EXPERIMENTAL INVESTIGATION OF A DUAL-FUEL COMPRESSION IGNITION ENGINE FOR IMPROVEMENT OF EMISSIONS AND THERMAL EFFICIENCY

A Thesis
Submitted in Partial Fulfillment of the Requirement for
the Award of the Degree of

DOCTOR OF PHILOSOPHY

By

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Dedicated to my beloved parents…

Late Lungaigaliu Kamei

Ajiliu Kamei

Kandai Kamei

Declaration

I hereby certify that the material presented in this thesis entitled, **"Experimental investigation of a dual-fuel compression ignition engine for improvement of emissions and thermal efficiency"**, is my own research work carried out under the supervision of Prof. Niranjan Sahoo and Dr. V.V.D.N. Prasad, and to the best of my knowledge the thesis contains no materials which have been written or published by any other person, except otherwise where stated. This thesis or any part of it has not been previously submitted for the award of a degree or diploma at IIT Guwahati or any Institution.

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It is certified that the work contained in the thesis entitled, "**Experimental investigation of a dual-fuel compression ignition engine for improvement of emissions and thermal efficiency"** by **Wittison Kamei**, a student in the Department of Mechanical Engineering, Indian Institute of Technology Guwahati, India, for the award of the degree of **Doctor of Philosophy** has been carried out under our supervision. This research work has not been submitted elsewhere for the award of any diploma or degree.

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Abstract

Alternative fuels such as natural gas, liquefied petroleum gas (LPG), and alcohols can be utilised in diesel engines through the dual-fuel mode of combustion. In the present research, LPG and dimethyl ether (DME) are studied as fuels for dual-fuel combustion with conventional diesel. LPG is a widely used alternative fuel with millions of auto LPG vehicles driven worldwide. DME is a promising alternative fuel as it has excellent properties such as high cetane number and high oxygen content, and it can be produced from different feedstocks such as coal and biomass. DME and LPG are miscible. Further, DME can be handled like LPG. Therefore, there is a potential to use the two fuels together.

Dual-fuel engines fumigated with gaseous hydrocarbon fuels such as natural gas or LPG face the challenges of low thermal efficiency and high emissions of carbon monoxide (CO) and unburnt hydrocarbon (HC) at low loads. For natural gas or methane fumigated dual-fuel engines, HC reductions are difficult with diesel oxidation catalysts. Therefore, in the present research, LPG-diesel dual-fuel engine is studied for engine performance, emissions, and combustion characteristics, with emphasis on the reductions of CO and HC at low loads. Two different oxidation catalytic converters (OCCs) – a commercial OCC and a customised OCC - are used to study CO and HC reductions, from low-load conditions to 70% load. Further, the thermal efficiency and emissions of the LPG-diesel dual-fuel mode are improved upon by adding dimethyl ether (DME) as a co-fumigant with LPG.

In the present research, a constant speed single-cylinder diesel engine is converted into the dual-fuel mode. At 20%, 40% and 70% load, diesel energy replacement rates of about 36%, 46% and 64% by the gaseous fuels are selected for detailed study of engine performance, emissions, and combustion characteristics. The three load conditions correspond to brake mean effective pressures (BMEPs) of 1.3 bar, 2.6 bar, and 4.4 bar. The LPG-diesel dual-fuel mode suffers from low brake thermal efficiency (BTE) at below half-load operations. However, at 70% load, the engine performance of the LPG-diesel dual-fuel mode is the same as that of the diesel mode. At the lower loads, the LPG-diesel dual-fuel combustion is associated with poor combustion characteristics. At the lowest tested load, with 36% LPG substitution, CO and HC emissions increase by up to 208%

and 173%, respectively. While CO emissions are readily oxidised by either of the OCCs at all the tested loads, the dual-fuel mode HC conversion efficiencies with the commercial OCC are limited to about 60% at 20-40% loads. The customised OCC achieves 77-85% HC conversion efficiencies at the 20-40% loads. With the customised OCC, the resulting dual-fuel mode HC emissions at all the tested loads are 38-87% lower than those of the diesel mode. Further, smoke opacity is reduced by up to 96% as compared to the diesel mode.

The addition of DME as a co-fumigant with LPG results in significant improvements of BTE. The DME-LPG-diesel dual-fuel mode BTEs are higher than those of the diesel mode. At below half-load operations, the DME-LPG-diesel dual-fuel mode achieves BTEs, which are 23.58-24.09% higher than those of the LPG-diesel dual-fuel mode, and 3.48-14.29% higher as compared to the diesel mode. The improvements in BTE are associated with improved combustion characteristics. The DME-LPG-diesel dual-fuel mode combustion exhibits low temperature reaction (LTR) and high temperature reaction (HTR) followed by diesel diffusion combustion. Further, compared with the diesel mode, CA50 and CA90 advance significantly. Exergy analysis shows that the addition of DME as a co-fumigant with LPG significantly improves exergy efficiency. With the combined approach of using the DME-LPG co-fumigation and the customised OCC, HC emissions are 85-89% lower and CO emissions are 89-94% lower as compared to the diesel mode. Further, smoke opacity values are 78-94% lower than those of the diesel mode. Therefore, the DME-LPG co-fumigated dual-fuel engine, equipped with the customised oxidation catalyst, outperforms the diesel mode and achieves lower emissions. These findings on the improvement in engine performance and the reduction in emissions are new contributions of the present research which show that the DME-LPG co-fumigated dual-fuel engine can achieve lower emissions and higher thermal efficiency when compared with the diesel mode.

Keywords: Dual-fuel engine, combustion, performance, emissions, oxidation catalyst, exergy, liquefied petroleum gas, dimethyl ether.

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26 June 2021 **Wittison Kamei**

Contents

Nomenclature

Abbreviations

ASTM	American Society for Testing and Materials
aTDC	After Top Dead Centre
BMEP	Brake Mean Effective Pressure (bar)
BP	Brake Power (kW)
bTDC	Before Top Dead Centre
BTE	Brake Thermal Efficiency
CA	Crank Angle
CA10	Crank Angle of 10% of Total Cumulative Heat Release
CA50	Crank Angle of 50% of Total Cumulative Heat Release
CA90	Crank Angle of 90% of Total Cumulative Heat Release
CF	Correction Factor
CHNS	Carbon, Hydrogen, Nitrogen, Sulfur
CMGT	Cylinder Mean Gas Temperature
CNG	Compressed Natural Gas
cpsc	Cells Per Square Centimeter
CR	Compression Ratio
DEE	Diethyl Ether
DME	Dimethyl Ether
EGR	Exhaust Gas Recirculation
ESC	European Steady-state Cycle
FS	Full Scale
GC	Gas Chromatography
GHG	Greenhouse Gas
GHSV	Gas Hourly Space Velocity (per hour)
HHV	Higher Heating Value (kJ/kg)
HTR	High Temperature Reaction
IMEP	Indicate Mean Effective Pressure (bar)
ITE	Indicated Thermal Efficiency

IVC	Intake Valve Closure
LHV	Lower Heating Value (kJ/kg)
LPG	Liquefied Petroleum Gas
LTR	Low Temperature Reaction
MFC	Mass Flow Controller
NDIR	Non-Dispersive Infrared
NHRR	Net Heat Release Rate (J/°CA)
NMHC	Non-Methane Hydrocarbon
OCC	Oxidation Catalytic Converter
PGM	Platinum Group Metal
ppm	Parts Per Million
PTFE	Polytetrafluoroethylene
RCCI	Reactivity Controlled Compression Ignition
RCE	Relative Cycle Efficiency
RPM	Revolutions Per Minute
SCFM	Standard Cubic Feet Per Minute
SCMH	Standard Cubic Meter Per Hour
SI	Spark Ignition
SOF	Soluble Organic Fraction
TDC	Top Dead Centre
VCR	Variable Compression Ratio

Notations

Cp	Specific heat capacity (kJ/kgK)
Ex	Specific chemical exergy (kJ/kg)
Exo	Molar chemical exergy (kJ/kmol)
k	Polytropic index
$\dot{m}$	Mass flow rate (kg/s)
P	In-cylinder pressure (kPa)
q	Heat added (J/degree CA)
R	Specific gas constant (kJ/kgK)

$\bar{R}$ Universal gas constant (kJ/kmolK)

T Cylinder mean gas temperature (K)

Tr Torque (Nm)

U Uncertainty

V Volume (m^3)

x Mole fraction

Greek Symbols

Y Number of moles per kg of gas mixture

ΔH Enthalpy of combustion (kJ/kmol)

ϒ Specific heat ratio

θ Crank angle (degree)

ρ Density (kg/m^3)

ω Engine speed in revolutions per minute

Subscripts

ci Coolant-in

co Coolant-out

eg Exhaust gas

Superscripts

o Dead state

std Standard state

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Chapter – 1

Introduction

Overview: *Alternative fuels have the potential to lessen both environmental and economic burdens. Alternative gaseous fuels can be used along with diesel in dual-fuel engines. Two such alternative gaseous fuels, liquefied petroleum gas (LPG) and dimethyl ether (DME), are selected for the present study. The motivation to conduct research on the use of these alternative fuels in the dual-fuel mode is discussed in this chapter. Further, the objectives of the present research are presented. At the end of the chapter, the thesis organisation is explained.*

1.1 Motivation

Alternative fuels have the potential to mitigate environmental pollutants and greenhouse gas (GHG) emissions. The alternative fuels such as natural gas and LPG have higher hydrogen to carbon ratio, which implies that there is potential for lower levels of CO_2 emissions from the combustion of these fuels. Further, alternative fuels can lessen the burden of crude oil imports. Applications of gaseous fuels and alcohols in spark ignition (SI) engines are well established. Such alternative fuels can also be utilised in compression ignition engines through the dual-fuel mode of combustion. In dual-fuel engines, high volatility fuel or gaseous fuel is fumigated at the intake manifold or intake port while diesel is directly injected into the combustion chamber. The injected diesel fuel acts as the source of ignition. A dual-fuel engine can also have directly injected natural gas instead of fumigation [Westport, 2013]. It is well established that the dual-fuel mode of combustion has potential to reduce smoke and nitrogen oxide (NOx) emissions, when compared with diesel engines. Further, the dual-fuel mode allows the combustion of gaseous fuels or alcohols at diesel engines' compression ratios (CRs). Therefore, combusting the gaseous fuels or alcohols in the dual-fuel mode has the advantage of higher compression ratio, as compared to their combustion in SI engines. Zheng et al. [2019] report that a natural gas-diesel dual-fuel engine's fuel consumption was lower than that of a natural gas SI engine.

Currently, dual-fuel engine technologies are not widely used. However, information is available on various dual-fuel technologies developed by different manufacturers. In 2019, the Volvo FH420

LNG-diesel, a Euro VI certified dual-fuel truck, passed the formal European in-service conformity test for the regulated emissions [Vermeulen, 2019]. In December 2013, Westport launched their next-generation high pressure direct injection system, Westport™ HDPI2.0, which can replace over 90% diesel with natural gas [Westport, 2013]. In September 2020, Weichai Westport received the China VI certification for its 12 litre engine equipped with the Westport™ HDPI2.0 [Westport, 2020]. Cummins produces QSK50 natural gas-diesel dual-fuel engine for land-based oil and gas drilling applications [Cummins, 2020].

Literature shows that dual-fuel mode of combustion, utilising alternative fuels such as natural gas [Mitchell and Oslen 2018; Mahla et al., 2018] and ethanol [Pedrozo et al., 2018], resulted in low thermal efficiency and high levels of carbon monoxide (CO) and unburnt hydrocarbon (HC) emissions at low-load operations. Therefore, effective oxidation catalysts are required to control the CO and HC emissions from dual-fuel engines. Dual-fuel engines such as the Volvo D13C-Gas engines used a dedicated oxidation catalyst for methane oxidation [Souto et al., 2014]. While oxidation catalysts can readily oxidise CO, results from the literature indicate that HC emissions from methane or natural gas fumigated dual-fuel engines are difficult to reduce with oxidation catalysts [Di Iorio et al., 2016; Johnson et al., 2018].

LPG is a high-octane gaseous fuel that can be readily fumigated for dual-fuel combustion. LPG is the most popular alternative fuel used in vehicles [Mintz et al., 2014]. From the year 2000 to 2019, the number of auto LPG driven vehicles grew around four times globally to reach over 27 million vehicles [World LPG Association, 2019]. LPG, being in the liquid form, has high energy density and, therefore, requires lower storage volume than methane and natural gas. Moreover, the storage, handling, and distribution infrastructures for LPG is well established. The majority of LPG comes from processing during natural gas and oil extraction from the earth [European LPG Association, 2013].

It is reported that LPG-diesel dual-fuel engines have low thermal efficiency and high CO and HC emissions at low-load operations [Polk et al., 2014; Gibson et al., 2011; Ashok et al., 2015; Ganesan and Ramesh, 2001]. The addition of DME as a co-fumigant with propane significantly improved brake thermal efficiency (BTE) and reduced HC emissions of a propane-diesel dual-fuel engine at low load [Prabhakar et al., 2015]. DME is a potential alternative fuel with excellent properties such as high cetane number [Ohno, 2001], high oxygen content, and an absence of carbon-carbon bond in the fuel molecule. DME can be produced from biomass [Nakyai et al., 2020]. Further, DME can be handled like LPG [Shikada et al., 1998], and they are miscible. Therefore, DME and LPG have the potential to be used together as fuels.

Based on the above discussion, there is a need to study LPG-diesel dual-fuel combustion with emphasis on reduction of CO and HC emissions by using a suitable oxidation catalyst. Further, there is a need to improve the thermal efficiency of LPG-diesel dual-fuel engines at low loads. As discussed above, DME addition as a co-fumigant to propane in dual-fuel mode has the potential to improve BTE and reduce HC emissions at low load. However, limited data are available on this topic. Therefore, there is a need for research on the dual-fuel mode of combustion with DME and LPG co-fumigation.

1.2 Objectives of the Present Investigation

The overall objective of the present research is to improve thermal efficiency and reduce emissions of a dual-fuel compression ignition engine using LPG and DME as the fumigants. The research work is carried out for two different modes of fumigation: LPG fumigation and DME-LPG co-fumigation. For the LPG-diesel dual-fuel mode, the present research work is targeted at a detailed study of the engine performance, combustion, and emissions, with emphasis on reduction of CO and HC emissions at low-load conditions. The reductions of CO and HC emissions are studied with two different oxidation catalytic converters (OCCs). In the DME-LPG-diesel dual-fuel mode, the research work is targeted not only at improving the emissions but also to achieve thermal efficiency higher than that of the LPG-diesel dual-fuel mode. The specific objectives of the present research work are given as follows.

1. Reduction of emissions and investigation of engine performance and combustion for the LPG-diesel dual-fuel mode of combustion:

   The first part of the research is targeted at the reduction of emissions and investigation of the engine performance and combustion for an LPG-diesel dual-fuel engine, with emphasis on low-load conditions. Reductions of CO and HC are targeted with a customised oxidation catalyst. The performance of the customised oxidation catalyst is compared with that of a commercial one.

2. Improvement of thermal efficiency and reduction of emissions by DME-LPG co-fumigation and oxidation catalysts:

   The second part of the research is targeted at the improvement of thermal efficiency by co-fumigation of DME with LPG and the reduction of emissions with the use of the oxidation catalysts. The engine performance and emissions are investigated along with a detailed combustion study.

3. The present research is also targeted at quantifying improvements in exergy efficiencies of the DME-LPG-diesel dual-fuel mode with reference to the LPG-diesel dual-fuel mode and the diesel mode.

1.3 Organisation of the Thesis

Chapter 1 is an introduction to the present thesis. The motivation to carry out the present research is discussed. Further, the objectives of the present research are presented. At the end of the chapter, the thesis organisation is briefly described.

Chapter 2 presents the literature review. The review discusses the challenges of dual-fuel engines in terms of thermal efficiency and emissions of CO and HC. The chapter also discusses the literature on the improvement in thermal efficiencies and reductions of CO and HC emissions for dual-fuel engines. Further, the potentials of DME and LPG as fuels for dual-fuel engines are discussed. The importance of exergy analysis and the literature on exergy analyses of dual-fuel engines using various alternative fuels are also discussed. Research gaps from the literature review are presented at the end of the chapter.

Chapter 3 presents the details of the test engine, and the conversions carried out to convert the engine into the LPG-diesel dual-fuel mode and the DME-LPG-diesel dual-fuel mode. Further, the test set-up along with the test instruments and the details of oxidation catalysts, are described. The experimental test procedures and the experimental data analyses methods are also described in this chapter.

Chapter 4 presents the experimental investigation on the LPG-diesel dual-fuel engine. The dual-fuel engine performance, combustion and emissions characteristics are studied and compared with those of the diesel mode. Further, the emissions reductions of the dual-fuel mode by use of oxidation catalysts, with emphasis on CO and HC emissions, are presented.

Chapter 5 presents the experimental investigation on the DME-LPG co-fumigated dual-fuel engine. The engine performance, combustion and emissions of the DME-LPG-diesel dual-fuel mode are presented in this chapter. The dual-fuel emissions are further studied with the oxidation catalysts.

Chapter 6 presents the exergy analysis of the LPG-diesel dual-fuel mode and the DME-LPG-diesel dual-fuel mode. The exergetic performance and the exergy distribution for the LPG fumigation and the DME-LPG co-fumigation are presented in detail.

Chapter 7 discusses the conclusions of the present research works. It also presents the application potential of the outcomes of the present research. Finally, the future scope of research is presented.

Chapter – 2

Literature Review

Overview: *The technical challenges associated with dual-fuel engines, in terms of low thermal efficiency and high CO and HC emissions at low loads, are discussed in this chapter. Further, the use of DME and the deployment of oxidation catalyst as promising options to address such technical challenges are also presented. The need for an effective oxidation catalyst to oxidise dual-fuel mode HC emissions is discussed in detail. DME and LPG are presented as two potential fuels for dual-fuel engines. In this chapter, the use of exergy analysis as a useful method to analyse dual-fuel engine performance is also discussed.*

2.1 Challenges of Dual-fuel Engines in Terms of BTE and Emissions of CO and HC

Alternative fuels, such as natural gas, LPG, and alcohols, can be utilised in diesel engines through the dual-fuel mode. However, at low loads, high emissions of CO and HC are reported for dual-fuel engines fumigated with LPG [Ganesan and Ramesh, 2001; Saleh, 2008], ethanol [Pedrozo et al., 2018; Nord et al., 2017], biogas [Sarkar et al., 2018a; Bora et al., 2014], natural gas [Mitchell and Oslen 2018; Mahla et al., 2018] and methane [Shoemaker et al., 2012; Gibson et al., 2011]. The trend of high levels of CO and HC emissions are not reported for hydrogen fumigated dual-fuel engines [Verma et al., 2017; Fang et al., 2014], as hydrogen is not a hydrocarbon fuel.

Results of CO emissions of a propane-diesel dual-fuel engine are shown in Fig. 2.1 [Saleh, 2008]. The results show that CO increased by about 2.5 times for 40% propane fumigation at 10% load. The reason for the high CO emissions was attributed to incomplete combustion due to low charge temperature. Results of HC emissions for an LPG-diesel dual-fuel engine at 20% load are shown in Fig. 2.2 [Poonia et al., 1999]. The results show that HC increased drastically from about 500 ppm to about 2000 ppm when diesel injection quantity was reduced from 8.4 mg/cycle to 4.6 mg/cycle. The reason for the high HC emissions was attributed to poor combustion. Similar trends of high emissions of CO and HC at low loads are reported for propane fumigated dual-fuel engines [Polk et al., 2014; Shoemaker et al., 2012; Gibson et al., 2011; Prabhakar et al., 2015].

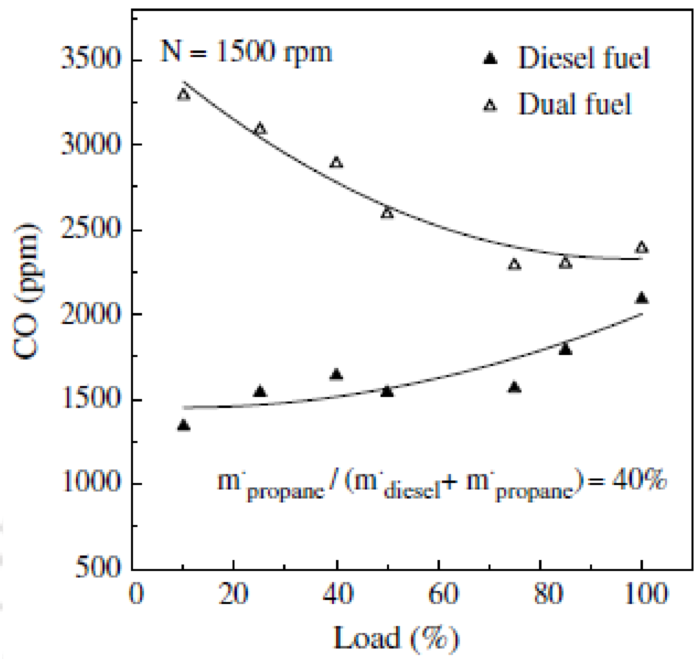


Fig. 2.1 CO emissions of a propane-diesel dual-fuel engine at different loads [Saleh, 2008].

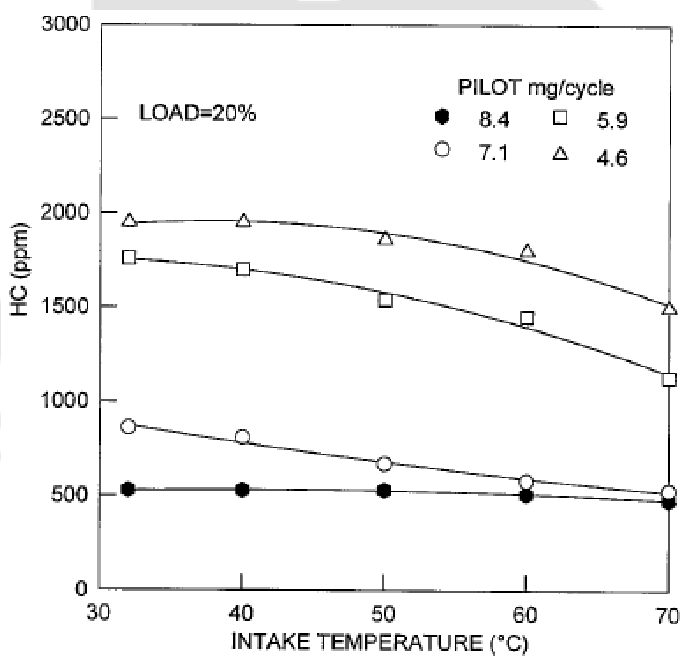


Fig. 2.2 HC emissions of an LPG-diesel dual-fuel at different intake temperatures [Poonia et al., 1999].

Apart from the high emissions of CO and HC, brake thermal efficiencies of dual-fuel engines also need improvement at low loads. The trend of low BTE at low loads is reported for dual-fuel engines fumigated with fuels such as propane [Gibson et al., 2011], LPG [Poonia et al., 1999; Ashok et al., 2015], natural gas [Mitchell and Oslen 2018; Mahla et al. 2018], biogas [Bora et al., 2014], and alcohol [Pedrozo et al., 2018; Imran et al., 2013]. BTE results of an LPG-diesel dual-fuel engine at 20% load are given in Fig. 2.3 [Poonia et al., 1999]. The results show that, at a given engine intake charge temperature, BTE reduced significantly when diesel pilot fuel quantity was decreased from 8.4 mg/cycle to 4.6 mg/cycle.

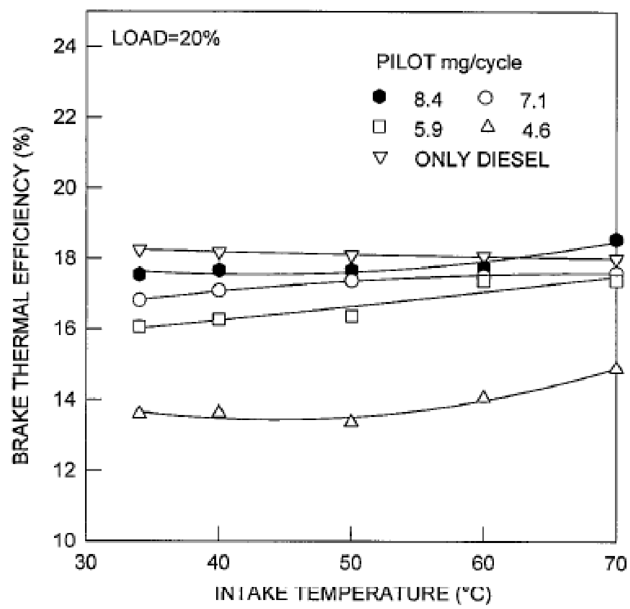


Fig. 2.3 BTE of LPG-diesel dual-fuel engine at different intake temperatures [Poonia et al., 1999].

2.2 Improvement of BTE and Reduction of CO and HC Emissions in Dual-fuel Engines

Various methods for improvements of dual-fuel engine performance and emissions have been reported in the literature. These methods include strategies such as optimisation of engine operating parameters [Sahoo et al., 2009; Bora et al., 2016; Papagiannakis et al., 2016], combustion chamber modification [Banapurmath et al., 2014], and heating of engine intake air [Poonia et al., 1999; Sarkar and Saha, 2018a; Sarkar and Saha, 2020]. Reductions of CO and HC emissions in dual-fuel engines by use of oxidation catalysts are also reported [Bittner and Aboujaoude 1992; Zhang et al., 2010; Chen et al., 2019].

Bora et al. [2016] varied the compression ratio and diesel injection timing of a biogas-diesel dual-fuel engine. The CR was varied from 16 to 18, while the injection timing was varied from 26 degrees crank angle (°CA) before top dead centre (bTDC) to 32 °CA bTDC. At 4.24 bar brake mean effective pressure (BMEP) and a high level of diesel replacement by biogas, they reported that the operating condition corresponding to CR of 18 and diesel injection timing of 29 ^{o}CA bTDC resulted in maximum BTE and least CO and HC emissions. However, their results show that the dual-fuel mode BTE and emissions of CO and HC at this operating condition still deteriorated when compared with the diesel mode. Papagiannakis et al. [2016] carried out a numerical evaluation of a turbo-charged methane-diesel dual-fuel engine. For high levels of diesel replacements at partial loads, their results indicated that increasing the CR and advancing the diesel injection timing could improve engine power output and reduce CO emissions. However, they reported that the use of a higher compression ratio and an advanced diesel injection timing with a lower level of diesel replacement by the gaseous fuel might be harmful to the engine structure due to a considerable increase in the maximum cylinder pressure. Their results also show that NOx increased with the higher CR and advanced injection timing. Carlucci et al. [2014] reported that CO and HC emissions of a producer gas-biodiesel dual-fuel engine reduced with pilot fuel split injection. Sudhir et al. [2003] advanced the diesel injection timing of an LPG-diesel dual-fuel engine from 23 ^{o}CA bTDC to 30 ^{o}CA bTDC. They observed that BTEs of the dual-fuel mode at 25-75% loads improved when the diesel injection timing was advanced, however, NOx increased.

Sarkar and Saha [2018b] reported that CO and HC emissions reduced by 29.41% and 14.21%, respectively and BTE increased by 5.72% when the intake charge of a biogas-diesel dual-fuel engine was preheated to 55 ± 2 ^{o}C at a part load of 4.36 Nm. The energy shares of biogas in total fuel (diesel + biogas) were 47.38% with the preheating and 46.69% without the preheating. Their results show that the dual-fuel mode CO and HC emissions after the reductions were still higher when compared with the diesel mode. Results of Poonia et al. [1999] show that, for an optimised pilot fuel quantity, BTE of an LPG-diesel dual-fuel engine at 20% load was improved from 17.5% to 19.4% with an optimal combination of different methods consisting of intake air heating, exhaust gas recirculation (EGR), and throttling of the intake. They observed a corresponding decrease of HC from 530 ppm to 350 ppm while CO increased. With intake air preheating in the dual-fuel mode, NOx emissions are reported to increase [Sarkar and Saha, 2018b; Sarkar and Saha, 2020].

Carpenter et al. [2017] varied intake manifold boost pressures and EGR rates in a propane-diesel dual-fuel engine at 25% load. They reported that HC emissions reduced with an increase in intake manifold boost pressure and EGR. The reason for the reduction in HC was attributed to an increase in bulk cylinder temperature when boost pressure and EGR were increased. They observed that higher EGR and boost pressure resulted in higher intake charge temperature and, further, burning of HC in recycled EGR also increased the bulk cylinder temperature. Their results show that CO also decreased with an increase in EGR. However, an increase in boost pressure increased the CO emissions.

Oxidation catalysts are reported to effectively reduce the CO emissions of methane or natural gas fumigated dual-fuel engines [Di Iorio et al., 2016; Dishy et al., 1995; Bittner and Aboujaoude 1992]. CO emissions reductions by oxidation catalysts in the range of 93% to almost 100% are reported for dual-fuel engines [Bittner and Aboujaoude 1992; Dishy et al., 1995]. However, low levels of HC conversion efficiencies by oxidation catalysts are reported for dual-fuel engines fumigated with methane or natural gas [Di Iorio et al., 2016; Dishy et al., 1995]. Dishy et al. [1995] studied an oxidation catalyst for a natural gas fumigated dual-fuel engine. They reported that the exhaust HC reduction with the oxidation catalyst was limited to 10-30% at low-load conditions. For a methane-diesel dual-fuel engine operated at full load, Bittner and Aboujaoude [1992] reported that 35% of non-methane hydrocarbon (NMHC) was reduced by an oxidation catalyst as compared to 85% reduction for diesel mode. Johnson et al. [2018] studied the reductions of dual-fuel mode methane and NMHC emissions by oxidation catalysts. They reported that oxidation catalysts reduced NMHC by more than 50%; however, there was no reduction of the methane emissions. The results of Di Iorio et al. [2016], as shown in Fig. 2.4, show that a platinum-based oxidation catalyst could not significantly reduce HC emissions from a methane-diesel dual-fuel engine. In their experiments, the same catalyst was able to reduce about 77% of HC in the diesel mode. Liu et al. [2001] studied an oxidation catalyst conversion of exhaust methane in a natural gas fumigated dual-fuel engine. Their results show that methane reduction by the oxidation catalyst was only 20% at 698 K exhaust temperature. Further, they achieved 77% methane conversion when the oxidation catalyst reactor temperature was enhanced to 850 K. The effect of low engine exhaust gas temperature resulting in a low level of HC conversion by oxidation catalyst was also observed for ethanol-diesel dual-fuel engine [Tsang et al., 2010]. Tsang et al. [2010] observed that the exhaust HC oxidation for an ethanol-diesel dual-fuel engine improved significantly to over 80% when exhaust gas temperature was increased from 200 ^{o}C to over 350 ^{o}C. However, at the 200 ^{o}C exhaust temperature, the HC reduction was less than 10%. Zhang et al. [2010] studied reductions of HC emissions by an oxidation catalyst for a methanol-diesel dual-fuel engine. When tested over the Japanese 13 Mode test cycle, they observed that the HC conversion was about 15-

51%, corresponding to methanol substitution levels of 10-30%. Chen et al. [2019] studied a methanol-diesel dual-fuel engine under the European Steady-state Cycle (ESC) conditions, excluding the idle mode. Their results show that diesel oxidation catalyst effectively reduced the HC emissions at all the test points. Ren et al. [2018] used a prototype oxidation catalyst, with Pt:Pd ratio of 4:1 and a catalyst loading of 1.1301 kg/m^3, in a gasoline-diesel dual-fuel engine. They reported effective conversions of exhaust HC when the exhaust gas temperature was above 250 ^{o}C. The results of Ren et al. [2018] for the 13-Mode ESC test is shown in Fig. 2.5. Effective conversion of HC emissions by oxidation catalyst is also reported for gasoline-diesel reactivity control compression ignition (RCCI) combustion [Piqueras et al., 2019].

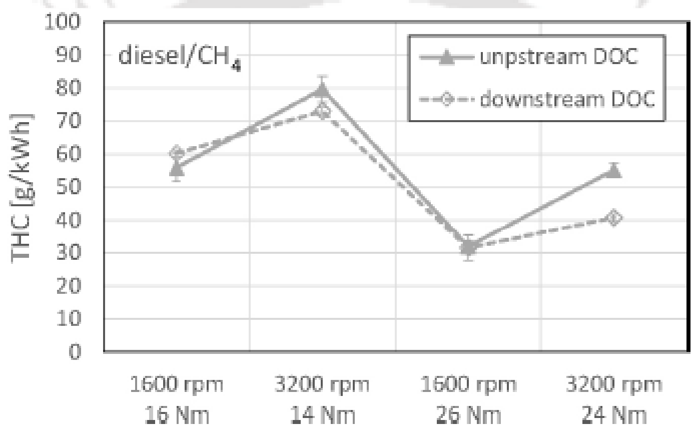


Fig. 2.4 Total hydrocarbon emissions of a methane-diesel dual-fuel engine with a platinum-based oxidation catalyst [Di Iorio et al., 2016].

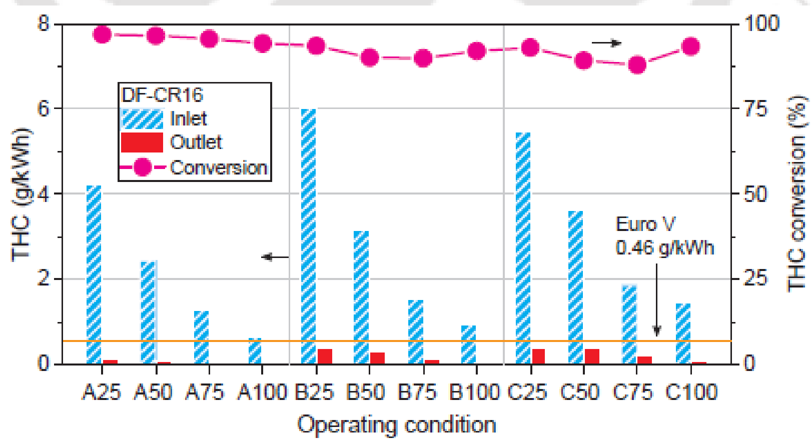


Fig. 2.5 Total hydrocarbon emissions at inlet and outlet of a Pt:Pd oxidation catalyst for a gasoline-diesel dual-fuel mode. A, B, C denote the three different speed conditions of the 13-Mode ESC test, and the numerical values given with these letters indicate the load percentages [Ren et al., 2018].

The above literature review indicates that HC conversion by oxidation catalyst is more efficient for a dual-fuel engine employing gasoline fumigation compared to dual-fuel engines employing natural gas or methane fumigation. For a methane-diesel dual-fuel engine, Bittner and Aboujaoude [1992] attributed the reason for low HC conversion by oxidation catalyst to the presence of higher concentrations of light hydrocarbons - propane and ethane - in the engine exhaust. For a methane-diesel dual-fuel engine installed with a conventional oxidation catalyst, Carlucci et al. [2020] attributed the reason for an increase in light-off temperatures of unburnt hydrocarbon conversion to an increase in short-chain unburnt hydrocarbon.

Therefore, the above literature review indicates that dual-fuel engines utilising gaseous hydrocarbon fuels such as LPG would require an effective oxidation catalyst to reduce HC emissions at low loads. García et al. [2019] reported that thermal inertia at transient engine operating conditions resulted in a better performance of oxidation catalyst at low-load conditions. For stationary engines operating under low-load steady-state conditions, the effect of thermal inertia will be absent. Therefore, for dual-fuel engines operating under low-load steady-state conditions, achieving an efficient performance of the oxidation catalysts could be more challenging. In India, stationary diesel engines are widely used for agricultural water pumping applications [Agriculture Census Division, 2021]. Such engines could be operated on the dual-fuel mode with effective oxidation catalysts.

2.3 DME as Fuel for Dual-fuel Engines

It is reported that co-fumigation of dimethyl ether [Prabhakar et al., 2015] or diethyl ether (DEE) [Barik and Murugan, 2016] with gaseous fuel improved engine performance in the dual-fuel mode. Prabhakar et al. [2015] studied a dual-fuel engine fumigated with DME and propane. Their results on BTE and total unburnt hydrocarbon emissions at 25% load and engine speed of 1800 revolutions per minute (RPM) are shown in Fig. 2.6 and 2.7. It can be seen from Fig. 2.6 that DME fumigation, in the neat form or as a co-fumigant with propane, resulted in improvement of BTE. Further, the results show that the highest BTE was for the co-fumigation with 20% DME and 30% propane. The results in Fig. 2.7 show that the addition of DME to propane significantly reduced the HC emissions of the dual-fuel mode. The CO and HC emissions with neat DME fumigation [Wang et al., 2013; Wang et al., 2014] or DME and propane co-fumigation [Prabhakar et al., 2015] in dual-fuel mode are reported to be higher as compared to diesel mode. With DME fumigation in diesel engines, the phenomena of low temperature reaction (LTR) and high temperature reaction (HTR) followed by diesel diffusion combustion are reported [Wang et al., 2013; Zhao et al., 2014]. Further, Wang et al. [2013] reported that DME fumigation in diesel engine resulted in an advancement of the position where the cumulated heat release reaches 50% of the total heat release (CA50).

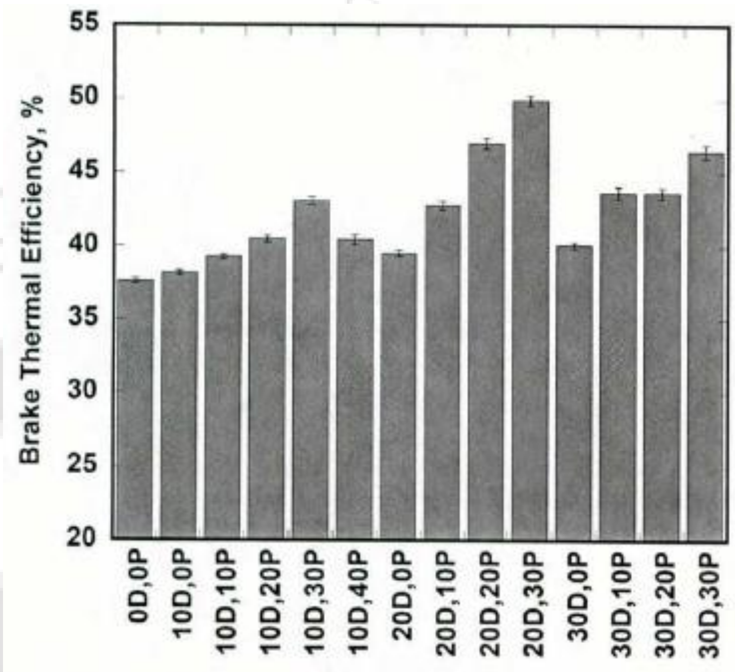


Fig. 2.6 Brake thermal efficiency for different DME and propane substitution levels in dual-fuel mode. D and P denote DME and propane, and the numerical values given with these letters denote the substitution percentages [Prabhakar et al., 2015].

DME is a high cetane fuel, with cetane number in the range of 55-60 [Ohno, 2001], and free from sulfur [Kim et al., 2007]. It is also an oxygenate with 34.8% oxygen content, and the fuel molecule does not have a carbon-carbon bond. Apart from coal and petroleum-based feedstocks, renewable feedstock such as biomass can also be used for the production of DME [Nakyai et al., 2020; Clausen et al., 2010; Shikada et al., 1998]. Based on an economic study, TOYO Engineering corporation estimated that DME price was economically competitive to LPG and LNG prices of 2001-2005 [Mii and Uchida, 2005]. Dimethyl production from shale gas was also estimated to be economically feasible [Mevawala et al., 2019]. DME can be stored and handled like LPG [Shikada et al., 1998]. However, compatible materials need to be used for the storage and handling of DME. Further, DME and LPG are miscible. Therefore, there is a potential to use these two fuels together as fuels.

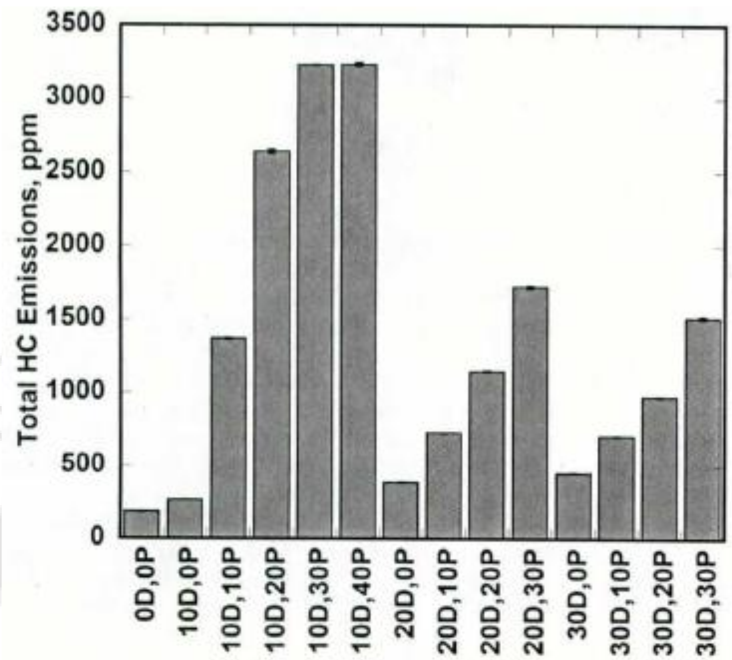


Fig. 2.7 Total hydrocarbon emissions for different DME and propane substitution levels in dual-fuel mode. D and P denote DME and propane, and the numerical values given with these letters denote the substitution percentages [Prabhakar et al., 2015].

2.4 Potential of LPG as Fuel for Dual-fuel Engines

LPG fumigated dual-fuel engines are reported to have engine performance better than that of diesel mode at above mid-load operations [Ganesan and Ramesh, 2001; Ashok et al., 2015]. The low thermal efficiency of LPG-diesel dual-fuel mode at low-load operations could be overcome by the addition of DME as a co-fumigant, as indicated by the literature review above. LPG is an alternative fuel that is widely used. LPG used in vehicles, called auto LPG, is a popular alternative fuel with global consumption reaching 26.835 million tonnes in 2017 [World LPG Association, 2019]. In 2017, the number of auto LPG vehicles reached over 27 million globally [World LPG Association, 2019].

As discussed earlier, LPG-diesel dual-fuel engines suffer from high CO and HC emissions. If the challenges, in terms of the high CO and HC emissions and the low thermal efficiency at low loads, can be improved, LPG can be effectively utilised in dual-fuel engines. Further, the higher H/C ratio of LPG, compared to diesel, also presents a potential to reduce CO_2 emissions from compression ignition engines.

2.5 Understanding Dual-fuel Engines through Exergy Analysis

Exergy or availability of a system is the amount of maximum theoretical work that can be obtained when the system attains equilibrium with its environment through reversible processes. The system is said to be in equilibrium with its environment when the condition of thermal, mechanical, and chemical equilibrium is attained. Exergy destructions in internal combustion engines can be identified and quantified through exergy analysis [Kanoglu et al., 2008]. Exergy analysis helps in more efficient utilisation of resource since it enables the locations, types and true magnitudes of waste and loss to be determined [Moran and Shaprio, 2010]. Exergy analysis with reference to various engine operating parameters helps identify the dominating parameters on different exergy losses [Ma et al., 2020]. When exhaust gas exergy is accounted for in the exergy efficiency estimation, the most efficient operating condition for an engine can be determined by considering both trends of exergy efficiency and the gross indicated fuel conversion efficiency [Mahabadipour et al., 2019].

Exergy analysis is widely used to understand the effect of alternative fuels on exergetic performance of dual-fuel engines. Verma et al. [2018a] carried out exergy analysis of a dual-fuel engine with fumigation of hydrogen and biogas. Their results show that exergy destruction decreased with an increase in hydrogen amount. At full engine load, a hydrogen-diesel dual-fuel engine is reported to have higher exergetic efficiency as compared to the diesel mode [Verma et al., 2018b]. Chintala and Subramanian [2014] reported that hydrogen-diesel dual-fuel mode of combustion resulted in lower irreversibilities of combustion and unburnt fuel. For biogas-diesel dual-fuel mode of operation, the exergetic efficiency is reported to be lower than that of diesel mode [Sarkar & Saha, 2020; Sahoo et al., 2012]. Sarkar and Saha [2020] studied a biogas fumigated dual-fuel engine through analyses of energy and exergy. They reported improvements in exergetic efficiency with intake charge preheating and use of oxygenated pilot fuels. Ma et al. [2020] reported that higher intake temperature increased the exergy efficiency of a methanol-diesel dual-fuel engine. Further, their results show that total exhaust exergy decreased with an increase in intake temperature, which was predominantly due to a decrease in the exhaust chemical exergy.

For the dual-fuel mode of combustion, the engine load has significant effects on the exergy efficiency. The exergy efficiency of a methanol-diesel dual-fuel engine is reported to be lower than that of diesel at low load and vice versa at higher load [Ma et al., 2020]. At low-load operating condition, low exergy efficiency is also reported for dual-fuel engines utilising gaseous fuels such as compressed natural gas (CNG) [Verma et al., 2017; Morsy et al., 2012] and biogas [Sarkar and Saha, 2020]. The low-load exergy efficiency with hydrogen fumigation is reported to be higher as compared to biogas or CNG fumigation [Verma et al., 2017]. Sarkar and Saha [2020] reported an exergy efficiency of 13.2% for biogas-diesel dual-fuel mode as compared to 16.11% for diesel mode at a low load of 4.36 Nm. Their results show that the corresponding exhaust gas exergy and exergy destruction were more for the dual-fuel mode as compared to those of the diesel mode. The results of Verma et al. [2017] on exergetic performance for fumigation of biogas, CNG, and hydrogen in a dual-fuel engine show that hydrogen fumigation resulted in the highest exergetic efficiency amongst the dual-fuel modes at both low-load and high-load operations. Further, their results show that CNG fumigation has slightly lower exergy efficiency than that of diesel at a low load, while it becomes comparable at a high load.

The above literature review shows that exergy efficiencies at low-load operations need to be improved for dual-fuel engines utilising gaseous hydrocarbon fuels and alcohols.

2.6 Summary

The literature review shows that dual-fuel engines fumigated with gaseous hydrocarbon fuels or alcohols emit high levels of CO and HC emissions at low loads. Further, such dual-fuel engines suffer from low BTE at low loads. Different methods, such as intake charge preheating, optimisation of CR, and advancement of diesel injection timing, improve BTEs and emissions of dual-fuel engines. Adding DEE or DME as a co-fumigant with gaseous fuel also improves BTEs and emissions of dual-fuel engines. DME can be stored and handled like LPG, and the two fuels are miscible. The addition of DME as co-fumigant with propane has the potential to improve BTE and HC emissions. The high levels of CO emissions in dual-fuel engines can be effectively controlled by oxidation catalysts. However, HC emissions from dual-fuel engines employing gaseous hydrocarbon fuels such as methane and natural gas are not readily oxidised by oxidation catalysts.

There is a lack of data on the use of customised oxidation catalysts in LPG-diesel dual-fuel engines. Amongst the gaseous hydrocarbon fuels, LPG is a popular alternative fuel that is widely used in automotive vehicles. LPG and DME have the potential to be used together as fuels. However, the literature review shows that there is a lack of data on the use of DME-LPG co-fumigation in dual-fuel engines. From the literature review, the following research gaps are identified:

- For dual-fuel engines fumigated with gaseous hydrocarbon fuels, low BTE and high emissions of CO and HC at low loads are major challenges. Dual-fuel engine CO emissions are effectively controlled by oxidation catalysts. However, for dual-fuel engines employing fumigation of gaseous hydrocarbon fuels, HC emissions are not readily oxidised by oxidation catalysts at low-load operating conditions. Further, the available data on the use of oxidation catalysts are mostly for natural gas and methane fumigations. Therefore, research on the use of customised oxidation catalysts in LPG-diesel dual-fuel engines would provide useful data.

- Diesel engine pumpsets are widely used in India for agricultural applications. However, there is no data on such engines operated on LPG-diesel dual-fuel mode with customised OCCs. Further, there is no data on comparative evaluations of emissions for customised OCC and commercial OCC on such dual-fuel engines.
- Limited data are available on the study of DME as a co-fumigant with propane in the dual-fuel mode. Such limited data show that the addition of DME to propane can improve BTE and reduce HC emissions at low load. Despite such potential, there is a lack of research on the use of DME as a co-fumigant with propane or LPG in the dual-fuel mode.
- The literature data on exergy analyses for gaseous fuel fumigated dual-fuel engines are mostly for fuels such as hydrogen, biogas, and CNG. Such analysis needs to be carried out for dual-fuel engines using LPG and DME.

The above research gaps are taken into consideration in the present research work. The present investigation is focused on improving the engine performance and emissions of the dual-fuel mode while utilising LPG and DME.

Chapter – 3

Experimental Set-up and Methodology

Overview: *A single-cylinder diesel engine is converted to operate in the dual-fuel mode. The conversion is carried out in two stages. Firstly, the diesel engine is converted into an LPG-diesel dual-fuel mode. Secondly, DME-LPG co-fumigation system is developed, and the diesel engine is converted into a DME-LPG-diesel dual-fuel mode. Two different oxidation catalytic converters are also used: a commercial OCC and a customised OCC. This chapter describes the whole experimental set-up and the dual-fuel conversions. The details of instruments and the overall experimental test procedures are also described in this chapter*.

3.1 Test Engine

The present research work is carried out using a variable compression ratio (VCR) research engine test bench (Product code: Im240, Apex Innovation, India). The experiments are carried out at the CSIR-Indian Institute of Petroleum, Dehradun. The research engine test bench is developed from a constant speed agricultural engine. The engine details are given in Table 3.1. The engine has a rated power of 5.2 kW at 1500 RPM, which corresponds to BMEP of 6.3 bar. The engine standard injection timing and standard compression ratio are used in the present research. Use of the engine manufacturer's standard injection timing allows for easy switching between diesel mode and dual-fuel mode. During the non-availability of gaseous fuels, the engine can automatically operate back to the diesel mode.

Constant speed diesel engines are widely used in India for agricultural applications. In India, 54.7% of the gross cropped area remained unirrigated in the year 2010-2011 [Agriculture Census Division, 2015]. The Agriculture Census 2016-17 [Agriculture Census Division, 2021] reports the use of diesel engine pumpsets for irrigation in over 6 million operational land holdings, under marginal size group having land area below 10,000 m^2 (1 hectare). Therefore, stationary diesel engines have a significant role in the Indian agriculture sector. The present research is carried out on a naturally aspirated constant speed agricultural engine.

Table 3.1: Engine Specifications.

Model and Make:	TV1, Kirloskar Oil Engines Ltd., India
Engine Type	1-cylinder, 4-stroke, water-cooled diesel engine
Stroke	110 mm
Bore	87.5 mm
Compression ratio	17.5:1
Displacement	661 cm^3
Engine speed	1500 RPM
Nozzle opening pressure	200-205 bar
Fuel injection timing by spill	23 ^{o}CA bTDC
Rated power	5.2 kW at 1500 RPM
Speed governing	Centrifugal type, Class B1

3.2 Conversion of the Diesel Engine to the LPG-diesel Dual-fuel Mode

The diesel engine is converted to the LPG-diesel dual-fuel mode. A fueling system for LPG fumigation is developed. Figure 3.1(a) depicts the LPG fueling system. The LPG is stored in liquid form in the LPG cylinder. The LPG cylinder is not fully filled with the liquefied fuel. Therefore, the upper portion of the LPG cylinder is filled with LPG vapour. The LPG continuously evaporates in the cylinder and is delivered in gaseous form to the gas line. The gaseous fuel line is constructed of seamless stainless-steel tubes. The gaseous fuel is passed through a gas filter before passing through a pressure regulator. Downstream of the pressure regulator is a valve to open or close the gas flow to a mass flow controller (MFC). The gaseous fuel flow rate is controlled by the MFC through an MFC controller. The gas flow line downstream of the MFC is fitted with a gas bleeding line. The bleeding line has a gas bleed valve to open or close the gaseous fuel to the atmosphere. When the gas bleed valve is closed, the gaseous fuel is directed towards the gas-air mixer through a needle valve. The needle valve is useful to avoid a sudden flow of the LPG into the engine via the gas air-mixer. The gas-air mixer is used for mixing the gaseous fuel and the engine intake air. The gas-air mixer is illustrated in Fig. 3.1(b). It is a closed concentric cylinder having gas delivery holes on the inner cylindrical wall. There are 87 gas delivery holes of 2 mm diameter each. As illustrated in Fig. 3.1(b), the gaseous fuel, which is delivered through the gas delivery holes, mixes with the engine intake air, and the gas-air mixture is further inducted into the combustion chamber through the intake port and intake valve opening.

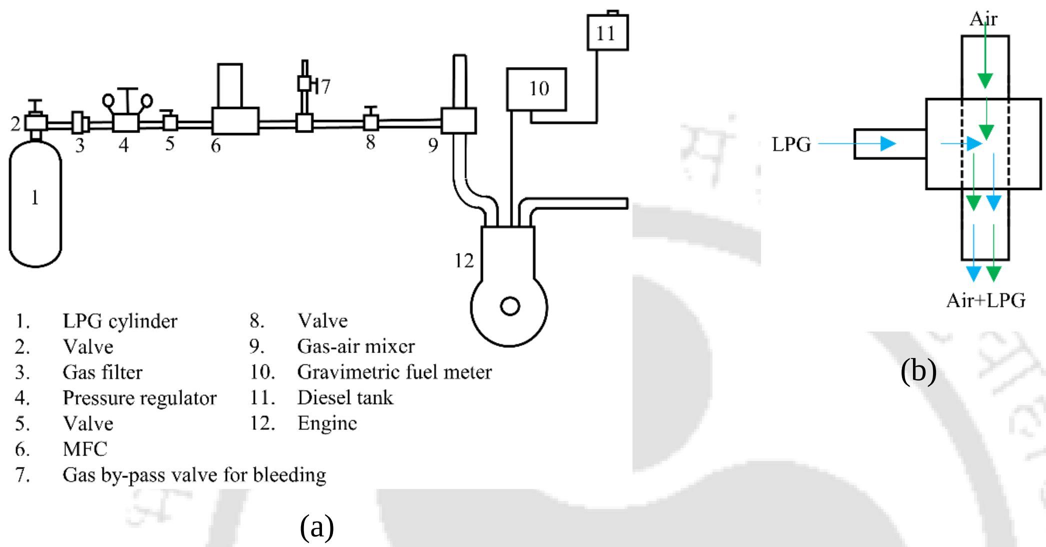
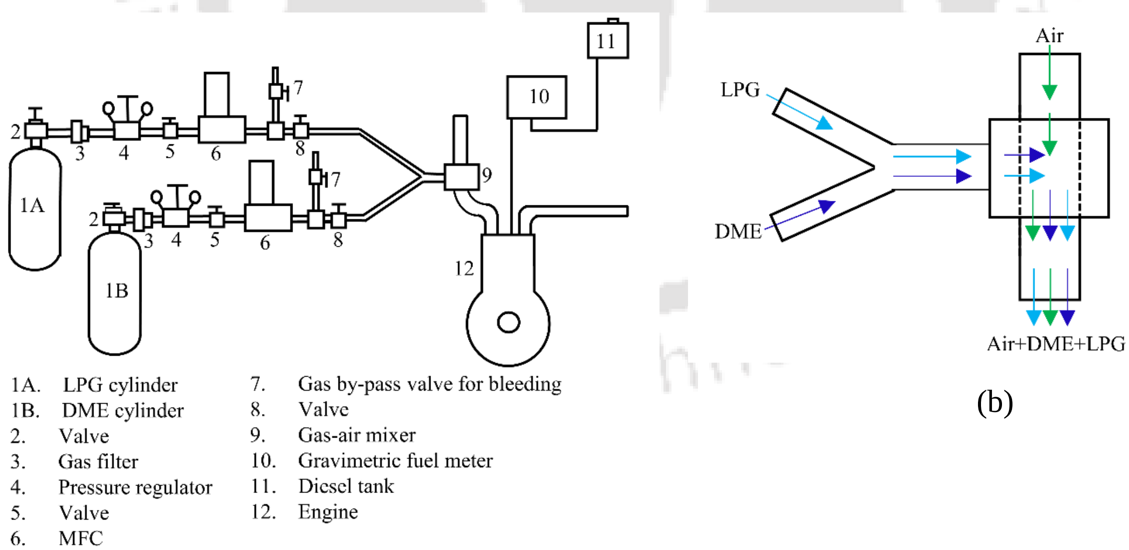


Fig. 3.1: LPG fueling system: (a) Schematic of LPG-diesel fueling system; (b) Fuel-air mixer illustration.

The diesel injection rate is controlled by the engine speed governor. At a given load, as the LPG flow rate is varied through the MFC, the diesel injection quantity is automatically adjusted by the diesel fuel pump through the engine speed governor. After the engine speed is stabilised by the governor, the speed is further fine-tuned by manually adjusting the engine governor lever position. After the engine speed stabilisation by the governor, the governor lever is temporarily clamped to a rod. The rod can be moved forward and backwards through a threaded mechanism, thereby changing the position of the governor lever.

3.3 Conversion of the Diesel Engine to the DME-LPG-diesel Dual-fuel Mode

The conversion of the diesel engine to the DME-LPG-diesel dual-fuel mode is based on the LPG-diesel dual-fuel mode. One fueling system for DME is added in addition to the LPG fueling system. Figure 3.2(a) shows the schematic of the DME and LPG fueling systems. LPG and DME are stored in liquid form in separate cylinders. However, the cylinders are not fully filled with the liquefied fuels. Therefore, the upper portion of each cylinder is filled with DME or LPG in gaseous form. The liquefied fuels continuously evaporate in the cylinders and are delivered in gaseous form to the gas lines. Both fuels coming out in the gaseous form from the cylinders are passed through gas filters. Downstream of the gas filters, pressure regulators regulate the pressures in the gaseous fuel lines. Two different MFCs are placed in the gas lines as shown in the schematic. Further, there is a provision for bleeding the gas from each of the gaseous fuel lines. The gas bleeding is done when the gaseous fuel is desired to be bled and released to the atmosphere. The DME fueling system from the DME cylinder to the inlet of the pressure regulator is made up of metals only. However, polytetrafluoroethylene (PTFE) materials are present in the DME MFC and the valves of the gas line downstream of the pressure regulator. The metal tubes used in the DME gas line are seamless stainless-steel tubes. The flow rates of DME and LPG are regulated by their respective MFCs. For a given load, as the mass flow rates of the two gaseous fuels are varied, the diesel fuel injection rate is automatically adjusted through the engine speed governor, as explained earlier. As shown in the schematic, the DME and LPG gas lines converge into a single line before entering the gas-air mixer. The gas-air mixer is already explained in the previous section. The flow and mixing of the gaseous fuels with the engine intake air are illustrated in Fig. 3.2(b). The photograph of DME-LPG fueling system is shown in Fig. 3.3.



(a) (b)

Fig. 3.2 DME-LPG Fueling system: (a) Schematic of DME-LPG-diesel fueling systems; (b) Fuel-air mixer illustration.

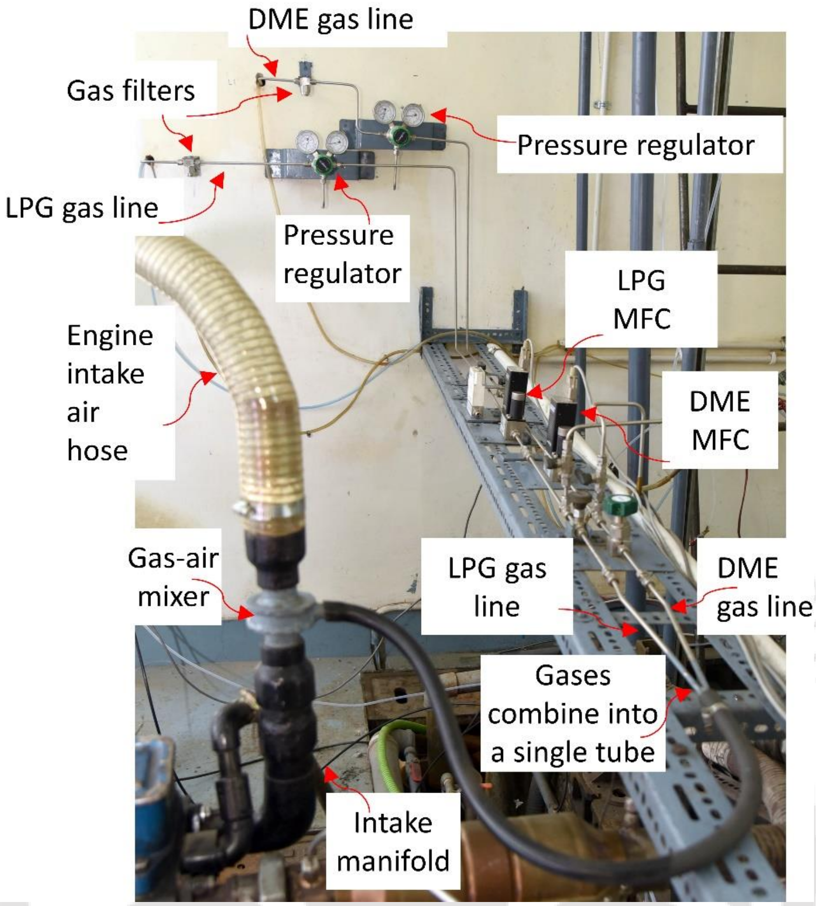


Fig. 3.3 Photograph of DME-LPG fueling system.

3.4 Test Set-up

3.4.1 Test Instrumentations

Table A1 of Appendix A summarises the major specifications of the instruments used in the study. The HC emissions are measured by a gas analyser (Model: OBS 2200; Horiba Ltd., Japan) using the heated flame ionisation detector (FID) principle. In a flame ionisation detector, ions are generated when hydrocarbons are introduced into a hydrogen flame. The instrument calculates the carbon atom from the number of ions generated. The emissions of CO and NOx are measured and displayed by an automotive emission analyser (Model: Mexa-584L; Horiba Ltd., Japan). The automotive emission analyser employs a nondispersive infrared (NDIR) sensor and an electrochemical sensor. Similar gas analysers are used by different researchers for reporting NOx and CO emissions [Jain et al., 2017; Ning et al., 2020; Zhang et al., 2020]. The electrochemical sensor employs a 3-electrode system to detect the concentration of nitric oxide in the gas sample. The nitric oxide concentration measured by the electrochemical sensor is displayed as NOx emissions on the gas analyser screen. The NDIR sensor in the emission analyser employs an infrared source and an infrared detector. Absorption of the infrared light by CO changes the amount of the infrared light reaching the infrared detector, and the CO concentration in the gas sample is determined by the amount of the infrared light detected. Smoke opacity is measured by a smoke opacimeter (Model: 439 Opacimeter; AVL List GmbH, Austria) that meets the standards such as the European directive 2005/78/EC and ISO 8178-9. Smoke in the measuring chamber of the opacimeter causes reduction in the intensity of light between a light source and a receiver. The instrument calculates the smoke opacity based on the Beer-Lambert Law by using the measured value of the reduction in light intensity between the source and the receiver.

The measurement of diesel fuel flow rate is done by a fuel meter (Model: 733S; AVL List GmbH, Austria) based on the gravimetric measuring principle. The diesel flow rate is displayed in kg/h. In the gravimetric fuel meter, fuel is supplied from a measuring vessel to the engine, and the weight of the measuring vessel is continuously measured. Averaging of the diesel flow rate is done by the fuel meter for a given measurement duration. Flow rates of DME and LPG are controlled and displayed by MFCs (Model: SLA5850; Brooks Instruments LLC, USA) in standard litre per minute (SLPM). In MFCs, temperature sensors measure the temperatures of gas upstream and downstream of a heating element. The increase in the gas temperature is converted into the gas flow rate based on the factory calibrations of the MFC. The air flow rate is measured using a laminar flow element (LFE) system (Model: 50MC2-2F; Meriam Instrument, USA), based on the calibration Eq. A8 given in Appendix-A. The engine is coupled to an eddy current dynamometer (Model: AG10; Saj Test Plant Pvt. Ltd., India) equipped with a load cell. A combustion analyser (Model: IndiMicro 602; AVL List GmbH, Austria) is used for the measurement of the in-cylinder pressure crank angle history. The combustion analyser system consists of a charge amplifier, a piezoelectric pressure transducer (Model: AVL GH14D; AVL List GmbH, Austria), and a crank angle encoder (Model: 365-C; AVL List GmbH, Austria). The in-cylinder pressure crank angle data are acquired through AVL IndiCom 2™ software. The processing of the in-cylinder pressure crank angle data for the combustion analysis is explained in Appendix A. Figure 3.4 shows the schematic of the test set-up. The photograph of the test set-up is shown in Fig. 3.5. A photograph showing the piezoelectric pressure transducer and the crank angle encoder fitted on the engine is shown in Fig. 3.6.

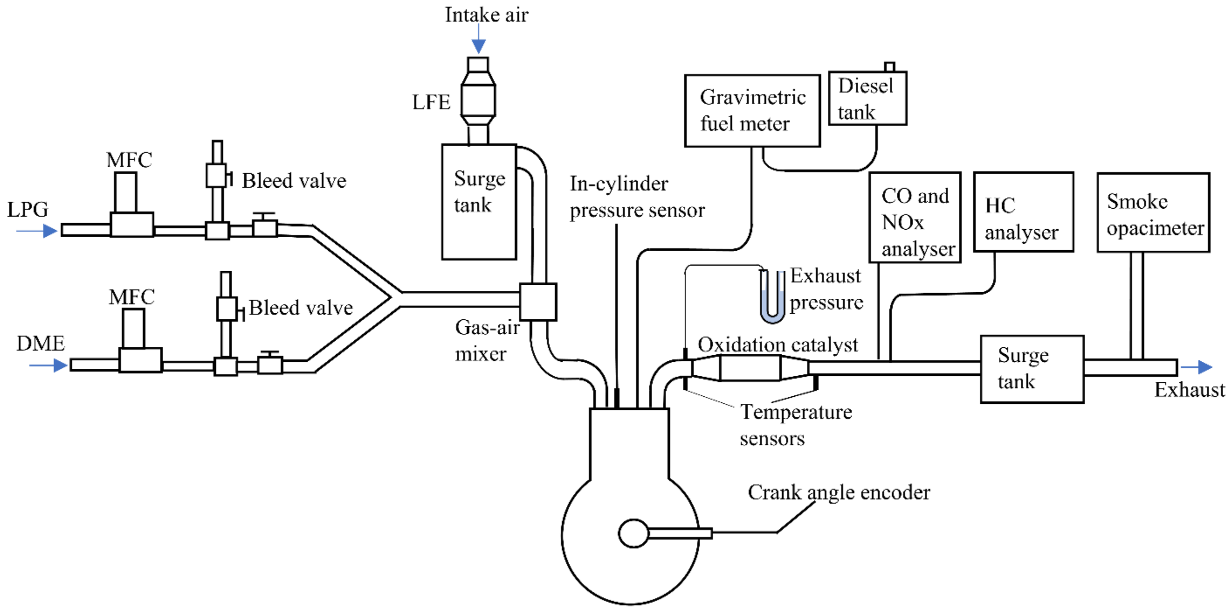


Fig. 3.4 Schematic representation of the experimental test bench.

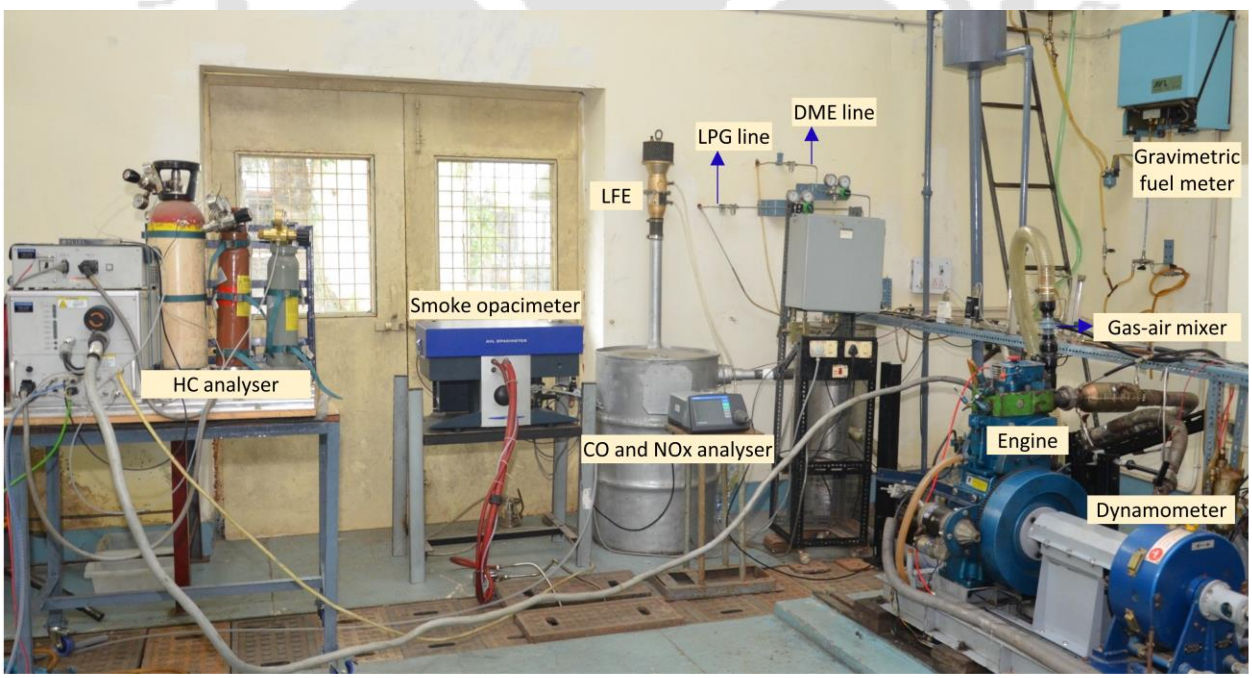


Fig. 3.5 Photograph of the experimental test bench.

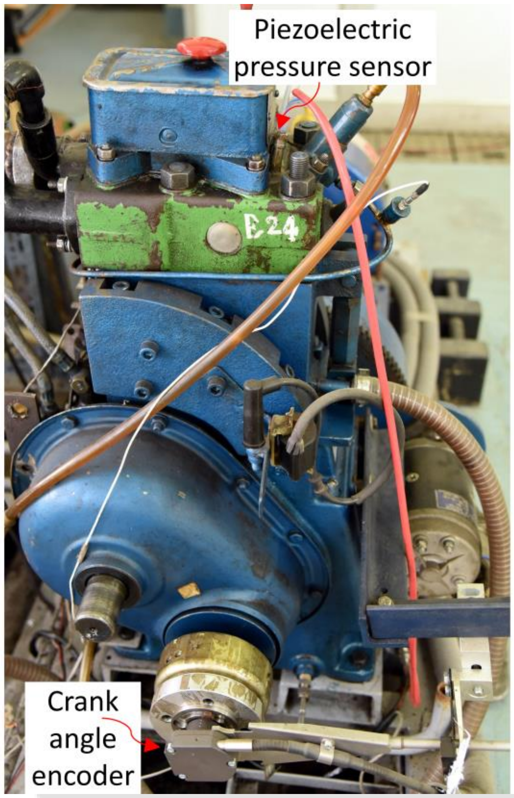


Fig. 3.6 Photograph showing the piezoelectric pressure sensor and the crank angle encoder fitted on the engine.

For exergy analysis, the exhaust gas specific heat capacity is required. During experiments to find the exhaust gas specific heat capacity at different loads, an exhaust gas calorimeter is connected the engine exhaust. The exhaust gas calorimeter is a shell and tube heat exchanger. The hot exhaust gas is cooled in the calorimeter by the cooling water.

3.4.2 Oxidation Catalysts

The dual-fuel mode emissions are studied with oxidation catalytic converters. A commercial OCC and a customised OCC, both supplied by Techinstro (Nashik, India), are used. The commercial OCC is Euro IV compliant. Both OCCs are the same in respects of dimension, support materials, ratio of platinum to palladium, and gas hourly space velocity (GHSV). The only difference between the two OCCs is the platinum group metal (PGM) loading. The customised OCC has a PGM loading of 1.766 kg/m^3. The PGM loading of the commercial OCC is not known. Both OCCs have a GSHV of 26,273 per hour. The OCCs have an aluminium oxide-based washcoat in a ceramic honeycomb substrate. Table 3.2 details the specification of the OCCs.

Table 3.2 Oxidation catalysts specifications.

	OCC volume (cm^3)	Cell density (cpsc)	Pt:Pd ratio	PGM loading (kg/m^3)
Customised OCC	951	62	3:1	1.766 kg/m^3
Commercial OCC	951	62	3:1	-

3.5 Test Procedures

3.5.1 Methods for Evaluation of Test Fuels Properties

The physico-chemical properties of the diesel fuel are evaluated as per ASTM methods. Density, kinematic viscosity, cetane index, flash point, and sulfur are evaluated as per ASTM D4052, ASTM D445, ASTM D4737, ASTM D93, ASTM D4294, respectively. The carbon and hydrogen content in the diesel fuel is evaluated through carbon, hydrogen, nitrogen, and sulphur (CHNS) elemental analysis. The higher heating value (HHV) of diesel is calculated using H and C composition as per the method given by DemirbaŞ [1998]. The LHV of LPG is calculated by weighted average using LHVs and molar concentrations of the constituent species. The molar concentrations of the constituent species of LPG are determined from gas chromatography (GC) analysis results.

3.5.2 Procedures for Evaluation of Engine Performance, Combustion, and Emissions

The engine performance, combustion and emissions are evaluated at different loads under steady state operations. There is a need to address the technical issues of low BTE and high levels of CO and HC emissions at low loads; therefore, the experiments are carried out covering 20% load (1.3 bar BMEP) and 40% load (2.6 bar BMEP). Further, the experiments are extended beyond half-load to cover 70% load (4.4 bar BMEP) so that the results at low loads can be compared with those at higher load. While taking the data of emissions and fuel consumption, averaging is done over 60 s duration. The emissions of CO, HC, NOx, and smoke are measured. NOx emissions of diesel engines consist of NO and NO_2. However, the results of NOx emissions presented in the present research do not include NO_2 measurements. The automotive emission analyser displays the NOx emissions of the engine exhaust based on the nitric oxide concentration measurements by the electrochemical sensor. Conversions to specific mass emissions of CO, HC, and NOx are done by the carbon balance method as per SAE J1003. The percentage reductions of CO and HC achieved by the OCs with reference to the CO and HC emissions without the OCs are taken as the OCs conversion efficiencies. The in-cylinder pressure crank angle data are recorded for 200 cycles. Averaging of 200 cycles of in-cylinder pressure data is used for the combustion analysis. The equations for BTE, brake mean effective pressure (BMEP), indicated thermal efficiency (ITE), relative cycle efficiency (RCE), and net heat release rate (NHRR) are described in Appendix A. The crank angles where cumulated heat release reaches 10%, 50%, and 90% of the total heat release, denoted by CA10, CA50, and CA90, respectively, are also estimated. The methods for exergy analysis are described in Appendix B. The method used for uncertainty analysis is given in Appendix C.

Chapter – 4

Investigation of LPG-diesel Dual-fuel Engine Performance, Combustion, and Emissions

Overview: *It is discussed in the literature review that LPG-diesel dual-fuel engines suffer from high emissions of CO and HC and low thermal efficiency at low-load conditions. In this chapter, the LPG-diesel dual-fuel mode of combustion is studied for engine performance, combustion, and emissions characteristics. The CO and HC emissions of LPG-diesel dual-fuel mode, at 20-70% load, are studied with the use of a commercial oxidation catalyst and a customised oxidation catalyst. Much of the literature on the use of oxidation catalysts in dual-fuel engines report the use of commercial oxidation catalysts with unknown PGM loading levels. Therefore, it is important to carry out such investigations with a customised oxidation catalyst of known PGM loading. The baseline LPG-diesel dual-fuel mode emissions of CO and HC, without the use of oxidation catalysts, are significantly high. The study shows that LPG-diesel dual-fuel combustion results in low BTE at below half-load operations. However, at the higher load, the LPG-diesel dual-fuel combustion results in engine performance the same as that of the diesel mode. The present study shows that the commercial oxidation catalyst is not effective in reducing HC emissions of the LPG-diesel dual-fuel mode at low-load operations. However, the customised oxidation catalyst effectively reduces the HC and CO at all tested loads*.

4.1 LPG Substitution Levels and Test Matrix

The fumigation rates of LPG are expressed in terms of LPG substitutions. LPG substitution is the percentage of LPG energy in total energy of the fuels (LPG + diesel). The LPG substitutions at different engine operating conditions are given in Table 4.1. The test matrix is shown in Table 4.2. At BMEPs of 1.3 bar (20% load), 2.6 bar (40% load), and 4.4 bar (70% load), LPG substitution levels are varied, and engine performance, combustion and emissions are studied. Two different LPG substitution levels are studied at each load. Parameters such as CO, HC, NOx, smoke, BTE, in-cylinder pressure, NHRR, cylinder mean gas temperature (CMGT), CA10, CA50, CA90, ITE, and RCE are investigated. Further, based on the results of engine performance and emissions, one of the two LPG substitution levels at each load is selected for study with the oxidation catalysts. In Table 4.1, the LPG substitution levels selected for the oxidation catalysts study are denoted as LPG, while those not selected are denoted as LPG-A.

As discussed in the literature review, LPG-diesel dual-fuel engines are associated with low BTE and high levels of CO and HC emissions at low loads. Therefore, at 20-40% loads, low LPG substitution rates are selected. Results of Sudhir et al. [2003] show that increase in brake specific fuel consumption and reduction in BTE became more rapid beyond 40% LPG substitution at 25% load. In the present research, 36-43% LPG substitutions are studied at 20% load. At 40% load, 39-48% LPG substitution levels are studied. Jian et al. [2001] reported that, at medium loads, LPG substitution level should be as high as approximately 40%. Their results show that CO and HC were increased by over 3 times when LPG substitution was about 35% at 40% load. Above mid-load operations, Ganesan and Ramesh [2001] reported that BTE remained almost the same for LPG substitution of about 20% to over 70% at 60% engine load. In the present research, 51-64% LPG substitution levels are studied at 70% engine load.

The properties of commercial diesel fuel used in the study are given in Table 4.3. The properties of butane and propane are given in Table 4.4. The composition of LPG used in the study is given in Table 4.5. The LHV of the LPG is calculated by weighted average using molar concentrations and LHVs of the constituent gases. The LPG composition is determined through GC analysis. For a given LPG substitution level, Elnajjar et al. [2013] reported that dual-fuel engine performance did not significantly change with variation in the percentage of propane and butane in the LPG.

Table 4.1: LPG substitution rates.

	BMEP (bar)	Diesel energy percentage (%)	LPG energy percentage (%)
LPG-A	4.4	49	51
	2.6	61	39
	1.3	57	43
LPG	4.4	36	64
	2.6	52	48
	1.3	64	36

Table 4.2: Test matrix of LPG-diesel dual-fuel engine study.

Parameters studied	Fuels	Engine Load
Engine performance and combustion characteristics: BTE, in-cylinder pressure, NHRR, CMGT, CA10, CA50, CA90, RCE, ITE	*100% diesel* *51% LPG + 49% diesel (LPG-A)* *64% LPG + 36% diesel (LPG)*	*4.4 bar BMEP (70% load)*
	100% diesel *39% LPG + 61% diesel (LPG-A)* *48% LPG + 52% diesel (LPG)*	*2.6 bar BMEP (40% load)*
	100% diesel *36% LPG + 64% diesel (LPG)* *43% LPG + 57% diesel (LPG-A)*	*1.3 bar BMEP (20% load)*
Emissions without OCC: CO, HC, NOx, smoke	*100% diesel* *51% LPG + 49% diesel (LPG-A)* *64% LPG + 36% diesel (LPG)*	*4.4 bar BMEP (70% load)*
	100% diesel *39% LPG + 61% diesel (LPG-A)* *48% LPG + 52% diesel (LPG)*	*2.6 bar BMEP (40% load)*
	100% diesel *36% LPG + 64% diesel (LPG)* *43% LPG + 57% diesel (LPG-A)*	*1.3 bar BMEP (20% load)*
Emissions with commercial OCC: CO, HC, NOx, smoke	*64% LPG + 36% diesel (LPG)*	*4.4 bar BMEP (70% load)*
	48% LPG + 52% diesel (LPG)	*2.6 bar BMEP (40% load)*
	36% LPG + 64% diesel (LPG)	*1.3 bar BMEP (20% load)*
Emissions with customised OCC: CO, HC, NOx, smoke	*64% LPG + 36% diesel (LPG)*	*4.4 bar BMEP (70% load)*
	48% LPG + 52% diesel (LPG)	*2.6 bar BMEP (40% load)*
	36% LPG + 64% diesel (LPG)	*1.3 bar BMEP (20% load)*

Table 4.3 Properties of commercial diesel test fuel.

Property	Test Method	Test result
Density at 15 ^{o}C (kg/m^3)	ASTM D4052	827.6
Kinematic viscosity at 40 ^{o}C (cSt)	ASTM D445	2.72
Cetane index	ASTM D4737	55
Flash point (^{o}C)	ASTM D93	47.5
Sulfur (ppm)	ASTM D 4294	25.87
H/C	CHNS analysis	1.95
Lower heating value (kJ/kg)		42909
Distillation properties 5% 50% 90%	ASTM D86	 172 ^{o}C 273.5 ^{o}C 345.7 ^{o}C

Table 4.4 Properties of propane and butane [Ohno, 2001; Saleh, 2008; Sorenson and Mikkelsen, 1995].

Property	Propane	Butane
Boiling point (^{o}C)	-42.0 – -42.1	-0.5
Vapour pressure at 20 ^{o}C (bar)	8.4	2.1
Liquid density: at 20 ^{o}C (kg/m^3)	490–501	610
Lower heating value (kJ/kg)	46300–46442	45740–45800
Ignition temperature at 1.01325 bar (^{o}C)	470	365

Table 4.5 Composition of LPG.

	i-Butane (C_4H_{10})	n-Butane (C_4H_{10})	Propane (C_3H_8)	t-2-Butene (C_4H_8)	1-Butene (C_4H_8)	i-Butene (C_4H_8)	Propylene (C_3H_6)
Mass fraction	0.258	0.260	0.317	0.047	0.074	0.020	0.024

4.2 Selection of Oxidation Catalysts

Two different oxidation catalysts are used in the present study. The oxidation catalysts details are given in Table 3.2 of Chapter 3. The physical dimensions of both OCCs are chosen to be the same. All specifications are the same for both OCCs, except the PGM loading levels. The PGM loading level of the commercial OCC is unknown. Oxidation catalysts PGM loading levels reported in the literature vary widely depending on the applications. High PGM loading levels are typical of light-duty diesel applications [Etheridge et al., 2015]. HeraPur® oxidation catalysts brochure of Haraeus [2012] shows a wide range of PGM loadings such as 0.530 kg/m^3 to 1.589 kg/m^3 for passenger cars and 1.589 kg/m^3 to 3.531 kg/m^3 for light-duty and heavy-duty vehicles. Majewski [2011] reported that PGM loadings as high as 2.825-3.531 kg/m^3 are used for control of PM and CO at low temperature. For stationary engines, the HeraPur® oxidation catalysts brochure of Haraeus [2012] shows that PGM loading levels of 0.530 kg/m^3 to 1.059 kg/m^3 are used. Piqueras et al. [2019] studied a prototype oxidation catalyst on a gasoline fumigated heavy-duty dual-fuel engine operated on the RCCI mode. Their results show that a PGM loading of 1.130 kg/m^3 was effective in controlling the CO and HC emissions. In the present research, LPG is used as the fumigated fuel. LPG is a lighter hydrocarbon as compared to fuels such as gasoline and diesel. Therefore, a PGM loading of 1.766 kg/m^3, which is higher than that used by Piqueras et al. [2019], is considered for investigation in the present study. The sizing of both OCCs is based on gas hourly spaced velocity (GHSV). Bittner and Aboujaoude [1992] used a GSHV of 15,000 per hour and 45,000 per hour for their study in a natural gas-diesel dual-fuel engine. In this study, a GHSV of 26,273 per hour is used for both OCCs. GHSV is expressed by Eq. 4.1 [Bittner and Aboujaoude, 1992].

$$GHSV\ (per\ hour) = \frac{Volumetric\ gas\ flow\ rate\ (SCMH)}{Catalyst\ volume\ (m^3)} \quad [4.1]$$

4.3 Engine Performance

The results of BTE for the diesel mode and the dual-fuel mode are shown in Fig. 4.1. At 70% load (4.4 bar BMEP), the LPG-diesel dual-fuel mode has engine performance the same as that of diesel. At this load, the BTE of the LPG-diesel dual-fuel mode, with 64% LPG substitution level, is

27.63% as compared to 27.56% for the diesel mode. Further, it is observed that the engine performance remains the same when the LPG substitution level is increased from 51% to 64%, indicating that engine performance is not a limiting factor for LPG substitutions at the 70% load. Ganesan and Ramesh [2001] also reported that there was no significant variation in engine performance when LPG substitution was widely varied at 60% load.

At below half-load operations, dual-fuel BTEs reduce significantly with LPG fumigation, and a decreasing trend with the increase in LPG substitution is observed. At 20% load, for 36% LPG substitution, the BTE is 13.89% as compared to 16.46% for the diesel mode. At the same load, when the LPG substitution is increased to 43%, the BTE is only 13.20%. The results indicate that further investigations with LPG substitution levels higher than 43% will not yield any useful results at 20% load, as the BTE is already significantly lower than that of the diesel mode.

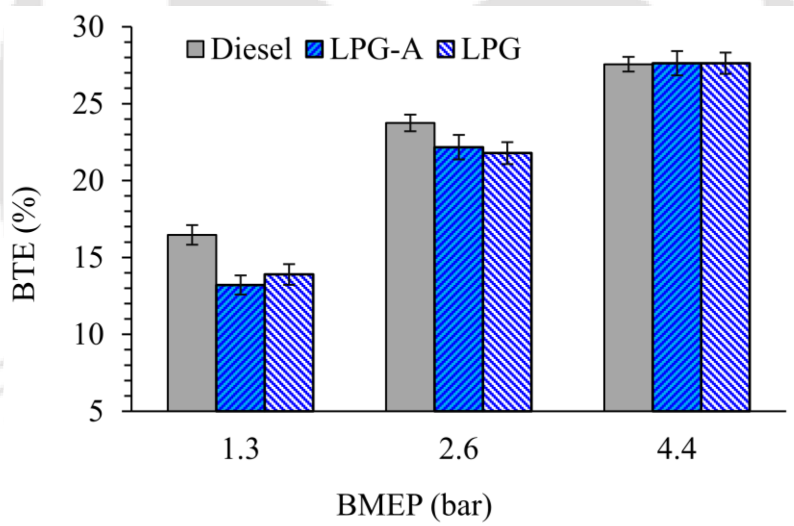


Fig. 4.1: BTE for diesel and LPG-diesel at different BMEPs.

The LPG-diesel dual-fuel mode BTE improves with the increase in load. At 40% load, for 39% LPG substitution, the BTE is 22.17% as compared to 23.73% for the diesel mode. When the LPG substitution is increased to 48%, the BTE is 21.78%. The improvement in the dual-fuel BTE with the increase in engine load is a result of improvement in the combustion characteristics, as shall be discussed in Sec. 4.4.

4.4 Combustion Characteristics

4.4.1 In-cylinder Pressure

The cylinder pressure results are shown in Fig. 4.2 through 4.4 for the dual-fuel mode and the baseline diesel mode. The results show that peak cylinder pressure values of the dual-fuel mode are significantly lower than those of the diesel mode. At a given load, the peak cylinder pressure decreases with an increase in LPG substitution level. At 20% load, the peak cylinder pressure for the diesel mode is 45 bar. At the same load, the LPG-diesel dual-fuel mode peak cylinder pressure values are 38 bar and 37 bar for LPG substitutions of 36% and 43%, respectively. Further, the dual-fuel mode peak pressure occurs 4-5 °CA later as compared to the diesel mode. Figure 4.2 through 4.4 show that the first rise of in-cylinder pressure, due to combustion, occurs significantly late as compared with the diesel mode. At 40% load, the LPG-dual-fuel mode peak cylinder pressure is lower by 6-9 bar, and its occurrence is delayed by 4-6 °CA as compared to the diesel mode. At 70% load, the peak cylinder pressure reduces with the LPG fumigation by as much as 11 bar. However, as discussed earlier, it is observed that BTE of the LPG-diesel mode is the same as that of the diesel at this load. The results show that despite the delay in peak cylinder pressure position and its lower value, the engine performance does not deteriorate at the higher load. The reason for the improvement in the dual-fuel engine performance at this load shall be explained in greater detail in Sec. 4.4.2.

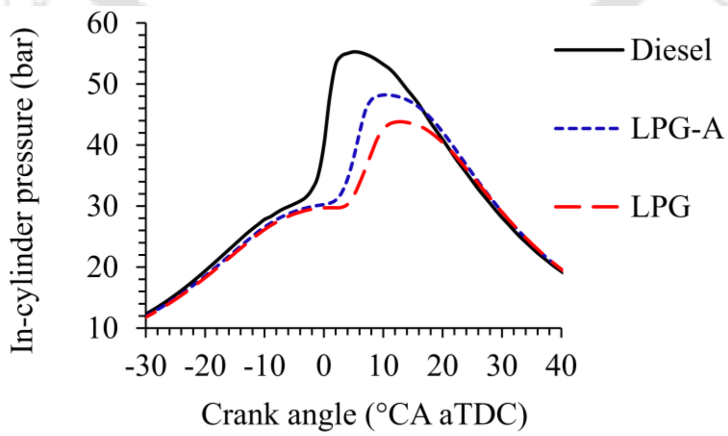


Fig. 4.2 In-cylinder pressure at 4.4 bar BMEP for diesel and LPG-diesel dual-fuel mode.

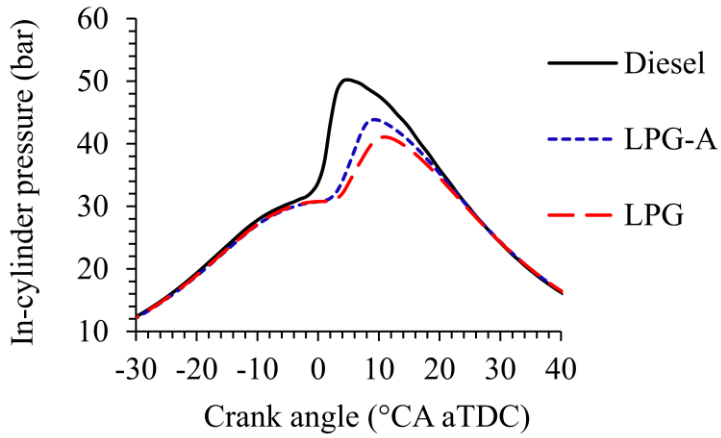


Fig. 4.3 In-cylinder pressure at 2.6 bar BMEP for diesel and LPG-diesel dual-fuel mode.

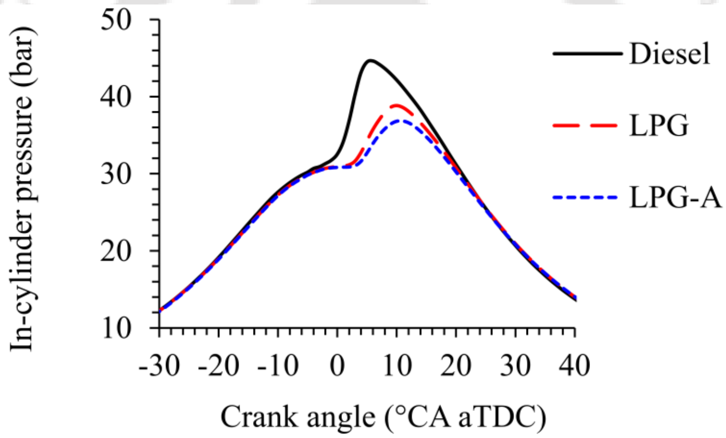


Fig. 4.4 In-cylinder pressure at 1.3 bar BMEP for diesel and LPG-diesel dual-fuel mode.

4.4.2 Net Heat Release Rate and Combustion Phasing

NHRR profiles are shown in Fig. 4.5 through 4.7. The cylinder mean gas temperature profiles are shown in Fig. 4.8 through 4.10. Further, CA10, CA50, and CA90 are shown in Fig. 4.11 through 4.13. At all tested loads, the NHRR peaks are lower for the LPG-diesel dual-fuel mode as compared to those of diesel. Further, at a given load, the peak NHRR decreases with an increase in LPG substitution. At 70% load, the peak NHRR is 44.35 J/°CA for 64% LPG substitution as compared to 79.26 J/°CA for diesel and occurred 7 °CA later than that of diesel. At lower loads, the reduction in peak NHRR with the dual-fuel mode is more prominent despite the lower levels of LPG substitution. In all the cases studied, CA10 and NHRR peak are observed to occur significantly late with dual fueling. LPG-air mixture has higher specific heat and, therefore, a lower charge temperature, thereby contributing to the delay in CA10. Another reason for the delay in combustion process is that the cetane number of LPG is much lower than that of diesel. The delay in CA10 increases with an increase in LPG substitution and is about 5 ^{o}CA at the higher LPG substitution levels at the tested loads of 20-70%.

As discussed earlier in Sec. 4.4.1, the rise of in-cylinder pressure due to combustion occurs significantly late with LPG fumigation. The results on CA10 also show that LPG fumigation results in a significant delay of CA10 at all tested loads. Therefore, LPG fumigation results in a late start of combustion at all loads. The late start of combustion results in late occurrences of NHRR peak and CA50. However, at 70% load, it is observed that the delays in both CA10 and NHRR peak do not result in drop of engine performance for the LPG-diesel mode at 70% load. As compared to the diesel mode, the dual-fuel mode has 2-3 ^{o}CA delay in CA50, but a significant advancement of CA90 is observed at this load. The CA90 is advanced by 10 ^{o}CA as compared to the diesel mode. Further, it is observed that the peak cylinder mean gas temperature after TDC is higher than that of diesel, and then it drops more rapidly afterwards as compared to that of diesel. For the 64% LPG substitution, the dual-fuel mode cylinder mean gas temperature becomes higher than that of diesel from 21 ^{o}CA after top dead centre (aTDC) onwards till 71 ^{o}CA aTDC after which it becomes lower than that of diesel. Despite the late CA10, Fig. 4.11 shows that the combustion duration, taken as CA10-90, is significantly shorter when compared with that of the diesel mode. The significantly advanced CA90, the cylinder mean gas temperature trend, and the significantly shorter combustion duration indicate rapid combustion of the LPG-diesel dual-fuel mode at 70% load, thereby resulting in better engine performance.

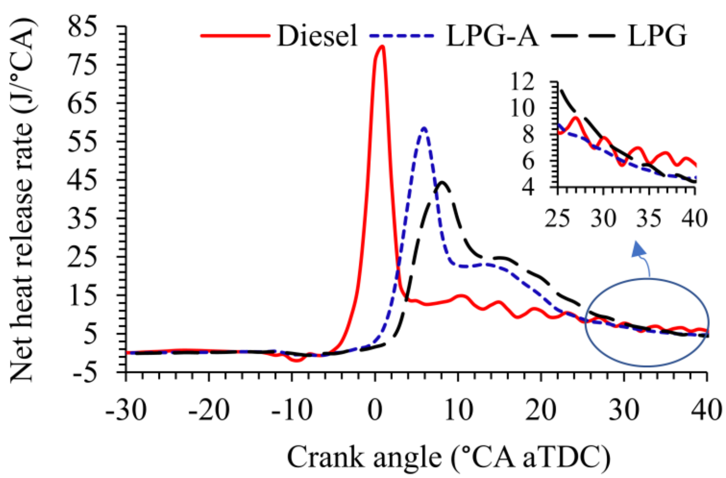


Fig. 4.5 NHRR profiles for LPG-diesel dual-fuel mode and diesel mode 4.4 bar BMEP.

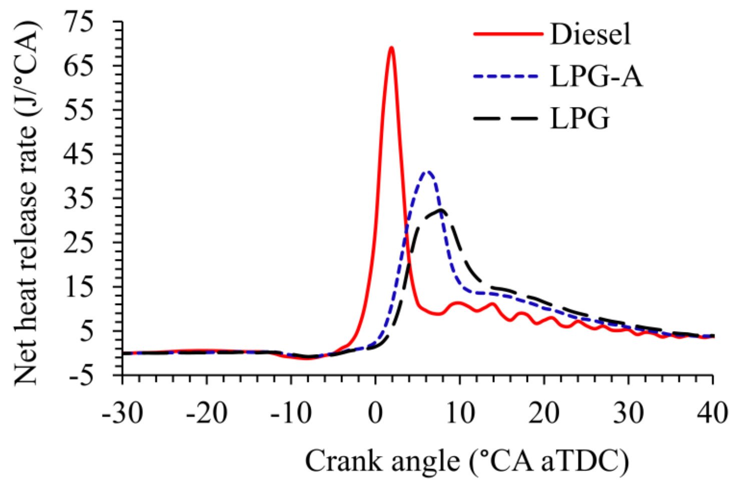


Fig. 4.6 NHRR profiles for LPG-diesel dual-fuel mode and diesel mode at 2.6 bar BMEP.

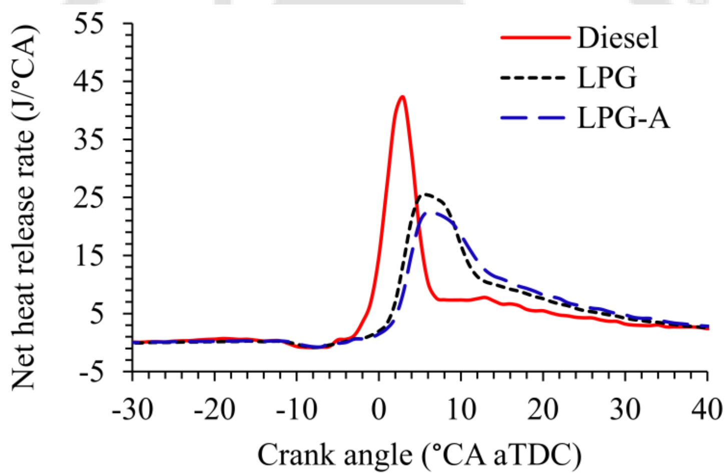


Fig. 4.7 NHRR profiles for LPG-diesel dual-fuel mode and diesel mode at 1.3 bar BMEP.

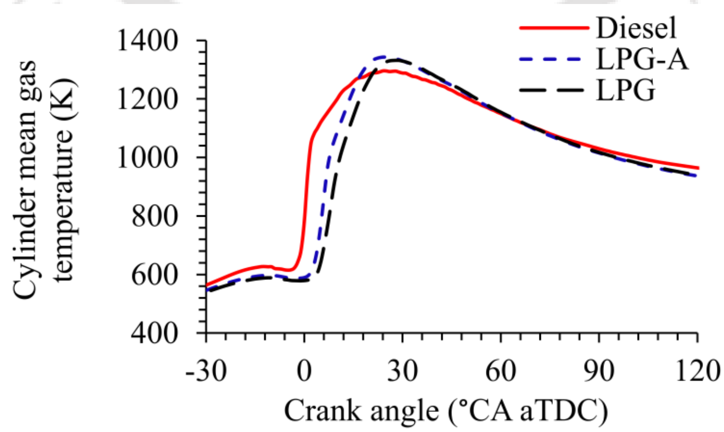


Fig. 4.8 CMGT at 4.4 bar BMEP for LPG-diesel dual-fuel mode and diesel mode.

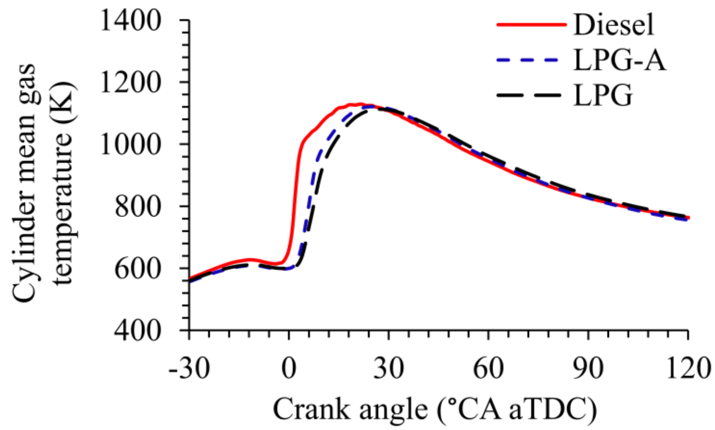


Fig. 4.9 CMGT at 2.6 bar BMEP for LPG-diesel dual-fuel mode and diesel mode.

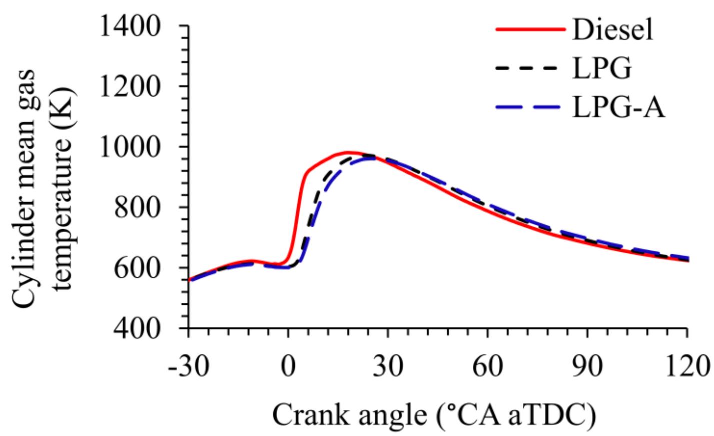


Fig.4.10 CMGT at 1.3 bar BMEP for LPG-diesel dual-fuel mode and diesel mode.

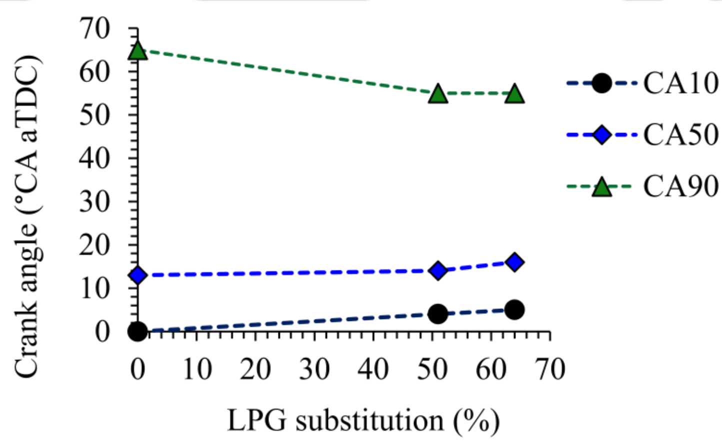


Fig. 4.11 CA10, CA50, and CA90 for diesel and LPG-diesel at 4.4 bar BMEP.

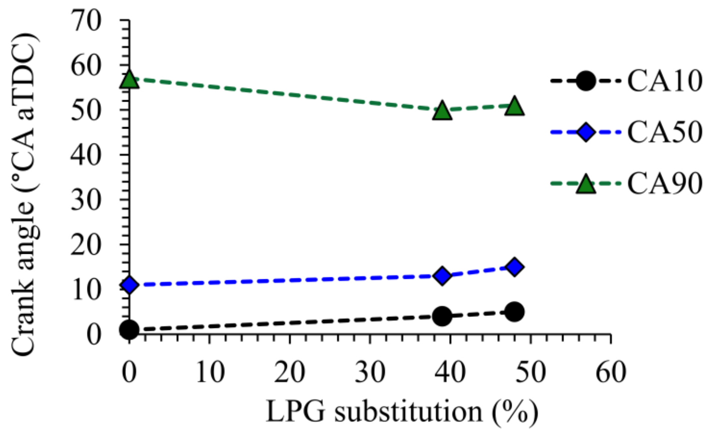


Fig. 4.12 CA10, CA50, and CA90 for diesel and LPG-diesel at 2.6 bar BMEP.

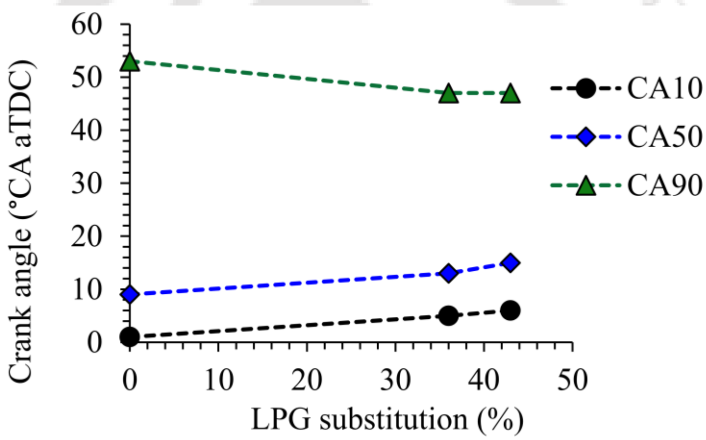


Fig. 4.13 CA10, CA50, and CA90 for diesel and LPG-diesel at 1.3 bar BMEP.

The delay in dual-fuel mode CA50 increases with the decrease in load. At below half-load operations, unlike the trend at 70% load, the peak cylinder mean gas temperature of the dual-fuel mode is lower than that of diesel. It is also observed that the advancement in CA90 with the LPG fumigation is not as significant as that at 70% load. These results indicate poor combustion of the LPG-diesel mode at below half-load operations. Consequently, the BTE is also lower as compared to the diesel mode.

4.4.3 Relative Cycle Efficiency and Indicative Thermal Efficiency

The concept of relative cycle efficiency (RCE) is found to be useful in understanding the LPG-diesel dual-fuel combustion. RCE is defined as the ratio of the thermal efficiency of heat added at a given crank angle to that at the top dead centre [Hsu, 1984]. Therefore, while calculating RCE, the amount of heat release, along with its position, is taken into consideration. Therefore, any change in parameters such as CA50 and CA90 will have an influence on the RCE. RCE and ITE are shown in Fig. 4.14 through 4.16. At 20-40% load, it is observed that the LPG-diesel dual-fuel combustion results in lower RCE and ITE as compared to the diesel mode. Further, the RCE decreases with an increase in LPG substitution. Both RCE and ITE follow similar trends, and test points having higher RCEs show correspondingly higher ITEs. At 20% load, ITE decreases from 30.02% to 28.59%, with an increase in LPG substitution from 36-43%. The ITE for diesel mode at this load is 35.34%. The dual-fuel mode performs better as the load is increased. At 70% load, LPG fumigation results in ITE of about 36%. At this load, the ITE of the dual-fuel mode are same as that of the diesel mode. The improved dual-fuel engine performance at this higher load is a result of the improved combustion characteristics, as discussed earlier.

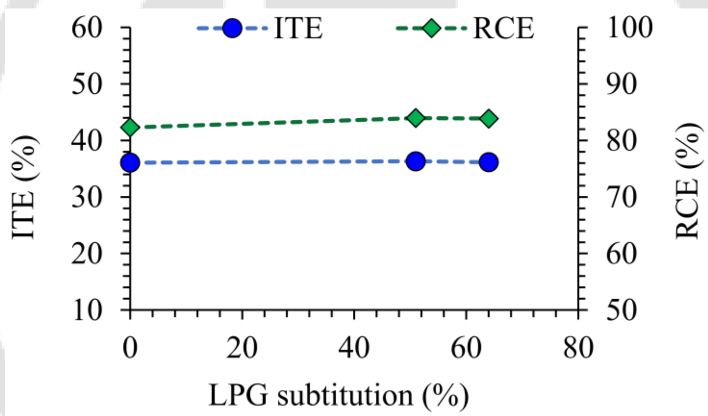


Fig. 4.14 RCE and ITE of LPG-diesel dual-fuel mode and diesel mode at 4.4 bar BMEP.

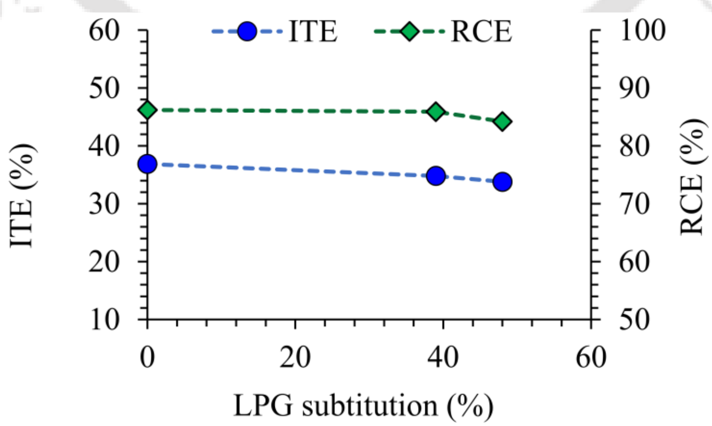


Fig. 4.15 RCE and ITE of LPG-diesel dual-fuel mode and diesel mode at 2.6 bar BMEP.

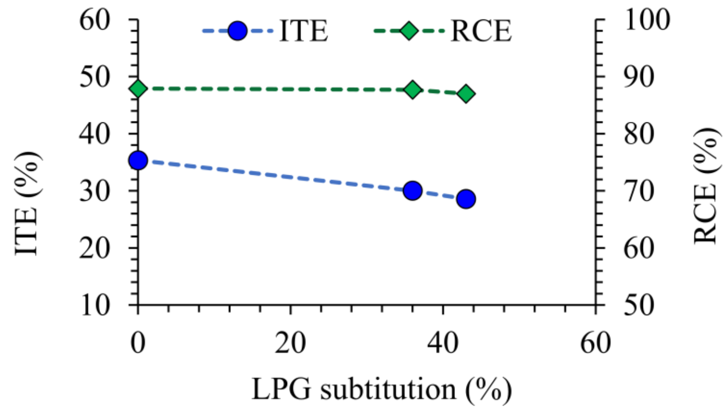


Fig. 4.16 RCE and ITE of LPG-diesel dual-fuel mode and diesel mode at 1.3 bar BMEP.

4.5 **Emissions**

4.5.1 HC Emissions

Figure 4.17 shows the HC emissions of the diesel mode and the dual-fuel mode. At below half-load operations, HC emissions of the dual-fuel mode are exceedingly higher than those of the diesel mode. At 20% load, for LPG substitutions of 36% and 43%, the dual-fuel mode HC emissions exceed those of the diesel mode by 173% and 240%, respectively. Therefore, a small increase in the LPG substitution is at the expense of significant deteriorations in HC emissions as well as BTE. Accordingly, the lower substitution level of 36% is selected for further emissions study with the oxidation catalytic converters. When the load is increased to 40%, the dual-fuel mode performs better in terms of the HC emissions, but the HC values are still significantly higher than those of the diesel mode. At this load, for LPG substitutions of 39% and 48%, the increase in HC emissions with reference to the diesel mode are 110% and 124%, respectively. The results indicate that, for 10% increase in LPG substitution level, the HC emissions do not deteriorate as severely as that is observed at 20% load. Therefore, the higher LPG substitution level of 48% is selected for further study with the oxidation catalytic converters.

The high emissions of HC in the dual-fuel mode at the lower loads are observed along with lower values of peak cylinder mean gas temperature. Literature attributes poor combustion as the reason for an increase in LPG-diesel dual-fuel mode HC emissions [Ganesan and Ramesh, 2001]. For an alcohol fumigated dual-fuel engine, Imran et al. [2013] attributed the reasons for high HC emissions to unburnt lean fuel-air regions and low combustion temperature.

At 70% load, the dual-fuel specific emissions of HC are considerably lesser than those at the lower loads. When LPG substitution level is increased from 51% to 64%, the HC emissions increased by 31% and 56%, respectively, with reference to the diesel mode. At this load, the peak cylinder mean gas temperature of the dual-fuel mode is higher than that of the diesel mode. Further, the CA90 is significantly advanced as compared to the diesel mode. Therefore, the dual-fuel mode HC emissions levels are considerably lower as compared to those at the lower loads. At this load, despite the higher LPG substitution, the dual-fuel engine performance is the same as that of the diesel mode, and the HC emissions are only moderately higher when compared with that of diesel. Therefore, the higher LPG substitution of 64% is selected for the emissions study with the oxidation catalytic converters.

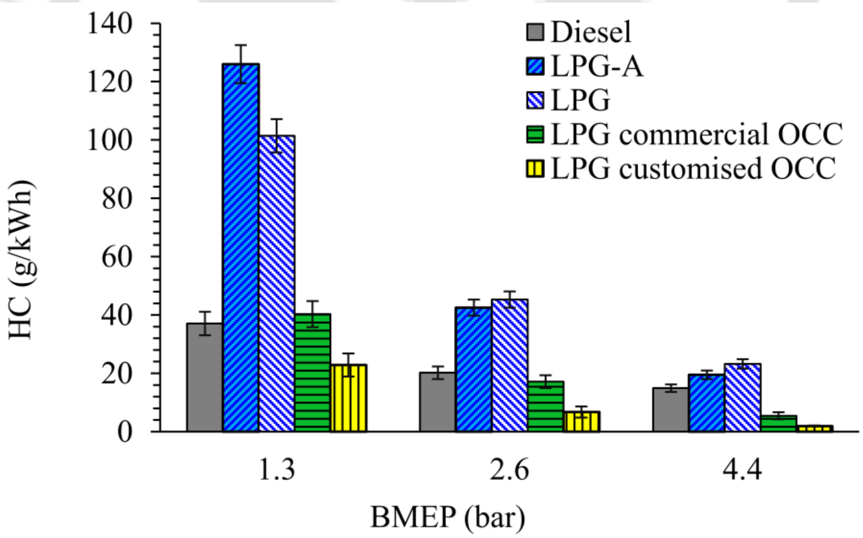


Figure 4.17: HC emissions for diesel and LPG-diesel at different BMEPs.

Figure 4.17 also shows the dual-fuel mode HC emissions with the commercial OCC and the customised OCC. At below half-load operations, the dual-fuel HC conversion efficiency with the commercial OCC is limited to about 60%. Consequently, the resulting HC emissions with the commercial OCC are 8.7% higher than those of the diesel mode at 20% load. At 40% load, the commercial OCC reduces the dual-fuel mode HC to a level slightly lower than that of the diesel mode. It is observed that the exhaust gas temperatures below the half-load conditions are about

230-300 ^{o}C only. The results indicate that the commercial OCC could not effectively oxidise the dual-fuel mode HC at this low range of exhaust gas temperature. At 70% load, the dual-fuel HC conversion efficiency with the commercial OCC improves to 77%.

The customised OCC performed significantly better as compared to the commercial OCC. At the low loads of 20-40%, the customised OCC achieves 77-85% dual-fuel HC conversion efficiency. The HC levels so achieved are 38% to 67% lower than those of the diesel mode. At 70% load, the dual-fuel HC conversion with the customised OCC is 92%, and the resulting HC is 87% lower than that of the diesel mode. The dual-fuel HC conversions achieved with the customised OCC are better than those reported in the literature for natural gas and alcohol fumigations [Bittner and Aboujaoude, 1992; Tsang et al., 2010; Zhang et al., 2010].

4.5.2 CO Emissions

Figure 4.18 shows the CO emissions of the diesel mode and the dual-fuel mode. At below half-load operations, the dual-fuel mode CO emissions are exceedingly high. At 20% load, an increase in the LPG substitution from 36% to 43% results in CO emissions 208% to 230% higher than those of the diesel mode. The dual-fuel mode CO emissions are consistently high for engine operations up to 40% load. At the 40% load, 39% to 48% LPG substitutions results in CO emissions 164% to 231% higher than those of the diesel mode. However, at 70% load, despite the high levels of LPG substitutions, the dual-fuel mode CO emissions are marginally lower by about 5% as compared to those of diesel. The reason for the exceedingly high CO emissions at below half-load operations is attributed to incomplete combustion. The dual-fuel combustion at these low loads is characterised by the lower cylinder mean gas temperature and the delayed CA10 and CA50, indicating poor combustion. It has been pointed out earlier that these low-load conditions are associated with poor BTE for the dual-fuel mode.

The results of dual-fuel mode CO emissions, as shown in Fig. 4.18, show that CO is readily oxidised by either of the OCCs. This is in line with results reported in the literature [Di Iorio et al., 2016; Dishy et al., 1995; Bittner and Aboujaoude, 1992]. The dual-fuel mode HC is reduced by about 97% with the use of either of the OCCs. The resulting CO emissions are less than those of the diesel mode by 89-94%.

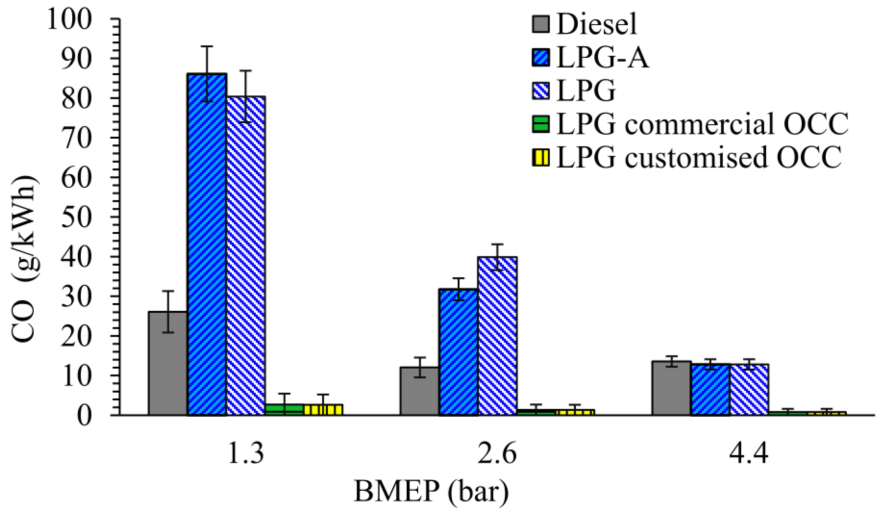


Fig. 4.18: CO emissions for diesel and LPG-diesel at different BMEPs.

4.5.3 NOx Emissions

Figure 4.19 shows the NOx emissions of the diesel mode and the dual-fuel mode. The dual-fuel mode NOx emissions reduce significantly at the lower loads, with higher LPG substitution levels resulting in higher reductions. A similar trend is reported in the literature for LPG-diesel dual-fuel combustion [Sudhir et al., 2003]. At the lowest load of 1.3 bar BMEP, for 36% LPG substitution, NOx emissions reduce by 64%. At 40% load, with LPG substitution of 39%, NOx emissions decrease by 45%. Further, at this load, when LPG substitution is 48%, the NOx emissions are 61% lower than those of the diesel mode. As the load is increased to 70%, 51% LPG substitution results in NOx emissions which are marginally lower than those of the diesel mode. At the same load, 64% LPG substitution results in 22% NOx reduction. Ganesan and Ramesh [2001] also reported reduction in NOx emissions with increasing LPG substitution levels at 20-80% loads.

Figure 4.19 also shows the NOx emissions with the use of the OCCs. Results of different engines compiled by Oslen et al. [2010] show that conversions of either NO to NO_2 or NO_2 to NO are possible in oxidation catalysts. NO_2 is a very effective oxidising agent [Oslen et al., 2010]. Conversions of NO_2 to NO are reported for two different oxidation catalysts at OCC temperatures below 275 °C and 375 °C, while the trend is opposite above these temperatures [Ambs and McClure, 1993]. Liu et al. [2012] reported that NO_2 emissions of a natural gas-diesel dual-fuel engine are highly dependent on engine load, and their results indicate that NO_2 is more than NO at low load, while the trend is opposite at high load. At low loads, the interaction of exhaust gas having elevated NO_2/NO ratio with an OCC could possibly result in a significant level of NO_2 to NO conversion. Any change in NO_2 and NO ratio will change the measurement value of the electrochemical sensor system as it measures only NO and not NO_2. In this study, dual-fuel mode exhaust NO_2 concentrations, estimated from additional emission measurement data, indicate that NO_2/NO ratio is elevated at low load. At two load points, approximate NO_2 concentrations in the engine exhaust were estimated indirectly by simultaneous measurements of NO and total of NO+NO_2, based on the electrochemical sensor principle and the chemiluminescence detector principle, respectively. Total concentration of NO+NO_2 measured by the chemiluminescence detector analyzer was subtracted by NO concentration value from the electrochemical sensor analyzer to get the approximate NO_2 concentration. It was observed that NO_2 increased significantly as the load was decreased from 70% to 20% loads, corresponding to 64% and 36% LPG substitutions, respectively. At 1.3 bar BMEP (20% load), the ratio of NO_2/(NO+NO_2) was about 0.71. However, at 4.4 bar BMEP, the NO_2/(NO+NO_2) ratio reduced drastically to about 0.25. Similar trend is reported by Liu et al. [2012]. Their results show that the ratio of NO_2 to total of NO+NO_2 for a natural gas fumigated dual-fuel compression ignition engine was about 0.64 at an operation below 20% load, while it drastically reduced to about 0.1 at full load. Substantial increase in NO_2 emissions at low load is also reported for H_2-diesel dual-fuel engine [Liu et al., 2011].

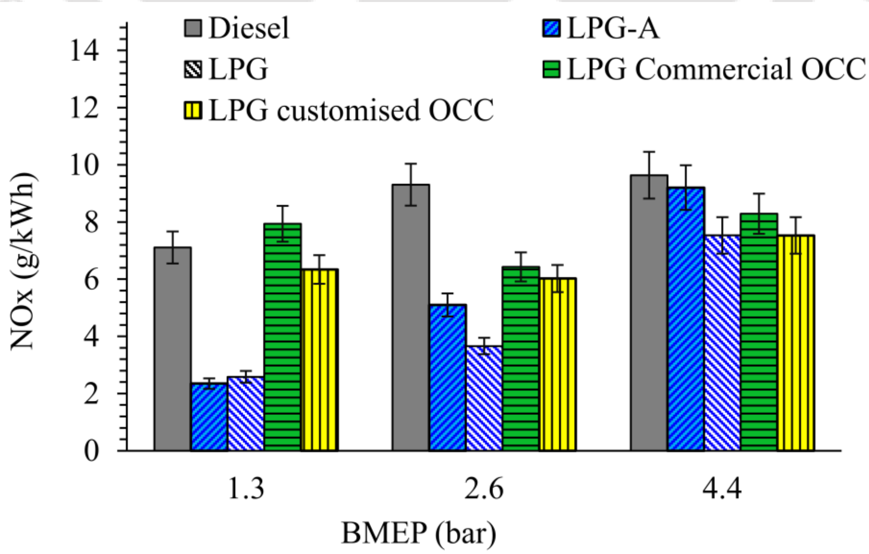


Fig.4.19: NOx emissions for diesel and LPG-diesel at different BMEPs.

4.5.4 Smoke

Figure 4.20 shows the smoke opacities for the diesel mode and the dual-fuel mode. The most prominent reduction of smoke opacity with the dual-fuel mode is observed at 70% load. At this load, the smoke opacity is reduced by 88% with 64% LPG substitution. It is noted that the smoke opacity value of the diesel mode at this load is significantly high, with opacity value of 39%. At 40% load, the smoke opacity reduction with the LPG-diesel is in the range of 50%. Further, at the lowest load tested, the dual-fuel mode reduces the smoke opacity by 31%. The levels of smoke reductions observed with the LPG fumigation agree with those reported in the literature [Sudhir et al., 2003]. Further, it is observed that the smoke levels achieved with the 36-48% LPG substitutions are reasonably low. Therefore, for 20-40% loads, LPG substitutions higher than the levels used here may not likely result in further significant benefits, more so as other aspects such as BTE, CO, and HC deteriorate at these loads. It is also reported in the literature that smoke reductions with an LPG-diesel dual-fuel engine diminishes after LPG substitution level of 40% [Jian et al. 2001].

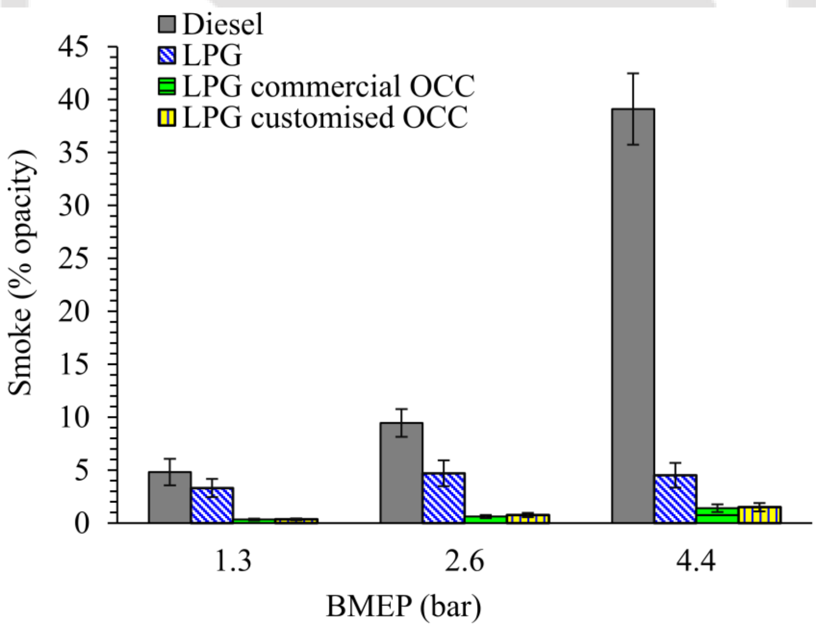


Fig 4.20: Smoke levels for diesel and LPG-diesel at different BMEPs.

Oxidation catalytic converters can oxidise the soluble organic fractions (SOFs) of engine exhaust particulate matters. The percentage of SOFs in the exhaust smoke increases with a decrease in load, and at low engine loads, corresponding to low exhaust temperatures, the exhaust smoke contains a high percentage of SOFs [Horiuchi et al. 1990]. In this study, with the combined effect of dual-fuel combustion and the OCCs, the smoke emissions are 92-96% lower when compared with the diesel mode.

4.6 Summary

LPG-diesel dual-fuel mode is studied for reductions of CO and HC at low loads with the use of oxidation catalysts. Evaluation of engine performance and combustion characteristics have also been done. The following summary is made from this study:

- At 20-40% load, the BTEs of the LPG-diesel dual-fuel engine are significantly lower than those of the diesel mode. However, at 70% load, the dual-fuel engine performance is the same as that of diesel. RCE and ITE show similar trends as the BTE.
- At below half-load operations, the LPG-diesel dual-fuel engine exhibits poor combustion characteristics. The dual-fuel combustion characteristics are observed to improve with the increase in engine load.
- CO and HC emissions of the LPG-diesel dual-fuel mode are exceedingly high at below half-load operations. At 20% load, the LPG fumigation increases the CO emissions by 208-230%. The corresponding increase in HC is 173-240%. At 40% load, the LPG fumigation increases the CO emissions by 164-231%, while the increase in HC is 110-124%.
- The dual-fuel mode CO and HC emissions levels at 70% load are considerably lower than those at the lower loads. At this load, the dual-fuel CO is marginally lower while HC is moderately higher by 31-56% as compared to the diesel mode.
- The dual-fuel mode CO emissions are readily oxidised by either of the OCCs. However, the dual-fuel HC conversion efficiency of the commercial OCC is limited to about 60% at 20-40% load. The customised OCC performs significantly better and achieves 77-85% dual-fuel HC conversion at the lower loads, and the conversion efficiency improves to 92% at 70% load. While the use of the customised OC results in dual-fuel mode HC emissions lower than those of the diesel, it is observed that the resulting specific emissions of HC at 20% load are comparatively higher than those at 40-70% loads.

- The LPG fumigation results in significant reductions of NOx and smoke emissions. At the highest tested load, the smoke reduction is as high as 88%. With the combined effects of LPG fumigation and OCCs, the smoke emissions are lower than those of diesel by 92-96%.

Table 4.6 LPG-diesel dual-fuel engine performance and combustion with reference to diesel.

Parameter	LPG-diesel dual-fuel mode with reference to diesel mode[a]		
	1.3 bar BMEP	2.6 bar BMEP	4.4 bar BMEP
BTE (% relative increase or decrease)	(-) 15.60	(-) 8.25	(+) 0.24
CA10 (^{o}CA delay)	(+) 4	(+) 4	(+) 5
CA50 (^{o}CA delay)	(+) 4	(+) 4	(+) 3
CA90 (^{o}CA advance)	(-) 6	(-) 6	(-) 10

a: Delay of CA10 and CA50 is denoted by (+); reduction and increase of BTE are denoted by (-) and (+) respectively; and advance of CA90 is denoted by (-).

Table 4.7 CO and HC conversion efficiencies of the OCCs.

Parameter	Conversion efficiency of commercial OCC (%)			Conversion efficiency of customised OCC (%)		
	1.3 bar BMEP	2.6 bar BMEP	4.4 bar BMEP	1.3 bar BMEP	2.6 bar BMEP	4.4 bar BMEP
CO	97	97	94	97	97	94
HC	60	62	77	77	85	92

Table 4.8 Comparative LPG-diesel dual-fuel emissions with reference to diesel.

Parameter	LPG-diesel dual-fuel mode with customised OCC with reference to diesel mode without OCC		
	1.3 bar BMEP	2.6 bar BMEP	4.4 bar BMEP
CO (% reduction)	90	89	94
HC (% reduction)	38	67	87
NOx (% reduction)	11	35	22
Smoke (% reduction)	93	92	96

Table 4.6 through 4.8 summarise the CO and HC conversion efficiencies of the OCCs and the overall LPG-diesel dual-fuel engine performance, combustion, and emissions with reference to the diesel mode. The results in Table 4.6 through 4.8 are for the LPG fumigation rates of 36%, 48%, and 64% at the BMEPs of 1.3 bar, 2.6 bar and 4.4 bar, respectively.

It is drawn from the above summary that LPG-diesel dual-fuel combustion results in lower BTE below half-load operations. However, BTE of the LPG-diesel dual-fuel mode is the same as that of diesel at 70% load. Smoke opacity is drastically reduced with LPG fumigation. Significant reductions in NOx emissions are observed at below half-load operations. However, CO and HC emissions are exceedingly high with the LPG fumigation. CO emissions are readily oxidised with the oxidation catalysts. However, the commercial oxidation catalyst is not effective for HC conversions at the low-load conditions. With the customised oxidation catalyst, the dual-fuel mode HC emissions are effectively reduced at all tested loads. However, there is a scope for further reduction of HC, especially at the lowest load, as the specific HC emissions values at this load are much higher than those at the higher load. Further, there is a need to improve the BTE of the LPG-diesel dual-fuel mode at below half-load operations. The addition of a high-cetane oxygenate fuel such as DME as a co-fumigant is expected to have significant effects on the LPG-diesel dual-fuel engine performance and emissions. Therefore, the use of DME as a co-fumigant with LPG is presented in the next chapter.

Chapter – 5

Investigation of DME-LPG Co-fumigated Dual-fuel Engine Performance, Combustion, and Emissions

Overview: *It is concluded in the previous chapter that the LPG-diesel dual-fuel mode of combustion results in high CO and HC emissions and low thermal efficiency at low engine loads. Further, it is observed that oxidation of the dual-fuel mode HC emissions is difficult with the commercial oxidation catalyst, while the customised oxidation catalyst is effective in the reduction of the dual-fuel mode HC. However, the results in the previous chapter show that there is a scope for further reduction of HC, especially at the lowest tested load. Further, there is a need to improve the thermal efficiency of the LPG-diesel dual-fuel mode below half-load operations. Therefore, in this chapter, a new approach consisting of a combined strategy using DME as a co-fumigant with LPG and deployment of the oxidation catalysts is presented. The objective of this study is to achieve dual-fuel engine thermal efficiency and emissions better than those of the diesel mode while utilising DME and LPG. It is observed in the literature review that DME has the potential to improve dual-fuel engine performance and emissions. Further, it has been discussed earlier that there is a scope to use DME and LPG together as fuels. However, data on dual-fuel engine studies on DME-LPG co-fumigation are limited. In the present study, comprehensive analyses of engine performance, combustion characteristics and emissions of CO, HC, NOx, and smoke are carried out. It is observed that DME, being a high-cetane oxygenate, exhibits interesting combustion characteristics in the dual-fuel mode, with the dual-fuel combustion resulting in low temperature and high temperature reactions and the combustion phasing showing significant improvement. The combined approach of DME-LPG co-fumigation and deployment of the customised oxidation catalyst is an effective strategy to drastically reduce emissions and achieve higher thermal efficiency as compared to the diesel mode*.

5.1 DME and LPG Substitution Levels and Test Matrix

The diesel energy substitution levels in the DME-LPG-diesel dual-fuel mode are approximately the same as those in the previous chapter for LPG-diesel dual-fuel mode. High gaseous fuels substitution levels are avoided at lower engine loads. The gaseous fuels substitution levels are given in Table 5.1. Selection of DME and LPG substitution levels with reference to the LPG-diesel dual-fuel mode provides the scope for comparative evaluation of the effect of DME-LPG co-fumigation with reference to the LPG alone fumigation. In this study, the diesel energy replacement rates of about 36%, 45% and 65% are used at 20%, 40% and 70% loads, respectively. These load points correspond to BMEPs of 1.3 bar, 2.6 bar, and 4.4 bar. At these loads, the indicated mean effective pressure (IMEP) values are 2.8 bar, 3.9 bar and 5.6 bar. While selecting the diesel energy replacement rate at the 70% load, the need to avoid combustion instability has been considered. DME being a high cetane fuel, a higher DME to LPG ratio in the dual-fuel mode could result in combustion instability. Even for a low-load operation, Prabhakar et al. [2015] reported that DME-propane-diesel dual-fuel mode combustion was unstable for DME substitutions higher than 30%. Therefore, at the 70% load, the total substitutions of gaseous fuels (DME+LPG) of about 65% provide the scope for a lower DME to LPG ratio while at the same time having a significant level of DME substitution.

Table 5.1 DME-LPG substitution rates in dual-fuel modes.

	BMEP (bar)	Diesel energy (%)	LPG energy (%)	DME energy (%)
DME-LPG-A	1.3	64	18	18
	2.6	53	24	23
	4.4	33	47	20
DME-LPG	1.3	64	11	25
	2.6	54	14	32
	4.4	36	52	12

At each of the tested engine loads, two different substitution levels of DME and LPG are studied, and the substitution levels that result in better combustion and engine performance are selected for further study on emissions. The test matrix is shown in Table 5.2. At 1.3 bar BMEP, for the 36% diesel energy replacement, DME substitution is varied from 18-25%, while that of LPG is varied from 11-18%. At 2.6 bar BMEP, the DME substitution is varied from 23-32% while that of LPG is varied from 14-24%. At 4.4 bar BMEP, the DME substitution is increased from 12-20%, and correspondingly the LPG substitution is decreased from 52% to 47%. At this load, the higher DME substitution level results in peak in-cylinder pressure about 10 bar higher than that of the diesel mode, and there is a significant increase in audible noise. Therefore, the lower DME substitution level is selected for the study on emissions. At 1.3 bar and 2.6 bar BMEPs, the combustion corresponding to the higher levels of DME substitutions results in better engine performance, and the engine operates smoothly. Therefore, the higher DME substitution levels are selected for the emissions study at 1.3-2.6 bar BMEP. For the sake of brevity, the sets of DME and LPG substitutions selected for the emissions study are identified as DME-LPG, while the other sets are denoted as DME-LPG-A. In Table 5.3, the properties of DME, LPG, and diesel used in the study are compared with the properties of DME, propane and butane from the literature.

Table 5.2: Test matrix.

Parameters studied	Fuels	Engine Load
Engine performance and combustion characteristics: BTE, in-cylinder pressure, NHRR, CMGT, CA10, CA50, CA90, RCE, ITE	*100% diesel* *18% DME + 18% LPG + 64% diesel (DME-LPG-A)* *25% DME + 11% LPG + 64% diesel (DME-LPG)*	*1.3 bar BMEP*
	100% diesel *23% DME + 24% LPG + 53% diesel (DME-LPG-A)* *32% DME + 14% LPG + 54% diesel (DME-LPG)*	*2.6 bar BMEP*
	100% diesel *12% DME + 52% LPG + 36% diesel (DME-LPG)* *20% DME + 47% LPG + 33% diesel (DME-LPG-A)*	*4.4 bar BMEP*
Emissions without OCC: CO, HC, NOx, smoke	*100% diesel* *25% DME + 11% LPG + 64% diesel (DME-LPG)*	*1.3 bar BMEP*
	100% diesel *32% DME + 14% LPG + 54% diesel (DME-LPG)*	*2.6 bar BMEP*
	100% diesel *12% DME + 52% LPG + 36% diesel (DME-LPG)*	*4.4 bar BMEP*
Emissions with the OCCs: CO, HC, NOx, smoke	*25% DME + 11% LPG + 64% diesel (DME-LPG)*	*1.3 bar BMEP*
	32% DME + 14% LPG + 54% diesel (DME-LPG)	*2.6 bar BMEP*
	12% DME + 52% LPG + 36% diesel (DME-LPG)	*4.4 bar BMEP*

Table 5.3 Comparison of properties of propane, butane, and DME from the literature [Huang et al., 1999; Kim et al., 2007; Ohno, 2001; Saleh, 2008; Sorenson and Mikkelsen, 1995; Ying et al., 2006] with the properties of the test fuels.

Property	Gaseous fuels from literature			Fuels used in the present study		
	Propane	Butane	DME	LPG	DME	Diesel
Boiling point (°C)	-42.0 – -42.1	-0.5	-24.9–-25.1	-	-24.9	273.5[a]
Vapour pressure at 20 °C (bar)	8.4	2.1	5.1	-	5.33	-
Liquid density: at 20 °C (kg/m^3) at 15 °C (kg/m^3)	490–501 -	610 -	660-670 -	- -	661 -	- 827.6
Lower heating value (kJ/kg)	46300–46442	45740–45800	28430–28870	45798	28712	42909
Ignition temperature at 1.01325 bar (°C)	470	365	235	-	350	-
Oxygen content (wt%)	-	-	34.8	-	34.8	-
Cetane number Cetane index	5 -	- -	55-60 -	- -	- -	- 55

a: 50% distillation.

5.2 Engine Performance

Figure 5.1 shows the BTEs of the diesel mode and the dual-fuel mode. With the DME-LPG co-fumigation, the BTEs of the dual-fuel mode are higher than those of the diesel mode. It is observed that BTE improves with an increase in the percentage of DME. At the lowest tested load, with 25% DME substitution and 11% LPG substitution, BTE is marginally higher than that of the diesel mode.

At 2.6 bar BMEP, the DME substitution levels are the highest. 23-32% DME substitution levels are used at this load. Correspondingly, the improvements in BTE with the DME-LPG co-fumigation are 8.75% to 14.29% as compared to the diesel mode. At the highest tested load, 12% DME substitution and 52% LPG substitution result in BTE 11.59% higher than that of the diesel mode. Further, the BTE improvement with reference to the diesel mode is 13.68% as the DME substitution is increased to 20% and the LPG substitution is decreased to 47%. The improvement in BTE with the increase in the DME substitution is attributed to the excellent properties of DME such as high cetane number, high oxygen content and easier to break fuel molecule due to the absence of carbon-carbon bond. As shall be discussed in Sec. 5.3, the DME addition improves the dual-fuel combustion characteristics, thereby achieving superior engine performance.

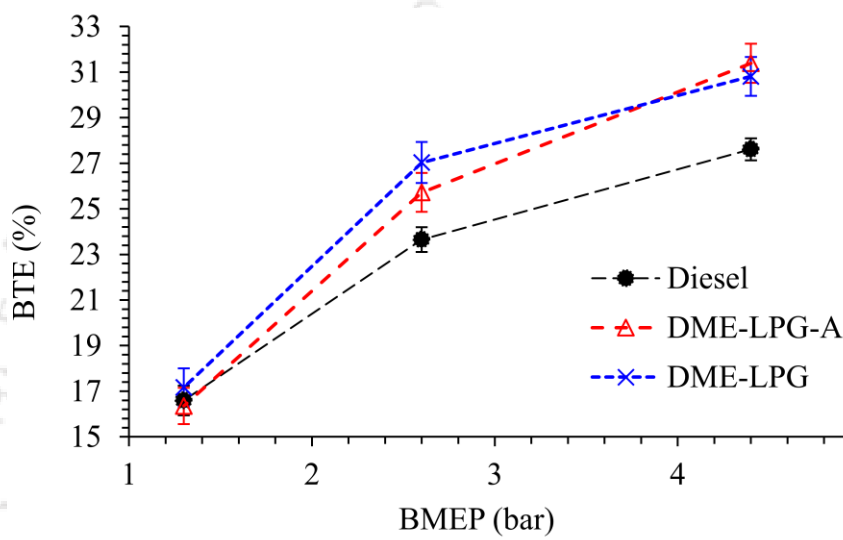


Fig. 5.1 Brake thermal efficiency variation with BMEP for DME-LPG-diesel dual-fuel mode and diesel mode.

5.3 Combustion Characteristics

5.3.1 In-cylinder Pressure

The in-cylinder pressure results are shown in Fig. 5.2 through 5.4 for the dual-fuel mode and the baseline diesel mode. The results show that the peak pressure of the dual-fuel mode increases with an increase in DME substitution. At 4.4 bar BMEP, the dual-fuel mode peak pressure increases from about 56 bar to about 66 bar as the DME substitution is increased from 12% to 20% and the LPG substitution is correspondingly decreased from 52% to 47%. At this load, the peak pressure with the higher DME substitution is about 10 bar higher than that of the diesel mode. Further, it is observed that the dual-fuel mode peak pressure is delayed and occurs at 6-10 ^{o}CA aTDC as compared to that of diesel mode, which occurs at 5 ^{o}CA aTDC. At 2.6 bar BMEP, as the DME substitution is increased from 20% to 32%, the cylinder peak pressure increases from about 51 bar to 53 bar. At this load, the dual-fuel mode peak pressure with the 32% DME substitution is higher than that of diesel by 1.5 bar and occurs at the same crank angle as that of diesel.

At the lowest load tested, for the 25% DME substitution and 11% LPG substitution, the peak pressure is identical to that of the diesel mode. However, for the DME and LPG substitutions of 18% each, the peak pressure is lower than that of the diesel mode by 2 bar. At this load, the peak pressures of the diesel mode and the dual-fuel mode occur at the same crank angle. It is observed from the results shown in Fig. 5.3 through 5.4 that the rapid rise of in-cylinder pressure occurs earlier for the dual-fuel mode with the higher DME substitution levels, the reason for which is attributed to early combustion, as shall be discussed in Sec. 5.3.3.

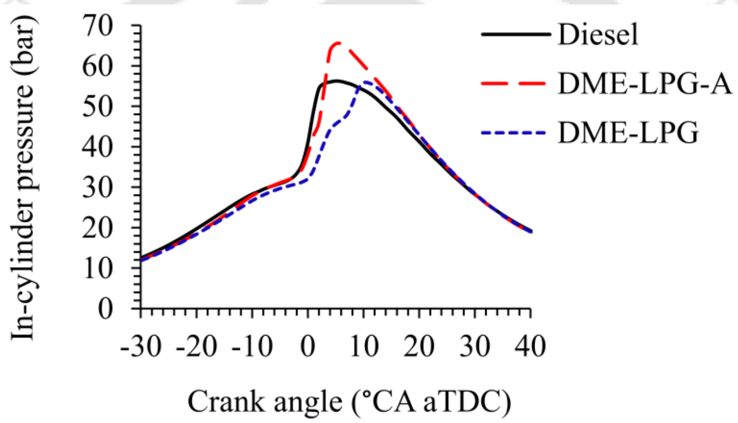


Fig. 5.2 In-cylinder pressure for DME-LPG-diesel dual-fuel mode and diesel mode at 4.4 bar BMEP.

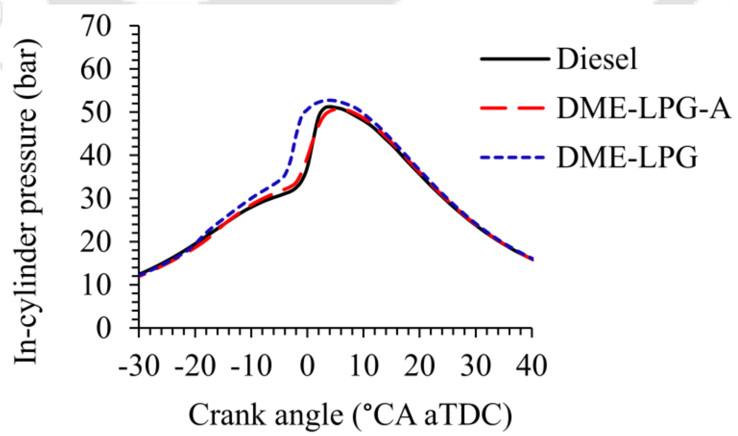


Fig. 5.3 In-cylinder pressure for DME-LPG-diesel dual-fuel mode and diesel mode at 2.6 bar BMEP.

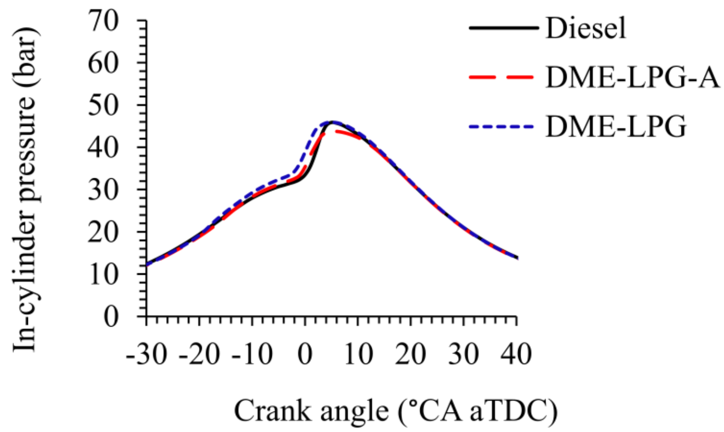


Fig. 5.4 In-cylinder pressure for DME-LPG-diesel dual-fuel mode and diesel mode at 1.3 bar BMEP.

5.3.2 Net Heat Release Rate

Figure 5.5 through 5.7 show the net heat release rates for the diesel mode and the dual-fuel mode. The results indicate that combustion of the dual-fuel mode is highly influenced by the DME fuel properties, such as the high cetane number. The dual-fuel combustion NHRR profiles are shown to exhibit LTR, HTR, and diesel diffusion combustion at all the tested loads. It is also reported in the literature that dual-fuel combustion, with DME fumigation or DME co-fumigation with LPG or propane, exhibits the LTR, HTR and diesel diffusion combustion [Prabhakar et al., 2015; Wang et al., 2013; Zhao et al., 2014].

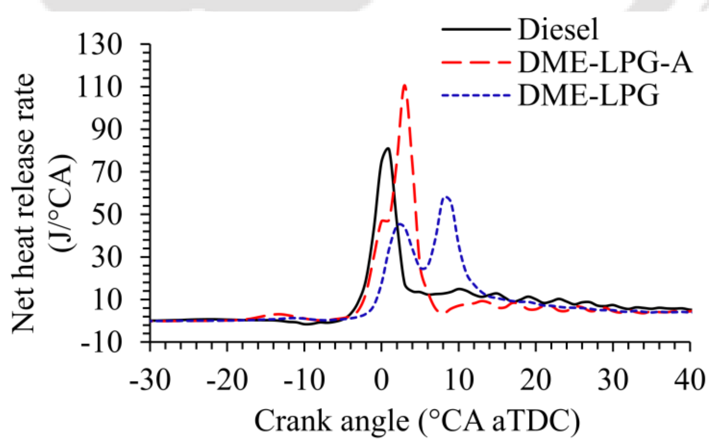


Fig. 5.5 NHRR for DME-LPG-diesel dual-fuel mode and diesel mode at 4.4 bar BMEP.

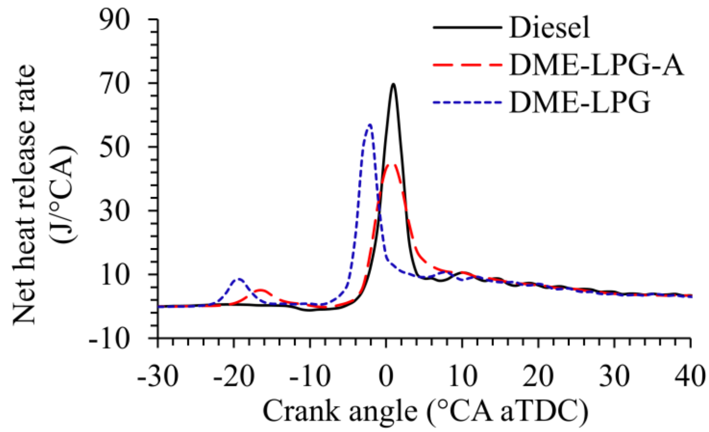


Fig. 5.6 NHRR for DME-LPG-diesel dual-fuel mode and diesel mode at 2.6 bar BMEP.

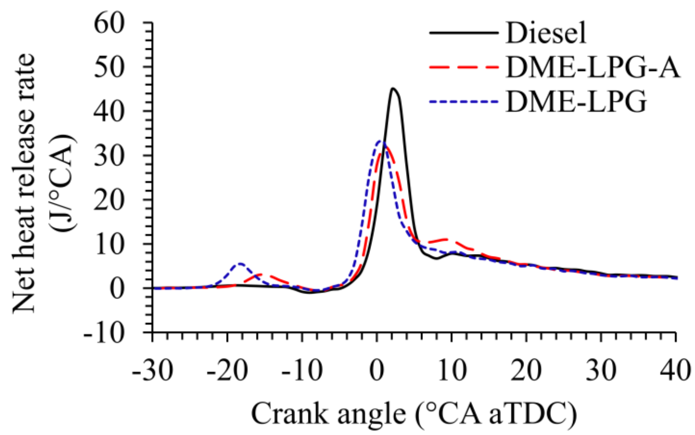


Fig. 5.7 NHRR for DME-LPG-diesel dual-fuel mode and diesel mode at 1.3 bar BMEP.

At the lowest tested load, for DME and LPG substitutions of 18% each, the LTR peak occurs at -15 ^{o}CA aTDC followed by the HTR peak at 1 ^{o}CA aTDC. As the DME substitution level is increased to 25% and that of LPG reduced to 11%, the LTR and HTR peaks are advanced and occur at -18 ^{o}CA aTDC and 0 ^{o}CA aTDC, respectively. At the BMEP of 2.6 bar, for 23% DME substitution and 24% LPG substitution, the LTR and HTR peaks occur at -17 ^{o}CA aTDC and 1 ^{o}CA aTDC, respectively. As the DME substitution is increased to 32%, the LTR and HTR peaks advance by 2-3 ^{o}CA.

At 4.4 bar BMEP, the LTR and HTR are not as prominent as those observed at the lower loads. For 12% DME substitution and 52% LPG substitution, the LTR and HTR peaks occur at -11 ^{o}CA aTDC and 8 ^{o}CA aTDC, respectively. The LTR and HTR peaks advance by 2 ^{o}CA and 5 ^{o}CA, respectively, as the DME substitution is increased to 20%.

5.3.3 Combustion Phasing and Combustion Duration

Figure 5.8 through 5.10 show the CA50 along with CA10 and CA90 for the diesel mode and the dual-fuel mode. Increasing the DME substitution level advances the CA50 at each of the tested loads. At the highest load, when the DME substitution is increased from 12% to 20%, CA50 is advanced by 3-8 ^{o}CA as compared to the diesel mode. At 2.6 bar BMEP, the advancement of CA50 with the DME-LPG co-fumigation is 2-3 ^{o}CA. At the lowest load, the CA50 with 18% DME substitution is the same as that of diesel. However, when the DME substitution is increased to 25%, the CA50 is advanced by 2 ^{o}CA. Advancement of CA50 with DME fumigation is also reported in the literature [Wang et al., 2013]. The DME-LPG co-fumigation results not only in the advancement of CA50, but also in the advancement of CA90. At 1.3-4.4 bar BMEPs, the CA90 of the dual-fuel mode is advanced by 7-15 ^{o}CA as compared to the diesel mode.

The cylinder mean gas temperature profiles are shown in Fig. 5.11 through 5.13. The results show that increasing the DME substitution results in higher values of peak cylinder mean gas temperature. At 2.6-4.4 bar BMEP, the peak cylinder mean gas temperature of the DME-LPG-diesel dual-fuel mode is consistently higher than that of diesel. At 1.3 bar BMEP, the peak cylinder mean gas temperature of the dual-fuel mode is almost identical to that of the diesel mode. It is also observed that the cylinder mean gas temperature profile, after its peak value, exhibits a more rapid rate of drop for the dual-fuel mode. The cylinder mean gas temperature trend and the highly advanced CA90 indicate that the combustion with the DME-LPG co-fumigation is more rapid than that of diesel.

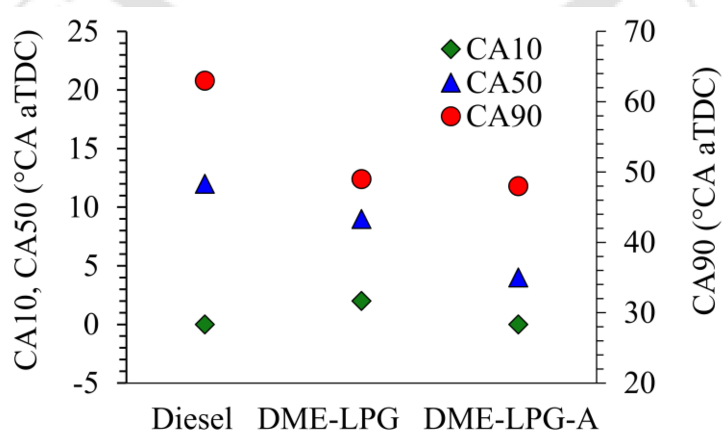


Fig. 5.8 CA10, CA50, and CA90 for DME-LPG-diesel mode and diesel mode at 4.4 bar BMEP.

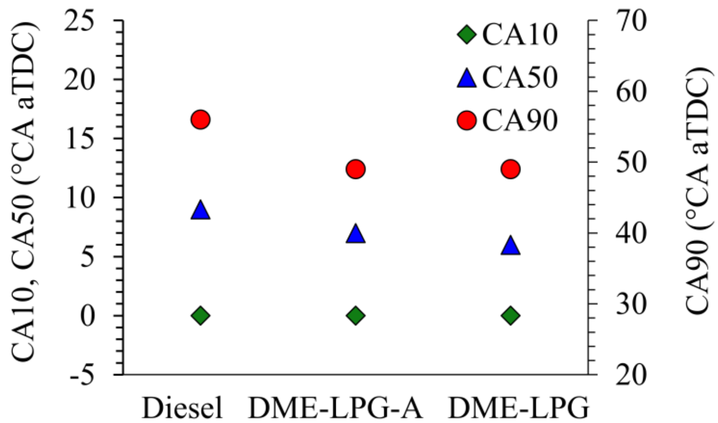


Fig. 5.9 CA10, CA50, and CA90 for DME-LPG-diesel mode and diesel mode at 2.6 bar BMEP.

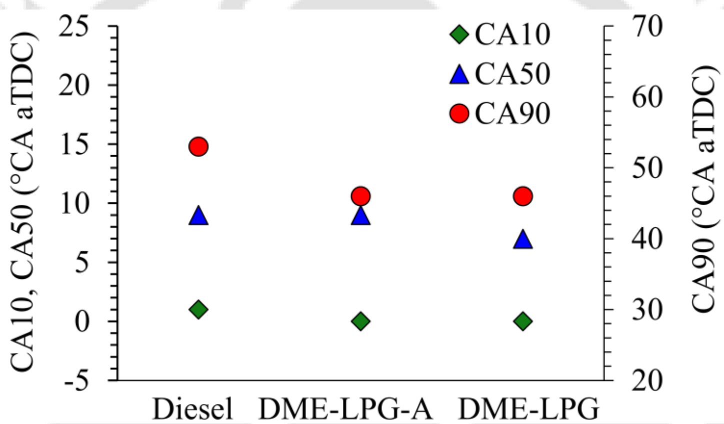


Fig. 5.10 CA10, CA50, and CA90 for DME-LPG-diesel mode and diesel mode at 1.3 bar BMEP.

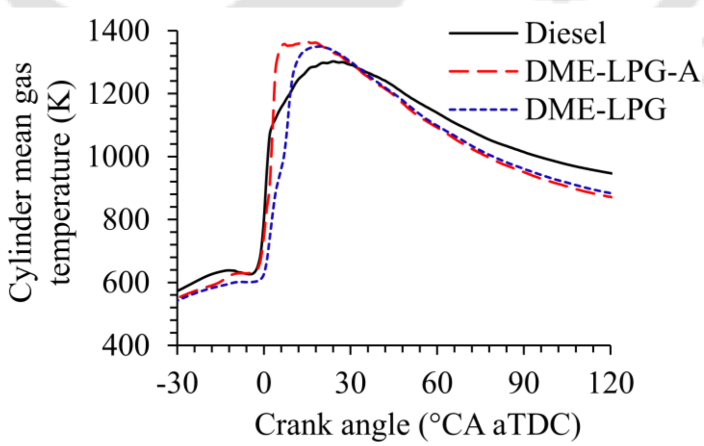


Fig. 5.11 Cylinder mean gas temperature for DME-LPG-diesel dual-fuel mode and diesel mode at 4.4 bar BMEP.

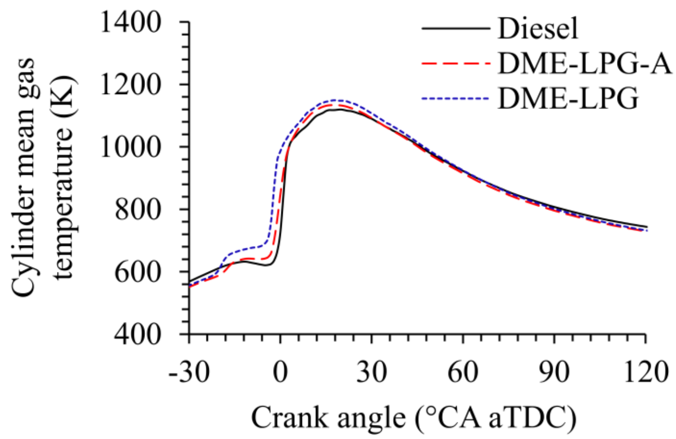


Fig. 5.12 Cylinder mean gas temperature for DME-LPG-diesel dual-fuel mode and diesel mode at 2.6 bar BMEP.

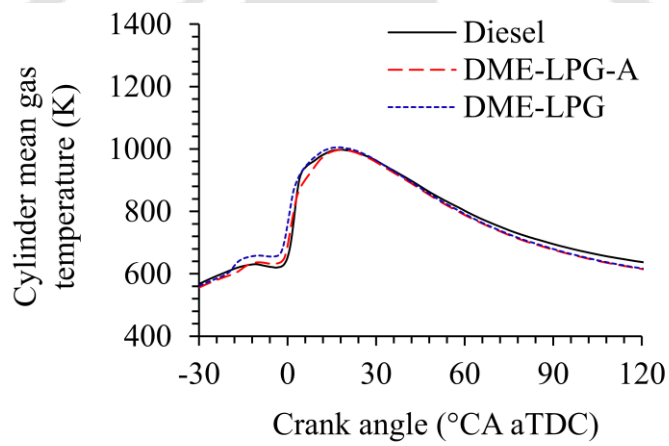


Fig. 5.13 Cylinder mean gas temperature for DME-LPG-diesel dual-fuel mode and diesel mode at 1.3 bar BMEP.

The combustion duration, taken as CA10-90, is shown in Fig. 5.14 through 5.16. The combustion duration significantly reduces with the DME and LPG co-fumigation. The significant reduction in the combustion duration with the dual-fuel mode is associated with the highly advanced CA90, indicating rapid combustion. At 4.4 bar BMEP, the dual-fuel combustion results in combustion durations of 47-48 ^{o}CA. At the same load, the combustion duration for the diesel mode is 63 ^{o}CA. At this load, the CA10 for both modes of combustion are comparable and, therefore, the main reason for the short combustion duration of the dual-fuel mode is the advanced CA90. As discussed earlier, the dual-fuel combustion at this load exhibits rapid combustion.

At 2.6 bar BMEP, the combustion duration of the dual-fuel mode is 50-53 ^{o}CA as compared to 56 ^{o}CA for the diesel mode. At this load, the CA10 for the dual-fuel mode is advanced by up to 4 ^{o}CA. This indicates that the combustion starts earlier and ends earlier as compared to the diesel mode. The high percentage of DME in the DME-LPG mixture – up to 70% DME in total gas – is responsible for the early CA10. At 1.3 bar BMEP, which corresponds to 20% load, the dual-fuel combustion results in CA10, which is 1-2 ^{o}CA advanced as compared to the diesel mode. However, the dual-fuel mode CA90 is significantly advanced. Therefore, the dual-fuel combustion durations are 46-47 ^{o}CA, while that of the diesel mode is 52 ^{o}CA. At this load, it is observed in the previous chapter that LPG-diesel combustion results in a delay of of CA10 by up to 5 ^{o}CA. The addition of DME as the co-fumigant – up to 70% DME in total gas - results in CA10 advancing by up to 7 ^{o}CA as compared to the LPG fumigation result presented in the previous chapter. This trend of CA10 shows that DME addition to LPG effectively advances the CA10.

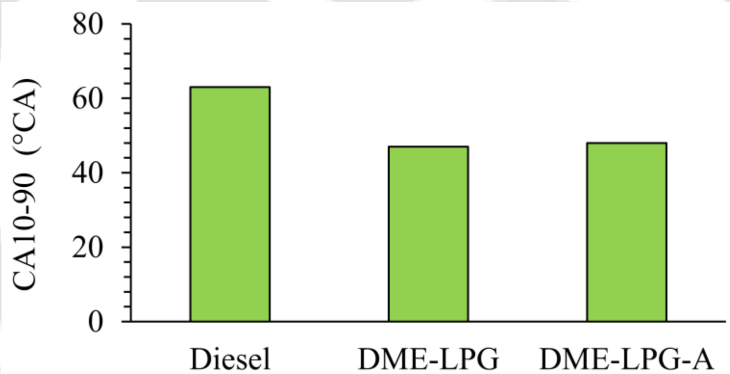


Fig. 5.14 Combustion duration for diesel and DME-LPG-diesel at 4.4 bar BMEP.

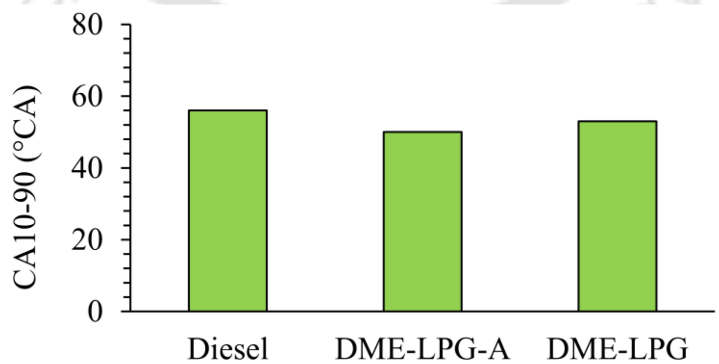


Fig. 5.15 Combustion duration for diesel and DME-LPG-diesel at 2.6 bar BMEP.

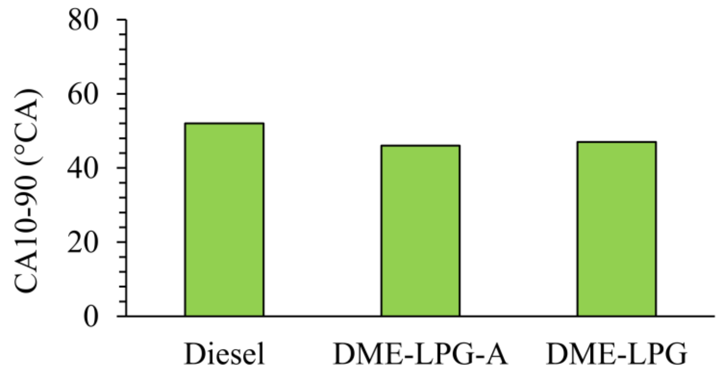


Fig. 5.16 Combustion duration diesel and DME-LPG-diesel at 1.3 bar BMEP.

5.3.4 Relative Cycle Efficiency and Indicated Thermal Efficiency

Figure 5.17 through 5.19 show the RCE and ITE of the diesel mode and the dual-fuel mode. RCE is the ratio of thermal efficiency of heat added at a given crank angle to that at TDC [Hsu, 1984; Poonia et al., 1998]. As discussed earlier, the DME-LPG co-fumigation has shown to have a significant influence on CA10, CA50 and CA90. Therefore, the quantity and position of heat release are different from that of diesel. The RCE calculation accounts for the effects of both quantity and position of heat release. At all the tested loads, RCEs improve with an increase in DME substitution. The same trend of improvement is also observed for ITE. Further, it is observed that RCE and ITE improve when both CA50 and CA90 are advanced. At 4.4 bar BMEP, the ITE of the dual-fuel mode is 39.77-40.47% as compared to 35.20% for the diesel mode.

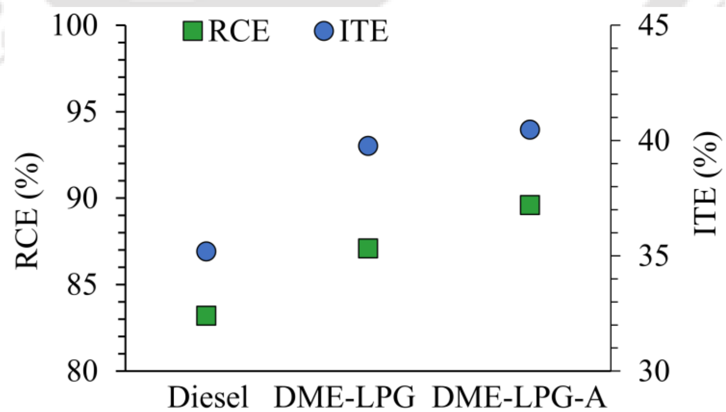


Fig. 5.17 RCE and ITE of DME-LPG-diesel dual-fuel mode and diesel mode at 4.4 bar BMEP.

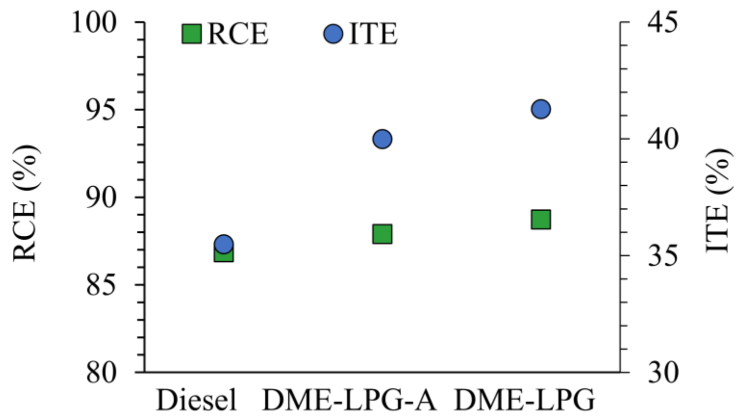


Fig. 5.18 RCE and ITE of DME-LPG-diesel dual-fuel mode and diesel mode at 2.6 bar BMEP.

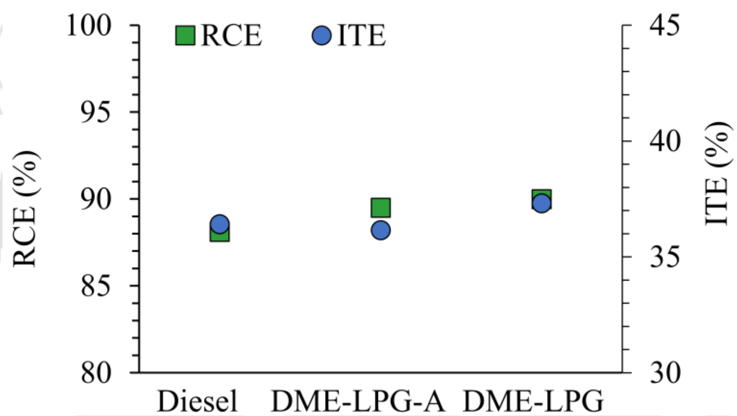


Fig. 5.19 RCE and ITE of DME-LPG-diesel dual-fuel mode and diesel mode at 1.3 bar BMEP.

At the lowest tested load, ITE of the dual-fuel mode with DME and LPG substitutions of 18% each is observed to be comparable to that of diesel mode. Further, with 25% DME and 11% LPG, the ITE is marginally higher than that of diesel. It has been observed in the previous chapter that ITEs of LPG-diesel dual-fuel mode are lower than those of the diesel mode below half-load operations. The results for DME-LPG co-fumigation show that the addition of DME as the co-fumigant with LPG effectively overcomes the issue of low ITE observed at lower loads for the LPG-diesel dual-fuel mode of combustion.

5.4 Emissions

5.4.1 HC Emissions

Figure 5.20 shows the HC emissions of the diesel mode and the dual-fuel mode. The HC emissions of the DME-LPG-diesel dual-fuel mode are within 11% of the diesel mode HC emissions. This is a very encouraging result as the HC emissions of the LPG-diesel dual-fuel mode, as discussed in the previous chapter, are exceedingly high. At 1.3 bar BMEP, the DME-LPG-diesel dual-fuel mode HC emissions level is slightly higher than that of diesel by 9%. At 2.6 bar BMEP, the energy share of DME is the highest (32% DME and 14% LPG), and consequently, the DME-LPG-diesel mode HC emissions level is marginally lower than that of diesel by 3%. At 4.4 bar BMEP, the DME-LPG-diesel mode HC emissions are 11% higher than those of diesel. The HC emissions of the DME-LPG-diesel dual-fuel mode are quite close to those of diesel because of better combustion. Low combustion temperature [Imran et al., 2013] and crevices [Königsson et al., 2013] contribute to high HC emissions in dual-fuel mode. For the DME-LPG co-fumigation, the peak cylinder mean gas temperature is higher than that of diesel. Moreover, the combustion phasing is better than that of the diesel mode. Therefore, achieving the reasonably moderate level of HC emissions for the DME-LPG-diesel mode – when compared with the baseline diesel mode – is due to the improved combustion characteristics.

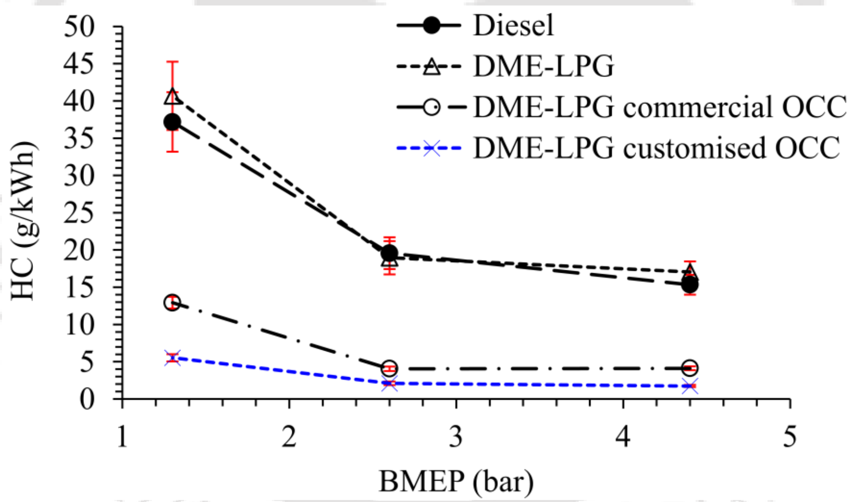


Fig. 5.20 HC emissions for DME-LPG-diesel dual-fuel mode and diesel mode at different BMEPs.

The DME-LPG-diesel dual-fuel mode HC emissions are reduced by the oxidation catalysts. The customised OCC performs better than the commercial OCC. At the lowest load tested, the dual-fuel HC conversion with the commercial OCC is limited to 68% only. At the same load, the dual-fuel HC conversion with the customised OCC is 86%, and the resulting HC emissions level is 85% lower than that of diesel. At the BMEP of 2.6 bar, the dual-fuel mode HC emissions conversion efficiency of the commercial OCC is 79% as compared to 89% for the customised OCC. At the highest load tested, the commercial OCC achieves 76% HC conversion efficiency. However, the customised OCC achieves a significantly higher conversion efficiency of 90%. At this load, the resulting dual-fuel mode HC emissions with the customised OCC are 89% lower than those of the diesel mode. The dual-fuel mode HC emissions with the customised OCC are within 2 g/kWh at 2.6-4.4 bar BMEP. Considering the results at all the tested loads, the dual-fuel mode HC emissions with the customised OCC are lower than those with the commercial OCC by 48-58%.

5.4.2 CO Emissions

The CO emissions for the diesel mode and the dual-fuel mode are shown in Fig. 5.21. At below half-load operations, the DME-LPG-diesel dual-fuel mode CO emissions are exceedingly high. At 1.3 bar BMEP, for 25% DME substitution and 11% LPG substitution, the increase in CO emissions is 382% as compared to that of diesel. At 2.6 bar BMEP, the increase in CO with the DME-LPG co-fumigation is 365%. However, at the higher load of 4.4 bar BMEP, the dual-fuel mode CO emissions level is lower than that of the diesel mode by 13%. At this load, the DME-LPG-diesel dual-fuel mode has higher peak cylinder mean gas temperature and rapid combustion, thereby resulting in CO emissions reduction. At the lower loads, the dual-fuel peak cylinder mean gas temperature values are either identical to that of diesel or only marginally higher. Therefore, at the lower loads, the combustion of the DME-LPG-diesel dual-fuel mode possibly requires a higher temperature for higher oxidation of CO. It is also reported in the literature that higher cylinder temperature reduces CO emissions in dual-fuel engines employing neat DME fumigation [Wang et al., 2014] and DME-propane fumigation [Prabhakar et al., 2015].

Both OCCs are effective in the dual-fuel mode CO conversions. It is also reported in the literature that CO emissions are effectively controlled by oxidation catalysts [Di Iorio et al., 2016; Dishy et al., 1995; Bittner and Aboujaoude, 1992]. In the present study, the CO conversion efficiencies with either of the OCCs are in the range of 93-98%, and the resulting dual-fuel mode CO emissions are 89-94% lower than those of the diesel mode. At the lowest load tested, the dual-fuel mode CO emissions are reduced by the customised OCC from 129 g/kWh to 2.7 g/kWh.

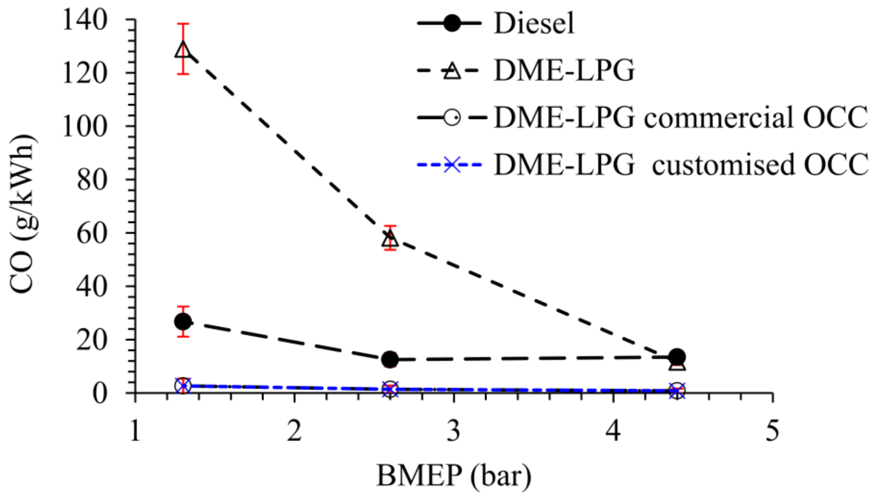


Fig. 5.21 CO emissions for DME-LPG-diesel dual-fuel mode and diesel mode at different BMEPs.

5.4.3 NOx Emissions

The NOx emissions for the diesel mode and the dual-fuel mode are shown in Fig. 5.22. At all the tested loads, the DME-LPG co-fumigation results in reductions of the NOx emissions. NOx emissions reduction of 47% is achieved with the dual-fuel mode at 1.3 bar BMEP. At 2.6 bar BMEP, the NOx emissions reduction with the dual-fuel mode is 51%. At 1.3 bar and 2.6 bar BMEP, the diesel energy replacements by the DME and LPG are 36% and 54%, respectively. For approximately the same level of diesel energy replacement by LPG alone, it is recalled from the previous chapter that the LPG-diesel dual-fuel mode NOx emissions are 61-64% lower than those of diesel. Therefore, the NOx emissions with the LPG-diesel dual-fuel mode are lower as compared to those with the DME-LPG-diesel dual-fuel mode. Further, at 4.4 bar BMEP, the NOx reduction with the DME-LPG co-fumigation is 16% as compared to the 22% reduction achieved with the LPG fumigation in the previous chapter. The higher oxygen content of DME contributes to the higher NOx emissions of DME-LPG co-fumigation as compared to the neat LPG fumigation. For a methanol-diesel dual-fuel engine, the high oxygen content of methanol is considered as a contributing reason for an increase in NOx at high load [Yao et al., 2008]. Simulation study also shows that DME fumigation results in higher NOx as compared to syngas fumigation [Kan et al., 2020].

It is discussed in the previous chapter that NO_2 to NO conversions possibly take place in the OCCs, thereby increasing the NO concentration detection by the electrochemical sensor. The same trend is observed with the use of the OCCs in the DME-LPG co-fumigated dual-fuel engine. It has been discussed in the previous chapter that the LPG-diesel dual-fuel combustion at low-load operations results in exhaust gases having elevated NO_2/NO ratios. At low loads, the interaction of exhaust gases having elevated NO_2/NO ratios with the OCCs could possibly result in significant NO_2 to NO conversions.

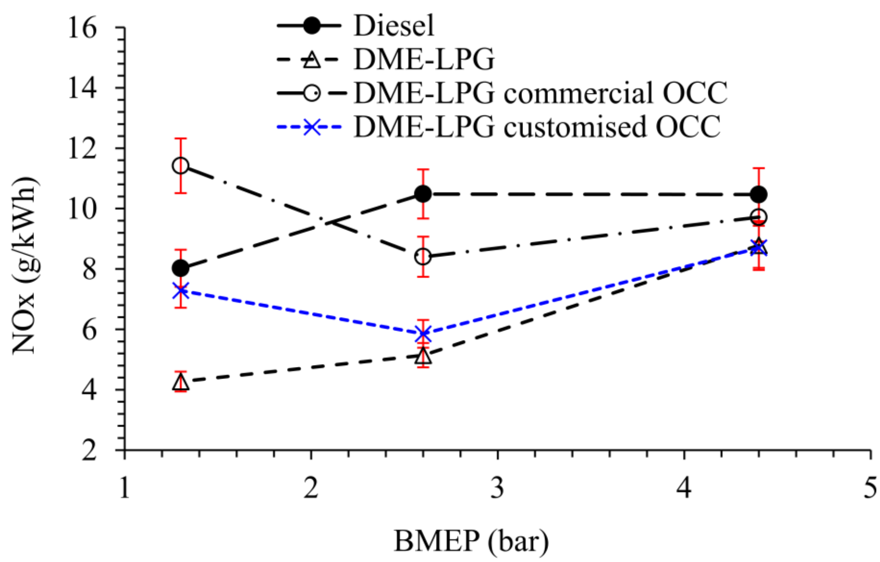


Fig. 5.22 NOx emissions for DME-LPG-diesel dual-fuel mode and diesel mode at different BMEPs.

5.4.4 Smoke

The smoke emissions for the diesel mode and the dual-fuel mode are shown in Fig. 5.23. Smoke opacity reduces with DME-LPG co-fumigation. A maximum smoke reduction of 89% is observed at 4.4 bar BMEP. At this load, the smoke opacity value for the diesel mode is 40%. At 1.3-2.6 bar BMEP, the smoke opacities of the dual-fuel mode are 43-54% lower than those of the diesel mode. The reduction of smoke emissions with the DME-LPG co-fumigation is attributed to the higher hydrogen to carbon ratio of the gaseous fuels, and the high oxygen content of the DME. When compared to the smoke reductions with the LPG-diesel dual-fuel mode from the previous chapter, the DME-LPG co-fumigation results in slightly more reductions of the smoke opacity at the lower loads. OCCs can reduce the soluble organic fractions of smoke. With the use of the OCCs, the dual-fuel mode smoke emissions are observed to reduce further. The dual-fuel mode smoke emissions with the OCCs are below 2.5% opacity at 1.3-4.4 bar BMEP. These opacity values are 78-94% lower than those of the diesel mode.

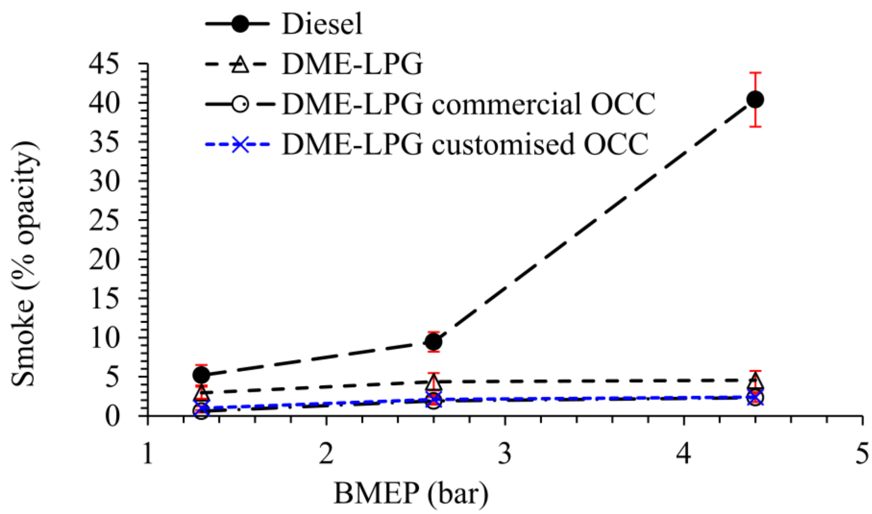


Fig. 5.23 Smoke opacity for DME-LPG-diesel dual-fuel mode and diesel mode at different BMEPs.

In Table 5.4, the performance and combustion characteristics of DME-LPG co-fumigation are compared with respect to the results obtained in the previous chapter for LPG alone fumigation. The comparisons shown in Table 5.4 correspond to the fumigation rates of LPG and DME-LPG selected for the detailed emissions studies.

The results in Table 5.4 show that DME-LPG co-fumigation achieves BTEs significantly higher than those of the LPG fumigation – about 24% higher at the lower loads. By the addition of DME as the co-fumigant with LPG, the low BTE issue of the LPG-diesel dual-fuel mode at below half-load operation is overcome. The improvement in BTE is associated with better combustion characteristics. CA50 is significantly advanced by 6-9 ^{o}CA with the addition of DME as the co-fumigant. The DME addition also advances the CA10 by 6-9 ^{o}CA at the lower loads. DME being a high-cetane oxygenate, significantly improves the dual-fuel combustion.

Table 5.4 Comparative engine performance and combustion characteristics of DME-LPG-diesel dual-fuel mode (substitution levels denoted as DME-LPG in Table 5.1) with reference to LPG-diesel dual-fuel mode (substitution levels denoted as LPG in Table 4.1).

Parameter	1.3 bar BMEP		Improvement for DME-LPG-diesel with respect to LPG-diesel
	LPG	DME-LPG	
BTE (%)	13.89	17.17	23.58 % relative increase
CA10 (^{o}CA aTDC)	5	-1	6 ^{o}CA advanced
CA50 (^{o}CA aTDC)	13	7	6 ^{o}CA advanced
CA90 (^{o}CA aTDC)	47	46	1 ^{o}CA advanced
	2.6 bar BMEP		
	LPG	DME-LPG	
BTE (%)	21.78	27.03	24.09 % relative increase
CA10 (^{o}CA aTDC)	5	- 4	9 ^{o}CA advanced
CA50 (^{o}CA aTDC)	15	6	9 ^{o}CA advanced
CA90 (^{o}CA aTDC)	51	49	2 ^{o}CA advanced
	4.4 bar BMEP		
	LPG	DME-LPG	
BTE (%)	27.63	30.81	11.51 % relative increase
CA10 (^{o}CA aTDC)	5	2	3 ^{o}CA advanced
CA50 (^{o}CA aTDC)	16	9	7 ^{o}CA advanced
CA90 (^{o}CA aTDC)	55	49	6 ^{o}CA advanced

5.5 Summary

DME-LPG-diesel dual-fuel mode is studied for engine performance, combustion, and emissions. The dual-fuel engine emissions are studied with and without the OCCs. The comparison of the dual-fuel mode with the diesel mode, in terms of engine performance, combustion and emissions, are summarised in Table 5.5.

Table 5.5 Comparison of the DME-LPG-diesel dual-fuel mode with the diesel mode.

	DME-LPG diesel with reference to diesel		
	1.3 bar BMEP	2.6 bar BMEP	4.4 bar BMEP
Performance and combustion:			
BTE (% relative increase)	(+) 3.48	(+) 14.29	(+) 11.59
CA50 (^oCA advance)	2	3	3
CA90 (^oCA advance)	7	7	14
Emissions[a]:			
CO (% change)	(-) 90	(-) 89	(-) 94
HC (% change)	(-) 85	(-) 89	(-) 89
Smoke (% change)	(-) 81	(-) 78	(-) 94
NOx (% change)	(-) 9	(-) 44	(-) 17

a; dual-fuel mode emissions with the customised OCC compared with the standard diesel mode emissions without OCC

The results of the DME-LPG-diesel dual-fuel mode are summarised as follows:

- The DME-LPG-diesel dual-fuel mode achieves better engine performance as compared to the diesel mode. Even at below half-load operations, the dual-fuel mode BTEs are higher than those of the diesel mode.
- The dual-fuel combustion exhibits LTR and HTR, which is due to the high cetane value of DME. As given in Table 5.5, the combustion phasing of the DME-LPG-diesel dual-fuel mode is better than that of the diesel mode. RCE and ITE improve with an increase in DME substitution. Further, RCE and ITE improve when both CA90 and CA50 are advanced. In such cases, the peak cylinder mean gas temperature is higher than that of diesel, and subsequently, it drops more rapidly as compared to the diesel mode.
- The HC emissions of the DME-LPG-diesel dual-fuel mode are within 11% of the diesel mode HC emissions. This is a very encouraging result as the HC emissions of the LPG-diesel dual-fuel mode, as discussed in the previous chapter, are exceedingly high.
- CO emissions are exceedingly high with the DME-LPG co-fumigation at below half-load operations. However, at 4.4 bar BMEP, the dual-fuel mode CO emissions are lower than those of the diesel mode by 13%.

- The customised OCC is more efficient than the commercial OCC in the dual-fuel mode HC conversions. At the lowest load tested, the dual-fuel HC conversion with the commercial OCC is limited to 68% only, while that of the customised OCC is 86%. At 2.6-4.4 bar BMEP, the customise OCC achieves 89-90% dual-fuel HC conversion efficiencies.
- Both OCCs are effective in the dual-fuel mode CO conversions, and the CO conversion efficiencies are in the range of 93-98%. At the lowest tested load, the dual-fuel mode CO emissions are reduced by the customised OCC from 129 g/kWh to 2.7 g/kWh.
- When compared to the results of LPG fumigation from the previous chapter, the DME-LPG co-fumigation has significantly higher BTE - about 24% higher at the lower loads. Therefore, it can be concluded that the addition of DME as the co-fumigant with LPG can overcome the low BTE issue of the LPG-diesel dual-fuel mode at below half-load operations. The improvement in BTE with the DME-LPG co-fumigation is associated with better combustion phasing. With DME-LPG-co-fumigation, CA10 and CA50 significantly advance as shown in Table 5.4.

It can be drawn from the above summary that DME-LPG co-fumigation results in engine performance superior to the diesel mode. It is concluded in the previous chapter that the LPG-diesel dual-fuel engine performance is poor at below half-load operations. This shortcoming is overcome by the addition of DME to LPG as the co-fumigant. Another prominent benefit observed is in the HC emissions. It is discussed in the previous chapter that the LPG-diesel dual-fuel combustion results in exceedingly high HC. The results presented in this chapter show that this issue of high HC is effectively addressed by the DME-LPG co-fumigation. The HC emissions of DME-LPG co-fumigation are remarkably close to those of diesel even without the oxidation catalyst. The combined strategy of DME-LPG co-fumigation and deployment of the customised oxidation catalyst has been shown to result in drastically reduced emissions and higher BTE as compared to the diesel mode. With the improvement in BTE and emissions achieved for the dual-fuel mode, further evaluation of the dual-fuel engine performance using exergy analysis is carried out and presented in the next chapter. The second law analysis, which helps identify the input fuel exergy and the location and quantity of the exergies resulting from the combustion of the fuel, is presented in detail in the next chapter.

Chapter – 6

Exergy Analysis for Dual-fuel Combustion with LPG Fumigation and DME-LPG Co-fumigation

Overview: *The results and discussion presented in the previous two chapters show that BTEs and emissions of the dual-fuel engine are significantly improved by the combined approach of DME-LPG co-fumigation and deployment of the customised oxidation catalyst. The results of BTEs and emissions obtained from the combined approach are even better than the baseline diesel mode. In this chapter, the exergy analysis is performed to identify and quantify the exergy distribution for the different options of the dual-fuel mode studied in the previous chapters. The second law analysis quantifies the exergetic performance and helps identify the fumigation option that exhibits superior engine performance. The exergy analysis is highly informative in that it provides an inclusive understanding of the input exergy, the output shaft work, and the other availabilities such as the exhaust exergy. The literature survey discussed earlier shows that there is a lack of data on exergy analysis of dual-fuel engines which utilise LPG and DME as the fumigants. Therefore, this chapter presents a detailed exergy analysis for the LPG-diesel dual-fuel mode, the DME-LPG-diesel dual-fuel mode, and the baseline diesel mode.*

6.1 LPG and DME Substitution Levels and Test Matrix

The diesel substitution rates by LPG and DME-LPG are given in Table 6.1. The engine performance, combustion, and emissions characteristics for these levels of LPG and DME substitutions have been discussed in detail in the previous chapters. The second law analysis is carried out to gain further insights into the exergetic performance with the LPG fumigation and the DME-LPG co-fumigation. The analysis is carried out as per the procedure presented in Appendix B. For each case of fumigation, the main parameters studied are input exergy, work exergy, exhaust exergy, coolant exergy, exergy destruction, and exergy efficiency. These parameters are studied at BMEPs of 1.3 bar, 2.6 bar and 4.4 bar.

Table 6.1 LPG and DME-LPG substitution rates in dual-fuel modes.

	BMEP (bar)	Diesel energy (%)	LPG energy (%)	DME energy (%)
LPG-diesel	1.3	64	36	-
	2.6	52	48	-
	4.4	36	64	-
DME-LPG-diesel	1.3	64	11	25
	2.6	54	14	32
	4.4	36	52	12

6.2 Exergy Distribution and Exergy Efficiency

6.2.1 Input Exergy

The input exergy is the exergy of the fuel used for combustion. When a mixture of fuel is used, the input exergy is the total of exergies of all the fuels. On combustion of a fuel in the internal combustion engine, a portion of the input exergy is utilised as the shaft work. Apart from the shaft work, the exhaust gas exergy is important as it has the potential to be recovered through methods such as waste heat recovery. The input exergies for all cases are shown in Fig. 6.1.

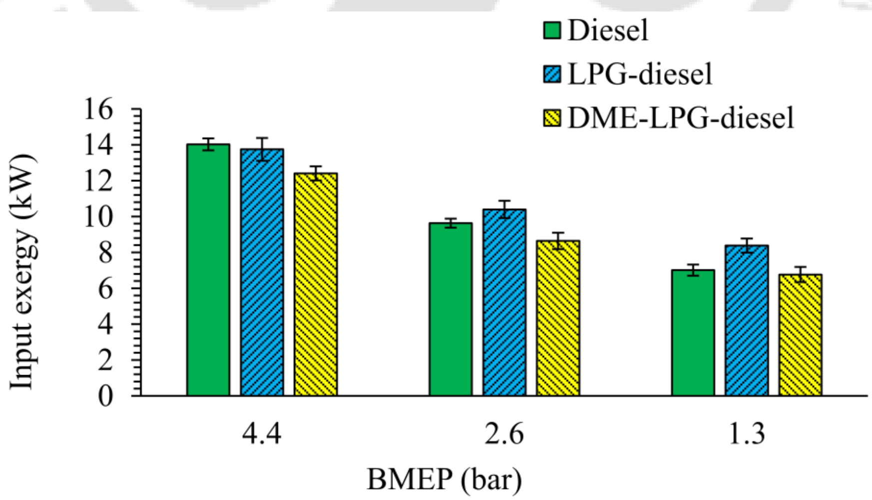


Fig. 6.1 Input exergies for diesel and dual-fuel modes.

At below half-load operations (1.3-2.6 bar BMEP), the input exergy of the LPG-diesel mode is the highest compared to the DME-LPG-diesel dual-fuel mode and the baseline diesel mode. However, at the higher load, the LPG-diesel dual-fuel combustion has marginally lower input exergy as compared to the baseline diesel mode. Of all the cases studied, DME-LPG co-fumigation results in the lowest input exergy. For a given shaft work, a reduction or an increase in input exergy can be correlated with the different exergies associated with exhaust gas, heat transfer and exergy destruction. The exergy distribution is discussed in detail in the following sections.

6.2.2 Exhaust Gas Exergy

The results of the exhaust gas exergy, both in magnitude and as a percentage of the input exergy, are shown in Fig. 6.2 and 6.3 for different cases of fumigations at different loads. The peak cylinder mean gas temperature and exhaust gas temperature for all cases are shown in Fig. 6.4 and 6.5, respectively. The LPG-diesel dual-fuel mode combustion results in lower exhaust gas exergy as compared to the diesel mode.

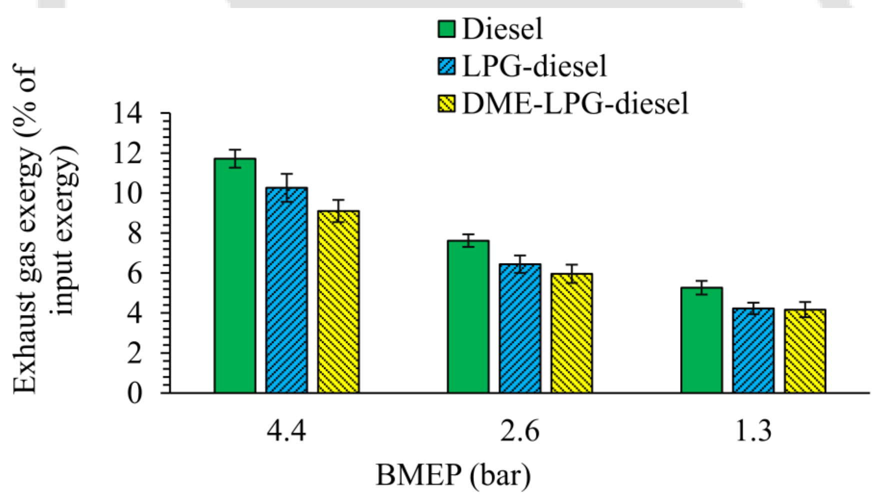


Fig. 6.2 Exhaust gas exergies for diesel and dual-fuel modes as percentages of input exergies.

The lower exhaust gas exergy with the LPG-diesel dual-fuel mode is due to early CA90. It is observed in the LPG-diesel dual-fuel combustion study that CA90 occurs earlier for the dual-fuel mode when compared with the diesel mode.

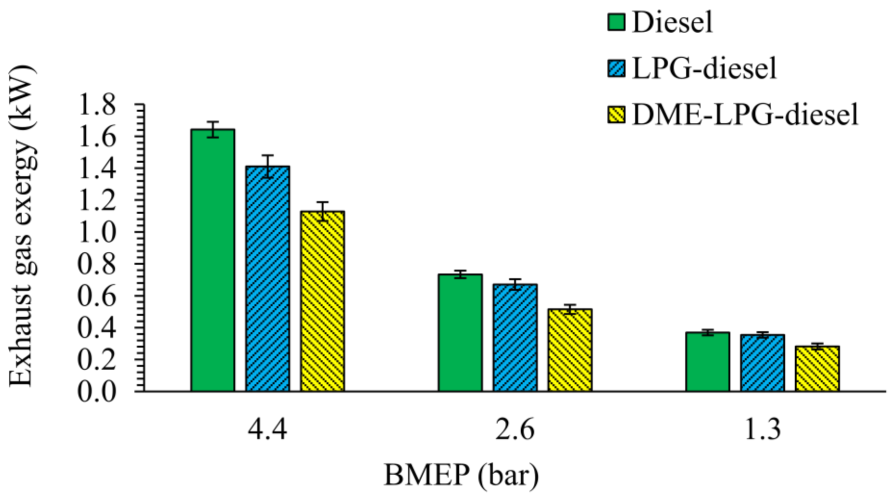


Fig. 6.3 Magnitudes of exhaust gas exergy for diesel and dual-fuel modes.

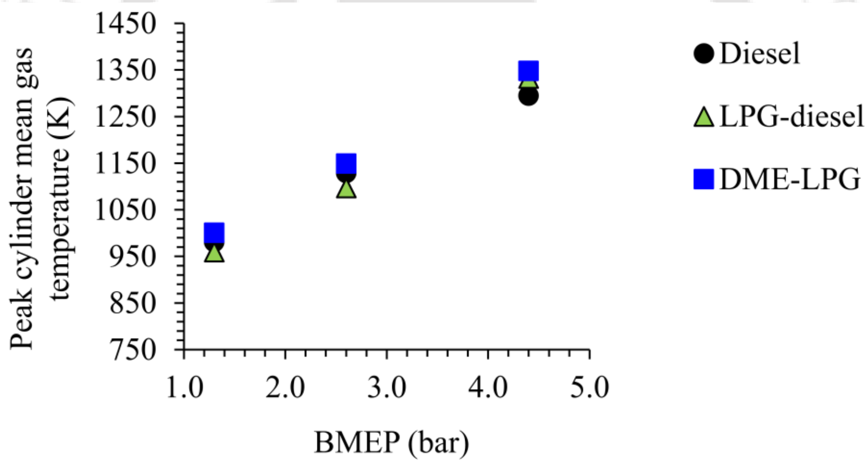


Fig. 6.4 Peak cylinder mean gas temperature for diesel and dual-fuel modes.

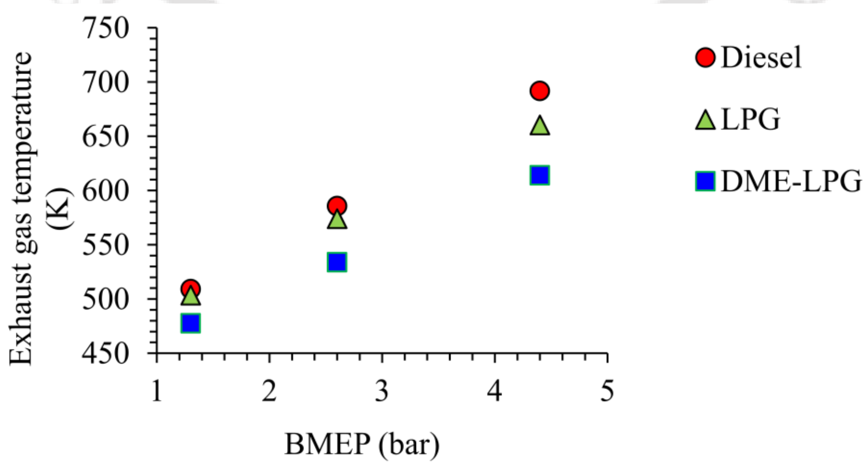


Fig. 6.5 Exhaust gas temperature for diesel and dual-fuel modes.

The exhaust gas exergy of the dual-fuel mode with the DME-LPG co-fumigation presents a consistent trend, with the DME-LPG co-fumigation resulting in the lowest exhaust gas exergy values when compared with the LPG-diesel mode and the baseline diesel mode. Further, these lower values of exhaust gas exergy are observed along with lower exhaust temperature and lower input exergy. The peak cylinder mean gas temperatures and exhaust gas temperatures are given in Fig 6.4 and 6.5. The peak cylinder mean gas temperature is the highest for the DME-LPG co-fumigation. Further, it has been discussed in the previous chapter that CA50 and CA90 significantly advance with DME-LPG co-fumigations. Higher peak cylinder mean gas temperature and advanced CA50 and CA90 indicate better combustion phasing and rapid combustion with DME-LPG co-fumigation, thereby resulting in lower exhaust gas temperatures. The lower exhaust gas temperatures imply lower exhaust gas exergies with DME-LPG co-fumigation as compared to the diesel mode and the LPG-diesel mode.

6.2.3 Coolant Exergy

The coolant exergy results, as percentages of input exergies, are shown in Fig. 6.6. The coolant exergy, as a percentage of the input exergy, follows the trend of peak cylinder mean gas temperature – experimental points with higher peak cylinder mean gas temperature showing higher coolant exergy. The higher peak cylinder mean gas transfer enhances heat transfer to the cooling water.

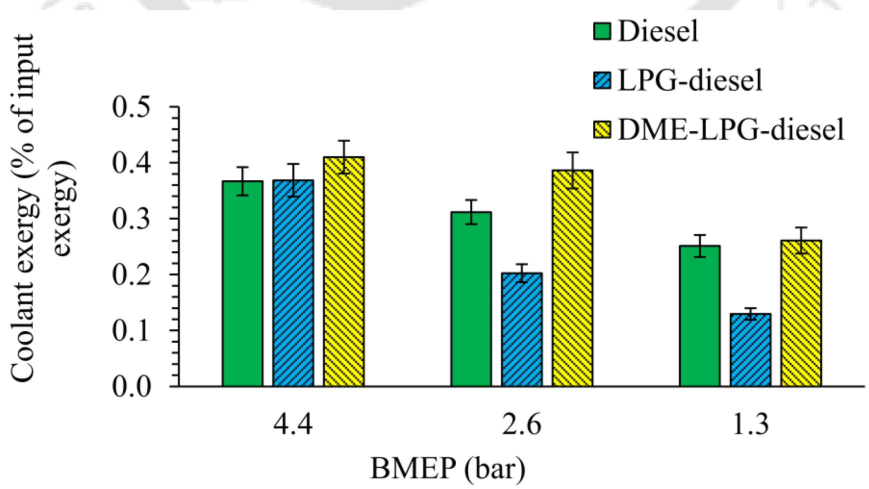


Fig. 6.6 Coolant exergies for diesel and dual-fuel modes as percentages of input exergies.

The coolant exergies, as percentages of input exergies, with the LPG-diesel dual-fuel mode are 0.13%, 0.20% and 0.37% at 1.3 bar, 2.6 bar and 4.4 bar BMEP, respectively. The corresponding coolant exergies for the diesel mode are 0.25%, 0.31 and 0.37% at 1.3-4.4 bar BMEP. DME-LPG co-fumigation has the highest exergy efficiency at each tested load, as shall be discussed in Sec. 6.2.5, and correspondingly the coolant exergy as a percentage of input exergy with the DME-LPG co-fumigation is the highest at each load. With the DME-LPG co-fumigation, the percentage of coolant exergy in the input exergy varies from 0.26% to 0.41% at 1.3-4.4 bar BMEP. In the experiments, cooling water enters the engine jackets, and the outlet cooling water is not recirculated into the engine. Therefore, the outlet cooling water temperature is relatively low, which contributes to lowering the values of the coolant exergy.

6.2.4 Exergy Destruction

The results of exergy destruction are shown in Fig. 6.7. At below half-load operations of 1.3-2.6 bar BMEP, the exergy destructions for the LPG-diesel dual-fuel mode are the highest. However, at 4.4 bar BMEP, the exergy destruction for the LPG-diesel dual-fuel mode is comparable to that of the diesel mode. The results indicate that LPG utilisation significantly improves above half-load operations. The exergy destruction needs to be reduced for the LPG fumigation at the low-load conditions. The exergy destructions with the LPG fumigation at the low-load conditions are significantly reduced by the co-fumigation of LPG with DME.

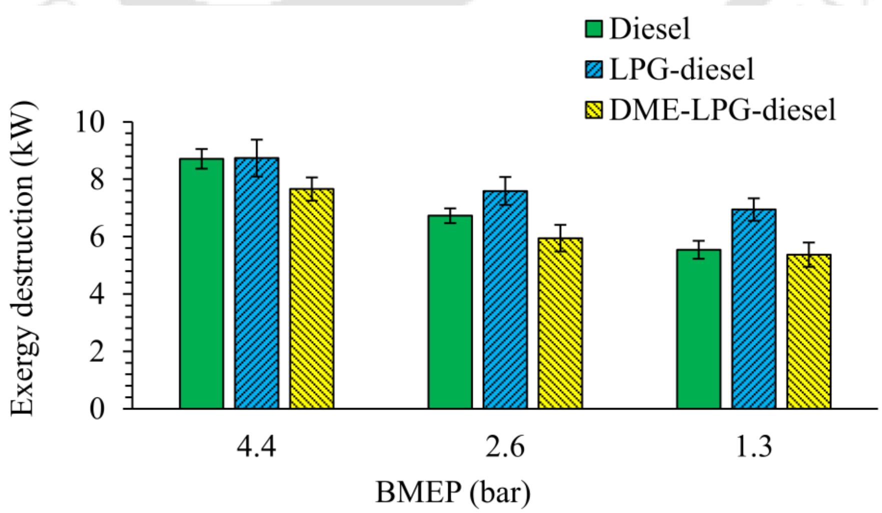


Fig. 6.7 Exergy destruction for diesel and dual-fuel modes.

As shown in Fig. 6.7, the DME-LPG co-fumigation results in the lowest magnitudes of exergy destruction at all the tested loads. At 1.3-4.4 bar BMEP, the exergy destruction of the DME-LPG-diesel dual-fuel mode is lower than those of the diesel mode by 3-12%. The results show that adding the high-cetane oxygenate as a co-fumigant with the LPG effectively lowers exergy destruction. High oxygen content and high cetane number are desirable properties that enhance combustion in compression ignition engines. Sarkar and Saha [2020] also report that adding oxygenate fuel, having a high cetane number, to the pilot fuel of a dual-fuel engine lowered the exergy destruction.

6.2.5 Exergy Efficiency

The exergy efficiency results are shown in Fig. 6.8. The exergy efficiency estimated in this study is the percentage of input exergy converted to shaft work [Verma et al., 2018b; Rakopoulos and Kyritsis, 2001]. The results show that the exergetic performance of the LPG-diesel dual-fuel mode improves with an increase in engine load. The operating points below half-load show poor exergetic performance. The exergy efficiencies of the LPG-diesel dual-fuel mode are lower than those of the diesel mode by 8.39% and 17.19% at the BMEPs of 2.6 bar and 1.3 bar, respectively. However, at 4.4 bar BMEP, corresponding to 70% engine load, the LPG-diesel dual-fuel mode exergetic performance is the same as that of the diesel mode. Therefore, it can be derived from the analysis that the exergetic performance of the LPG-diesel dual-fuel mode at below half-load operations needs improvement.

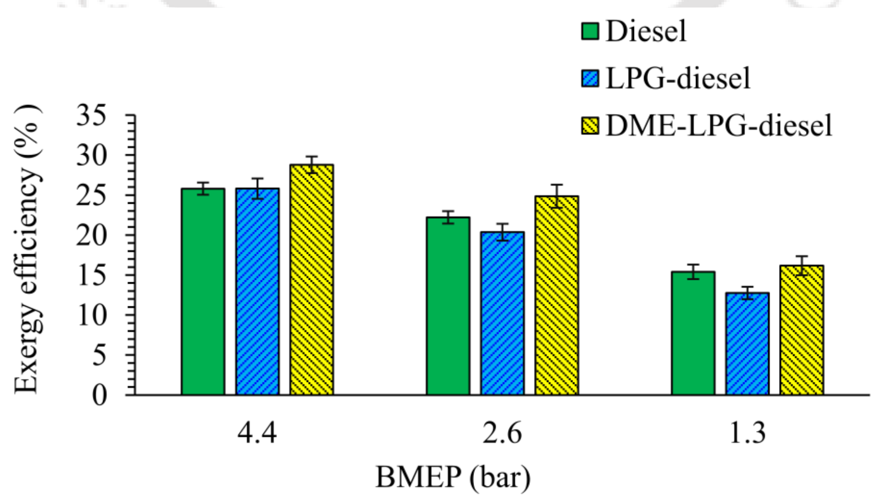


Fig. 6.8 Exergy efficiency for diesel and dual-fuel modes.

Further analysis with the DME-LPG co-fumigation yields good results in terms of the exergetic performance at all tested loads. The DME-LPG co-fumigation not only overcomes the low exergetic performance associated with the LPG-diesel dual-fuel mode at the low-load conditions but also outperforms the diesel mode. The DME-LPG co-fumigation results in exergy efficiencies which are 4.90-11.80% higher than those of the diesel mode. The corresponding exergy efficiency improvement of the DME-LPG co-fumigation with reference to the LPG-diesel dual-fuel mode is even better. At 2.6 bar BMEP, the DME-LPG co-fumigation results in exergy efficiency which is 22.05% higher as compared with that of the LPG-diesel dual-fuel mode. At 1.3 bar BMEP, the exergy efficiency of the DME-LPG co-fumigated dual-fuel engine is 26.68% higher than that of the LPG-diesel dual-fuel mode. The reason for the superior exergetic performance with the DME-LPG co-fumigation is attributed to the excellent properties of DME in terms of high cetane number, high oxygen content and the absence of carbon-carbon bond in the fuel molecule. In the previous chapter, the DME-LPG co-fumigation has been shown to exhibit improved combustion characteristics as compared to the LPG-diesel dual-fuel mode and the baseline diesel mode. Improvement in dual-fuel engine exergy efficiency is reported with utilisation of alternative fuels having high cetane number and high oxygen content [Sarkar and Saha, 2020; Rakopoulos and Kyritsis, 2001]. The DME-LPG co-fumigation has the highest exergy efficiency and the lowest exergy destruction at all tested loads.

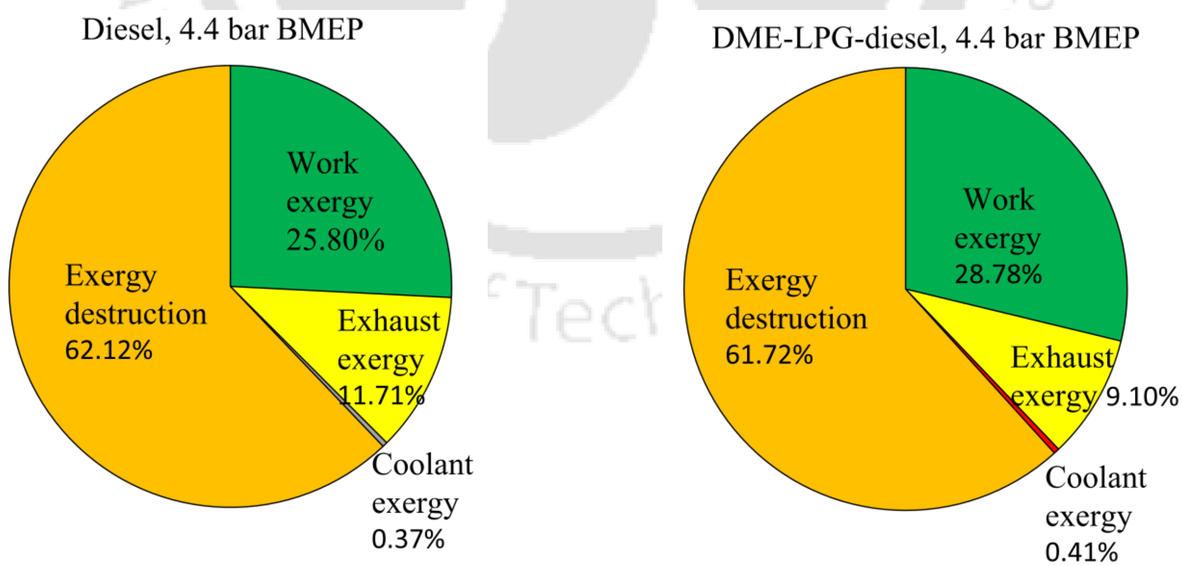


Fig. 6.9 Exergy distribution as percentage of input exergy for diesel and dual-fuel modes.

The exergy distributions, as percentages of the input exergies, for the DME-LPG-diesel dual-fuel mode and the diesel mode, at 4.4 bar BMEP, are shown in Fig. 6.9. The major differences between the comparison are in terms of the percentage of input exergy converted into the work exergy and the exhaust gas exergy. With the DME-LPG co-fumigation, the percentage of input exergy converted into work exergy is 28.78% as compared to 25.80% for the diesel mode. The percentage of input exergy converted to exhaust exergy with the DME-LPG co-fumigation is 9.10% as compared to 11.71% for the diesel mode.

6.3 Summary

Exergy analysis is carried out for the dual-fuel mode and the diesel mode. The exergetic performance and exergy distribution of the dual-fuel mode with LPG fumigation and DME-LPG co-fumigation are compared with those of the diesel mode. The following conclusions are made from this study:

- The exergetic performance of the LPG-diesel dual-fuel mode improves with the increase in engine load. At below half-load engine operations, the exergy efficiencies of the LPG-diesel dual-fuel mode are lower than those of the baseline diesel mode by 8.39-17.19%. However, at the higher load of 4.4 bar BMEP, the exergy efficiency of the LPG-diesel dual-fuel mode is the same as that of the diesel mode.
- DME-LPG co-fumigation results in the best exergetic performance amongst all the cases studied. The low exergy efficiencies of LPG-diesel dual-fuel mode at below half-load conditions are improved through the addition of DME as a co-fumigant. The DME-LPG co-fumigation results in exergy efficiencies higher than those of the LPG-diesel mode by 11.50-26.68%. The exergy efficiencies with the DME-LPG co-fumigation are 4.90-11.80% higher than those of the diesel mode.
- The exhaust gas exergies of the dual-fuel mode are lower than those of the diesel mode. The lower exhaust gas exergy observed with the dual-fuel mode is due to low exhaust gas temperature resulting from early CA90.
- The coolant exergy, as a percentage of the input exergy, follows the trend of peak cylinder mean gas temperature. Higher percentage of input exergy is converted to coolant exergy at operating points having higher peak cylinder mean gas temperature.

- Exergy destruction is the lowest for the DME-LPG co-fumigation at all tested loads. The magnitude of exergy destruction for the LPG-diesel dual-fuel mode is comparable with that of the diesel mode at 4.4 bar BMEP, while it is higher than that of the diesel mode at below half-load operations. The major differences in exergy distribution between the DME-LPG co-fumigation and the diesel mode are in terms of the percentage of input exergy converted into the work exergy and the exhaust gas exergy. As compared with the diesel mode and the LPG-diesel dual-fuel mode, the DME-LPG co-fumigation results in higher work exergy and lesser exhaust gas exergy.

Chapter – 7

Conclusions and Future Scope

Overview: *Alternative gaseous fuels and alcohols can partially replace diesel in compression ignition engines through the dual-fuel mode. However, such dual-fuel mode suffers from low thermal efficiency and high emissions of CO and HC, particularly at low loads. Therefore, in the present research work, experimental studies have been carried out to improve thermal efficiency and emissions of a single-cylinder dual-fuel engine, with emphasis on low-load operating conditions. The study is carried out on a category of diesel engine, which is widely used in India for agricultural water pumping applications. In the first part of the research work, an LPG-diesel dual-fuel mode is studied with emphasis on addressing the high levels of CO and HC emissions by using a commercial oxidation catalyst and a customised oxidation catalyst. The outcome of the first part of the research work indicates the necessity for improvement of the dual-fuel engine thermal efficiency at below-half load operations and the scope for further reductions of the emissions. Therefore, further study is carried out with DME-LPG co-fumigation and the oxidation catalysts, and the technique is found to be very promising in terms of improving the engine performance and reducing the exhaust emissions. The engine performance for the different fumigation options studied is further analysed in greater detail through the exergy analysis.*

7.1 LPG Fumigated Dual-fuel Engine

A diesel engine is converted to operate on the LPG-diesel dual-fuel mode. LPG substitution levels of 36-43%, 39-48%, and 51-64% are studied for performance, combustion, and emissions at BMEPs of 1.3 bar (20% load), 2.6 bar (40% load), and 4.4 bar (70% load), respectively. The LPG substitutions of 36%, 48% and 64% are further studied with the oxidation catalysts at 20%, 40% and 70% loads, respectively. The following conclusions are made from this study:

7.1.1 LPG-diesel Dual-fuel Engine Performance and Combustion Characteristics

- The LPG-diesel dual-fuel engine performance is the same as that of diesel at 70% load. The engine performance remains the same when the LPG substitution level is increased from 51%

to 64%, indicating that engine performance is not a limiting factor for LPG substitutions at the 70% load. At below half-load operations, BTEs of LPG-diesel dual-fuel mode are significantly lower than those of the diesel mode. There is a need to improve the BTE of the LPG-diesel mode at the lower loads.

- CA10 and NHRR peak are observed to occur significantly late in the dual-fuel mode when compared with the diesel mode. Further, peak cylinder pressure and NHRR peaks are significantly lower for the dual-fuel mode. The higher specific heat of the LPG-air mixture, which implies a lower charge temperature, contributes to the delay in CA10.
- The improved dual-fuel engine performance at the higher engine load of 70% is associated with improved combustion characteristics. At this load, CA90 is significantly advanced by 10 ^{o}CA with reference to the baseline diesel mode. Further, the peak cylinder mean gas temperature after TDC is higher than that of diesel, and then the cylinder mean gas temperature drops more steeply as compared to that of the diesel mode.

7.1.2 LPG-diesel Dual-fuel Engine Emissions without Oxidation Catalysts

- CO and HC emissions are exceedingly high for the LPG-diesel dual-fuel engine at below half-load operations. With 36% LPG substitution at 20% load, CO and HC emissions increase by 208% and 173%, respectively. Further increase of the LPG substitution by a mere 7% leads to a significant increase in the CO and HC emissions. However, at 40% load, a similar increase in LPG substitution from 39% to 48% produces only a marginal change in HC.
- The dual-fuel mode CO and HC emissions at 70% load are considerably lower than those at the lower loads. At the 70% load, CO is marginally lesser, and HC is moderately higher when compared with the diesel mode.
- NOx emissions decrease with an increase in LPG fumigation at all tested loads. At the lower loads, the NOx reduction with the LPG fumigation is more prominent. At the higher load of 70%, NOx reduction with LPG fumigation is only 4-22% as compared to the diesel mode. At 20% and 40% load, the LPG fumigation results in reductions of NOx emissions by up to 67% and 61%, respectively.
- Smoke opacity decreases with LPG fumigation. The highest smoke opacity reduction of 88% is observed with 64% LPG substitution at 70% load. At 40% load, 48% LPG substitution

results in 50% smoke opacity reduction. The smoke opacity reduction at 20% load is 31% for 36% LPG substitution.

7.1.3 LPG-diesel Dual-fuel Engine Emissions with Oxidation Catalysts

- Both oxidation catalysts are equally effective in oxidising CO emissions at all tested loads. The CO conversion efficiencies of the OCCs are in the range of 94-97%, and the resulting CO emissions are 89-94% lower than those of the diesel mode.
- The HC conversion efficiency of the commercial OCC is unsatisfactory at the lower loads. At 20-40% load, the HC conversion efficiency of the commercial OCC is limited to about 60%. The low HC conversion is associated with the low exhaust gas temperature at the lower loads. As the load is increased to 70%, the HC conversion efficiency is improved to 77%.
- The customised oxidation catalyst effectively controls the dual-fuel HC emissions at all the tested loads. The dual-fuel HC conversion efficiencies achieved are 77-85% at the lower loads. With these levels of HC conversions, the dual-fuel HC emissions are lower than those of the diesel by 38-67% at 20-40% load. The dual-fuel HC conversion efficiency reached 92% at 70% load, and the resulting HC emissions level is 87% lower than that of diesel. The results show that the customised OCC with the PGM loading level of 1.766 kg/m^3 performs significantly better than the commercial OCC.
- The use of the customised OCC results in the dual-fuel mode HC emissions lower than those of the diesel mode at all tested loads. However, it is observed that the specific HC emissions with the customised OCC at 20% load are still comparatively higher than those at 40-70% loads.
- With the combined effect of LPG fumigation and the OCCs, the smoke opacities are 92-96% lower than those of the diesel mode.

7.2 DME-LPG Co-fumigated Dual-fuel Engine

The LPG-diesel dual-fuel mode suffers from low BTE at the low-load operating conditions. There is also a need for further reduction of the dual-fuel HC emissions, especially at the lowest load. Therefore, further study is carried out by using DME, a high-cetane oxygenate, as a co-fumigant with LPG. The study is carried out at BMEPs of 1.3 bar (20% load), 2.6 bar (40% load), and 4.4 bar (70% load). The following conclusions are made from this study:

7.2.1 DME-LPG-diesel Dual-fuel Engine Performance and Combustion Characteristics

- The DME-LPG co-fumigated dual-fuel engine outperforms the diesel mode. The dual-fuel mode partially replaces the conventional diesel fuel while at the same time achieving higher thermal efficiencies. BTE increases with the increase in the DME substitution levels. At 4.4 bar BMEP, BTEs of the dual-fuel mode are 11.59-13.68% higher than that of the diesel mode. At 2.6 bar BMEP, the dual-fuel mode achieves 14.29% higher BTE as compared with the diesel mode. Even at the lowest load of 1.3 bar BMEP, the DME-LPG co-fumigation achieves a marginal increase of BTE as compared with the diesel mode.
- The low BTEs associated with the LPG-diesel dual-fuel engine at low loads are effectively addressed by the addition of DME as the co-fumigant. The BTEs of the DME-LPG-diesel dual-fuel mode are much higher than those of the LPG-diesel dual-fuel mode, especially at below half-load operations. At 20% load, co-fumigation of 25% DME and 11% LPG results in 23.58% higher BTE as compared to the LPG-diesel dual-fuel mode with 36% LPG fumigation. At 40% load, 32% DME and 14% LPG co-fumigation results in 24.09% higher BTE as compared to 48% LPG fumigation. At 70% load, 12% DME and 52% LPG co-fumigation results in 11.51% higher BTE as compared to the LPG-diesel mode with 64% LPG fumigation.
- The improvement in engine performance with the DME-LPG co-fumigation is due to improved combustion characteristics. DME-LPG co-fumigation results in better combustion phasing at all the tested loads. It is observed that improvements in ITE and RCE are strongly related to CA50 and CA90 taken together. When both CA50 and CA90 advance towards TDC, RCE and ITE show improvement. In such cases, a higher cylinder mean gas temperature after TDC is observed, and subsequently, this temperature drops more rapidly, indicating a rapid rate of combustion.
- The combustion durations of the DME-LPG co-fumigated dual-fuel engine are shorter than those of the diesel mode. At the higher load of 4.4 bar BMEP, the dual-fuel mode combustion duration is shorter than that of diesel by up to 16 ^{o}CA. At this load, the CA10 for both modes

of combustion are comparable and, therefore, the main reason for the short combustion duration of the dual-fuel mode is the advanced CA90. At the lower loads, the dual-fuel mode combustion duration is shorter than that of diesel by 3-6 ^{o}CA.

- The DME-LPG co-fumigation results in significant advancements of CA50, as compared with the diesel mode. At 4.4 bar BMEP, as the DME energy share in the total fuel is increased from 12% to 20%, the advancement in CA50, with reference to the diesel mode, are 3 ^{o}CA and 8 ^{o}CA, respectively. At the lower BMEPs, the DME-LPG co-fumigation results in the advancement of CA50 by up to 3 ^{o}CA.
- The DME-LPG co-fumigation results in significant advancement of CA90, as compared with the diesel mode. At 4.4 bar BMEP, CA90 advances by up to 15 ^{o}CA with reference to the diesel mode. At the lower BMEPs, CA90 advances by 7 ^{o}CA.
- CA10 and CA50 of the DME-LPG co-fumigated dual-fuel engine are significantly advanced as compared with the LPG-diesel dual-fuel mode. At the lower loads of 1.3 bar BMEP and 2.6 bar BMEP, the DME-LPG co-fumigated dual-fuel engine advances CA10 by up to 6 ^{o}CA and 9 ^{o}CA, respectively, as compared with the LPG-diesel dual-fuel mode. Correspondingly, CA50 also advances by up to 6 ^{o}CA and 9 ^{o}CA with reference to the LPG-diesel dual-fuel mode.
- An increase in DME energy share results in the advancement of the NHRR peak towards TDC. Except for the case with 20% DME energy share at 4.4 bar BMEP, all other cases of the DME-LPG co-fumigation exhibit lower NHRR peaks as compared with those of the diesel mode.
- DME being a high cetane fuel, the NHRR profiles of the DME-LPG-diesel dual-fuel mode exhibit LTR and HTR. At the lower loads, the LTR and HTR become more prominent due to the high ratio of DME to LPG (about 70% DME energy in total gaseous fuels energy).

7.2.2 DME-LPG-diesel Dual-fuel Engine Emissions

- The HC emissions of the DME-LPG co-fumigated dual-fuel engine are not significantly different from those of the diesel mode. This is a significant observation as the HC emissions of the LPG-diesel dual-fuel mode increases by as much as 124-240% at the lower loads.

- At the BMEPs of 1.3 bar and 4.4 bar, the DME-LPG-diesel dual-fuel mode HC emissions are slightly higher than those of diesel by 9% and 11%, respectively. However, at 2.6 bar BMEP, where the energy share of DME is the highest (32% DME and 14% LPG), the DME-LPG-diesel dual-fuel mode HC emissions are marginally lower than those of the diesel mode by 3%. The lower dual-fuel mode HC emissions, with the addition of DME as the co-fumigant, is due to the improved combustion phasing and higher combustion temperature.
- The dual-fuel mode CO emissions show a decreasing trend at the higher BMEP. However, at the lower BMEPs, the dual-fuel mode CO emissions increase drastically. At 4.4 bar BMEP, the dual-fuel mode CO emissions are 13% lower than those of the diesel mode. At 1.3-2.6 bar BMEPs, the DME-LPG-diesel mode CO emissions are over 4.7 times higher than those of the diesel mode. The lower dual-fuel mode CO at 4.4 bar BMEP is observed with peak cylinder mean gas temperature higher than that of the diesel mode. At the lower BMEPs, the dual-fuel cylinder mean gas temperature values are only marginally higher than those of the diesel mode. Therefore, at the lower loads, the combustion of the DME-LPG-diesel dual-fuel mode possibly requires a higher temperature for higher oxidation of CO.
- The rate of NOx reduction with the DME-LPG co-fumigation reduces with an increase in load. The DME-LPG co-fumigation results in a reduction of NOx by 47-51% at the lower BMEPs. The NOx reduction is relatively less at 4.4 bar BMEP.
- DME-LPG co-fumigation results in a significant reduction of smoke. With the increase in BMEP from 1.3 bar to 4.4 bar, smoke opacity reduces by 43% to 89% as compared with those of the diesel mode. Better combustion, higher hydrogen to carbon ratio and high oxygen content of DME contributed to this reduction.

7.2.3 DME-LPG-diesel Dual-fuel Engine Emissions with Oxidation Catalysts

- The customised OCC achieves significantly higher HC conversions when compared with the commercial OCC. At the lowest BMEP of 1.3 bar, the HC conversion efficiency of the customised OCC is 86%, while that of the commercial OCC is limited to 68%. At 2.6 bar BMEP, the HC conversion efficiencies for the commercial OCC and the customised OCC are 79% and 89%, respectively.

- The HC conversion improves with load and, at 4.4 bar BMEP, the commercial OCC and the customised OCC achieve 76% and 90% HC conversion efficiencies. Overall, the dual-fuel mode HC emissions with the commercial OCC are 1.9-2.4 times higher as compared to those with the customised OCC. The final dual-fuel mode HC emissions achieved with the customised OCC are lower than those of the diesel mode by 85% to 89% at 1.3-4.4 bar BMEP.
- Both OCCs are equally effective in oxidising CO emissions. At the lower BMEPs of 1.3-2.6 bar, CO conversion efficiency of 98% is achieved. At 4.4 bar BMEP, the CO conversion efficiency is 93%. The resulting dual-fuel mode CO emissions are 89-94% lower than those of the diesel mode.
- With the combined effects of the DME-LPG co-fumigation and the customised OCC, the dual-fuel mode smoke opacity values are 78-94% lower than those of the diesel mode.

7.3 Exergy Analysis

The exergy analysis is carried out for the different cases of the dual-fuel mode and the diesel mode. The LPG-diesel dual-fuel mode is studied with 36%, 48%, and 64% LPG substitutions at the BMEPs of 1.3 bar, 2.6 bar, and 4.4 bar, respectively. The DME-LPG co-fumigation studied has the fumigation rates of 11% LPG and 25% DME at 1.3 bar BMEP; 14% LPG and 32% DME at 2.6 bar BMEP; and 52% LPG and 12% DME at 4.4 bar BMEP. The following conclusions are made from this study:

- The LPG-diesel dual-fuel mode suffers from low exergy efficiency at below half-load operations. The LPG-diesel dual-fuel mode exergy efficiencies are 8.39% and 17.19% lower than those of the diesel mode at the BMEPs of 2.6 bar and 1.3 bar, respectively. However, at 4.4 bar BMEP, the exergy efficiency of the LPG-diesel dual-fuel mode improves and is the same as that of the diesel mode. The results indicate that there is a need to improve the exergetic performance of the LPG-diesel dual-fuel mode at the lower loads.
- The exergetic performance of the DME-LPG-diesel dual-fuel mode is the best amongst all the cases studied. The exergy efficiencies with the DME-LPG co-fumigation are 4.90-11.80% higher than those of the diesel mode. The reason for the superior exergetic performance with the DME-LPG co-fumigation is attributed to the properties of DME in terms of high cetane number, high oxygen content and absence of carbon-carbon bond. It is concluded earlier in the

combustion study that the DME-LPG co-fumigation results in improved combustion as compared to diesel and LPG-diesel.

- The challenge of low exergy efficiency of the LPG-diesel dual-fuel mode at the low-load conditions is effectively addressed by the addition of DME as the co-fumigant. The DME-LPG co-fumigation results in exergy efficiencies higher than those of the LPG-diesel mode by 11.50-26.68%.
- The DME-LPG co-fumigation results in the lowest values of exergy destruction, as compared with the diesel mode and the LPG-diesel dual-fuel mode. At 1.3-4.4 bar BMEPs, the exergy destruction of the DME-LPG-diesel dual-fuel mode is lower than those of the diesel mode by 3.11-12.06%. For the LPG-diesel dual-fuel mode, the magnitude of exergy destruction at 4.4 bar BMEP is comparable with that of the diesel mode, while it is higher than that of diesel at below half-load operations.
- The major differences in exergy distribution between the DME-LPG co-fumigation and the diesel mode are in terms of the percentage of input exergy converted into the work exergy and the exhaust gas exergy. The DME-LPG co-fumigation has higher work exergy and lesser exhaust gas exergy.

It can be drawn from the above conclusions that the DME-LPG co-fumigated dual-fuel engine outperforms the LPG-diesel dual-fuel mode and the diesel mode. The low BTEs associated with the LPG-diesel dual-fuel engine at the lower loads are effectively addressed by the DME-LPG co-fumigation. The dual-fuel mode emissions of CO and HC are effectively reduced by the customised oxidation catalyst. The DME-LPG co-fumigated dual-fuel engine, employed with the customised oxidation catalyst, achieves drastically reduced emissions and higher thermal efficiency as compared with the diesel mode.

7.4 Application Potential

7.4.1 LPG-diesel Dual-fuel Engine

The present research work is conclusive in terms of emissions reductions for the LPG-diesel dual-fuel engine with LPG substitutions of 36%, 48%, and 64% at 20%, 40%, and 70% loads, respectively. The customised oxidation catalyst shows clear potential in reducing an exceedingly high level of LPG-diesel dual-fuel mode HC emissions at low exhaust gas temperature condition

(20% load). The customised oxidation catalyst HC conversions at 20-70% load are 77-92%. In the present research work, the customised oxidation catalyst is intended to be evaluated at exceedingly high levels of HC and CO, and therefore, the high emissions levels at 20% load are chosen. In actual applications, the HC conversion efficiency can be further enhanced by options such as (i) reducing the LPG substitution levels, (ii) enhancing the low-load dual-fuel engine performance through diesel injection timing optimisation or the addition of co-fumigant such as DME.

With the customised oxidation catalyst, the LPG-diesel dual-fuel mode CO and smoke are drastically reduced. Therefore, the present research work indicates that customised oxidation catalysts can be used in LPG-diesel dual-fuel engines to achieve low levels of emissions. The research work is relevant for stationary engine applications such as diesel engine pumpsets for agricultural water pumping. However, non-stationary engines which operate on transient cycles could also benefit from such oxidation catalyst, and the thermal inertia associated with transient operation could enhance HC and CO conversions at low load.

7.4.2 DME-LPG-diesel Dual-fuel Engine

The combined approach of DME-LPG co-fumigation and deployment of the customised oxidation catalyst achieves high thermal efficiency and low emissions. The thermal efficiency of this dual-fuel mode is better than that of diesel. The emissions of CO, HC and smoke are drastically reduced as compared to the diesel mode.

DME can be produced from feedstocks such as coal and biomass. DME and LPG are miscible, and they can be handled and stored similarly, with the exceptions that DME compatible materials need to be used. Therefore, DME and LPG have the potential to be used together as fuels. The present research work indicates that DME-LPG co-fumigation is a potential option for dual-fuel combustion where both fuels can be made available.

7.5 Future Scope of Research

7.5.1 Optimisation of DME and LPG Substitution Levels

The DME and LPG substitution levels used in the present research work are limited to two sets at each load. Research work can be carried out to encompass a greater number of DME and LPG substitution levels. Further, the present research work covers engine loads from 20-70%, with a major focus on emissions reductions and BTE improvement at low loads. Research works need to be carried out covering up to the full load operation.

In the present research, as explained in the experimental set-up, DME and LPG are supplied from two different gas streams. The two gas streams converge through the Y-shape tube into a single tube. Downstream of the Y tube, the two gases pass through the single tube and enter the gas-air mixer for fumigation at the intake manifold. However, there can be other configurations to fumigate the DME and LPG. DME and LPG can be blended in liquid form, and the blended mixture can be fumigated at the intake manifold. In such cases, the ratio of DME to LPG in the blend shall remain fixed at all engine loads. Therefore, for such blends, finding the optimised DME/LPG ratio will be important. Further, it will be necessary to study different fumigation levels for such blend at different loads. Research work can be carried out for such studies on DME-LPG blends.

7.5.2 Simulation Studies

Combustion simulation studies on DME-LPG-diesel dual-fuel mode using detailed reaction mechanisms could be carried out. Such studies could significantly contribute towards the optimisation of DME and LPG substitution levels. Further, optimisation studies based on methods such as the artificial neural network could be carried out to optimise the emissions and performance of DME-LPG-diesel dual-fuel engines.

7.5.3 Optimization of Diesel Injection Timing for LPG-diesel Dual-fuel Mode

In the present research work, the engine manufacturer specified standard diesel injection timing is used in the dual-fuel mode. In the event of a non-availability of the gaseous fuel, the engine should preferably operate back on neat diesel mode. Therefore, to account for this fallback option, the standard diesel injection timing is used in the present study. Research work can be carried out on the LPG-diesel dual-fuel mode by optimising the diesel injection timing for a dedicated dual-fuel operation. The effects of oxidation catalysts on emissions of such dedicated dual-fuel mode can also be studied.

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Appendix - A

Engine Performance and Combustion Parameters

The brake power (BP) is given by Eq. A1.

$$BP(kW) = \frac{2 \times \pi \times \omega(rev/min) \times Tr(Nm)}{60{,}000} \tag{A1}$$

where, Tr is the torque and ω is the engine speed. The brake mean effective pressure (BMEP) is given by Eq. A2.

$$BMEP(bar) = \frac{2 \times \pi \times no.of\ revolution\ per\ power\ stroke \times Tr(Nm) \times 10^{-2}}{V_{swept}(dm^3)} \tag{A2}$$

where, V_{swept} is the swept volume. The brake thermal efficiency (BTE) is given by Eq. A3.

$$BTE(\%) = \frac{BP(kW)}{\sum \dot{m}_{fuel}(\frac{kg}{s}) \times LHV_{fuel}(\frac{kJ}{kg})} \times 100 \tag{A3}$$

where, $\dot{m}_{fuel}$ is the mass flow rate of fuel and LHV_{fuel} is the lower heating value of the fuel. The LPG substitution rate in LPG-diesel dual-fuel mode is given by Eq. A4.

$$LPG\ substituion(\%) = \frac{\dot{m}_{LPG}(\frac{kg}{s}) \times LHV_{LPG}(\frac{kJ}{kg})}{\dot{m}_{diesel}(\frac{kg}{s}) \times LHV_{diesel}(\frac{kJ}{kg}) + \dot{m}_{LPG}(\frac{kg}{s}) \times LHV_{LPG}(\frac{kJ}{kg})} \times 100 \tag{A4}$$

The LPG substitution rate in DME-LPG-diesel dual-fuel mode is given by:

$$LPG\ substituion(\%) = \frac{\dot{m}_{LPG}\left(\frac{kg}{s}\right) \times LHV_{LPG}\left(\frac{kJ}{kg}\right) \times 100}{\dot{m}_{diesel}\left(\frac{kg}{s}\right) \times LHV_{diesel}\left(\frac{kJ}{kg}\right) + \dot{m}_{LPG}\left(\frac{kg}{s}\right) \times LHV_{LPG}\left(\frac{kJ}{kg}\right) + \dot{m}_{DME}\left(\frac{kg}{s}\right) \times LHV_{DME}\left(\frac{kJ}{kg}\right)} \tag{A5}$$

The DME substitution rate in DME-LPG-diesel dual-fuel mode is given by:

$$DME\ substituion(\%) = \frac{\dot{m}_{DME}(\frac{kg}{s}) \times LHV_{DME}(\frac{kJ}{kg}) \times 100}{\dot{m}_{diesel}(\frac{kg}{s}) \times LHV_{diesel}(\frac{kJ}{kg}) + \dot{m}_{LPG}(\frac{kg}{s}) \times LHV_{LPG}(\frac{kJ}{kg}) + \dot{m}_{DME}(\frac{kg}{s}) \times LHV_{DME}(\frac{kJ}{kg})} \tag{A6}$$

The air flow measurement in standard cubic feet per minute (SCFM) is carried out with a Meriam laminar flow element (LFE) system. The air flow calibration equation is a function of pressure drop across the LFE, and is given in Eq. A7. The flow calibration equation along with the correction factors (CFs) is given in Eq. A8.

$$SCFM = A \times \mathrm{h}^2 + B \times \mathrm{h} \tag{A7}$$

$$SCFM = [A \times \mathrm{h}^2 + B \times \mathrm{h}] \times CF_{temperature} \times CF_{pressure} \times CF_{viscosity} \times CF_{relative\ humidity} \tag{A8}$$

In Eq. A7 and A8, *SCFM* is the air flow rate (standard ft^3/min), A and B are calibration constants, *h* is the pressure drop across the LFE (inch water column), and $CF_{temperature}$, $CF_{pressure}$, $CF_{viscosity}$, and $CF_{relative\ humidity}$ are the correction factors for air temperature, pressure, relative humidity, and viscosity, respectively. The CFs values are taken from the LFE manual.

As the flow rate in the above equation (A8) is in SCFM, therefore mass flow rate can be obtained by multiplying the SCFM by the standard air density (ρ_{std}). Further, applying suitable unit conversions, the air flow rate $\dot{m}_{air}$ (kg/h) is given as follows.

$$\dot{m}_{air} = \mathrm{SCFM} \times \rho_{standard} \times 0.0283168 \times 60 \tag{A9}$$

where, $\rho_{standard}$ is the standard air density (kg/m^3).

The averaging of cylinder pressure crank angle data using different number of cycles such as 50 cycles to 250 cycles are reported for combustion analysis [Sharma et al., 2020; Venkatesan and Kadiresh, 2019; Das et al., 2015; Mack et al., 2009]. The averaging of 200 cycles has been used in the present study. The net heat release is calculated using Eq. A10 [Stones, 2012]. Motored heat release is subtracted from the combustion heat release [Truedsson et al., 2012]. Eq. A11 [Brunt and Platts, 1999] is used for the estimation of specific heat ratio (Ɣ).

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} v \frac{dP}{d\theta} \tag{A10}$$

where, $\frac{dQ}{d\theta}$ is the net heat release rate (J/^{o}CA), ϴ is the crank angle (^{o}CA), V and P are cylinder volume (m^3) and pressure (Pa), Ɣ is the specific heat ratio, $\frac{dV}{d\theta} = \frac{(V_{i+1} - V_{i-1})}{(\theta_{i+1} - \theta_{i-1})}$ and $\frac{dP}{d\theta} = \frac{(p_{i+1} - p_{i-1})}{(\theta_{i+1} - \theta_{i-1})}$.

$$\Upsilon = 1.35 - 6.0 \times 10^{-5} \times T + 1.0 \times 10^{-8} \times T^2 \quad \text{(A11)}$$

where, T is the cylinder mean gas temperature (K). The cylinder mean gas temperature at any i^{th} crank angle after intake valve closure (IVC), is estimated from cylinder volume and pressure at IVC, intake air temperature, and cylinder pressure and volume at the i^{th} crank angle using Eq. A12 [Sarkar and Saha, 2018b].

$$T_i = \frac{(P_i \times V_i)}{(P_{IVC} \times V_{IVC})} \times T_{intake\ air} \quad \text{(A12)}$$

CA10 to CA90 is taken as the combustion duration [Walker et al., 2015]. The relative cycle efficiency (RCE), which is defined as the ratio of thermal efficiency of heat added at a given crank angle to that at TDC, is calculated using Eq. A13 [Hsu, 1984; Poonia et al., 1998].

$$RCE_i = \frac{CR^{k-1} - \left(\frac{V_i}{V_{TDC}}\right)^{k-1}}{CR^{k-1} - 1} \quad \text{(A13)}$$

where, CR is the compression ratio, k is the polytropic index, and V_i and V_{TDC} are cylinder volume (m^3) at i^{th} crank angle and TDC, respectively. When q_i amount of heat is added at a given i^{th} crank angle (J/^{o}CA), the overall RCE of all heat added up to n^{th} crank angle ($RCE_{average}$) is expressed as:

$$RCE_{average} = \frac{\sum_{i=1}^{n}(q_i \times RCE_i)}{\sum_{i=1}^{n} q_i} \quad \text{(A14)}$$

The slope of the logarithm plot between pressure and volume is used for estimation of the polytropic index [Prashant et al., 2016, Shivapuji and Dasappa, 2012]. For the estimation of expansion and compression polytropic index, linear least-squares fit for logarithm plot between pressure and volume plot has been used [Thurnheer and Soltic, 2012]. In the present study, polytropic index values become stable at about 10 ^{o}CA after IVC. The compression polytropic index is estimated from -105 ^{o}CA (aTDC) to -20 ^{o}CA aTDC. For determining the crank angle interval for expansion polytropic index, a method used by Brunt and Platts [1999], based on an estimated end of combustion, has been used.

The indicated mean effective pressure is given by equation A15 [Nagashima and Tsuchiya, 2002].

$$IMEP = \frac{1}{V_{swept}} \sum_{j=1}^{n-1} \frac{P_{j+1} + P_j}{2} \left(V_{j+1} - V_j\right) \times 10^{-5} \tag{A15}$$

where, *IMEP* is the indicated mean effective pressure (bar), V_{swept} and V_j are swept volume and j^{th} displacement volume (m^3), P_j is the j^{th} cylinder pressure (Pa), and n = number of pressure samples per cycle. The net indicated power is calculated from the *IMEP*. The indicated thermal efficiency (ITE), also called as the indicated fuel conversion efficiency, is calculated as follows.

$$ITE(\%) = \frac{Net\ indicated\ power(kW)}{\sum \dot{m}_{fuel}(\frac{kg}{s}) \times LHV_{fuel}(\frac{kJ}{kg})} \times 100 \tag{A16}$$

Appendix - B

Exergy Analysis

The specific chemical exergy of diesel fuel is given as follows [Kotas, 1985].

$$Ex_{diesel}(\frac{kJ}{kg}) = LHV_{diesel}(\frac{kJ}{kg}) \times \left[1.0401 + 0.1728\,\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{s}{c} \times \left(1 - 2.0628\frac{h}{c}\right)\right] \quad \text{(B1)}$$

where, h, c, o, and s are mass fractions of H, C, O, and S respectively.

Considering the mixture of the gaseous fuels as mixture of ideal gases, the molar chemical exergy of the gas mixture (Exo_{gases}^{molar}) is calculated as described in Kotas [1985] and is given as follows.

$$Exo_{gases}^{molar}(\frac{kJ}{kmol}) = \sum_i x_i\, Exo_i^o(\frac{kJ}{kmol}) + \bar{R}(\frac{kJ}{kmolK})T^o(K)\sum_i x_i \ln x_i \quad \text{(B2)}$$

In the above equation, x_i is the mole fraction of i^{th} species of the mixture, $\bar{R}$ is the universal gas constant, Exo_i^o is the molar chemical exergy of the i^{th} species at dead state (P^o, T^o). Temperature correction is applied to the standard molar chemical exergy (Exo_i^{std}) for each species at $T^{std} =$ 298.15 K and P^{std} =1.01325 bar to get Exo_i^o, as given in Eq. B3 [Kotas, 1985]. The values of standard molar chemical exergy (Exo_i^{std}), used in Eq. B3, are taken from literature [Kotas, 1985].

$$Exo_i^o(\frac{kJ}{kmol}) = Exo_i^{std}(\frac{kJ}{kmol}) \times \frac{T^o(K)}{T^{std}(K)} - \Delta H_i^{std}(\frac{kJ}{kmol}) \times \left(\frac{T^{std}(K) - T^o(K)}{T^{std}(K)}\right) \quad \text{(B3)}$$

In the above equation, the values of standard enthalpy of combustion (ΔH_i^{std}) for the constituent species of the mixture are taken from the NIST Standard Reference Database 69: NIST Chemistry WebBook [2020] and Zwolinski and Wolhoit [1972]. The molar chemical exergy of the gas mixture from the Eq. B2 is multiplied by the number of moles per kg of the gas mixture (Y) to get the specific chemical exergy of the gas mixture (Ex_{gases}) as follows.

$$Ex_{gases}(\frac{kJ}{kg}) = Exo_{gases}^{molar}(\frac{kJ}{kmol}) \times Y\left(\frac{kmol}{kg}\right) \quad \text{(B4)}$$

The total input exergy, Ex_{input}, is given as follows.

$$Ex_{input}(kW) = \dot{m}_{diesel}\left(\frac{kg}{s}\right) Ex_{diesel}\left(\frac{kJ}{kg}\right) + \dot{m}_{gases}\left(\frac{kg}{s}\right) Ex_{gases}\left(\frac{kJ}{kg}\right) \quad \text{(B5)}$$

where, $\dot{m}_{diesel}$ and $\dot{m}_{gases}$ are the mass flow rates of the diesel and gaseous fuels (kg/s).

The exhaust gas exergy, $Ex_{exhaust}$, is calculated by considering it as an ideal gas [Sarkar and Saha, 2020; Verma et al., 2018a] and the equation is as follows.

$$Ex_{exhaust}(kW) = \left[\dot{m}_{eg}\left(\frac{kg}{s}\right) \times Cp_{eg}\left(\frac{kJ}{kgK}\right) \times \left(T_{eg}(K) - T^0(K)\right)\right] - \left[\dot{m}_{eg}\left(\frac{kg}{s}\right) \times T^0(K) \times \left\{Cp_{eg}\left(\frac{kJ}{kgK}\right) \times \ln\left(\frac{T_{eg}(K)}{T^0(K)}\right) - R\left(\frac{kJ}{kgK}\right) \times \ln\left(\frac{P_{eg}(kPa)}{P^0(kPa)}\right)\right\}\right] \quad \text{(B6)}$$

where, $\dot{m}_{eg}$ = mass flow rate of exhaust gas, Cp_{eg} = specific heat capacity of exhaust gas, R is the specific gas constant, T_{eg} is the exhaust gas temperature.

The specific heat of the exhaust gas is estimated by using an exhaust gas calorimeter in the exhaust. The exhaust gas specific heat is determined at each load for the diesel mode and the same value has been used for the dual-fuel mode also.

The coolant exergy ($Ex_{coolant}$) is calculated as follows [Sarkar and Saha, 2020; Sahoo et al., 2012].

$$Ex_{coolant}(kW) = \dot{m}_{coolant}\left(\frac{kg}{s}\right) \times Cp_{coolant}\left(\frac{kJ}{kgK}\right) \times \left[\left(T_{co}(K) - T_{ci}(K)\right) - T^0(K) \times \ln\left(\frac{T_{co}(K)}{T_{ci}(K)}\right)\right] \quad \text{(B7)}$$

where, $\dot{m}_{coolant}$ = mass flow rate of coolant, $Cp_{coolant}$ = specific heat capacity of coolant, T_{co} and T_{ci} are the coolant-out and coolant-in temperatures, respectively.

The work exergy (Ex_{work}) is the brake power. The exergy destruction is given by Eq. B8.

$$Ex_{destruction}(kW) = Ex_{input}(kW) - [Ex_{work}(kW) + Ex_{exhaust}(kW) + Ex_{coolant}(kW)] \quad \text{(B8)}$$

The exergy efficiency is calculated as follows [Verma et al., 2018b; Rakopoulos, 2001].

$$Exergy\ efficiency(\%) = \frac{Ex_{work}\,(kW)}{Ex_{input}(kW)} \times 100 \tag{B9}$$

Appendix - C

Instruments Specifications and Experimental Uncertainties

The major specifications of the instruments used in the experiments are given in Table C1. The photographs of gas analysers and smoke opacimeter are given in Fig. C1.

Table C1 Instruments specifications.

Parameter acquired/displayed	Instrument	Range	Accuracy
DME flow rate	DME MFC	0-15 SLPM	± 0.18% full scale (FS)
LPG flow rate	LPG MFC	0-15 SLPM	± 0.18% FS
Diesel flow rate	Diesel fuel meter	0-100 kg/h	± 0.81%
Airflow rate	LFE system	6-100 SCFM	± 0.86%
Engine load	Load cell system	0-100 Nm	± 1.11%
HC NOx CO	Gas analysers	0-10,000 ppmC 0-1000 ppmC 0-4000 ppm NO 0-10 vol.% CO	± 2.5% FS ± 2.5% FS ± 5% Repeatability: 0.01 vol.%
Smoke	Opacimeter	0-100% opacity	Sensitivity: ± 0.1% opacity
In-cylinder pressure	Piezoelectric sensor	0-250 bar	Linearity: ≤ ± 0.3 % Sensitivity: 18.85 pC/bar
Engine speed and crank angle	Crank angle encoder	50-20,000 RPM	Resolution: 0.5 ^{o}CA

The uncertainties are estimated based on the methods described by Moffats [1988]. If R is a result of an experiment calculated from a set of measurements x_1, x_2, ...x_n (independent variables), then the relation between the result and the measurements can be expressed by Eq. C1.

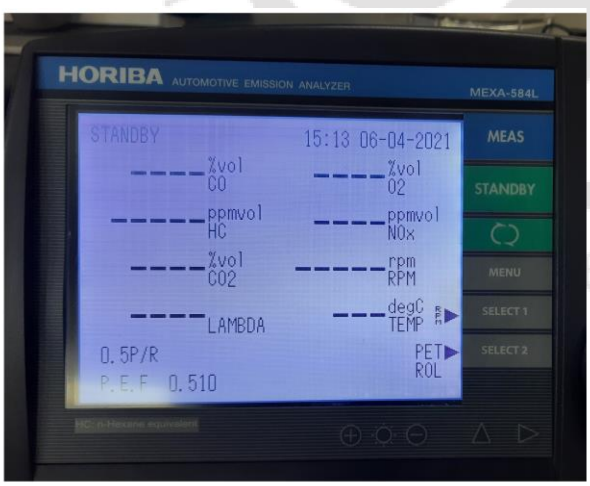
$$R = R(x_1, x_2, x_3, \ldots \ldots . \ldots . x_n) \tag{C1}$$

The uncertainty of the result R denoted by U_r can be calculated from the uncertainties of the independent variables denoted by U_{x1}, U_{x2}, U_{x3}, … U_{xn} by the following relation.

$$U_r = \left[\sum_{i=1}^{n}\left(\frac{\partial R}{\partial xi}U_{xi}\right)^2\right]^{1/2} \tag{C2}$$

The uncertainties are shown as error bars in the plots showing the results of engine performance, emissions, and exergies.

(a) Gas analyser for HC



(b) Gas analyser for CO and NOx

(c) Smoke opacimeter

Fig. C1 Photographs of: (a) gas analyser for HC; (b) gas analyser for CO and NOx; (c) smoke opacimeter.

Publications

Kamei W, Sahoo N, and Prasad VVDN, (2021), Dimethyl ether and liquefied petroleum gas co-fumigation and oxidation catalyst exhaust aftertreatment: a synergy for improvement of thermal efficiency and emissions in a dual-fuel engine, *ASME Journal of Energy Resources Technolo*gy, 143(11), 112301. https://doi.org/10.1115/1.4049601.

Kamei W, Sahoo N, and Prasad VVDN, Investigation of engine performance and combustion and use of oxidation catalysts in an LPG-diesel dual-fuel engine, *Under Review (Journal of Energy Engineering).*

Kamei W, Sahoo N, and Prasad VVDN, Improvement of the exergetic performance of a dual-fuel compression ignition engine by co-fumigation of liquefied petroleum gas and dimethyl ether, *Manuscript Under Preparation for International Journal of Exergy*.