *Fig 5.43* for a graphical representation). The study reveals a pattern akin to that observed in conventional diesel engines, where BSEC and BSFC initially decrease until reaching a specific speed, beyond which they start to rise.

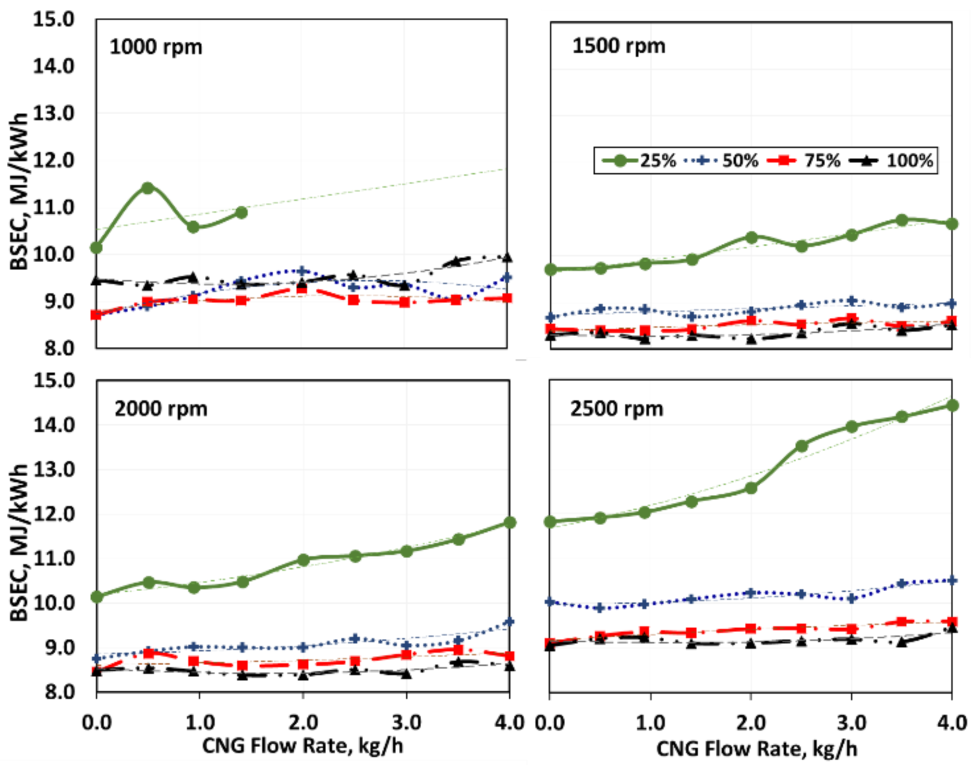


Fig 5.42: Variations in brake-specific energy consumption at varying loads and speeds.

The investigation revealed that when the engine's load rises, there was a simultaneous decrease in both BSEC and BSFC across all speeds, indicating enhanced efficiency. Notably, when operating at 25% load, BSEC and BSFC demonstrated a positive correlation with CNG's flow rate. As the rate of CNG flow increased at all speeds, both BSEC and BSFC also increased. However, when operating at full load (100%), BSEC and BSFC initially decreased until reaching a specific range of CNG flow rates, specifically between 1.41 and 2.0 kg/hr. Beyond this range, BSEC and BSFC began to increase.

Likewise, under DDF operation, BSEC and BSFC exhibited consistent behavior within the partial load range of 50% to 75%. The research indicated that by utilizing CNG with a higher LHV and replacing a smaller mass of CNG in DDF, significant reduction in BSFC and BSEC were achieved. Nevertheless, when the replacement rates were increased, the Brake Power (BP) decreased, leading to reductions in BSEC and BSFC.

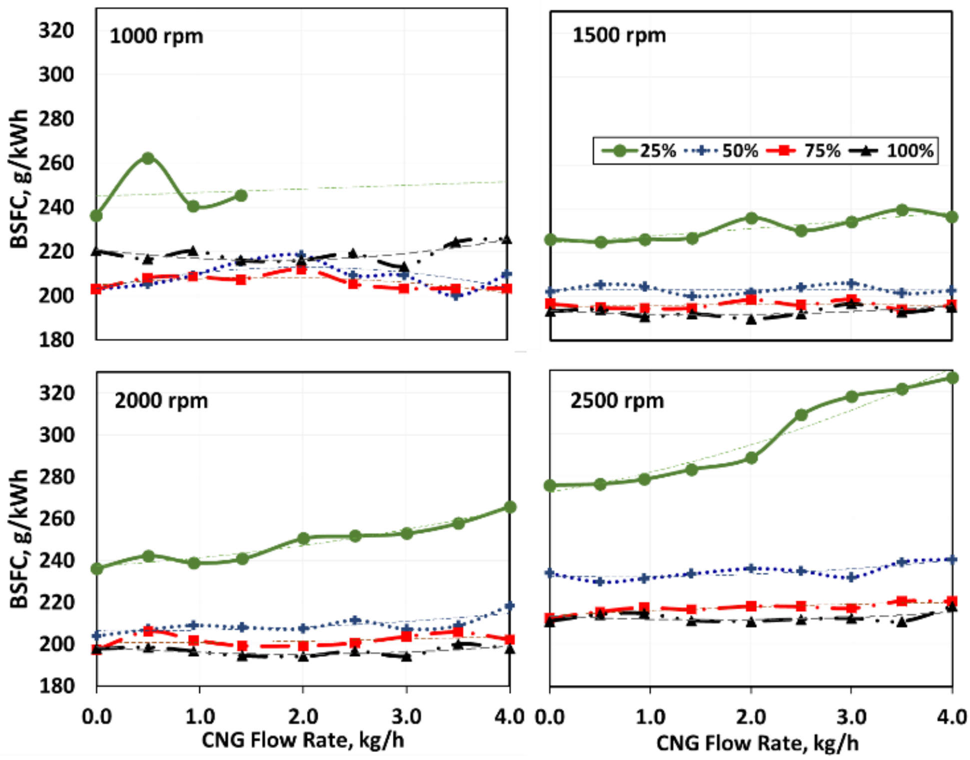


Fig 5.43: Variations in brake-specific fuel consumption at varying loads and speeds.

5.3.3. Mass Emission Performance

As mentioned earlier, the measurement of regulated emissions was conducted both before and after implementing selective catalytic reduction (SCR) through the utilization of a mass emission analyzer. To enhance comprehension with regards to regulatory requirements, the emission values initially recorded in parts per million (ppm) were subsequently converted to a mass basis.

In the light and heavy-duty vehicle categories, SCR is utilised to reduce NOx emissions, whereas, in the passenger car segment, EGR or LNT is employed. A liquid reductant agent, such as ammonia solution, is injected via a specific catalyst in an SCR to reduce NOx, and the degree of reduction is determined by flow rate of DEF. This section primarily addresses the impact of the engine's emission performance on the SCR system, which is commonly employed to mitigate NOx emissions.

5.3.3.1. CO Emission

When comparing diesel engines to gasoline engines, it can be observed that diesel engines typically employ lean combustion, characterized by lower carbon monoxide (CO) emissions and a higher air-fuel ratio. However, during lean conditions, there may be a slight formation of CO due to chemical kinetic effects, diesel's larger droplet size, or inadequate swirl/turbulence in the combustion chamber [55]. Moreover, SCR's effects on CO emissions have hardly ever been studied, and research has pointed out that its influence on CO reduction is relatively insignificant in comparison to its effect on nitrogen oxide (NOx) reduction [56]. *Fig 5.44* & *Fig 5.45* depicts the alterations in CO emissions during DDF combustion and the consequent impact of SCR

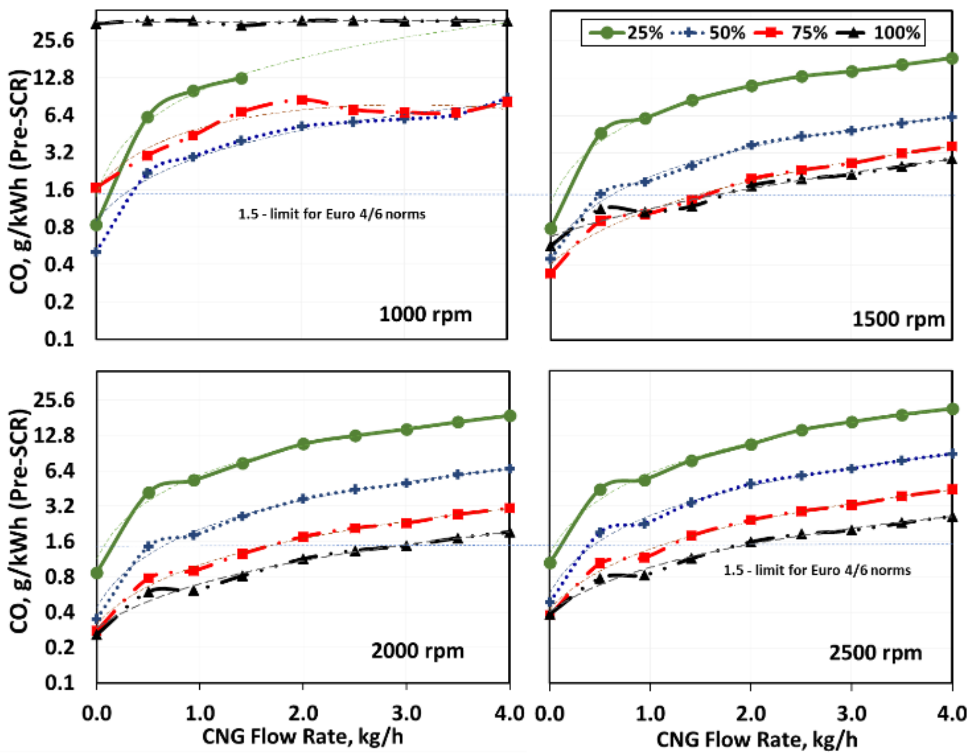


Fig 5.44: CO Emission under Pre-SCR at varying loads and speeds.

According to the findings, it is evident that at lower speeds, there is a notable initial level of CO emissions, which gradually decreases as the speed increases. Nevertheless, when the engine functions under various load circumstances, with the exception of 1000 rpm, CO emissions decrease as the load increases at all speeds. Additionally, at 1000 rpm, CO emission decreases under intermittent loads but increases under full load. Furthermore, it was discovered, as cited in references [35] [68] [78] [49] [63] that when operating in DDF mode, increasing the CNG flow rate causes CO emissions to rise.

For heavy-duty engines, emission legal testing under Euro 6 and Euro 6 requirements involves using the World Harmonized Stationary Cycle and the European Steady-state Cycle (13-mode). These regulations impose a CO emission limitation of 1.5 g/kWh. Although this restriction is not directly relevant to the study undertaken, it is important to acknowledge that CO emissions were comparatively lower during particular DDF operations when specific speed and load conditions were maintained. These emissions were well within the legal limits defined by the legislation. It is crucial to emphasize that the extended duration of the intake stroke within the cylinder can result in the entrapment of CNG within the crevices by the incoming CNG-air mixture. This phenomenon could potentially influence the emission patterns that are observed. The trapped CNG participates in the post-combustion process as it is released during the expansion stroke, where it cannot completely oxidise because of the lower temperature, and thus CO emissions increase as the flow rate of CNG increases [71]. The CO emissions exhibited a significant disparity depending on the fuel and operating conditions. During the testing conducted at 2000 rpm and load condition of 100%, the CO emissions from running on neat diesel fuel were found to be remarkably low, measuring only 0.26 g/kWh. However, when CNG was utilized at a 2.5 kg/hr flow rate, 1000 rpm, and under full load conditions, the CO emissions were alarmingly high, reaching as much as 37.02 g/kWh.

Contrary to what is claimed in the literature, which states that CO emissions were not reduced after SCR, *Fig 5.45* shows that SCR conversion efficiency decreased as engine speed and load increased. The emission pattern of CO closely resembled the measurements taken before the SCR system. The CO values for neat diesel were within the permissible limits set by regulations. Even when using lower flow rate of CNG substitution during DDF operation, the CO levels remained within acceptable limits, especially during intermittent and high load conditions. However, the maximum CO emissions for neat diesel fuel increased post SCR. Nevertheless, the increase in CO levels was notably mitigated with increase in CNG flow rate. After the SCR, there was an average increase of 33.65% in CO emissions when using neat diesel fuel. However, when a 2.5 kg/hr CNG flow rate was used, the rise in CO emissions was significantly lower at 6.36%. Comparing the SCR system's average conversion efficiency of CO emissions to measurements taken directly from the engine, it was found to be (-) 11.84% due to the specific DDF operation.

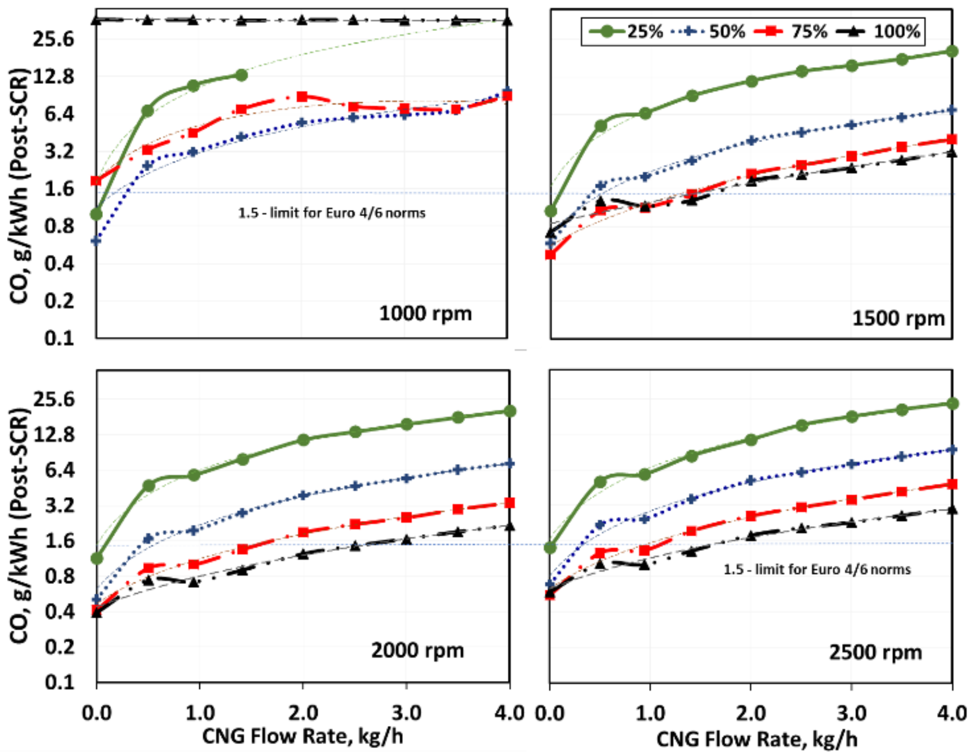


Fig 5.45: CO Emission under Post-SCR at different loads and speeds.

5.3.3.2. NMHC Emission

At temperatures below 1200 K, incomplete combustion of diesel fuel results in hydrocarbon (HC) emissions [109]. The trend of non-methane hydrocarbon (NMHC) emissions varying with DDF before and after SCR is evident in *Fig 5.46* & *Fig 5.47*.

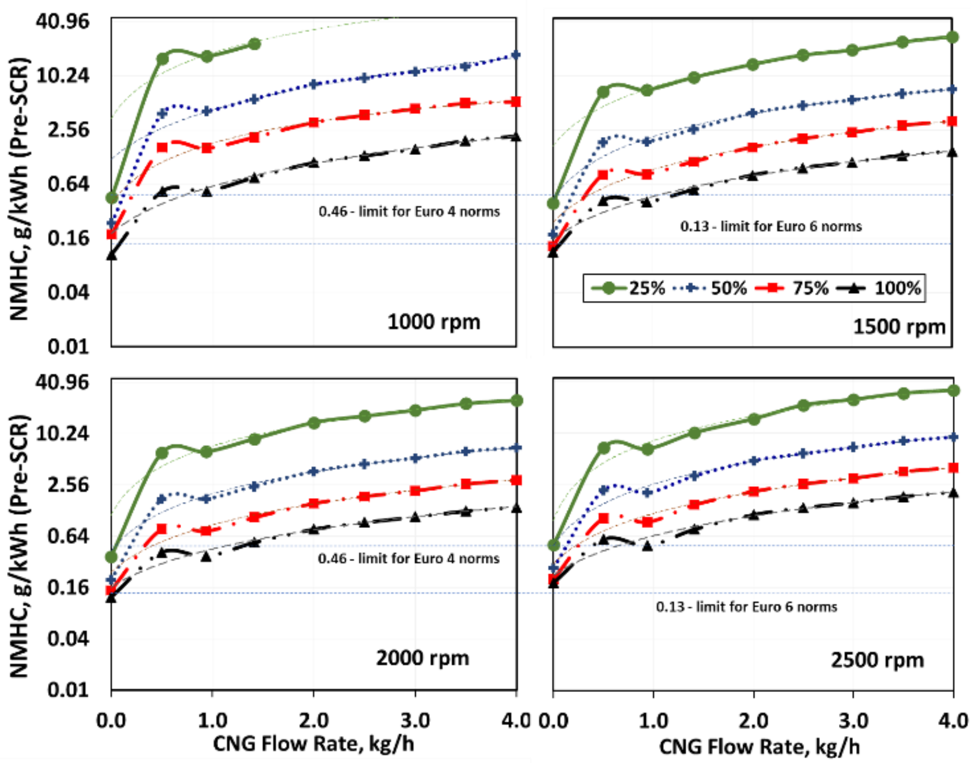


Fig 5.46: NMHC Emission under Pre-SCR at varying engine speeds and loads.

The NMHC was shown to decrease with increasing load, but it increased at lower speeds and remained nearly constant at higher speeds. However, when CNG flow rate was increased, NMHC emission also increased across all engine speeds and loads. Although neat diesel fuel complied with Euro 4 norms, the flow rates of diesel and DDF systems were generally higher, leading to potentially higher emissions [68] [78] [49] [63] [72] [40] [80] [69]. Due to the flame quenching action, CNG becomes trapped in crevices and participates in the later stages of combustion, resulting in unburned hydrocarbons and an increase in NMHC. Moreover, under lower load conditions, the temperature within the combustion chamber is reduced, while the RAFR is elevated, as depicted in *Fig 5.39* & *Fig 5.36.* This combination of factors can hinder the complete propagation of the flame throughout the charge, leading to a higher concentration of NMHC due to the presence of unburned fuel mixture.

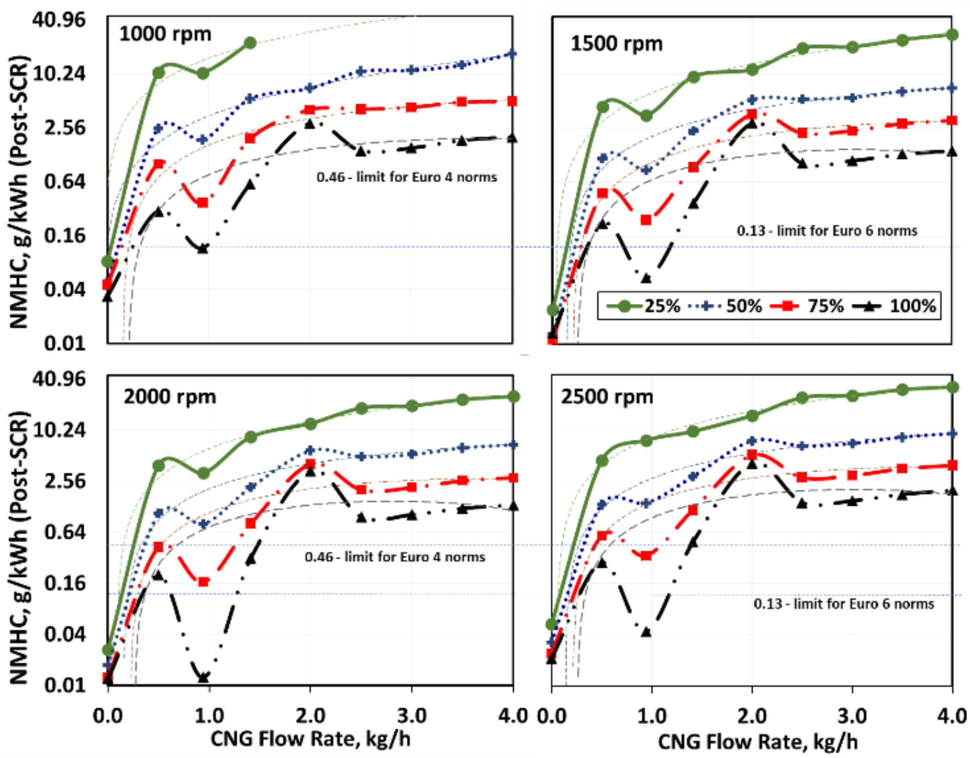


Fig 5.47: NMHC Emission under Post-SCR at different engine speeds and loads.

The impact of SC) on the reduction of NMHC under DDF mode of operation is illustrated in *Fig 5.47*. The system has been specifically tailored for diesel fuel application, and the results demonstrate a significant average reduction of 86.90% in NMHC emissions when utilizing regular diesel fuel. With increasing load, SCR conversion efficiency drops, but with normal diesel, improved conversion is found in the speed range 1500-2500 rpm. When under DDF, the efficiency of SCR improves as the load increases while the substitution rates decrease.

For CNG, there was an unexpected rise in NMHC emissions at a flow rate of 2.0 kg/hr. Moreover, when higher substitution rates were used, there was an elevation in NMHC levels at low loads, but the conversion rate remained negligible at high loads. The SCR system achieved an average NMHC conversion efficiency of 86.9%, 40.68%, 59.92%, 15.8%, -105.48%, -9.64%, 0.7%, 0.95%, and 2.34% when utilising neat diesel and CNG flow rates between 0.65 and 4.0 kg/hr. With SCR technology, the neat diesel was even able to comply with Euro 6 standards. Furthermore, the engine successfully adhered to Euro 4 standards under different speeds and loads when the CNG flow rates fell between the range of 0.65-0.94 kg/hr.

Furthermore, it was noted that following an initial increase in NMHC emissions during DDF operation, the NMHC levels for higher CNG flow rates remained relatively stable across different speed-load conditions. As CNG is the main factor contributing to elevated NMHC emissions during DDF operation, an option to deal with this issue would be the implementation of a dedicated catalyst system positioned optimally before the SCR system, specifically designed for the HC emissions.

5.3.3.3. NOx Emission

Diesel engines emit nitrogen dioxide (NO_2) and nitrogen monoxide (NO) as part of NOx emissions, which pose significant hazards and contribute to air pollution. While NO_2 poses a significantly higher threat, being greater by five times hazardous than NO, NO is responsible for over 90% of NOx among these emissions. Thermal, prompt and fuel related process are the three primary mechanism by which NOx is generated during combustion. The functioning of these mechanisms relies on various elements, including temperature, surplus oxygen, duration of stay, and the level of turbulence inside the combustion chamber [10]. By controlling the combustion in-cylinder parameters, it is possible to affect other pollutants like CO, CO_2, soot, and HC. Hence, managing NOx through after-treatment systems is preferred. Effective reduction of NOx emissions has been demonstrated in passenger cars and heavy duty vehicles using lean NOx traps and SCR. SCR systems commonly employ zeolites and metal oxides as catalysts. However, the specific catalyst coating on the SCR is unclear in this experiment, which involves a commercial diesel engine. The variations in NOx emissions during DDF operation are shown in *Fig 5.48* & *Fig 5.49* under various loads and speeds.

When measured before SCR system, the NOx emissions from the experimental engine exceeded the Euro IV emission limits for the fuels tested across various speeds and loads, as depicted in *Fig 5.48.* This is to be expected, given that in-cylinder characteristics were not adjusted, which would have had a negative impact on other emissions that are handled by the SCR system. The study examined the connection between speed, load, and NOx emissions. It was observed that when operating at lower and intermittent loads, NOx emissions decreased as the speed of the engine increased. However, for higher loads, NOx emissions initially rose up to 1500 rpm before beginning to decline. Interestingly, at 1000 rpm, NOx emissions remained relatively steady as the load increased, but they decreased at maximum load. Conversely, at different speeds, NOx emissions increased until reaching 75% load and then started to decrease.

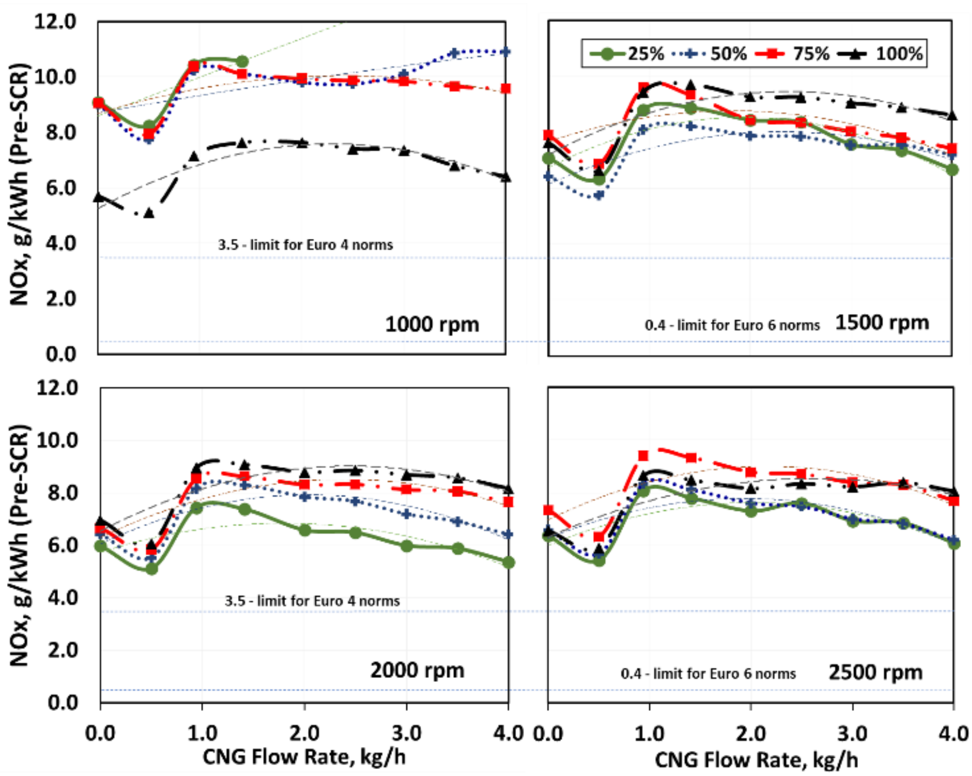


Fig 5.48: NOx Emission under Pre-SCR at different loads and speeds.

The findings presented in *Fig 5.48* indicate that during DDF operation, compared to conventional diesel, a 0.65 kg/hr flow rate of CNG produced fewer NOx emissions. Furthermore, the NOx emissions increased with increasing flow rate of CNG up to 1.0 kg/hr before starting to decrease, regardless of the load speed. The increased NOx emissions witnessed during DDF operation can be ascribed to the thermal-NO process, that is affected by factors such as temperature of in-cylinder, air composition, and the rate of air-fuel mixture formation (*Fig 5.36* & *Fig 5.39*). It was discovered that switching to CNG slightly boosted the RAFR [10] [110]. The higher in-cylinder temperature and atomic oxygen mole fraction in DDF contribute to a higher NO mole fraction, resulting in increased NOx emissions [70]. Conversely, an increase in flow rate of CNG results in a rise in combustion mixture's heat capacity, causing the average temperature to be lower at the compression stroke's end. This decrease in temperature ultimately leads to a reduction in NOx generation.

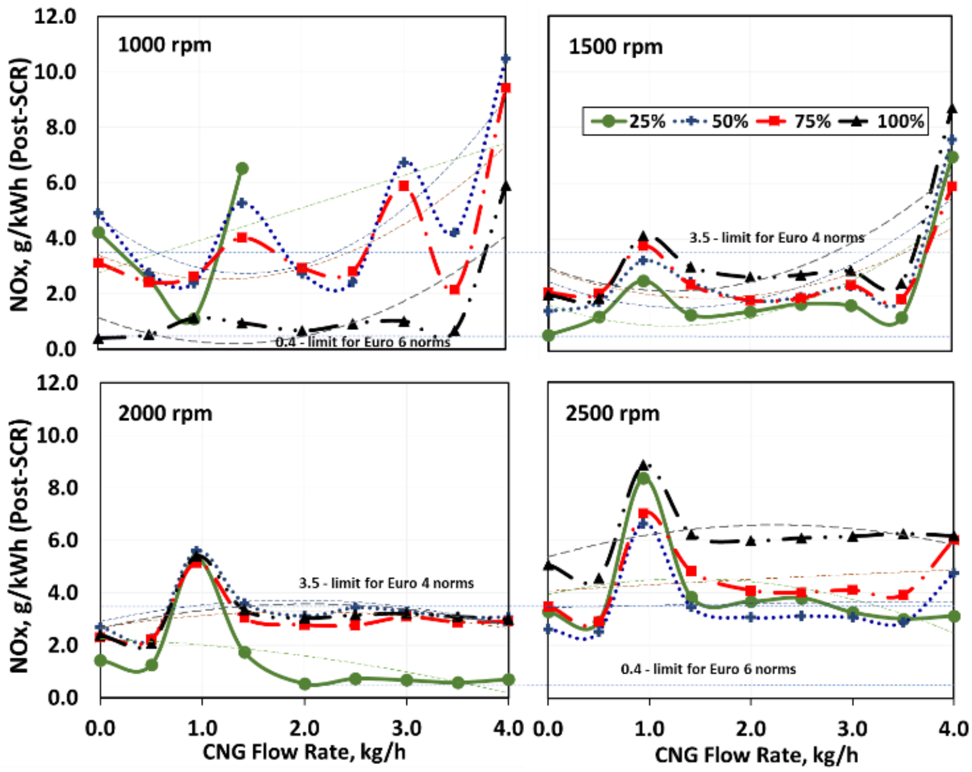


Fig 5.49: NOx Emission under Post-SCR at different loads and speeds.

As demonstrated in *Fig 5.49*, NOx emissions have been considerably decreased for each test fuel with the SCR system. There has been no discernible trend in NOx emissions following SCR since NOx emission reduction is mostly attributable to the injection of DEF, which is regulated by an additional ECU. The DEF is limited based on the NOx/O_2 levels detected by the O_2/NOx sensors placed at the end of SCR system. The DEF flow rate is managed by the secondary ECU to ensure compliance with the Euro 4 limit. It is worth mentioning that, except for specific load conditions with 0.94 and 4.0 kg/hr CNG flow rates, the NOx emissions have decreased for all fuels at lower speeds, specifically 1000 and 2500 rpm, remaining within the Euro 4 limits.

Under normal operating conditions, the maximum conversion efficiency achieved was 93% at high load and low speed. However, during DDF (specific conditions), the conversion efficiency was maximuma at 91.7% with a 2.0 kg/hr CNG flow rate. Additionally, among all the fuels tested, the SCR system demonstrated the highest mean conversion efficiency, achieved at 67.79% at a 2.0 kg/hr CNG flow rate and offering the highest conversion efficiency at 1500 rpm.

5.3.3.4. Methane Emission

Methane (CH_4) is the shortest hydrocarbon, and it is a constituent of THC as an emission. Methane emissions are an issue if utilised in IC engines; while the result of methane emissions is less in terms of ozone formation potential than other hydrocarbon emissions, they are a potent greenhouse gas emission (GHG) with a larger warming potential than CO_2 [111]. Studies of THC and NMHC are common, but methane emissions under DDF operation has been scarcely studied, and *Fig 5.50* & *Fig 5.51* shows how methane under DDF affects. As expected, methane emissions with DDF are significantly greater than those of regular diesel, that is essentially non-existent and troublesome [74]. From *Fig 5.50*, it can be observed that as speed increased, methane emission increased with normal diesel while it reduced with DDF; nevertheless, as load went up, methane emission reduced for all the fuels tested. Methane emissions for DDF were also found to be much higher in low load conditions than for conventional diesel, additionally under other load scenarios. In accordance with the earlier mentioned information, because of the increased CNG flow rate, there was a rising link between methane emissions. The highest emission levels were detected when the flow rate of CNG reached 4.0 kg/hr across various speeds and loads, except under the 25% load condition. Notably, at 25% load and 1000 rpm, when the CNG flow rate was 1.41 kg/hr, methane emissions were measured at 25.78 g/kWh. This occurrence can be ascribed to the heterogeneous combustion process occurring within diesel engines, wherein the ignition of diesel fuel results in the combustion of a mixture of CNG and air. Consequently, methane is released from crevices as a byproduct.

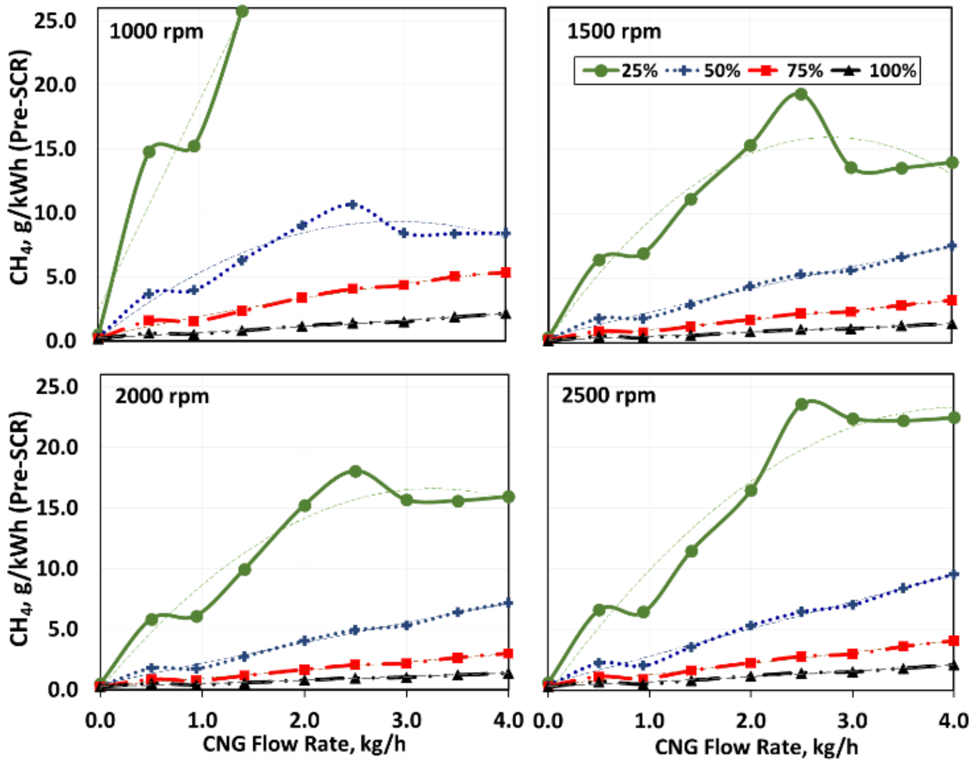


Fig 5.50: CH_4 Emission under Pre-SCR at different loads and speeds.

Methane emissions have dropped with DDF at lower CNG replacement ratios and are practically negligible with regular diesel, as shown in *Fig 5.51*. In the study conducted, it was noted that at increased rates of CNG flow, the recorded levels of methane emissions prior to SCR which was higher either remained unchanged or experienced minimal increases. This observation was particularly evident when the flow rates reached 3.0 kg/hr and beyond. It is worth mentioning that SCR systems are not primarily designed to address methane emissions. Nevertheless, during DDF operation, a maximum conversion efficiency of 57.4% has been observed, while regular diesel combustion has achieved conversion efficiencies as high as 90% when methane concentrations were very low. Due to the decreased efficiency at higher speeds, DDF mode is expected to result in higher methane emissions. Hence, the most efficient strategy for maximizing the use of CNG in DDF mode involves running the engine at a reduced substitution rate [112].

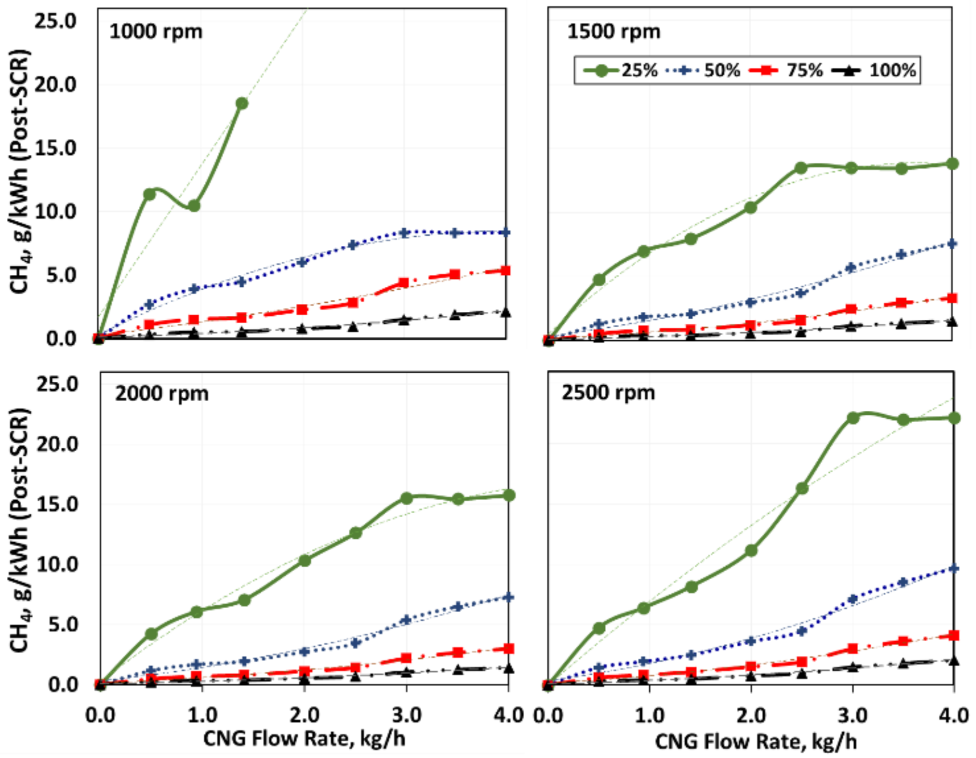


Fig 5.51: CH_4 Emission under Post-SCR at different loads and speeds.

5.3.3.5. CO_2 Emission

CO_2 levels in exhaust emissions are mostly determined by the fuels' chemical constitution and oxygen's availability during combustion. The most energy-efficient engine is the diesel engine, which can reach near-zero emissions while having a higher fuel efficiency, suggesting reduced CO_2 emissions.

Fig 5.52 & *Fig 5.53*, shows how CO_2 emissions vary with DDF, demonstrating that as speed increased, the levels of CO_2 emissions exhibited a decline up to 1500 rpm, after which they experienced an upward trend across all loads and CNG substitution [58]. DDF analysis revealed that increasing CNG substitution lowered CO_2 emissions for all load and speed conditions [79] [80]. This might be connected to CNG's reduced carbon content, as well as reduced pumping and friction losses [81]. CO_2 emissions decreased with increasing load conditions in general, but at lower speeds, for all fuels, they decreased until 50% load, after which they marginally increased. Furthermore, in comparison to every other load situation, 25 % load produced significantly more CO_2.

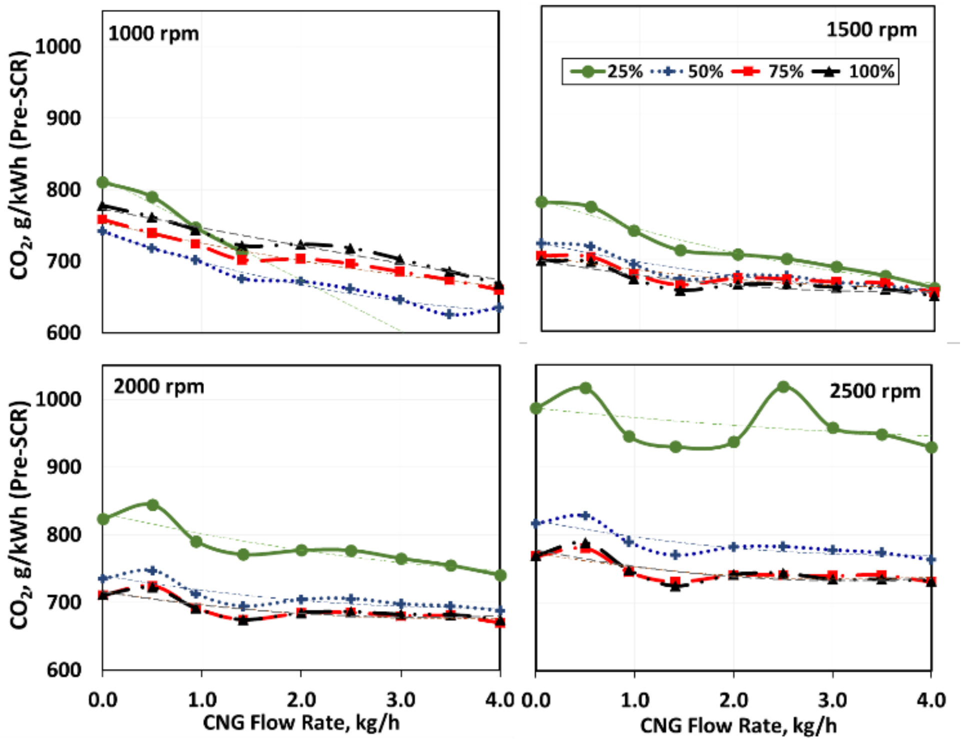


Fig 5.52: CO_2 Emission under Pre-SCR at different loads and speeds.

In an SCR system, the hydrolysis of DEF results in the production of CO_2, however with DDF, as shown in *Fig 5.53*, there is a minor decrease in CO_2 emissions after SCR with all test fuels. In terms of CO_2, the SCR conversion efficiency was 3.3–10.2 %, with better conversion at 2500 rpm, 25% load, with a 2.5 kg/hr CNG flow rate. Moreover, there was a positive correlation between the augmentation in conversion efficiency and the rise in CNG flow rate. The most substantial reduction in emissions was notices at a 4.0 kg/hr CNG flow rate, which also contributed to lower CO_2 emissions. Additionally, the use of CNG lowered the carbon content of the air-fuel blend by 20%, which decreased CO_2 emissions in the DDF system [58] [82].

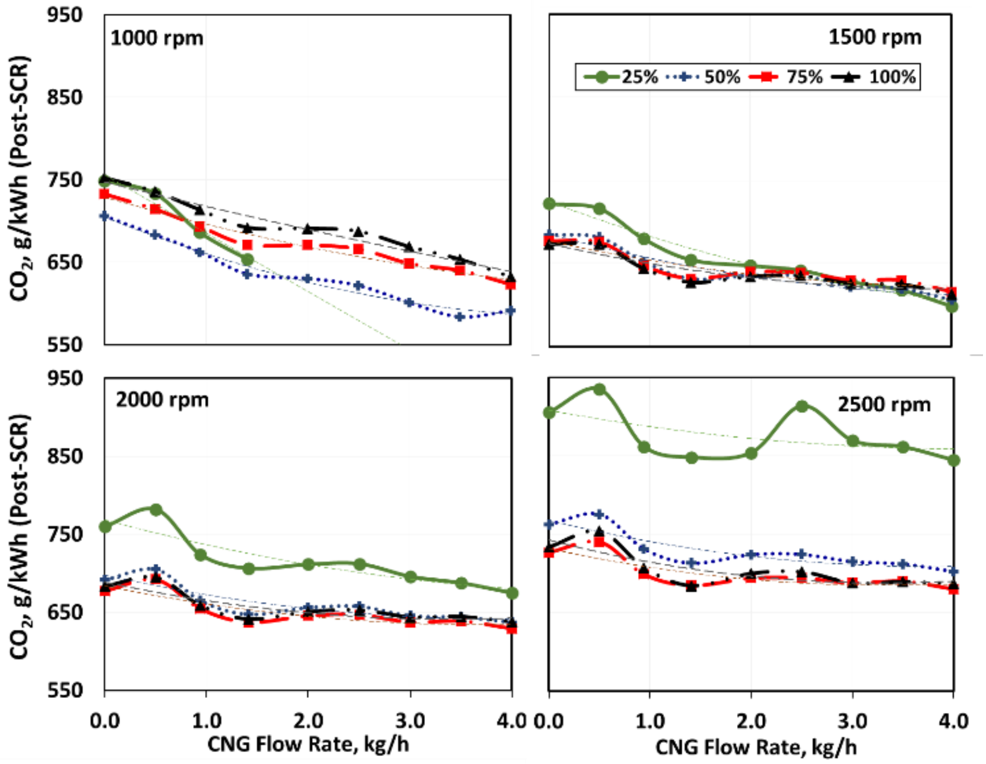


Fig 5.53: CO_2 Emission under Post SCR at different loads and speeds.

5.3.4. Optimization of Diesel-CNG blend/ratio

In the preceding sections, we conducted a comprehensive analysis of the impact of DDF on engine performance and emissions. To accommodate diverse operating situations, performance optimisation was also done for varied CNG flow rate combinations. These optimizations were based on several parameters such as BSEC, BSFC, power, VE, BTE, post-SCR THC, CO, NOx, methane, and CO2 emissions. To assess how these variables affect engine performance, we employed the linear normalization technique as under:-

$$x' = \frac{x - \min(x)}{\max(x) - \min(x)} \quad \text{Eqn. (6.7)}$$

Where, x = specific value for the given condition, i.e., speed and load, x' = ranges between 0-1 for the given condition.

Table 5.3 is a computation instance for neat diesel.

Table 5.3: Normalisation Illustration.

Parameters	Min	Max	Neat Diesel	Normalised Value
BSEC	10	11	10.1	0.00
BSFC	236	262	236	0.00
Power	16	17	16.2	1.00
BTE	32%	35%	35%	0.00
VE	222%	225%	225%	0.00
CO Post	1	13	1.0	0.00
THC Post	0	23	0.08	0.00
NOx	1	7	4.2	0.57
CH_4	0	19	0.1	0.00
CO_2	654	749	749	1.00
Normalised Data for Neat Diesel				2.57

Table 5.4 displays the cumulative normalized data obtained after performing computations on individual fuel samples under various operating conditions, and Table 5.5 shows the feasible DDF combinations for various operating conditions based on the lowest value. For DDF engines, the green colour denotes an ideal CNG flow rate and a lower normalised value in terms of the performance of engine and emission.

Table 5.4: Normalized Data of DDF under different loads and speeds.

Engine Speed, rpm	**% Load**	**Neat Diesel**	**CNG flow rate in kg/hr**							
			0.65	**0.93**	**1.41**	**2.1**	**2.5**	**3.0**	**3.5**	**4.0**
1000	25	2.57	6.24	4.60	7.09					
	50	3.31	3.32	3.89	5.80	5.94	5.55	6.25	4.76	6.80
	75	2.20	3.97	4.30	4.72	6.35	4.69	4.97	4.48	5.33
	100	2.95	2.32	3.42	2.50	3.64	4.29	2.91	6.27	7.29
1500	25	2.00	3.98	3.80	2.67	4.68	5.74	7.05	5.92	7.15
	50	2.08	2.19	2.22	3.31	5.76	4.42	6.71	4.88	6.89
	75	2.57	4.38	3.03	3.35	3.48	4.78	5.97	5.11	7.04

	100	2.22	2.38	3.06	3.88	4.45	6.22	5.06	7.07	7.74
2000	25	1.86	3.16	4.35	4.23	4.44	5.18	4.55	4.57	6.26
	50	1.50	4.29	4.44	2.96	3.60	4.76	5.82	7.20	5.61
	75	1.50	4.29	4.44	2.96	3.60	4.76	5.82	7.20	5.61
	100	3.50	3.74	4.26	2.98	3.26	4.53	3.86	6.16	5.27
2500	25	1.25	1.94	2.91	2.77	2.57	6.99	6.89	6.59	7.15
	50	2.31	2.55	3.42	3.70	4.37	4.56	4.82	5.69	6.81
	75	1.16	4.11	5.09	4.11	5.35	4.76	4.73	6.36	6.49
	100	1.42	2.64	3.81	3.04	3.18	3.69	4.20	4.56	6.71

Table 5.5: DDF Combination that has been optimised for different conditions of operation.

Engine Speed	1000	1500	2000	2500
% Load	CNG flow rate in kg/hr			
25	Neat diesel	1.41	1.41	2.1
50	0.93	1.41	0.65	1.41
75	Neat diesel	1.41	2.1	Neat diesel
100	3.0	2.1	2.1	1.41

The objective of the study was to enhance the efficiency and emission characteristics of a diesel engine operating on DDF mode, while keeping the engine components and OEM settings unchanged, and maintaining comparable performance levels. The objective was to ensure that the normalised value of the DDF engines remained the lowest or deviated no more than 40% from the performance of diesel engines. In the subsequent section, the detailed information regarding the validation process for the enhanced CNG flow rate matrix is presented. The results indicate that the DDF engine exhibited comparable performance to diesel operation, while the emissions remained well within the boundaries defined by Euro 4 regulations.

5.4. OPTIMIZATION OF DDF ENGINE BASED ON ENGINE PERFORMANCE AND EMISSIONS UNDER EUROPEAN STATIONARY CYCLE

In this segment, we delve into the impact of various CNG flow rates evaluated under ESC, focusing on both engine performance and emission performance. Subsequently, by evaluating the overall performance across different test modes, a statistical tool was employed to optimize the CNG flow rate that matches or exceeds the performance of diesel fuel operation. Finally, this process was concluded by finalizing an implementation matrix.

5.4.1. Emission Legislation and its importance in India

Automotive emission legislation plays a crucial role in India, aiming to control and reduce the harmful pollutants emitted by vehicles. In a country with a rapidly growing population and increasing vehicular population, addressing vehicle emissions is of utmost importance to combat air pollution and its detrimental effects on public health and the environment [113].

Automotive emissions legislation in India has undergone significant developments over time [114]. In 1991, the government introduced emission standards known as Bharat Stage (BS) norms, initially based on European regulations. These norms were periodically revised to address the increasing pollution levels. In 2010, the implementation of BS IV across major cities and BS III in other regions aimed to reduce vehicle emissions [115]. Further, in 2017, the government announced a leap to BS VI, equivalent to Euro 6 standards, to combat pollution effectively [116]. This transition mandated the adoption of advanced emission control technologies, such as DPF and SCR, leading to cleaner and more environmentally friendly vehicles [117].

ESC (European Stationary Cycle) testing for heavy-duty diesel engines is mandated by the BS-IV emission standards [114]. The performance and emission parameters of the engine are evaluated in this test using simulated real-world driving situations. A number of transient and steady-state operating modes, such as idle, acceleration, deceleration, and constant speed segments, make up the ESC cycle. Different loads and speeds are applied to the engine to simulate various driving situations. The test evaluates a number of pollutants, including nitrogen oxides (NOx), particulate matter (PM), hydrocarbons (HC), and carbon monoxide (CO). Heavy-duty diesel engines that adhere to the BS-IV requirements are guaranteed to adhere to strict emission limitations, decreasing air pollution and enhancing overall environmental quality.

5.4.2. Effect of Engine Performance under ESC

Directive 1999/96/EC, which governs Euro III emissions, implemented in 2000, brought about significant changes in the measurement of emissions from heavy-duty diesel engines. The ESC test cycle, along with the ETC and ELR tests, were introduced as a result of these modifications. The R-49 test was superseded by the ESC, a steady-state procedure with 13 modes. Previously, the test was also known as the ACEA cycle or OICA cycle.

On an engine dynamometer, the engine is put through a series of steady-state modes for testing *Fig 5.54.*

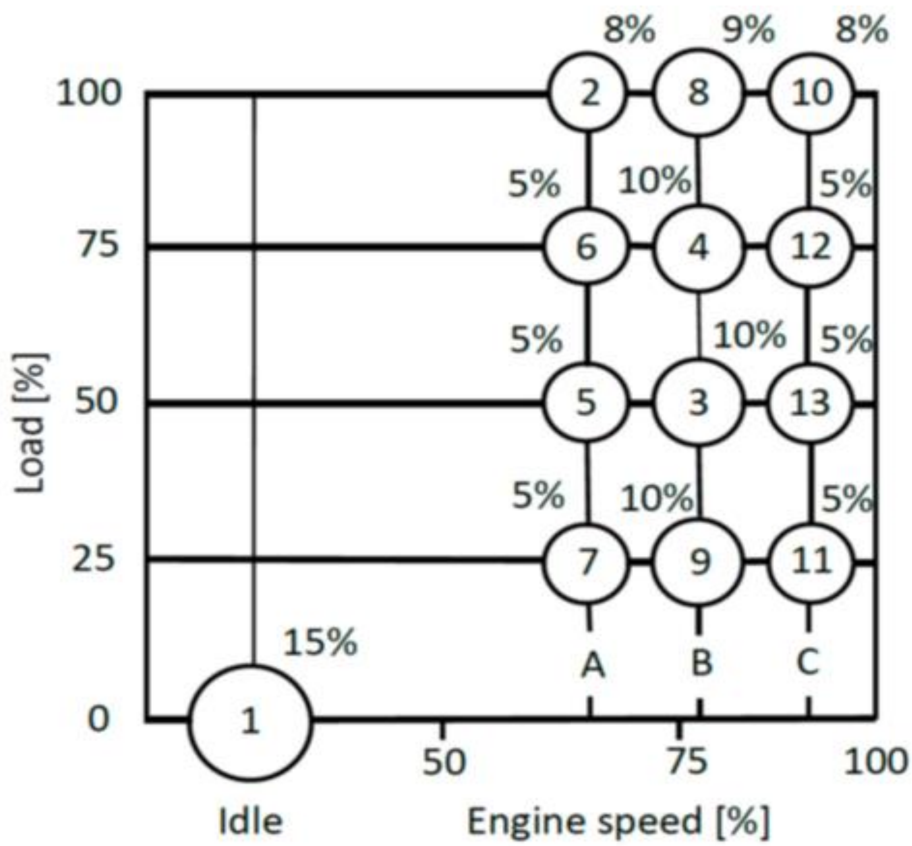


Fig 5.54: European Stationary Cycle Modes.

It is crucial to ensure that alterations in engine speed and load are accomplished within the initial 20 seconds of operation, and the engine must consistently operate for the designated duration in each mode. In order to comply with the testing regulations, it is necessary to maintain the specified torque levels and speed within a tolerance range of 2% of the maximum torque and 50 rpm at the testing speed, respectively. Throughout the cycle, emissions are assessed in each mode and aggregated using a series of weighting parameters. On a single filter, emissions of PM from all 13 modes are monitored. Results for the final emissions are provided in g/kWh.

The following definitions apply to engine speeds:

- The high-speed n_{hi} is calculated at a maximum net power requirement of 70%. The maximum engine speed (more than the rated speed) on the power curve at which this power value is present is referred to as n_{hi}.
- Using 50% of the maximum net power, the low-speed n_{lo} equation is derived. On power curve, n_{lo} is the engine's lowest speed less than the rated speed at which this power value occurs.
- The engine speeds A, B, and C for the test are then determined using the formulas below:

$$A = n_{lo} + 0.25(n_h i - n_{lo})$$

$$B = n_{lo} + 0.50(n_{hi} - n_{lo})$$

$$C = n_{lo} + 0.75(n_{hi} - nl_o)$$

The calculated engine speed and related torque for each mode are shown in *Table 5.6* based on the aforementioned equations. The ESC test is characterised by exceptionally high exhaust gas temperatures and high average load factors.

Table 5.6*: ESC Modes and DDF Engine Operating Conditions.*

ESC Mode	Time, s	Engine Speed, rpm	Torque, Nm	Weightage, %
Idle, 1	240	600	0	15
2	360	1510	812	8
3	480	1889	406	10

4	600	1889	603	10
5	720	1510	403	5
6	840	1510	603	5
7	960	1510	201	5
8	1080	1889	811	9
9	1200	1889	201	10
10	1320	2267	692	8
11	1440	2267	173	5
12	1560	2267	519	5
13	1680	2267	346	5

5.4.2.1. Engine Power under ESC

Fig 5.55 displays the power produced during ESC at various CNG flow rates One could notice that no significant change is observed with respect to the power. This is to be expected as the engine was operated in speed-torque mode while running at a partial load to maintain the desired speed and torque. The engine power as generated will fall as per the power curve of the engine.

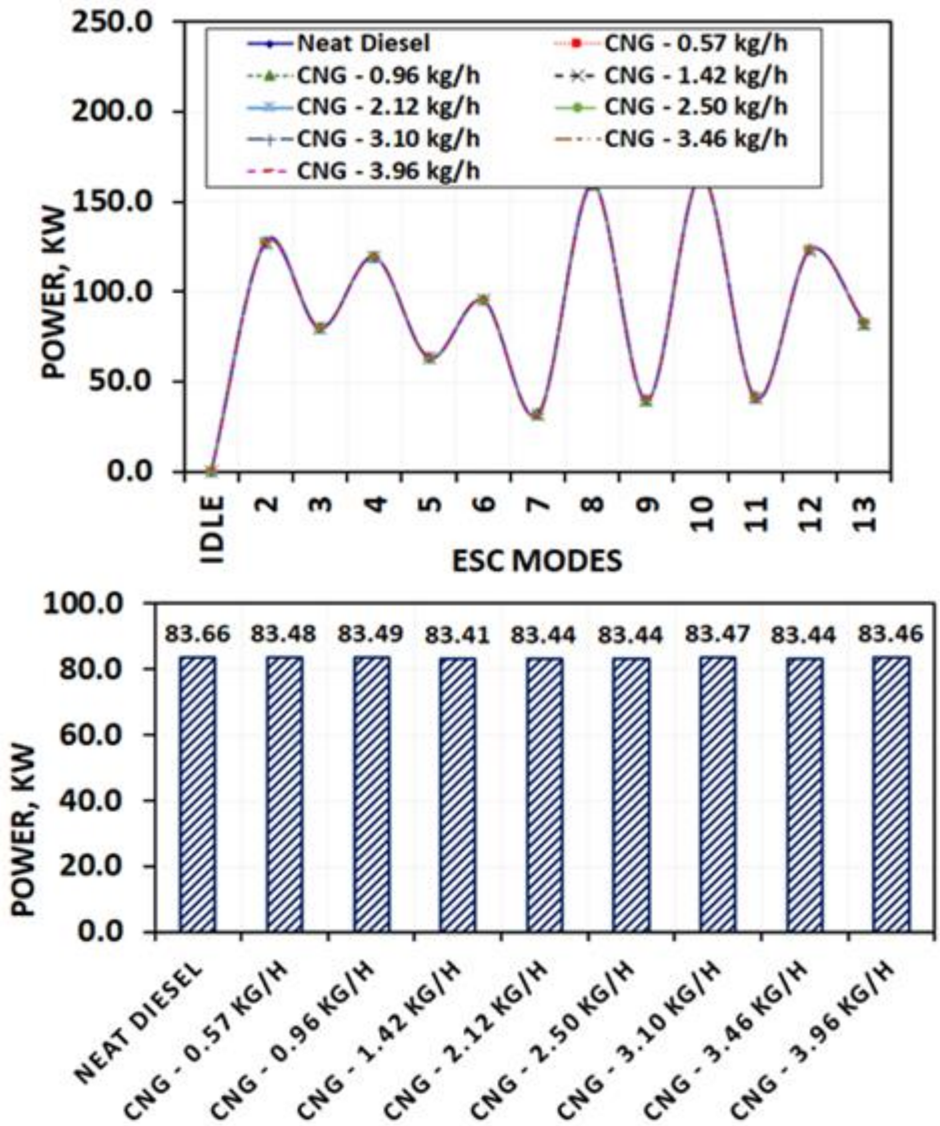


Fig 5.55: Power under ESC for varying CNG Flow Rate.

5.4.2.2. Efficiencies under ESC

The brake thermal and volumetric efficiency of neat diesel and different CNG flow rate under ESC are shown in *Fig 5.56* and *Fig 5.57*. BTE was reduced at different ESC modes with an increase in the CNG flow rate, however, the engine net/weighted BTE of ESC cycle under the DDF's operating mode was almost constant. As CNG flow rate increases, it is anticipated that BTE would decrease because better premixed combustion at higher CNG replacement minimises flame propagation [89]. The volumetric efficiency exhibited an increase of up to 2.1 kg/hr in the flow rate of CNG, while experiencing a decrease in various

ESC modes and the net/weighted volumetric efficiency of ESC was almost constant as seen from figure.

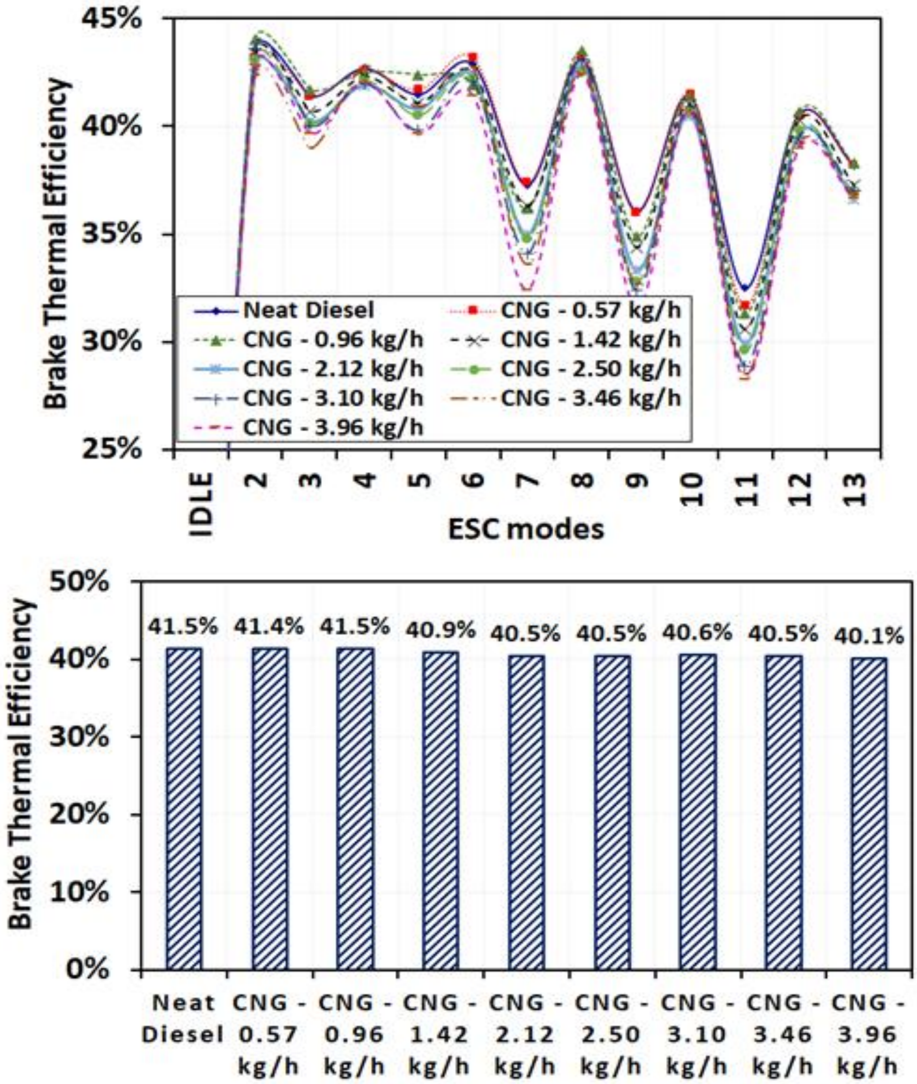


Fig 5.56: Brake Thermal Efficiency under ESC for varying CNG Flow Rates.

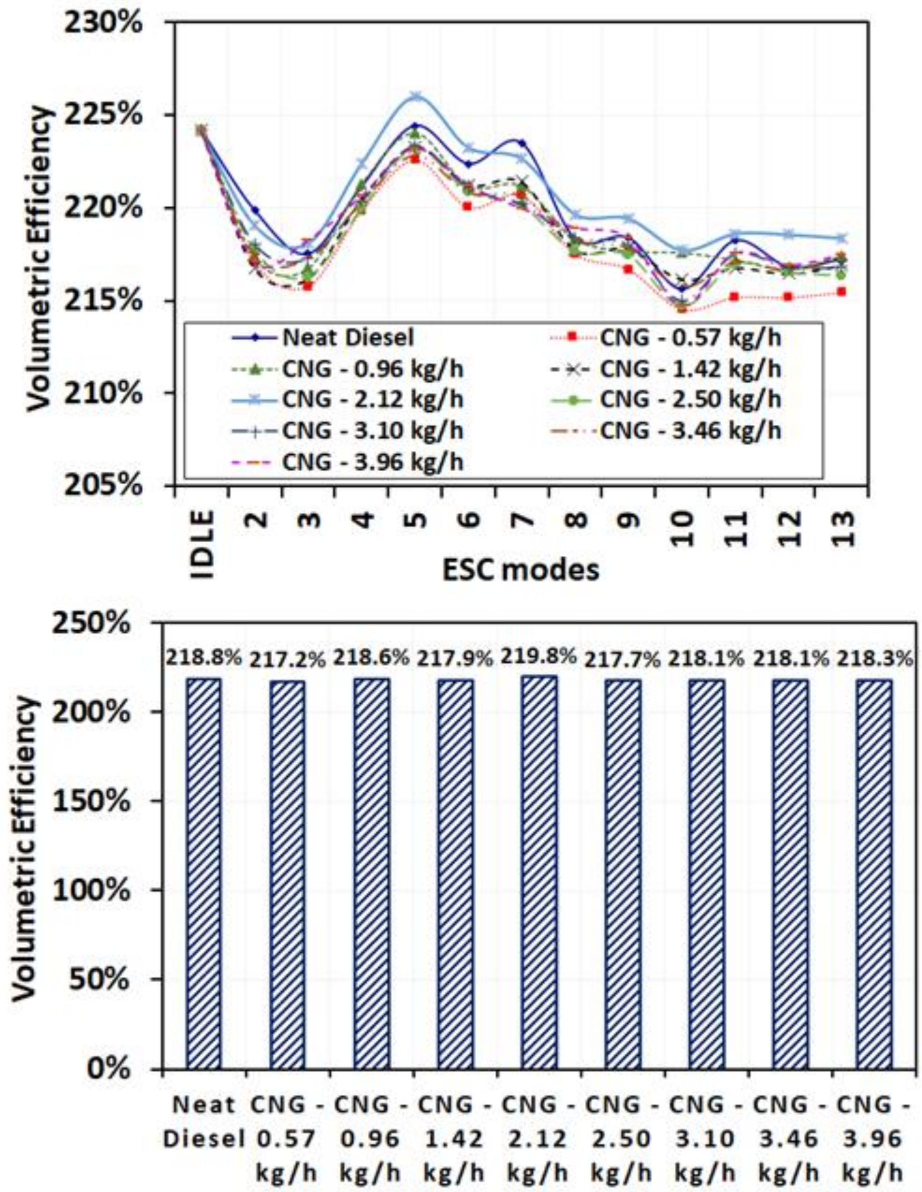


Fig 5.57: Volumetric Efficiency under ESC for varying CNG Flow Rates.

5.4.2.3. Specific Fuel and Energy Consumption under ESC

While an engine's fuel efficiency is expressed in BSFC, when two or more fuels of different energy contents are used, it is expressed in BSEC. BSFC and BSEC of DDF engine running under ESC operation with different CNG flow rate are illustrated in *Fig 5.58* and *Fig 5.59*. The data illustrates a consistent trend wherein both BSEC and BSFC display an upward trajectory in response to higher CNG flow rates. While the weighted BSEC and BSFC are considered, only a marginal increase in the range of 2-3.5% is observed.

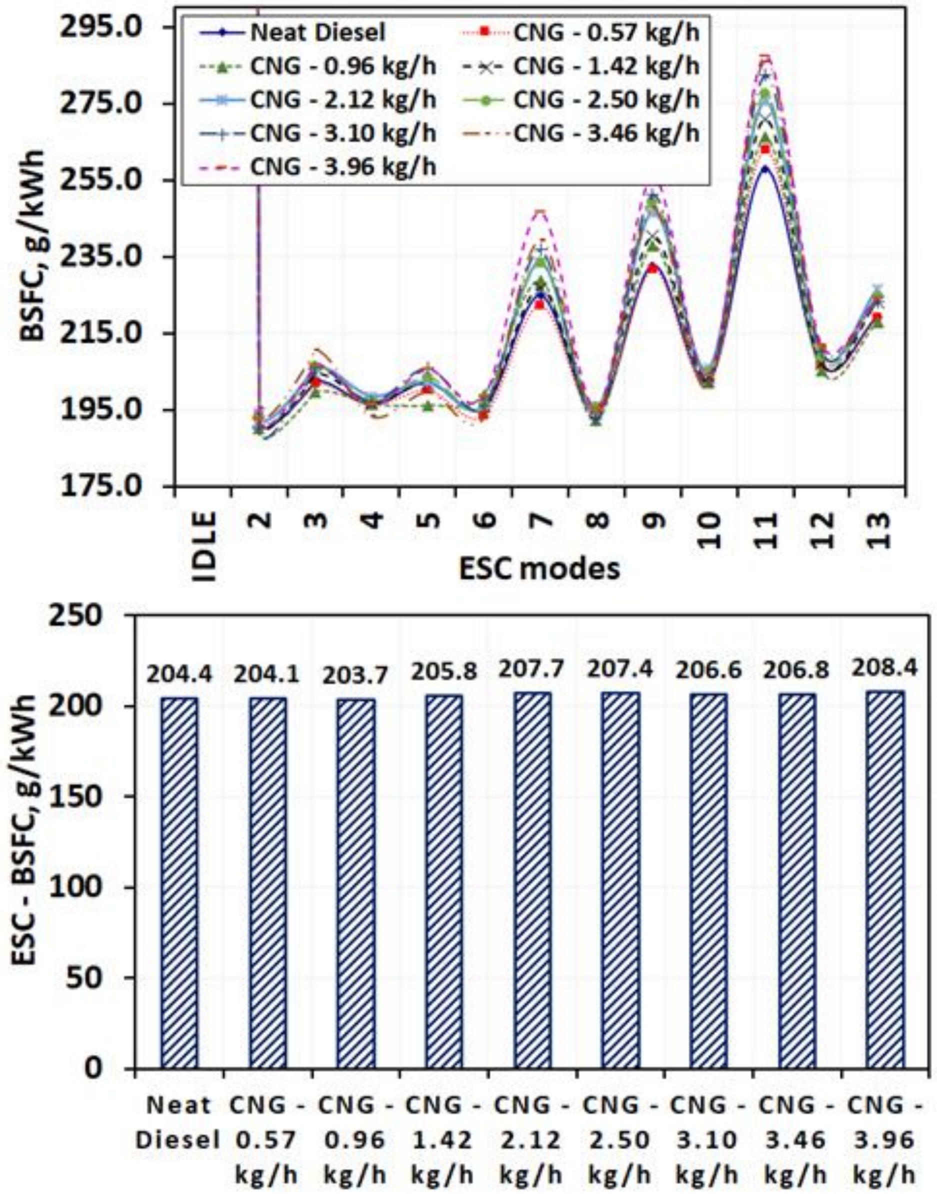


Fig 5.58: Brake-Specific Fuel Consumption under ESC for varying CNG Flow Rates.

5.4.3. Effect of Emission Performance under ESC

The emission was measured after SCR using a mass emission analyzer, as previously described. The ESC is run with different CNG flow rates (constant flow) through an automated system of the engine test bench. During the cycle test, the emission is measured after sufficient stabilisation in a different modes for 30 sec before the end of each mode. This section discusses the emission performance of DDF operation with different CNG flow rates tested under ESC.

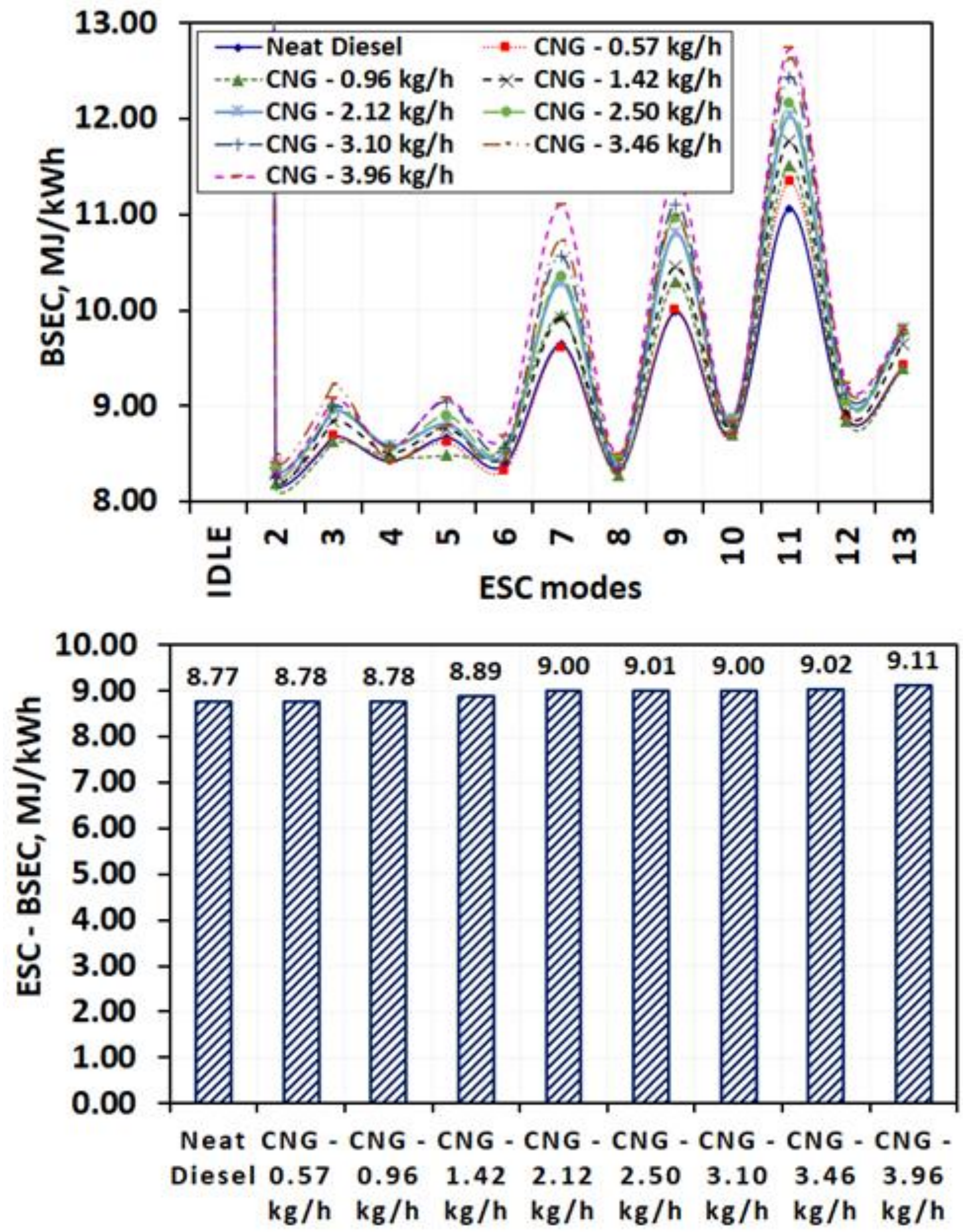


Fig 5.59: Brak-Specific Energy Consumption under ESC for varying CNG Flow Rates.

5.4.3.1. <u>CO Emission under ESC</u>

The trend of CO emission with different CNG flow rates run under ESC is depicted in *Fig 5.60*. The emission of CO exhibits a direct correlation with the escalation of CNG flow rate across all ESC modes. Moreover, the specific CO emission (weighted) of ESC also demonstrates a corresponding augmentation as the CNG flow rate rises. This increase in CO emission is attributed to the soluble organic fraction's partial oxidation and hydrocarbon through the SCR catalyst [59]. Moreover, as the flow rate of CNG increased, there was a corresponding increase in the emission of CO. However, it should be noted that the CO emission stayed below the permissible level of 1.5 g/kWh for flow rates up to 0.96 kg/hr. For higher flow rates, the CO emission was observed to be even higher. This can be attributed to the presence of trapped CNG, which undergoes incomplete oxidation during the post-combustion process due to lower temperatures in the expansion stroke, leading to an increase in CO emissions [71].

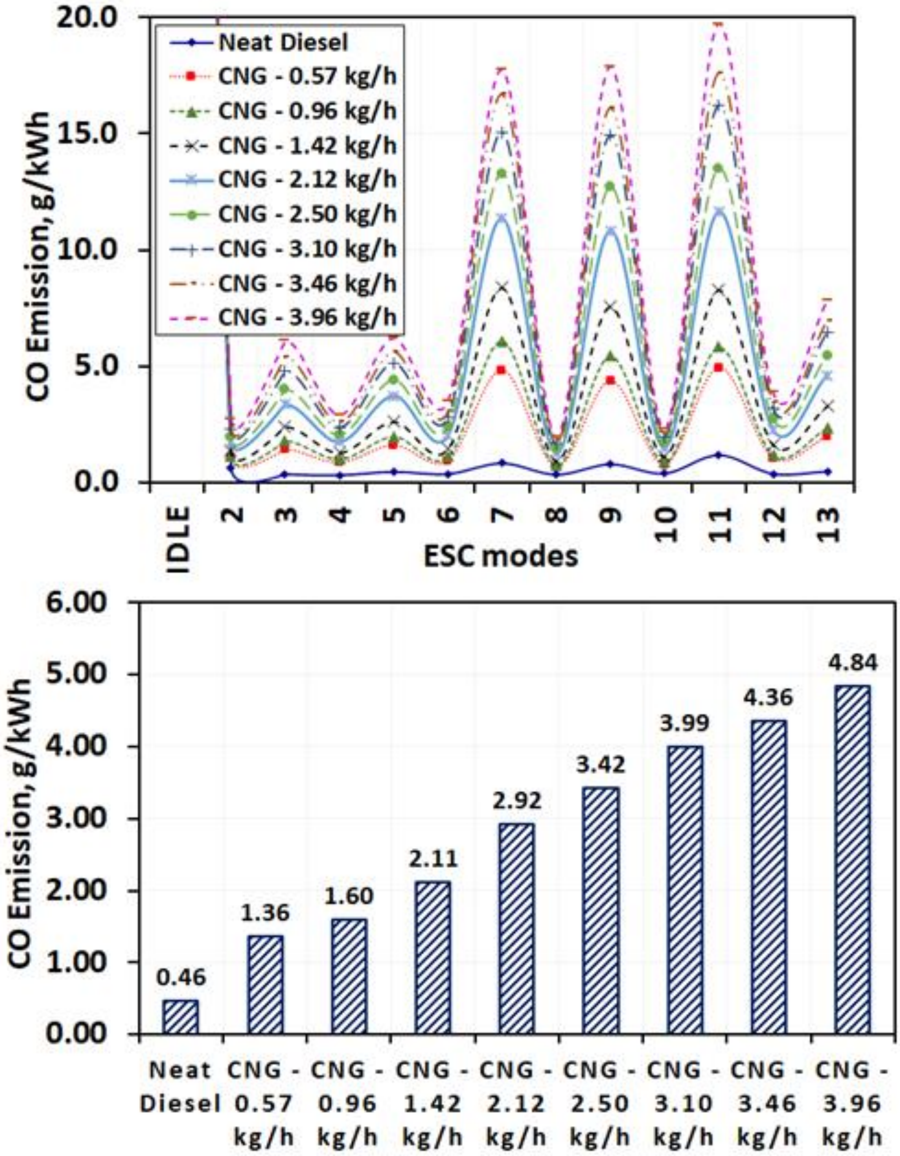


Fig 5.60: CO Emission under ESC for varying CNG Flow Rates.

5.4.3.2. THC Emission under ESC

The variation of THC emission with varying flow rate of CNG run under ESC is shown in *Fig 5.61*.

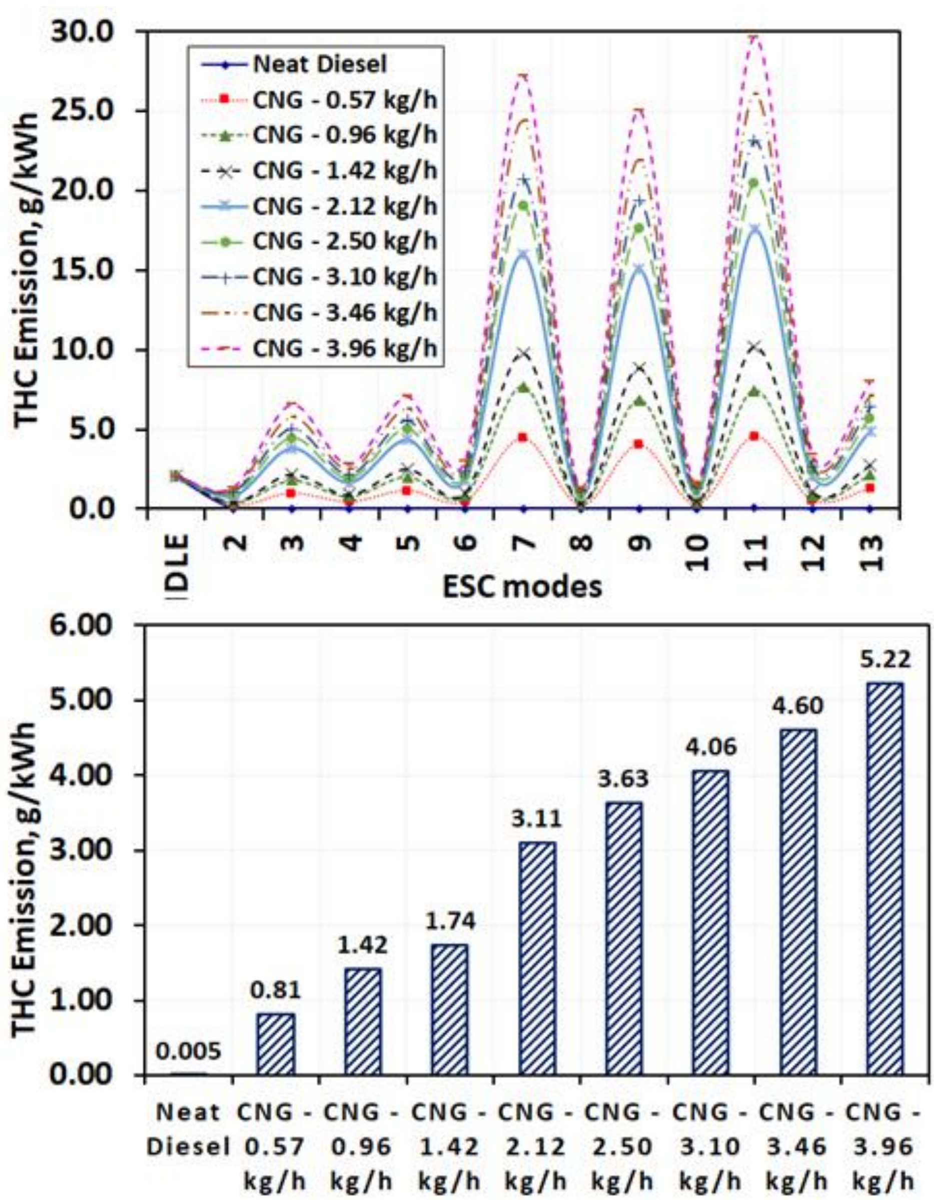


Fig 5.61: THC Emission under ESC for varying CNG Flow Rates.

Similar to CO emission, the emissions of THC also showed an upward trend when the flow rate of CNG increased across all ESC modes. Furthermore, the weighted THC emissions of ESC also exhibited an increase with the CNG flow rate [68] [78] [49] [63] [72] [40] [80] [69]. Furthermore, an escalation in the CNG flow rate resulted in a corresponding elevation in the emission of THC within the permissible limit of 0.46 g/kWh, which was only observed in neat diesel. However, for all CNG flow rates, the THC emission was found to be higher. This increased THC emission due to the CNG flow rate increment can be attributed to three primary factors: (i) quenching of the flame, (ii) the entrapment of the fuel mixture in the crevices of the cylinder during the compression strock process, and (iii) the presence of a higher air/fuel ratio [63].

5.4.3.3. NOx Emission under ESC

The trend of NOx emission under ESC with different CNG flow rate runs is shown in *Fig 5.62.*

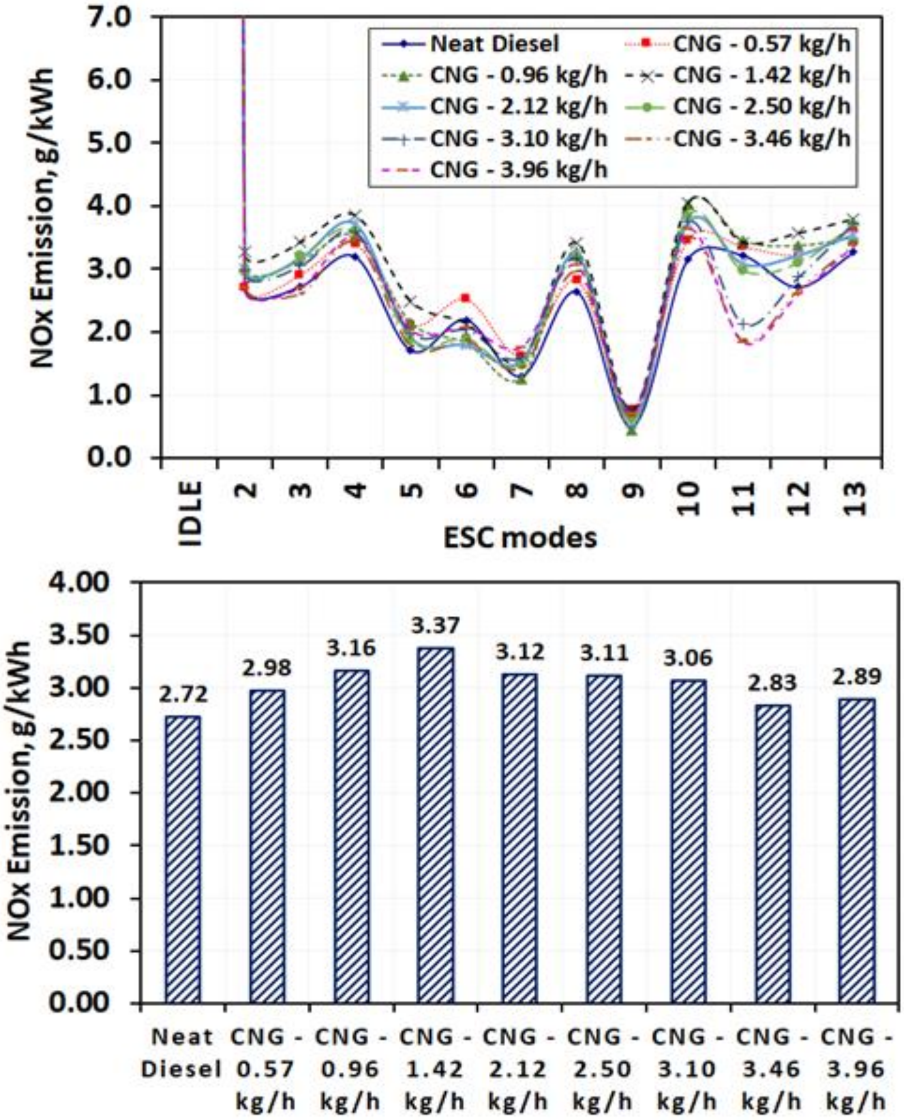


Fig 5.62: NOx Emission under ESC for varying CNG Flow Rates.

It was clear from the figure that NOx emission though marginally increased with an increase in CNG flow rate at different ESC modes, for all fuels, it did not exceed the 3.5 g/kwh regulatory limit. Since the engine is fitted with an SCR system, it is expected that the NOx emission is maintained within the legislation limit with sufficient doping of DEC in the system. Although the NOx emission remained within the legal limits, it experienced an initial rise at a 1.42 kg/h CNG flow rate and subsequently exhibited a decline.

5.4.3.4. CO_2 and Methane Emission under ESC

CO_2 and methane emission with different CNG flow rate run under ESC is depicted in *Fig 5.63* and *Fig 5.64* respectively. It is noticeable that CO_2 has reduced with all of the CNG flow rates and at different ESC modes. The lower CO_2 emission is expected under DDF operation because of lower carbon content in CNG fuel [81]. Further, the carbon content is lower by 20% in the DDF air-fuel ratio and therefore reduction in CO_2 emission [58] [82].

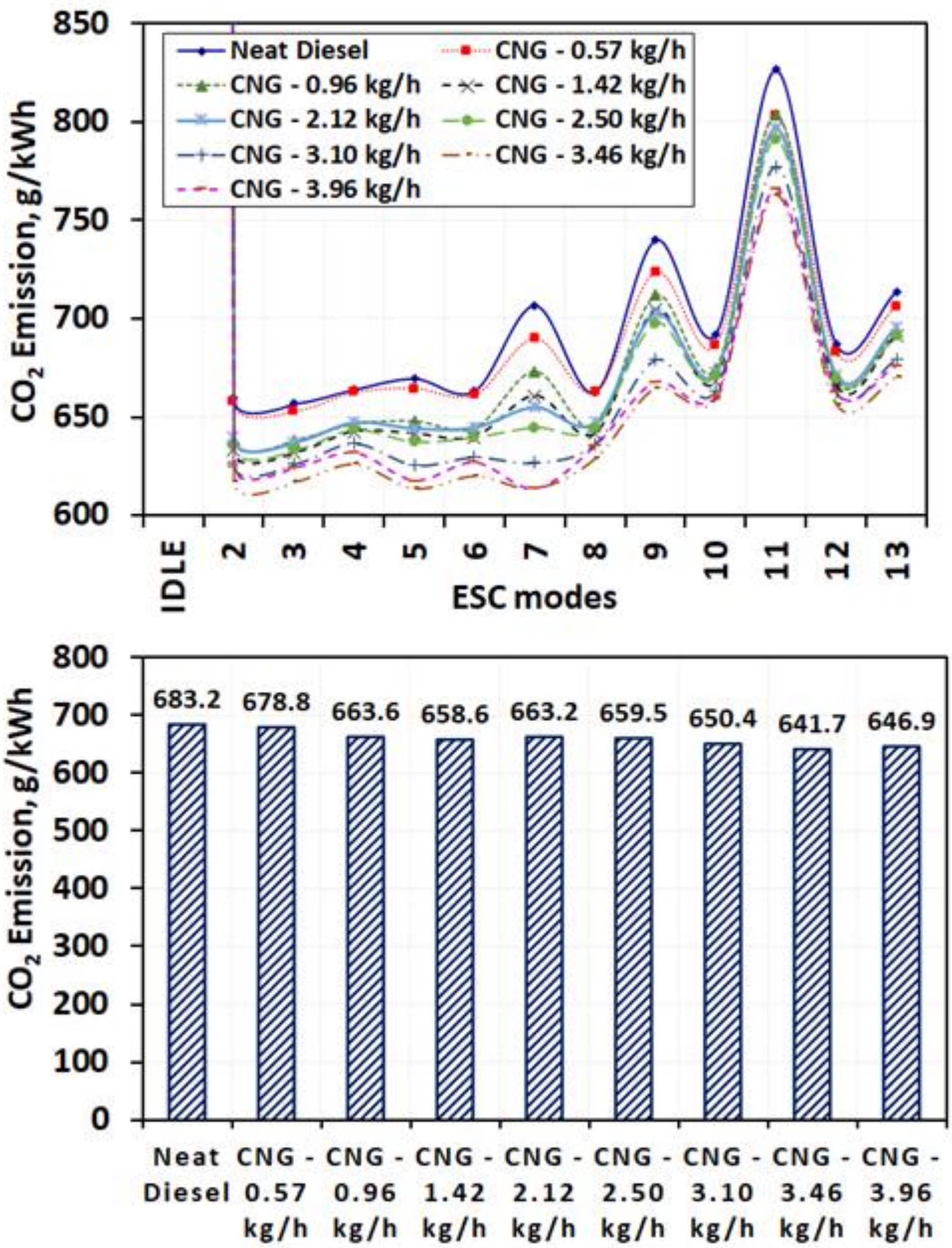


Fig 5.63: CO_2 Emission under ESC for varying CNG Flow Rates.

Alternatively, CH_4 emission increased under ESC modes for all the CNG flow rates. Anticipated increase in CH_4 emissions is foreseen in the context of DDF due to the combustion chamber being filled with premixed CNG, whereby the CNG-air mixture eludes at the overlap interval through the exhaust valve [118]. Additionally, higher CH_4 emissions are attributed to factors such as lower combustion temperature, lowered lean mixture concentration in the cylinder wall and CNG trapping in the crevice area [75] [76]. Although the specific/weighted CO_2 under ESC decreased, methane emissions rose in tandem with the CNG flow rate as it was increased.

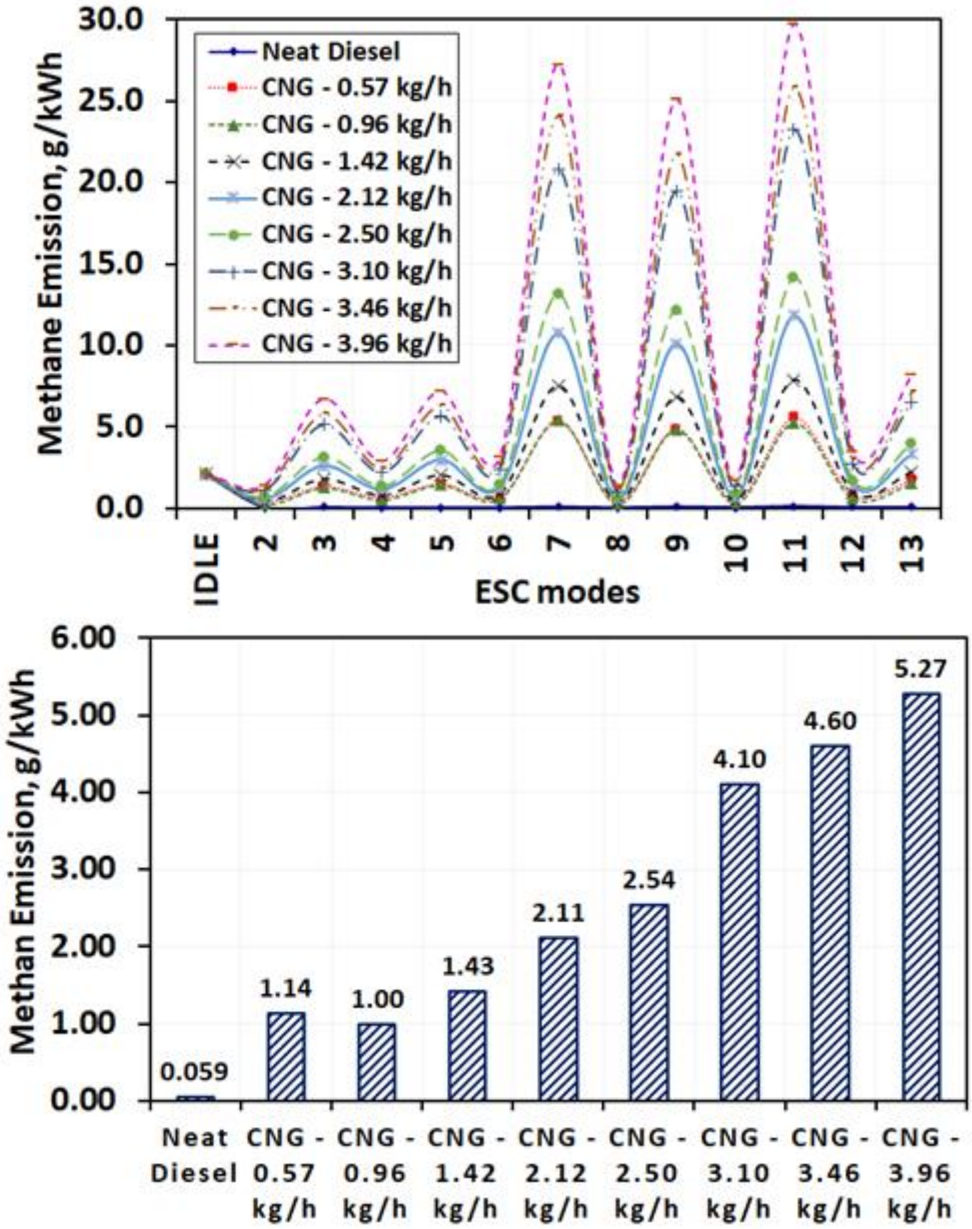


Fig 5.64: Methane Emission under ESC for varying CNG Flow Rates.

5.4.4. Optimization of CNG Flow in respect of ESC and Development of DDF Kit

According to the guidelines outlined in Section **5.3.4**, the optimization of DDF involved experimentation with various combinations of CNG flow rates for ESC Modes while considering different performance parameters. An evaluation of engine performance was conducted using a linear normalization technique, which influenced the results. Table 5.7 displays the aggregated normalized data obtained for each fuel by considering various operating conditions. After conducting an analysis of the lowest observed value, Table 5.8, proposes a potential combination of DDF for ESC mode. The cells highlighted in green indicate the preferred normalized values and the optimal CNG flow rate for achieving the best engine performance and emissions.

Table 5.7: ***Normalized Data of DDF under ESC.***

	Neat Diesel	CNG - 0.57 kg/h	CNG - 0.96 kg/h	CNG - 1.42 kg/h	CNG - 2.12 kg/h	CNG - 2.50 kg/h	CNG - 3.10 kg/h	CNG - 3.46 kg/h	CNG - 3.96 kg/h
2	*1.99*	*5.13*	*3.59*	*5.37*	*6.15*	*5.98*	*5.12*	*7.55*	*7.00*
3	*1.92*	*3.57*	*2.65*	*4.63*	*5.31*	*6.48*	*6.28*	*6.84*	*6.49*
4	*3.28*	*4.33*	*3.46*	*5.40*	*5.88*	*6.17*	*6.53*	*4.31*	*7.07*
5	*2.73*	*3.90*	*2.73*	*5.05*	*5.15*	*6.20*	*6.86*	*5.32*	*7.50*
6	*2.59*	*3.85*	*4.62*	*3.96*	*4.62*	*5.02*	*5.32*	*4.27*	*7.81*
7	*2.05*	*3.78*	*3.63*	*4.18*	*4.23*	*5.92*	*6.18*	*7.06*	*8.50*
8	*2.91*	*4.69*	*3.50*	*6.32*	*5.67*	*7.61*	*5.55*	*6.68*	*7.99*
9	*1.90*	*3.72*	*3.07*	*5.15*	*4.32*	*6.07*	*6.45*	*6.02*	*7.58*
10	*3.24*	*3.00*	*3.57*	*5.27*	*6.62*	*7.64*	*5.64*	*5.04*	*6.76*
11	*2.53*	*4.07*	*4.26*	*4.92*	*5.22*	*5.45*	*6.59*	*6.92*	*7.07*
12	*2.00*	*4.08*	*2.38*	*4.60*	*4.94*	*5.44*	*5.53*	*5.91*	*6.68*
13	*2.57*	*4.30*	*2.78*	*5.76*	*6.50*	*7.09*	*6.86*	*6.17*	*6.35*

Table 5.8: ***Optimized DDF Combination under ESC.***

Mode No	CNG Flow Rate in kg/h
Idle	**Neat Diesel**
2	**0.96**
3	**0.96**
4	**3.46**
5	**0.96**
6	**1.41**
7	**0.96**
8	**0.96**

9	**0.96**
10	**0.57**
11	**Neat Diesel**
12	**0.96**
13	**0.96**

The study's major goal was to increase the diesel engine's efficiency when using DDF while taking engine performance and emission levels into account. Importantly, this optimisation was completed without modifying any engine parts or OEM settings, ensuring adherence to emission standards. The objective of the research was to obtain the lowest normalised value among the DDF choices or, alternatively, to accept a maximum departure from diesel performance of 28%.

5.4.4.1. Development of DDF Kit

The DDF engine was made to operate based on the optimized matrix of CNG flow rates both under ESC and different load/speeds, accordingly, a DDF kit was developed.

Fig 5.65: DDF Engine Setup.

Four CNG injectors were installed on the intake manifold as indicated in *Fig 5.65* & *Fig 5.66* that are are part of the DDF kit and the CNG flow rates were controlled through ECU based on the optimal CNG flow rates for different speed and load including ESC modes that was generated during the engine and emission performance.

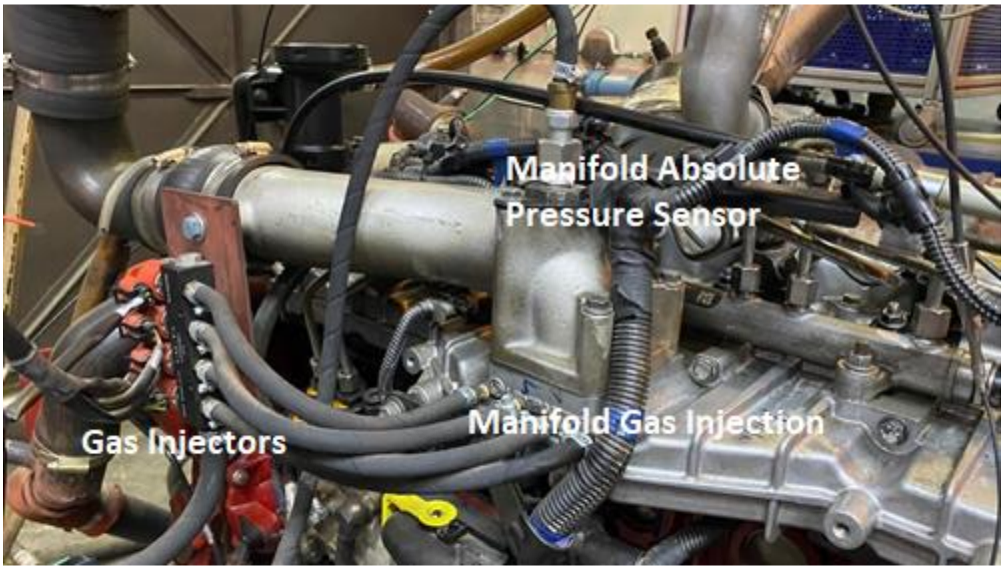


Fig 5.66: DDF Engine CNG Injector Setup.

5.5.OPERATION OF DDF ENGINE

The DDF engine with the optimized CNG flow rate was run for 100 hours of operation at different engine speeds/loads and an analysis was conducted on the used engine oil both at the beginning and end stages. The engine and emission performance of the DDF engine in comparison to a diesel operation under ESC and performance in respect of lubricant are covered in this section.

5.5.1. Engine and Emission Performance of DDF Engine under ESC

The base data with neat diesel operation of the heavy duty engine for 100 hrs was available in the laboratory and comparative performance evaluation of DDF engine has been carried out. Engine performance and emission assessment have been carried out after completion of 100 hrs of operation with the optimized matrix of CNG flow under ESC to ascertain compliance to BS-IV emission legislation.

It is clear from *Fig 5.67* & *Fig 5.68* that marginal reduction in BSFC and BSEC is seen with DDF engine in comparison to that of operation of neat diesel. The lower BSFC/BSEC can be attributed to the higher LHV of CNG which enables to consumption of less fuel for the same power [118].

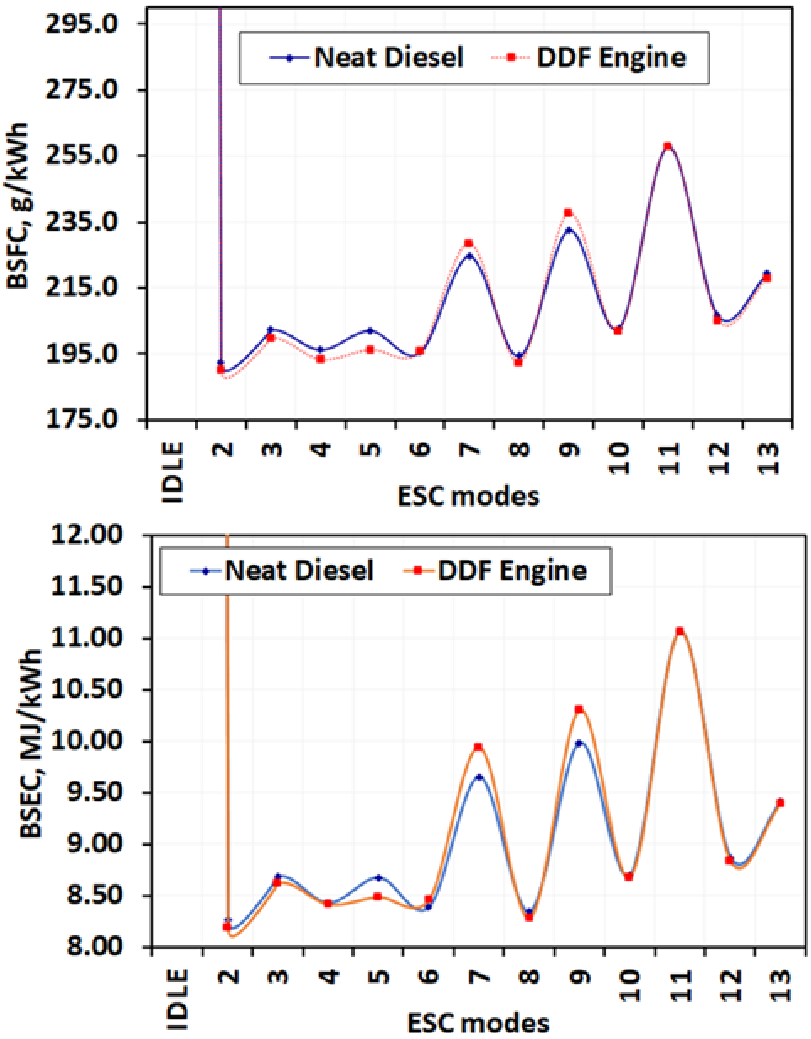


Fig 5.67: BSFC & BSEC of Neat Diesel and DDF Operation under ESC Mode.

Fig 5.69 & *Fig 5.70* depicts the efficiencies of the DDF engine in comparison to neat diesel operation under ESC mode. While there has been no significant change in BTE, there has been a marginal reduction in VE.

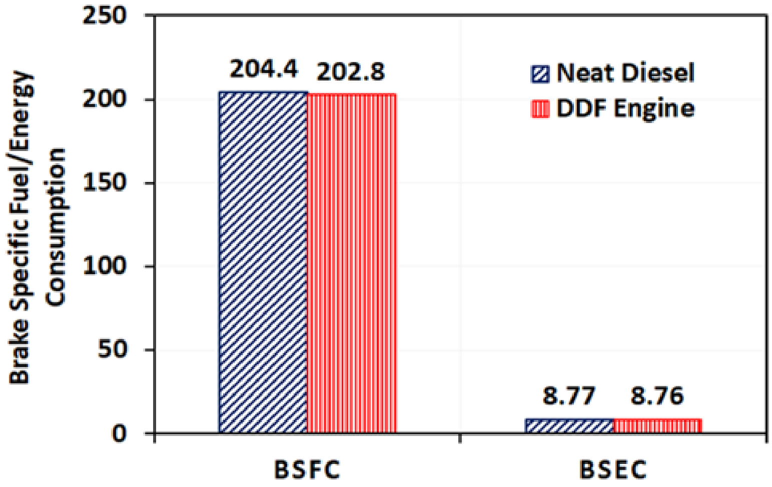


Fig 5.68: BSFC & BSEC Comparison of Neat Diesel and DDF Operation under ESC Mode.

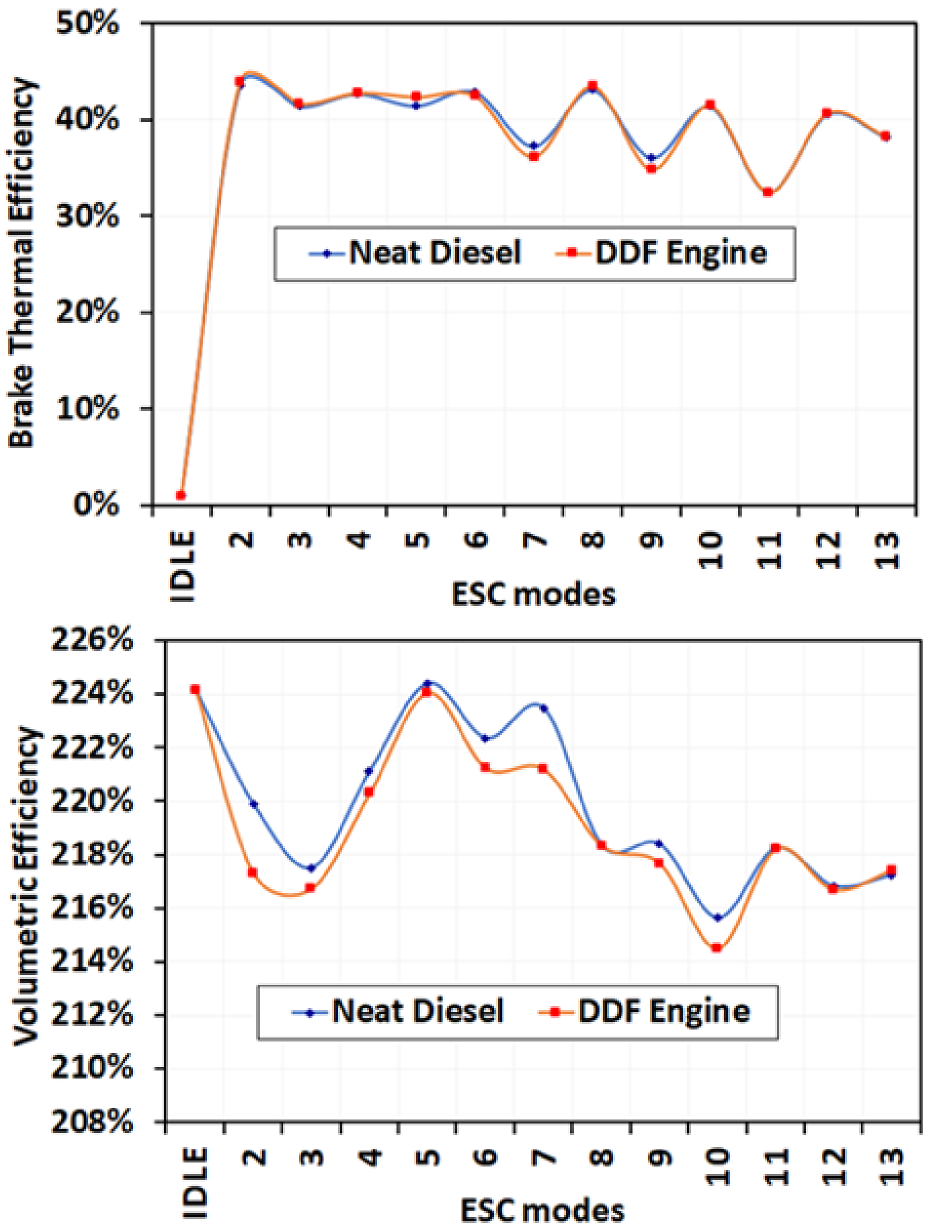


Fig 5.69: BTE & VE of Neat Diesel and DDF Operation under ESC Mode.

It is speculated that the addition of CNG to the intake manifold is what caused the decline in VE. This is because the engine must compress both air and CNG, resulting in increased compression work and subsequently lower VE [119].

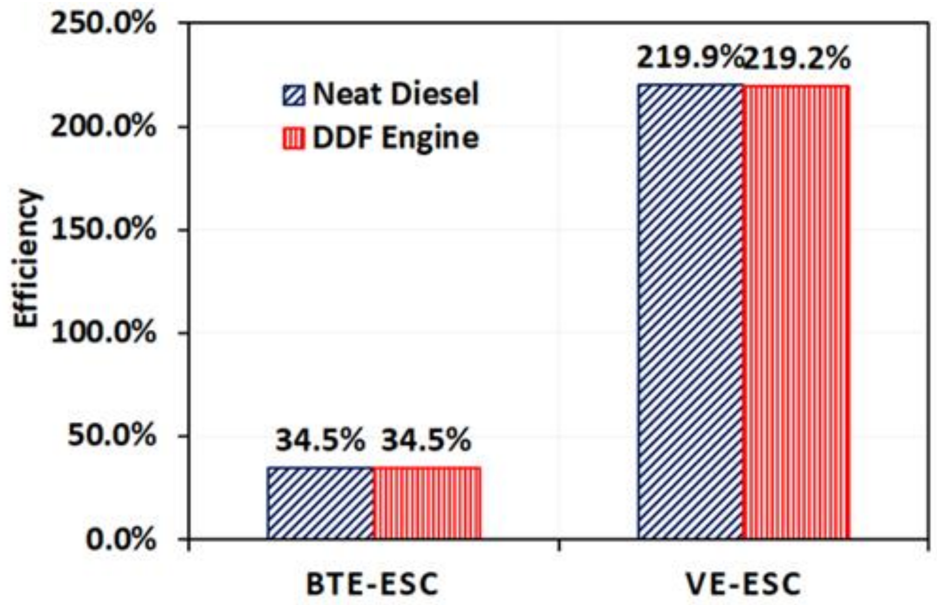


Fig 5.70: Brake Thermal Efficiency & Volumetric Efficiency Comparison of Neat Diesel and DDF Operation under ESC Mode.

The emission behaviour of the DDF engine is depicted in *Fig 5.71*, *Fig 5.72*, *Fig 5.73* & *Fig 5.74* as compared to neat diesel operation.

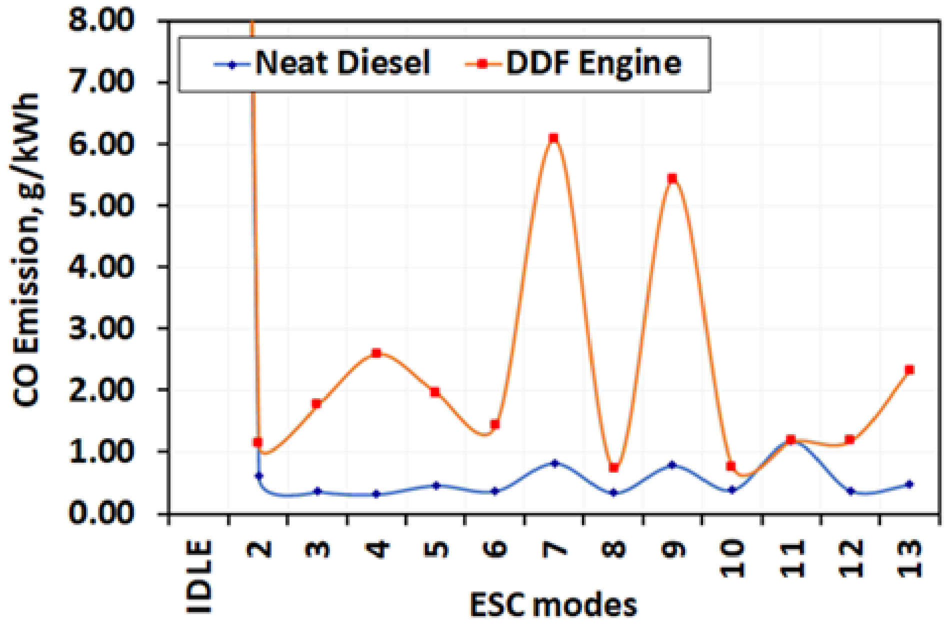


Fig 5.71: CO Emission of Neat Diesel and DDF Operation under ESC Mode.

It could be observed that all the mass emissions have increased in DDF engines in comparison to neat diesel fuel operation. While CO emission was marginally higher by 0.22 kg/kWh than the emission legislation limit of 1.5 g/kWh, THC emission was three times higher than the legislation limit of 0.46 g/kWh with DDF engine. However, the NOx emission was under the 3.5 g/kWh legislative limit, lower by 0.43 g/kWh. Since the engine is fitted with an SCR system with a primary objective of NOx reduction, NOx emission is maintained within the legislation limit. Nevertheless, the effectiveness of this post-treatment devices in mitigating CO and THC emissions remains limited.

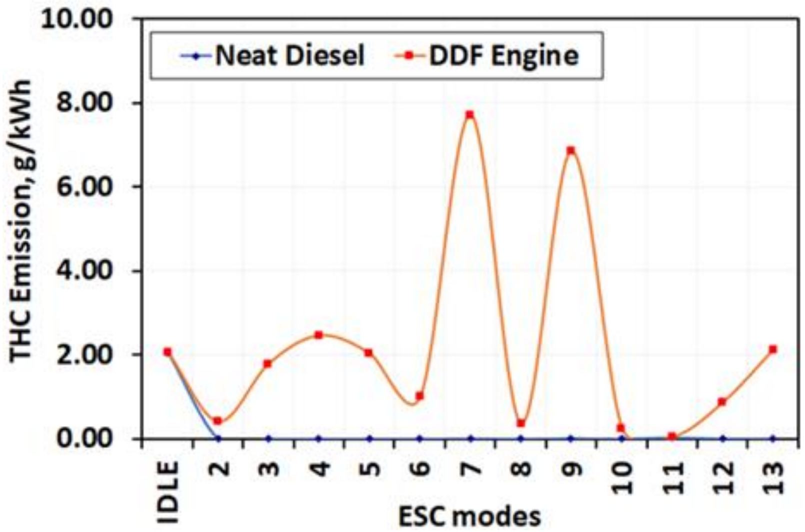


Fig 5.72: THC Emission of Neat Diesel and DDF Operation under ESC Mode.

While it has been noted in a lot of publications that DDF engine CO and THC emissions increase, less attempt has been made to study the application of diesel oxidation catalyst (DOC). As a future scope of work, research activity can be taken up for studying the performance of DOC in reduction of emission under DDF mode of operation without any modification of engine components, ECU setting, etc.

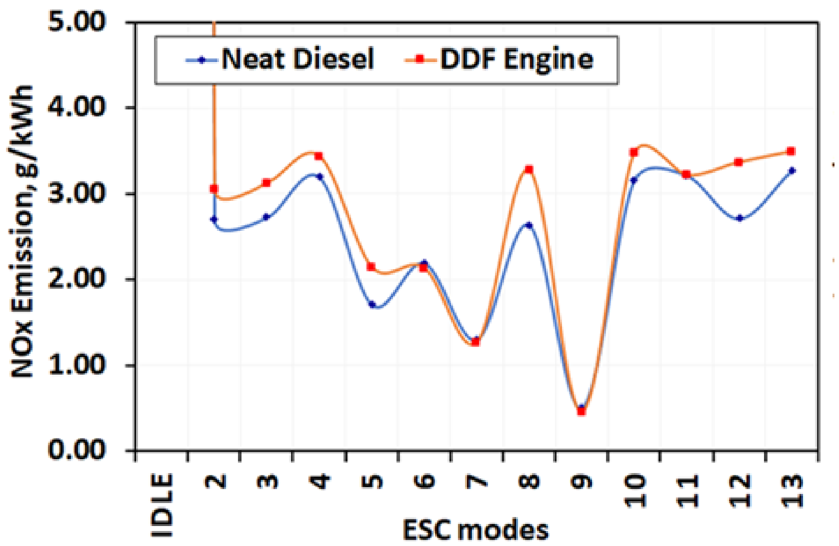


Fig 5.73: NOx Emission of Neat Diesel and DDF Operation under ESC Mode.

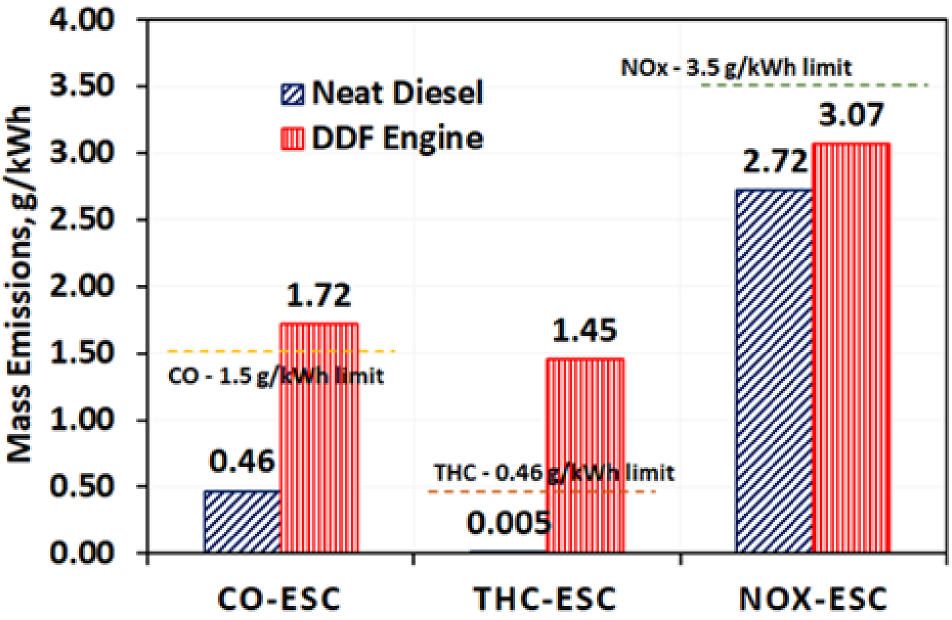


Fig 5.74: Regulated Emission Comparison of Neat Diesel and DDF Operation under ESC Mode.

The greenhouse gases such as methane and CO_2 emission are depicted in *Fig 5.75*, *Fig 5.76* & *Fig 5.77.* Similar to THC emission, methane emission also has increased in DDF engine while CO_2 emission has reduced. Because diesel ignites the CNG-air combination and discharged into the exhaust, the methane present in the crevices is not involved in the combustion process [119].

Additionally, because CNG has less carbon and due to minimised pumping and friction losses, the DDF engines emit less CO_2 [82].

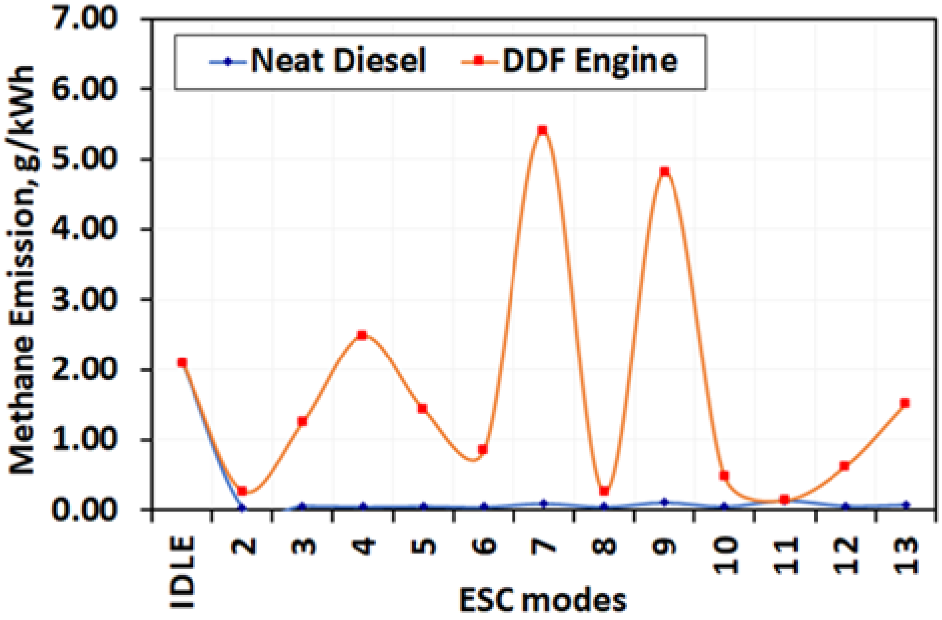


Fig 5.75: Methane Emission Comparison of Neat Diesel and DDF Operation under ESC Mode.

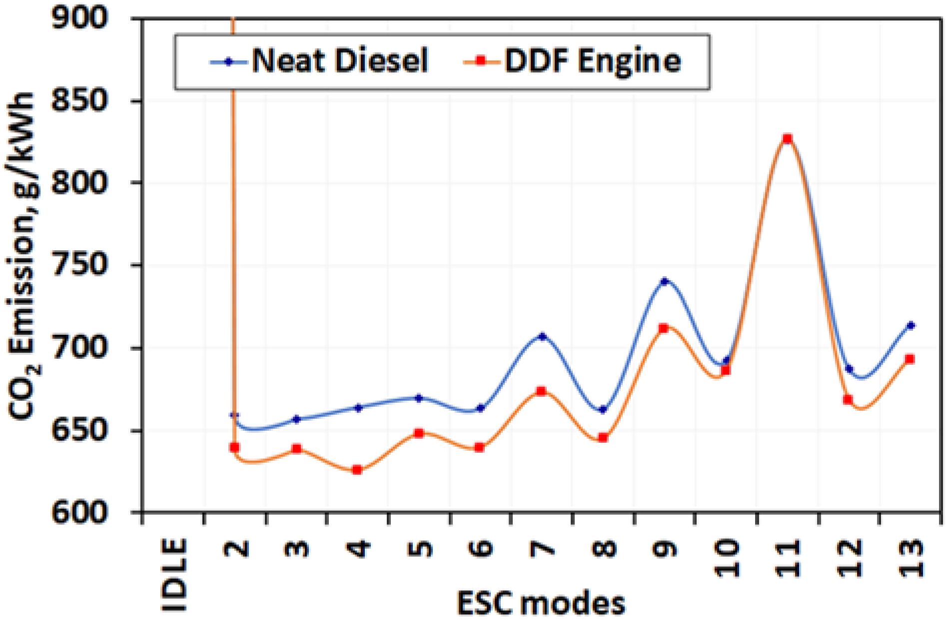


Fig 5.76: CO_2 Emission Comparison of Neat Diesel and DDF Operation under ESC Mode.

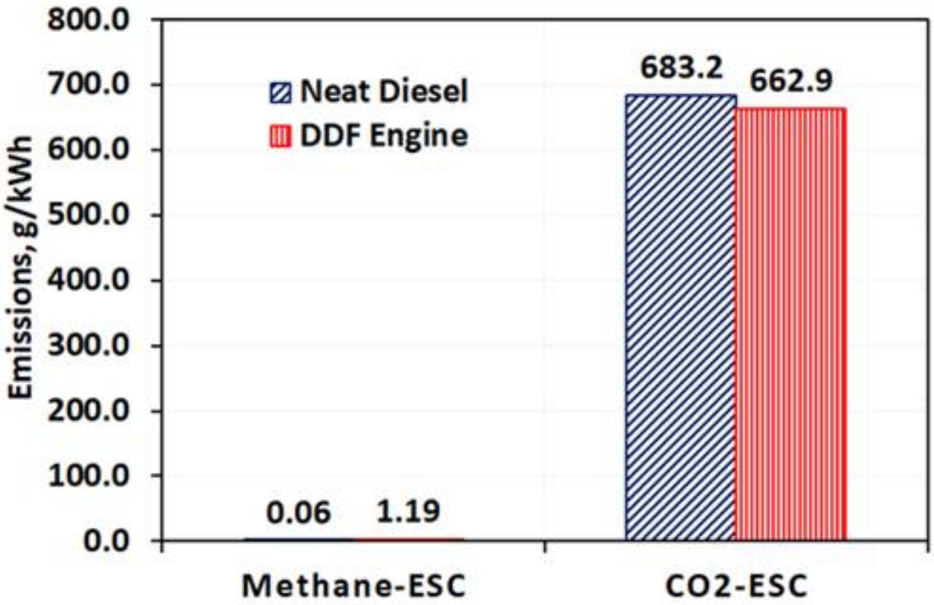


Fig 5.77: Methane & CO_2 Emission Comparison of Neat Diesel and DDF Operation under ESC Mode.

5.5.2. Used Engine Oil Analysis of DDF Engine

Commercial CK4 10W30 engine was used both for diesel fuel operation as well as DDF engine. A handheld portable oil analyser was used for monitoring the engine oil that utilizes InfraRed spectroscopy to test oil chemistry, including Nitrition, Oxidation, Sulfation, Total Base Number, Total Acid Number and Water. And for viscosity measurement, a handheld viscometer that is solvent-free and temperature-controlled was used for measuring kinematic viscosity at 40^{o}C and 100^{o}C. Additionally for fuel dilution, by taking a sample of the "headspace" in a vial, a portable Surface Acoustic Wave (SAW) vapour sensor was utilised to gauge the amount of fuel in used oil samples. *Fig 5.78* shows the used engine oil analysis of neat diesel and DDF engines for a monitoring period of 100 hours.

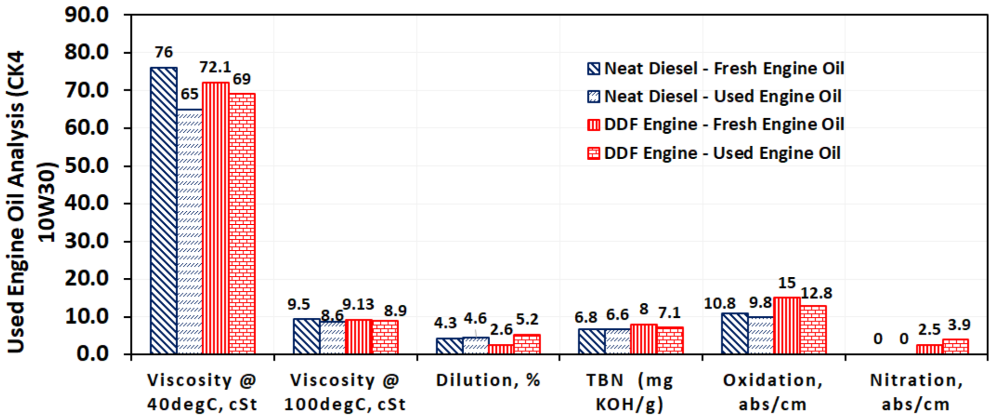


Fig 5.78: Used Engine Analysis of Neat Diesel and DDF Operation.

CK4 10W30 was used in both engine testing following a change at 100 hours. The monitoring of used engine oil samples revealed a satisfactory condition of oil for both diesel and DDF operation except for a marginal increase in dilution and nitration under DDF engine. This effect on engine oil with DDF is expected considering the seeping of methane into the engine oil and increasing the fuel-oil dilution. Further, it was observed under DDF, a marginally higher rate of nitration in engine oil. This is due to the marginal increase in in-cylinder NOx emission [118] [119].

CHAPTER

6. CONCLUSION

An investigation was conducted on a commercially available, heavy duty, inline turbocharged and inter-cooled diesel engine to assess its engine performance, emissions characteristics, and the impact on the SCR system when operated in DDF mode. The major finding of this research activity has been concluded in four different namely (i) Full Throttle Condition, (ii) Part Throttle/Varying Load Condition, (iii) European Stationary cycle testing and (iv) Engine Control Unit (ECU) as under.

(i) Utilising CNG in diesel engines, specifically as a DDF with a maximum CNG flow rate of 1.5 kg/h, has been determined to enhance the performance of a heavy duty multi-cylinder commercial diesel engine operating at full load. This was established by identifying the appropriate operating conditions and substituting energy sources, leading to improved efficiency.

- To cater to various engine speeds, a controlled flow of CNG was introduced into the intake manifold post-intercooler. The CNG flow rates were maintained at three distinct constant levels: 0.67, 0.94 & 1.54 kg/h. These flow rates corresponded to ESRs ranging from 1.8% to 10.2% and MFSRs ranging from 1.8% to 9.4%.

- The brake power demonstrated a rise proportional to the increase in the rate of CNG substitution, reaching its peak at a flow rate of 1.54 kg/h. The power gain amounted to 8.9%, attributed to the elevated combined heating value. Additionally, there was an observed torque increase of approximately 9.0%.

- The selected engine demonstrates a RAFR range of 1–3 and is renowned for its efficient and environmentally-friendly combustion process. When the engine operates at lower speeds, fuel combustion is optimized, leading to significantly higher exhaust temperatures in contrast to higher speeds during DDF.

- This engine's exceptional capability to transform potential energy resulted in a remarkable maximum BTE of 43% when supplied with a fuel flow rate of 0.94 kg/h, accounting for an energy share of 6–7%. Notably, the engine achieved its highest volumetric efficiency at 1250 rpm with a CNG flow rate of 1.54%. However, it is worth mentioning that diesel fuel exhibited the highest volumetric efficiency overall.

- At a1.54 kg/hr CNG flow rate, the BSFC decreased by up to 1% due to CNG's higher LHV, whereas BSEC decreased by 0.2 %. Under DDF mode, engine speeds between 1250 and 2250 rpm were ideal for enhanced fuel economy.

- CO emissions have decreased at all engine speeds under DDF mode, except for the lowest and highest engine speeds, due to the lower carbon content of CNG. The emission of THC demonstrated an upward trend as the proportion of CNG increased in both pre and SCR evaluations. Additionally, higher engine speeds were found to contribute to increased THC emissions. This can be attributed to the impact of fuel composition on the production of UBHC and the prolongation of ignition delay.

- Because of the higher combustion temperature, NOx emissions have increased with an increased percentage of CNG used in diesel fuel mode has increased.

- Under dual fuel mode, as expected, CH_4 emissions increased as the percentage of CNG increased. Under full load conditions, CO2 emissions decreased by 4.5 %.

- In dual fuel mode, SO_2 and propane emissions decreased, whereas formaldehyde levels increased. Under DDF operation, there was a slight rise in acetylene and ethylene emissions, as well as traces of formic acid and ammonia slip, but no significant changes.

- The efficiency of THC emissions conversion using SCR technology ranged from 10% to 86%. It was shown to be higher when utilising a diesel fuel system and lower when using a CNG dual fuel system. The SCR system effectively reduced NOx emissions by 50% to 67%. Importantly, all of the tested fuels successfully complied with the Euro 4 legislation limit. Even though the SCR system was not designed to reduce methane emissions, a reduction of 12.12–27.77 % was discovered.

(ii) The research results have significantly advanced our comprehension of the basic combustion properties of the CNG energy fraction. Additionally, the study delved into the in-cylinder combustion of dual fuel engines using diesel and CNG, specifically examining the effects of increasing energy fraction on emissions at different speeds and loads. The investigation aimed to identify the emissions associated with this dual fuel configuration.

- The CEF ranged from 1.9 to 60.2 % for the various CMFR, with lower CEF at low load and speed, and higher CEF at high load and speed. When operating at 1500 rpm and full load,a a comparison of BSEC revealed that the minimum BSEC was recorded at 4% CEF. Conversely, the maximum BSEC was noted at 33% CEF during operation at 2500 rpm and partial load of 25%. BSEC reduced as CEF increased, especially at lower loads, although a slight drop was seen at higher loads.

- The rise in in-cylinder pressure was observed as the speed increased, up until it reached a moderate range between 1500 and 2000 rpm. Subsequently, the pressure began to decrease, with the highest value occurring at a point away from TDC. At 2000 rpm, neat diesel recorded a maximum pressure of 145.17 bar, whereas the 4.0 kg/h CMFR exhibited a peak pressure of 152.89 bar at 1500 rpm. As engine speed increases, peak HRR reduces as well for all fuels, while peak HRR increases when CMFR lowers, i.e. as pilot diesel fuel quantity decreases. The relationship between engine speed and PRR under DDF exhibits a negative correlation, meaning that as the engine speed increases, the PRR decreases. Conversely, the PRR shows a positive correlation with CMFR, reaching its highest point at 4.0 kg/h. As the load increases, the duration of combustion also increases. Simultaneously, the engine speed exhibits an upward trend until it reaches 1500 rpm, after which it experiences a decline. Under DDF, as CEF increased, ID decreased at 100% load, while ID increased slightly at most engine speeds for other load conditions.

- The study on the impact of combustion duration, FBR, DID, and IMEP on raw emissions under DDF revealed notable changes in pollutant levels at different loads and consistent speeds. Specifically, the findings indicated an increase in emissions of CO and THC, whereas CO_2 and NOx emissions exhibited a decrease. THC and CO emissions dropped as load conditions increased, whereas CO_2 and NOx increased. The emissions of THC and CO exhibited a decline as the combustion duration, FBR, and DID increased, and were comparatively lower during higher IMEP. Conversely, decreasing the combustion duration, FBR, and DID, along with reducing the IMEP, led to decreased emissions of NOx and CO_2.

(iii)To improve the DDF engine's efficiency and emissions, the impact of substituting CNG was investigated under the speed-torque mode. This analysis led to the optimization of the DDF engine and produced the following outcomes.

- The study achieved a maximum replacement of 4.0 kg/hr of CNG, resulting in a 58% substitution rate. However, it was not possible to achieve a higher MFSR under lower load-speed conditions owing to the excessively high RAFR and challenges in combustion. Furthermore, the ESR varied between 1.9% and 60.2%.

- The utilization of DDF led to exhaust temperatures ranging from approximately 236°C to 536°C, which proved to be advantageous for the proper functioning of the exhaust after-treatment device.

- The brake power exhibited a slight improvement until CNG flow rate of 2.1 kg/hr, after which it declined when subjected to DDF. However, the torque remained consistent with the diesel baseline curve throughout.

- Under DDF, brake thermal efficiency decreased as the CNG flow rate increased; however, better efficiency of 44 % was recorded at CNG flow rate of 0.94 kg/hr. For every speed-load scenario, the VE experienced a decrease of up to 1.5 kg/hr in CNG flow rate before subsequently rising.

- As CNG's flow rate was increased, the outcomes for BSFC and BSEC exhibited a varied response across different speeds and loads.

- The analysis of mass emissions conducted using the DDF indicated that the emission of CO escalated with the higher flow rate of CNG measured prior to SCR. However, it remained within the limits defined by Euro 4 regulations at a specific flow rate. After the SCR experiment, it was observed that the increase in CO emissions was the lowest when the flow rate of CNG was 2.5 kg/hr, with a percentage of 6.36% in comparison to the other fuels tested.

- The emission of THC exhibited an increase during the DDF system for various speed and load scenarios in the pre-SCR testing phase. Nevertheless, following SCR, the THC emission experienced a substantial reduction of up to 86.9% when normal diesel was used. Additionally, it was noted that as CNG flow rate increased, the conversion efficiency decreased. The CNG flow rate during DDF operation was in the range of 0.65-0.94 kg/hr and complied with Euro 4 criteria.

- The SCR system is primarily used to reduce NOx emissions; prior to SCR, Euro 4 standards was not met of any of the tested fuels, while NOx emissions increased under DDF up to a 0.94 kg/hr CNG flow rate and then decreased. Following SCR, NOx emissions experienced a substantial decrease across all types of fuels, showcasing a noteworthy maximum conversion rate of 91.7% specifically in the case of DDF fuel.

- Higher methane emissions are expected under DDF due to the use of CNG as a fuel, and the incidence of this phenomenon has escalated in direct correlation with the rise in CNG flow rate, but normal diesel emissions were essentially undetectable when tested before and after SCR. Despite not being specifically designed for it, the DDF with SCR system achieved a remarkable conversion rate of 57.4%.

- CO_2 emissions exhibited a consistent reduction when employing the DDF technique, accompanied by an increase in CNG substitution, regardless of the speed and load variations. The maximum reduction observed was 10.2% when utilizing the SCR system.

- The data was standardized and analyzed considering multiple performance parameters. To determine the ideal CNG flow rate for the DDF engine, the sum of individual fuel at different loads and speeds was computed. This approach ensured that the engine could operate optimally within the existing setup and without any adjustments, while also meeting the stringent Euro 4 emission standards. Consequently, a matrix was developed, presenting an optimized range of CNG flow rates, ranging from 0.65 kg/hr to 3.0 kg/hr.

(iv) Similar to performance evaluation at different speed/load conditions, at CNG mass flow rates ranging from 0.65 to 4.0 kg/h, CNG was inducted for the test condition conducted as per Euro 4/BS-VI emission norms, i.e. under ESC engine test cycle.

- The engine performance with DDF has been similar to that of normal diesel in terms of brake-thermal efficiency, brake-specific energy consumption, brake-specific fuel consumption, power, and volumetric efficiency.

- However, in the case of emission performance, CO and THC emissions increased when the CNG flow rate increased while NOx increased up to 1.42 kg/h and decreased thereafter. While CO & THC emissions were higher than Euro 4 emission norms, NOx emission was complying with the Euro 4 emission level.

- As expected methane emission increased with an increase of CNG flow rate and CO_2 reduced. Based on the above performance, a normalised matrix was prepared to determine the optimal flow rate of CNG for the different ESC modes.

(v) A DDF induction kit was developed wherein four CNG injectors were connected to the intake manifold and the CNG flow rates were controlled through ECU based on the optimal CNG flow rates for different speeds and loads including ESC modes that was generated during the engine and emission performance.

- The DDF engine was thereafter run for 100 hours of operation at different engine speeds/loads and engine oil analysis was carried out at initial and final in addition to the emission testing as per Euro 4 emission norms.

- The DDF engine performed comparably in terms of brake thermal and volumetric efficiency, with a marginal decrease in the brake fuel and energy consumption.

- Under the regulated emissions, CO and THC increased under DDF though CO emission was marginally higher than Euro 4 emission limit, THC emission was much higher. NOx emission though increased under DDF, it was under the 3.5 g/kWh Euro 4 limit.

- There was a marginal increase in methane emission while CO_2 was reduced under DDF.

- The used engine oil of neat diesel and DDF engine revealed that the lubricant performance is comparable in terms of kinematic viscosity at 40°C & 100 °C and TBN, while there has been a marginal increase in fuel dilution, oxidation and nitration though within the limit.

(vi) With the development of the DDF kit, it has been established that the CNG flow rate if optimized based on many parameters such as engine performance and emissions, a commercial engine can be run on diesel-CNG dual fuel without affecting engine performance and meeting the emission norms.

(vii) The DDF kit developed through this research is a novel work considering that a commercial heavy-duty engine meeting BS-IV emission norms that were legislated in 2017 was modified to run on CNG-diesel dual fuel technique. Based on the recommendations of the National Green Tribunal, the Government of India during 2019 has advised the conversion of stationary diesel generators to run either on neat CNG or under DDF mode for reducing PM emissions by 70%. However, diesel engine manufacturers are promoting the use of retrofit after-treatment devices to reduce the same due to difficulty in conversion for DDF operation. This research, however, focused on the optimization of the CNG flow rates, taking into account many parameters of the engine such as performance and emission, and modified the commercial engine to run under DDF mode.

CHAPTER

7. FUTURE WORK

It is conceivable to highlight a number of specific prospects for extending or improving the research experiment presented in this thesis. This study was successful in determining the effects of CNG-Diesel Dual Fuel on a heavy-duty turbocharged diesel engine's performance and emissions as well as the impact on the after-treatment devices. Based on the literature on dual fuel technology and the current study, the upcoming suggestions are made for future research in this area that can be conducted to reduce emissions:

1. To further reduce CO and THC emission, diesel oxidation catalyst (DOC) can be employed along with exhaust gas recirculation for reduction of NOx emission.

2. With the availability of BS-VI heavy-duty commercial engines, the optimization of the engine with respect to the latest emission legislation can be taken up.

3. The effect of particulates under DDF can be evaluated considering that the BS-VI commercial heavy-duty engines employ Diesel Particulate Filter.

4. The effect of oil dilution, oxidation and nitration can be addressed through nano-based lubricants and suitable formulation of engine oil can be taken up.

REFERENCES

[1] "Consumption of Petroleum Products," [Online]. Available: https://www.ppac.gov.in/content/147_1_ConsumptionPetroleum.aspx.

[2] S. H. Yoon and C. S. Lee, "Experimental investigation on the combustion and exhaust emission characteristics of biogas-diesel dual-fuel combustion in a CI engine," *Fuel Processing Technology,* vol. 92, 2011.

[3] J.-O. Müller, D. Su, R. Jentoft, W. U. and R. Schlögl, "Diesel engine exhaust emission: Oxidative behavior and microstructure of black smoke soot particulate," *Environ Sci Technol.,* vol. 40, pp. 1231-1236, 2006.

[4] J. Zhu, B. Aikawa and R. Pigeon, "Measurement of fine particles in diesel emissions using a real time aerosol monitor," *J Air Waste Manage Assoc.,* vol. 55, p. 978–983, 2005.

[5] K. Black, F. Divita, R. Margiotta and R. Guensler, "Correlating Particulate Matter Mobile Source Emissions to Ambient Air Quality," 2004.

[6] "Energy Statistics India 2021," 2021. [Online]. Available: http://mospi.nic.in/publication/energy-statistics-india-2021.

[7] T. Korakianitis, A. M. Namasivayam and R. J. Crookes, "Natural-Gas Fueled Spark-Ignition (SI) and Compression-Ignition (CI) Engine Performance and Emissions," *Progress in Energy and Combustion Science,* vol. 37, no. 1, pp. 89-112, 2011.

[8] A. Demirbas, Methane Gas Hydrate, Springer, 2010.

[9] British Petroleum, "Statistical Review of World Energy," 2021. [Online]. Available: https://www.bp.com/content/dam/bp/business-sites/en/global/corporate/pdfs/energy-economics/statistical-review/bp-stats-review-2021-full-report.pdf.

[10] J. B. Heywood, Internal Combustion Engine Fundamentals, McGraw-Hill, 1988.

[11] L. Turrio-Baldassarri, C. Battistelli, L. Conti, R. Crebelli, B. D. Berardis and A. Iamiceli, "Evaluation of emission toxicity of urban bus engines: compressed natural gas and comparison with liquid fuels," *Science of The Total Environment*, vol. 355, no. 1-3, pp. 64-77, February 2006.

[12] "Overview of Greenhouse Gases, Methane Emissions," 2013. [Online].

[13] A. Lampinen, "Development and governance of renewable methane use in transport applications," ACTA Academia Stromstandiensis, 2013.

[14] B. S. Lemke, "Performance and Efficiency of a Bi-Fuel Bio Methane/Gasoline Vehicle," in *International Conference on Renewable Energies and Power Quality (ICREPQ'11)*, Las Palmas de Gran Canaria (Spain), 2011.

[15] S. Thipse, A. Kulkarni, S. J. Vispute and S. Rairikar, "Development of Dual Fuel (Diesel-CNG) Engine for SUV Application in India," *SAE Int. J. Engines*, vol. 8, no. 1, pp. 341-349, 2015.

[16] M. Muralidharan, A. Srivastava and M. Subramanian, "A Technical Review on Performance and Emissions of Compressed Natural Gas - Diesel Dual Fuel Engine," *SAE Technical Paper 2019-28-2390*, 2019.

[17] B. Borman and K. Ragland, Combustion engineering, WCB/McGraw-Hill, 1998.

[18] K. Bhandari, "Performance and emissions of natural gas fueled internal combustion engine," *JSIR*, vol. 64, pp. 333-338, 2005.

[19] R. Singh and M. Sagar, "Dual fueling of a twin-cylinder compression ignition engine with diesel and CNG," *Journal of engineering and applied sciences*, vol. 7, pp. 90-99, 2012.

[20] R. M. Modarres and G. Karim, "Examination of the dual-fuel engine performance using low Btu gaseous fuels," in *13th International Conference on Thermal Engineering and Thermogrammetry*, Hungari, 2003.

[21] A. Carlucci, A. d. Risi, D. Laforgia and F. Naccarato, "Experimental Investigation and Combustion Analysis of a Direct Injection Dual-Fuel Diesel-Natural Gas Engine," *Energy*, vol. 33, no. 2, p. 256–263, 2008.

[22] C. Gibson, "Comparison of Propane and Methane Performance and Emissions in a Turbocharged Direct Injection Dual Fuel Engine,"

Journal of Engineering for Gas Turbines and Power, vol. 133, September 2011.

[23] F. Königsson, P. Stålhammar and H. Ångström, “Combustion Modes in a Diesel-CNG Dual Fuel Engine,” *SAE Technical Paper 2011-01-1962,* 2011.

[24] R. Hanson, S. Kokjohn, D. Splitter and R. Reitz, “Fuel Effects on Reactivity Controlled Compression Ignition (RCCI) Combustion at Low Load,” *SAE Int. J. Engines,* vol. 41, pp. 394-411, 2011.

[25] R. D. Reitz, “DOE Project: Optimization of Advanced Diesel Engine Combustion Strategies,” University Research in Advanced Combustion and Emissions Control Office of Freedom CAR and Vehicle Technologies, October 2012.

[26] M. Olofsson, L. Erlandsson and K. Willner, “Enhanced Emissions Support and Fuel Efficiency for HD Methane Engines,” A Report Prepared for International Energy Agency - Advanced Motor Fuels, 2014/05.

[27] “Bosch Launches Heavy-Duty Natural Gas Dual-Fuel Conversion Kit in Brazil,” 2008. [Online]. Available: http://www.greencarcongress.com/2008/06/bosch-launches.html.

[28] Westport Fuel Systems, “Westport HPDI 2.0,” 2016. [Online]. Available: https://wfsinc.com/our-solutions/hpdi-2.0.

[29] A.-I. Nafis, M. K. G. Babu and A. Ramesh, “Experimental Investigations of Different Parameters Affecting the Performance of a CNG-Diesel Dual Fuel Engine,” *SAE Technical Paper No. 2005-01-3767,* 2005.

[30] M. Y. Selim, “Effect of engine parameters and gaseous fuel type on the cyclic variability of dual fuel engines,” *Fuel,* vol. 84, no. 7-8, pp. 961-971, 2005.

[31] J. Shen, J. Qin and M. Yao, “Turbocharged diesel/CNG Dual-Fuel Engines with Intercooler: Combustion, Emissions and Performance,” *SAE Transaction No. 2003-01-3082,* 2003.

[32] P. M. Duc and K. Wattanavichien, “Study on biogas premixed charge diesel dual fuelled engine,” *Energy Conversion & Management,* vol. 48, 2007.

[33] S. H. Yoon and C. S. Lee, “Experimental investigation on the combustion and exhaust emission characteristics of biogas–biodiesel dual-fuel combustion in a CI engine,” *Fuel Process Technol.*, vol. 92, pp. 992-1000, 2011.

[34] A. P. Carlucci, D. Laforgia, R. Saracino and G. Toto, “Combustion and emissions control ina diesel-methane dual fuel engines: The effects of methane supply method combined with variable in-cylinder bulk motion,” *Energy Conversion and Management*, vol. 52, 2011.

[35] R. Papagiannakis, C. Rakopoulos, D. Hountalas and D. Rakopoulos, “Emission characteristics of high speed, dual fuel, compression ignition engine operating in a wide range of natural gas/diesel fuel proportions,” *Fuel*, vol. 89, no. 7, p. 1397–1406, 2010.

[36] M. Y. Selim, “Pressure-time characteristics in diesel engine fueled with natural gas,” *Renewable Energy*, vol. 22, no. 4, pp. 473-89, 2001.

[37] T. Korakianitis, A. M. Namasivayam, R. J. Crookes, S. Imran, B. Ihracska, A. Diez and N. A. Malik, “Compression-Ignition Engine Performance and Emissions in Single and Dual Fuelling Modes with Sustainable Fuels,” in *Proceedings of the IASTED International Conference on Power and Energy Systems and Applications, PESA 2011*, USA, 2011.

[38] M. Y. Selim, “Sensitivity of dual fuel engine combustion and knocking limits to gaseous fuel composition,” *Energy Conversion and Management*, vol. 45, no. 3, pp. 411-25, 2004.

[39] A. S. Namasivayam, “Combustion, Performance and Emissions Characteristics of Compression-Ignition Engines Fuelled by Sustainable Fuels,” Ph.D. Thesis, 2011.

[40] R. Papagiannakis and D. Hountalas, “Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas,” *Energy Conversion and Management*, vol. 45, no. 18-19, pp. 2971-87, 2004.

[41] R. K. Saray, “Enhancement of Combustion Process in Dual Fuel Engines at Part Loads by Using Suitable Techniques,” *IJE Transaction B: Applications*, vol. 22, no. 1, pp. 77-90, 2009.

[42] M. Y. Selim, "Effect of exhaust gas recirculation on some combustion characteristics of dual fuel engine," *Energy Conversion and Management,* vol. 44, no. 5, pp. 707-21, 2003.

[43] G. McTaggart-Cowan, H. Jones, S. Rogak, W. Bushe, P. Hill and S. Munshi, "Direct-Injected Hydrogen-Methane Mixtures in a Heavy-Duty Compression Ignition Engine," in *SAE 2006 World Congress & Exhibition*, Detroit, USA, SAE Technical Paper 2006-01-0653, 2006.

[44] J. Mattson, C. Langness and C. Depcik, "An Analysis of Dual Fuel Combustion of Diesel with Compressed Natural Gas in a Single Cylinder Engine," *SAE Technical Paper No. 2018-01-0248,* 2018.

[45] B. Sahoo, N. Sahoo and U. Saha, "Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines-A critical review," *Renew Sustain Energy Rev,* 2009.

[46] B. Yang, L. Wang, L. Ning and K. Zeng, "Effects of pilot injection timing on the combustion noise and particle emissions of a diesel/natural gas dual-fuel engine at low load," *Applied Thermal Engineering,* vol. 102, pp. 822-828, 2016.

[47] G. McTaggart-Cowan, H. Jones, S. Rogak, W. Bushe, P. Hill and S. Munshi, "The effects of high-pressure injection on a compression-ignition, direct injection of natural gas engine," *Journal of Engineering for Gas Turbines and Power,* vol. 129, no. 2, pp. 579-88, 2007.

[48] J. You, Z. Liu, Z. Wang, D. Wang, Y. Xu, G. Du and X. Fu, "The exhausted gas recirculation improved brake thermal efficiency and combustion characteristics under different intake throttling conditions of a diesel/natural gas dual fuel engine at low loads," *Fuel,* vol. 266, 2020.

[49] K. Cheenkachorn, C. Poompipatpong and C. Gyeu, "Performance and emissions of a heavy-duty diesel engine fuelled with diesel and LNG (liquid natural gas)," *Energy,* vol. 53, pp. 52-57, 2013.

[50] MotorTrend, "Exhaust Gas Temperature Basics – Diesel Tech," 2006. [Online]. Available: www.trucktrend.com/cool-trucks/0607dp-exhaust-gas-temperature-basics/.

[51] S. Bari and S. Hossain, "Performance of a diesel engine run on diesel and natural gas in dualaefv-fuel mode of operation," *Energy Procedia,* vol. 160, pp. 215-222, 2019.

[52] W. Pulkrabek, Engineering Fundamentals of the Internal Combustion Engine, Prentice Hall, 2004.

[53] M. Moran, H. Shapiro, D. Boettner and M. Bailey, Fundamentals of Engineering Thermodynamics, John Wiley & Sons, 2010.

[54] J. Livengood, A. Rogowski and C. Taylor, "The Volumetric Efficiency of Four-Stroke Engines," *SAE Technical Paper 520259,* 1952.

[55] I. A. Resitoglu, K. Altinisik and A. Keskin, "The pollutant emissions from diesel-engine vehicles and exhaust aftertreatment systems," *Clean Technologies and Environmental Policy,* vol. 17, no. 1, pp. 15-27, 2014.

[56] M. Smith, C. Depcik, J. Hoard and S. Bohac, "The Effects of CO, H2, and C3H6 on the SCR Reactions of an Fe Zeolite SCR Catalyst," *SAE Technical Paper 2013-01-1062,* 2013.

[57] M. Ergeneman, C. Sorusbay and A. G. Goktan, "Exhaust Emission and Fuel Consumption of CNG Diesel Fueled City Buses Calculated Using a Sample Driving Cycle," *Journal Energy Sources,* vol. 21, no. 3, 1999.

[58] T. F. Yusaf, D. Buttsworth, K. H. Saleh and B. Yousif, "CNG-diesel engine performance and exhaust emission analysis with the aid of artificial neural network," *Applied Energy,* vol. 87, no. 5, pp. 1661-1669, 2010.

[59] R. van Helden, M. van Genderen, M. van Aken and R. Verbeek, "Engine Dynamometer and Vehicle Performance of a Urea SCR-System for Heavy-Duty Truck Engines," *SAE Technical Paper 2002-01-0286,* 2002.

[60] S. Xiaoyan, Y. Yunbo, H. Hong, S. Shijin, D. Hongyi and L. Rulong, "Combination of biodiesel-ethanol-diesel fuel blend and SCR catalyst assembly to reduce emissions from a heavy-duty diesel engine," *Journal of Environmental Sciences,* no. 20, p. 177–182, 2008.

[61] R. Stone, Introduction to Internal Combustion Engines, Macmillan, 1992.

[62] S. Roy, A. K. Das, R. Banerjee and P. K. Bose, "A TMI based CNG dual-fuel approach to address the soot-NOx-BSFC trade-off characteristics of a CRDI assisted diesel engine - an EPA perspective," *Journal of Natural Gas Science and Engineering,* vol. 20, pp. 221-240, 2014.

[63] J. Liu, F. Yang, H. Wang, M. Ouyang and S. Hao, “Effects of pilot fuel quantity on the emissions characteristics of a CNG/DDF engine with optimized pilot injection timing,” *Appl. Energy,* vol. 110, no. 0, p. 201–206, 2013.

[64] O. Lim, N. Iida, G. Cho and J. Narankhuu, “The Research about Engine Optimization and Emission Characteristic of Dual Fuel Engine Fueled with Natural Gas and Diesel,” *SAE Technical Paper 2012-32-0008,* 2012.

[65] C. Koythong, N. Nuntapap, A. Pal, S. Songchon and Y. Laoonual, “Investigation of Emission Characteristics of DDF (DDF) Engine with Multi-Point Natural Gas Injection System,” in *The 3rd TSME International Conference on Mechanical,* Chiang Rai, 2012.

[66] T. Ballinger, J. Cox, M. Konduru, D. De and W. Manning, “Evaluation of SCR Catalyst Technology on Diesel Particulate Filters,” *SAE International Journal of Fuels and Lubricants,* vol. 2, no. 1, pp. 369-374, 2009.

[67] G. S. Hebbar, “NOx from diesel engine emission and control strategies - a Review,” *International Journal of Mechanical Engineering and Robotics Research,* vol. 3, no. 4, 2014.

[68] J. Egúsquiza, S. Braga and C. Braga, “Performance and gaseous emissions characteristics of a natural gas/diesel dual fuel turbocharged and aftercooled engine,” *Journal of the Brazilian Society of Mechanical Sciences and Engineering,* vol. 31, no. 2, pp. 142-150, 2009.

[69] S. Imran, D. Emberson, A. Diez, D. Wen, R. Crookes and T. Korakianitis, “Natural gas fueled compression ignition engine performance and emissions maps with diesel and RME pilot fuels,” *Applied Energy,* vol. 124, pp. 354-365, 2014.

[70] J. Shu, J. Fu, Y. Zhang, M. Xie, J. Liu, J. Liu and D. Zeng, “Influences of natural gas energy fraction on combustion and emission characteristics of a diesel pilot ignition natural gas engine based on a reduced chemical kinetic model,” *Fuel,* vol. 261, 2020.

[71] L. Wei and P. Geng, “A review on natural gas/DDF combustion, emissions and performance,” *Fuel Processing Technology,* vol. 142, p. 264–278, 2016.

[72] R. Papagiannakis and D. Hountalas, “Experimental investigation concerning theeffect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine,” *Appl. Therm. Eng.,* vol. 23, no. 3, p. 353–365, 2003.

[73] Z. Sun, X. Wang, X. Wang and J. Zhou, “Combustion and emission analysis of heavyduty vehicle diesel engine,” in *AIP Conference Proceedings 1820*, 2017.

[74] P. Napolitano, C. Guido, C. Beatrice and N. Del Giacomo, “Application of a Dual Fuel Diesel-CNG Configuration in a Euro 5 Automotive Diesel Engine,” *SAE Technical Paper 2017-01-0769,* 2017.

[75] K. Subramanian, Biofueled Reciprocating Internal Combustion Engines, Boca Raton: CRC Press, 2017.

[76] H. Huang, Z. Zhu, Y. Chen, Y. Che, D. L. J. Zhu and T. Ouyang, “xperimental and numerical study of multiple injection effects on combustion and emission characteristics of natural gas–diesel dual-fuel engine,” *Energy Conversion and Management,* vol. 183, p. 84–96, 2019.

[77] B. M, I. J, T. A and K. H, “Emissions Characterization from Different Technology Heavy-Duty Engines Retrofitted for CNG/Diesel Dual-Fuel Operation,” *SAE Int. J. Engines,* vol. 8, no. 3, 2015.

[78] T. Gatts, S. Liu, C. Liew, B. Ralston, C. Bell and H. Li, “An experimental investigation of incomplete combustion of gaseous fuels of a heavy-duty diesel engine supplemented with hydrogen and natural gas,” *International Journal of Hydrogen Energy,* vol. 37, no. 9, pp. 7848-7859, 2012.

[79] C. Guido, P. Napolitano, V. Fraioli and C. Beatrice, “Assessment of Engine Control Parameters Effect to Minimize GHG Emissions in a Dual Fuel NG/Diesel Light Duty Engine,” *SAE Technical Paper 2018-01-0266*, 2018.

[80] M. Abdelaal and A. Hegab, “Combustion and emission characteristics of a natural gas-fueled diesel engine with EGR,” *Energy Conversion and Management,* vol. 64, p. 301–312, 2012.

[81] T. Grigoratos, G. Fontara, G. Martini and C. Peletto, “A study of regulated and green house gas emissions from a prototype heavy-duty

compressed natural gas engine under transient and real life conditions," *Energy*, vol. 103, pp. 340-355, 2016.

[82] T. Yusaf and M. Z. Yusoff, "Development of a 3D CFD model to investigate the effect of the mixing quality on the CNG-diesel engine performance," in *The International Conference and Exhibition and Natural Gas Vehicles*, Yokohama, Japan, 2000.

[83] A. K. Agarwal, P. C. Shukla, C. Patel, J. G. Gupta, N. Sharma, R. K. Prasad and R. A. Agarwal, "Unregulated emissions and health risk potential from biodiesel (KB5, KB20) and methanol blend (M5) fuelled transportation diesel engines," *Renewable Energy*, vol. 98, pp. 283-291, 2016.

[84] K. Takada, F. Yoshimura, Y. Ohga and J. Kusaka, "Experimental Study on Unregulated Emission Characteristics of Turbocharged DI Diesel Engine with Common Rail Fuel Injection System," *SAE Technical Paper 2003-01-3158*, 2003.

[85] T. W. Hesterberg, C. A. Lapin and W. B. Bunn, "A Comparison of Emissions from Vehicles Fueled with Diesel or Compressed Natural Gas," *Environ. Sci. Technol.*, vol. 42, no. 17, p. 6437–6445, 2008.

[86] L. Li, G. Liu, X. Cui, X.Sun, S. Li and Y. Qian, "Effects of the variation in diesel fuel components on the particulate matter and unregulated gaseous emissions from a common rail diesel engine," *Fuel*, vol. 232, p. 279–89, 2018.

[87] B. Tesfa, R. Mishra, C. Zhang, F. Gu and A. Ball, "Combustion and performance characteristics of CI (compression ignition) engine running with biodiesel," *Energy*, vol. 51, pp. 101-115, 2013.

[88] E. Uludamar, V. Karaman, S. Yildizhan and H. Serin, "Improvement of Density, Viscosity and Cold Flow Properties of Palm Oil Biodiesel by Alcohol Addition," *International Journal of Engineering Technologies IJET*, vol. 2, no. 2, pp. 64-67, 2016.

[89] C.-f. Lee, Y. Pang, H. Wu, K. Nithyanandan and F. Liu, "An optical investigation of substitution rates on natural gas/diesel dual-fuel combustion in a diesel engine," *Applied Energy*, vol. 261, 2020.

[90] A. Srna, B. v. Rotz, K. Herrmann, K. Boulouchos and G. Bruneaux, "Experimental investigation of pilot-fuel combustion in dual-fuel

engines, Part 1: Thermodynamic analysis of combustion phenomena," *Fuel,* vol. 255, 2019.

[91] F. Payri, J. Luján, J.Martín and A. Abbad, "Digital signal processing of in-cylinder pressure for combustion diagnosis of internal combustion engines," *Mech. Syst. Signal Process.,* vol. 24, no. 6, pp. 1767-1784, 2010.

[92] N. N. Mustafi, R. R. Raine and S. Verhelst, "Combustion and emissions characteristics of a dual fuel engine operated on alternative gaseous fuels," *Fuel,* vol. 109, pp. 669-678, 2013.

[93] J. Stewart, A. Clarke and R. Chen, "An experimental study of the dual-fuel performance of a small compression ignition diesel engine operating with three gaseous fuels," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering,* vol. 221, no. 943, 2007.

[94] S. Ghaffarzadeh, A. N. Toosi and V. Hosseini, "An experimental study on low temperature combustion in a light duty engine fueled with diesel/CNG and biodiesel/CNG," *Fuel,* vol. 262, 2020.

[95] M. Muralidharan, A. Srivastava and M. Subramanian, "In-cylinder combustion studies of diesel-CNG dual fuel with increasing energy fraction and its effect on emissions," *SAE Int. J. Engines,* vol. 15, no. 5, 2022.

[96] M. F. J. Brunt and K. C. Platts, "Calculation of Heat Release in Direct Injection Diesel Engines," *Journal of Engines,* vol. 108, no. 3, pp. 161-175, 1999.

[97] B. Ashok and K. Nanthagopal, "15-Eco friendly biofuels for CI engine applications," *Advances in Eco-Fuels for a Sustainable Environment,* pp. 407-440, 2019.

[98] J. Zhou, C. Cheung and C. Leung, "Combustion, performance and emissions of a diesel engine with H2, CH4 and H2-CH4 addition," *International Journal of Hydrogen Energy,* vol. 39, pp. 4611-4621, 2014.

[99] N. Nuntapap, H. Singh and Y. Laoonual, "Investigation of Abnormal Combustion Characteristics of Dual Fuel Engine using Natural Gas and

Diesel as Fuels," in *The 4th TSME International Conference on Mechanical Engineering, AEC-2010*, Pattaya, Chonburi, 2013.

[100] J. A. Massey, "Indication of Cylinder Pressure Rise Rate by means of vibration and acoustic emissions of an internal combustion engine," 2008.

[101] S. Mahla, L. Das and M. Babu, "Effect of EGR on Performance and Emission Characteristics of Natural Gas Fueled Diesel Engine," *Jordan Journal of Mechanical and Industrial Engineering,* vol. 4, no. 4, p. 523–530, 2010.

[102] L. Zhou, Y.-F. Liu and C.-B. Wu, "Effect of the diesel injection timing and the pilot quantity on the combustion characteristics and the fine-particle emissions in a micro-diesel pilot-ignited natural-gas engine," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering,* vol. 227, no. 8, pp. 1142-1152, 2013.

[103] J. Lee, S. Lee, C. Kim and S. Lee, "Characteristic of energy fractions and emissions under natural gas/diesel dual-fuel heavy-duty engine in terms of the combustion parameters," *International Journal of Automotive Technology,* vol. 21, no. 1, pp. 103-113, 2020.

[104] A. Peter, S. Riess, M. Wensing, J. Fruehhaber and T. Lauer, "Investigation of Mixture Formation and Flammability of Natural Gas and Diesel under Dual Fuel Operating Conditions in the Limits of Flame-quenching and Knocking," in *The Ninth International Conference on Modeling and Diagnostics for Advanced Engine Systems (COMODIA 2017),* Okayama, Japan, 2017.

[105] M. Mikulski and S. Wierzbicki, "Numerical investigation of the impact of gas composition on the combustion process in a dual-fuel compression-ignition engine," *Journal of Natural Gas Science and Engineering,* vol. 31, pp. 525-537, 2016.

[106] C. Langness, J. Mattson and C. Depcik, "Moderate Substitution of Varying Compressed Natural Gas Constituents for Assisted Diesel Combustion," *Combustion Science and Technology,* vol. 189, no. 8, pp. 1354-1372, 2017.

[107] K. Tanin, D. Wickman, D. Montgomery and S. Das, "The Influence of Boost Pressure on Emissions and Fuel Consumption of a Heavy-Duty

Single-Cylinder D.I. Diesel Engine," *SAE Technical Paper 1999-01-0840*, 1999.

[108] R. Mamat, N. R. Abdullah, H. Xu, M. L. Wyszynski and A. Tsolakis, "Effect of Boost Temperature on the Performance and Emissions of a Common Rail Diesel Engine Operating with Rapeseed Methyl Ester (RME)," in *Proceedings of the World Congress on Engineering 2010 Vol II, WCE 2010*, London, U.K., 2010.

[109] J. Mattson, C. Langness and C. Depcik, "Exergy Analysis of Dual-Fuel Operation with Diesel and Moderate Amounts of Compressed Natural Gas in a Single-Cylinder Engine," *Combustion Science and Technology,* vol. 190, no. 3, pp. 471-489, 2018.

[110] S. Hill and L. D. Smoot, "Modeling of nitrogen oxides formation and destruction in combustion systems," *Progress in Energy and Combustion Science,* vol. 26, no. 4-6, pp. 417-458, 2000.

[111] F. Tong, P. Jaramillo and I. M. L. Azevedo, "Comparison of Life Cycle Greenhouse Gases from Natural Gas Pathways for Medium and Heavy-Duty Vehicles," *Environmental Science & Technology,* vol. 49, no. 12, p. 7123–7133, 2015.

[112] L. Wang, Z. Chen, B. Yang and K. Zeng, "Control Strategy Development of Natural Gas/Diesel Dual Fuel Engine for Heavy Duty Vehicle," *SAE Technical Paper 2016-01-0628,* 2016.

[113] M. D. Gajbhiye, S. Lakshmanan, N. Kumar, S. Bhattacharya and S. Nishad, "Effectiveness of India's Bharat Stage mitigation measures in reducing vehicular emissions," *Transportation Research Part D: Transport and Environment,* vol. 115, p. 103603, 2023.

[114] S. Singh, M. J. Kulshrestha, N. Rani, K. Kumar, C. Sharma and D. K. Aswal, "An Overview of Vehicular Emission Standards," no. 38, pp. 241-263, 2023.

[115] Ministry of Heavy Industries and Public Enterprises, "Automotive Mission Plan 2006–2016, A Mission for Development of Indian Automotive Industry," Government of India, New Delhi, 2006.

[116] Ministry of Heavy Industries and Public Enterprises, "Automotive Mission Plan, 2016–26, A roadmap for Indian Automotive industry-

automobiles, tractors and auto components," Government of India, New Delhi, 2016.

[117] S. Naik, D. Johnson, L. Fromm, J. Koszewnik, F. Redon, G. Regner and N. Abani, "Achieving Bharat Stage VI Emissions Regulations While Improving Fuel Economy with the Opposed-Piston Engine," *SAE International Journal of Engines*, vol. 10, no. 1, pp. 17-26, 2017.

[118] M. Muralidharan, A. Srivastava and M. Subramanian, "Full Load Investigation of CNG—Diesel Dual-Fuel Heavy-Duty Engine with Selective Catalytic Reduction on Engine Performance and Emissions for Its Potential Use," *SAE Int. J. Engines*, vol. 15, no. 3, 2022.

[119] M. Muralidharan, A. Srivastava and M. Subramanian, "Optimization of CNG-Diesel Dual Fuel in Heavy Duty Multi-Cylinder Engine with After-treatment Device for the Performance and Emission under Different Load Condition," *SAE Int. J. Engines*, vol. 15, no. 4, 2022.

[120] G. Rhys-Tyler, W. Legassick and M. Bell, "The significance of vehicle emissions standards for levels of exhaust pollution from light vehicles in an urban area," *Atmospheric Environment*, vol. 45, no. 19, pp. 3286-3293, 2011.

APPENDIX

Equations used in the Abstract

$$ESR = \frac{\dot{m}_{CNG} \times LHV_{CNG}}{\dot{m}_{CNG} \times LHV_{CNG} + \dot{m}_{D} \times LHV_{D}} \times 100\ \%$$

$$MFSR = \frac{\dot{m}_{CNG}}{\dot{m}_{CNG} + \dot{m}_{D}} \times 100\ \%$$

$$CHV = \frac{\dot{m}_{CNG} \times LHV_{CNG} + \dot{m}_{D} \times LHV_{D}}{\dot{m}_{CNG} + \dot{m}_{D}}$$

$$RAFR = \frac{\dot{m}_{air}}{\dot{m}_{D} \times \left(\frac{A}{F}\right)_{st.D} + \dot{m}_{CNG} \times \left(\frac{A}{F}\right)_{st.CNG}}$$

$$\text{Brake Thermal Efficiency (BTE)} = \frac{3600 \times BP}{\dot{m}_{CNG} \times LHV_{CNG} + \dot{m}_{D} \times LHV_{D}} \times 100\ \%$$

$$VE = \frac{(\dot{m}_{air} + \dot{m}_{CNG}) \times n_{r}}{\rho_{a} \times V_{d} \times N_{e}}$$

$$bsfc = \frac{\dot{m}_{CNG} + \dot{m}_{D}}{BP} \times 1000$$

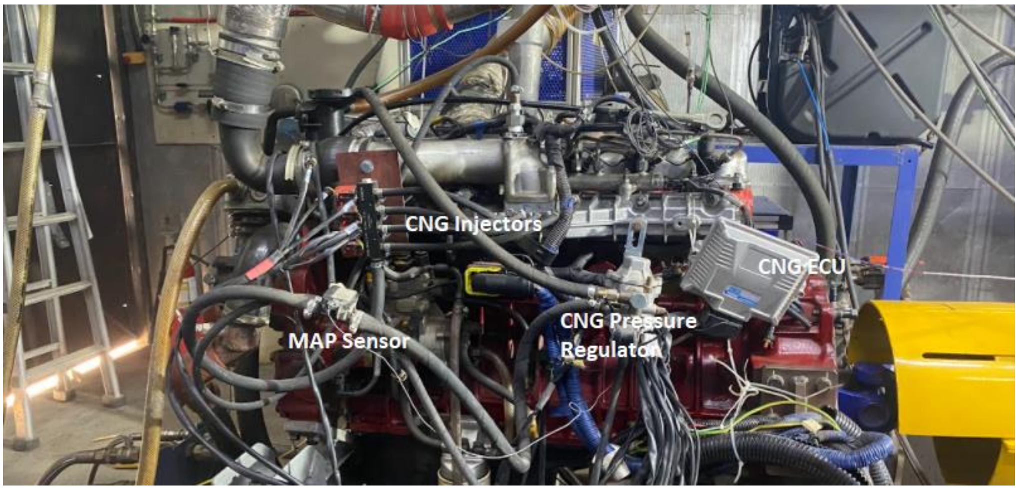
$$bsec = \frac{\dot{m}_{CNG} \times LHV_{CNG} + \dot{m}_{D} \times LHV_{D}}{BP}$$

$$C_xH_y + a(O_2 + 3.76N_2) \rightarrow bCO_2 + cH_2O + a3.76N_2$$

Where, $\dot{m}_{CNG}$ (kg/h) and $\dot{m}_{D}$ (kg/h) represent the mass flow rate of CNG and diesel respectively, while LHV_{CNG} (MJ/kg) and LHV_{D} (MJ/kg) denote the LHV of methane and diesel, BP represents brake power. $\dot{m}_{air}$ represent the mass flow rate of air and $\left(\frac{A}{F}\right)_{st.}$ refers to the stoichiometric equivalence ratio of diesel or methane, ρ_a is air density, n_r *is* number of crankshaft rotations for a complete engine cycle (for 4-stroke engine $n_r = 2$) N_e is engine speed and V_d is theoretical volume of engine/cylinder.

Additional Photographs of Test Set-up

Development of CNG Kit Photographs


CNG Injectors
CNG ECU
CNG Pressure Regulator
MAP Sensor

Thesis

ORIGINALITY REPORT

11%	6%	10%	1%
SIMILARITY INDEX	INTERNET SOURCES	PUBLICATIONS	STUDENT PAPERS

PRIMARY SOURCES

1	M. Muralidharan, Ajay Kumar, M. Subramanian. "Investigation of Performance and Exhaust Emissions of Compressed Natural Gas-Diesel Dual-Fuel Engine under Different Engine Speed and Load Conditions", SAE International Journal of Engines, 2021 Publication	2%
2	M. Muralidharan, Ajay Srivastava, M. Subramanian. "Full Load Investigation of CNG–Diesel Dual-Fuel Heavy-Duty Engine with Selective Catalytic Reduction on Engine Performance and Emissions for Its Potential Use", SAE International Journal of Engines, 2021 Publication	1%
3	M. Muralidharan, Ajay Kumar, M. Subramanian. "In-Cylinder Combustion Studies of Diesel-Compressed Natural Gas Dual Fuel with Increasing Energy Fraction and Its Effect on Emissions", SAE International Journal of Engines, 2021 Publication	<1%

4	M. Muralidharan, Ajay Srivastava, M Subramanian. "A Technical Review on Performance and Emissions of Compressed Natural Gas - Diesel Dual Fuel Engine", SAE International, 2019 Publication	<1 %
5	Submitted to University of Petroleum and Energy Studies Student Paper	<1 %
6	dr.ddn.upes.ac.in:8080 Internet Source	<1 %
7	qmro.qmul.ac.uk Internet Source	<1 %
8	Homa Hosseinzadeh-Bandbafha, Shahin Rafiee, Pouya Mohammadi, Barat Ghobadian et al. "Exergetic, economic, and environmental life cycle assessment analyses of a heavy-duty tractor diesel engine fueled with diesel–biodiesel-bioethanol blends", Energy Conversion and Management, 2021 Publication	<1 %
9	worldwidescience.org Internet Source	<1 %
10	utpedia.utp.edu.my Internet Source	<1 %
11	Soo-Young No. "Application of Liquid Biofuels to Internal Combustion Engines", Springer	<1 %