

Study on effects of LPG addition and EGR application on the combustion and emission characteristics in a dual-fuel diesel engine

ARTICLE INFO

The purpose of this study is to analyze the combustion and exhaust emission characteristics of a four-cylinder dual-fuel diesel engine (AJM 1.9 TDI), operating with diesel as the base fuel and LPG as a premixed, port-injected complement. The engine was modified suitably to run in dual-fuel mode with minimal modifications, without altering the original diesel fuel supply system, engine electronic control logic, and operation algorithms, nor any other factory-installed components. Experiments were conducted under two conditions: with and without the Exhaust Gas Recirculation system (EGR-on and EGR-off). Under each EGR condition, multiple speed/load scenarios were evaluated, replacing diesel with LPG, with a share ranging from 0 to approximately 30% of the total fuel energy. Throughout the experimental runs, in-cylinder pressure (p) data and emissions of NO_x , CO , CO_2 , O_2 , and λ were logged. Based on the collected data, the combustion metrics such as gross mean effective pressure (GMEP), mean in-cylinder temperature (T) traces, net heat release rate (NHRR) profiles, mass fraction burn (MFB) profiles, burn rate (BR) dynamics, crank angle timings ($\phi_{\text{MFB}10}$, $\phi_{\text{MFB}50}$, $\phi_{\text{MFB}90}$) at key MFB stages, combustion duration (CD) and crank angle timing ($\phi_{p,\text{max}}$) of the maximum in-cylinder pressure (p_{max}) were computed and evaluated.

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1. Introduction

Global energy trends are gradually shifting towards decarbonization and zero-emission targets, driven by tighter emission regulations, growing renewable energy deployment (Fig. 1), and the rise of electric and hydrogen fuel cell vehicles. Yet, traditional internal combustion engines, including those from older generations, still hold a large share of the global market.

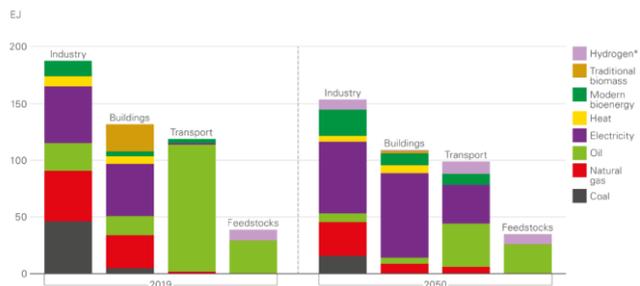


Fig. 1. Energy demand by sector and fuel [3] (* Includes hydrogen-derived fuels)

Diesel engines are widely utilized across various sectors due to their superior thermal efficiency, high power output, durability, and better fuel economy compared to spark-ignition engines. However, a large proportion of internal combustion engines (ICE) in these sectors continue to run on conventional fossil fuels. This leads to serious environmental problems and contributes to the depletion of non-renewable resources, potentially resulting in energy crises and political instability. The transportation sector accounts for approximately 23–28% of global air pollution [1, 4]. Negative consequences of air pollution on human health and the environment have triggered global efforts to reduce conventional ICE's emissions. These concerns, combined

with rising interest in ecological security, have ushered the development of advanced combustion and novel after-treatment technologies. Although these approaches offer effective solutions for reducing harmful compounds in exhaust gases, their implementation is often hindered by economic feasibility and technical complexity, making them less viable for older ICE designs.

Diesel engines are well known for their high output of NO_x and PM pollutants. Combustion in traditional diesel engines, operating in a single-fuel mode (diesel only), initiates with the fuel injection event. It occurs in an environment characterized by a spatially non-uniform distribution of temperature and species concentration, which significantly contributes to NO_x and particulate matter (PM) emissions [24]. The in-cylinder temperature, oxygen concentration, and residence time at high thermal conditions are considered the primary factors contributing to NO_x formation [14, 24]. As highlighted by the author [14, 19], the most commonly used methods to control NO_x emissions include exhaust gas recirculation (EGR), injection timing retardation, staged fuel injection, water injection, catalytic after-treatment, and reduction of premixed burn fraction by reducing ignition delay. EGR is widely regarded as one of the most effective methods currently employed to reduce NO_x emissions in diesel engines. By redirecting a portion of the exhaust gases back into the combustion chamber, the intake charge is diluted, resulting in lower combustion temperatures and, consequently, reduced NO_x formation.

Diesel PM is a complex mixture of solid and liquid phase materials, primarily consisting of carbonaceous material known as soot, a variety of HC's, sulfur compounds and other species [17]. As reported in studies [11, 19, 24], the key contributors to diesel PM formation in the combustion chamber are poor liquid fuel preparation, insufficient air-fuel mixing, and the occurrence of locally rich fuel

regions. It is evident that employing EGR typically leads to increased PM emissions. The trade-off between performance and NO_x -PM emissions in the diesel engine development is widely recognized as a key technical constraint. Relying solely on EGR, without integrating advanced and costly after-treatment systems, prevents older generations of diesel engines from meeting stringent emission regulations. In this context, utilizing alternative fuels in dual-fuel mode (diesel and gaseous fuel), combined with EGR, can provide an effective, practical, and affordable solution for this type of diesel engines.

The dual-fuel strategy involves introducing a secondary fuel, typically in gaseous form, into the intake manifold, where it mixes with the intake air before entering the combustion chamber. This mixture is ignited inside the cylinder by a pilot injection of diesel at the end of the compression stroke. The conversion of lightweight diesel engines to dual-fuel systems requires limited alterations, making the process straightforward and economically viable. Additionally, these engines retain the capability to operate solely on diesel fuel. The operational concept of the dual-fuel supply strategy has been detailed in the previous study [18]. However, for the sake of clarity and completeness, a brief overview is provided in the present paper. In dual-fuel operation, gas flow is controlled by the dual-fuel electronic control unit (DFECU). In contrast, diesel fuel delivery remains under the control of the engine's stock electronic control unit (ECU). According to the user-defined secondary fuel maps, the DFECU adjusts its flow rate and tunes sensor signals from accelerator pedal position (APP), mass air flow (MAF), and manifold absolute pressure (MAP), and transmits these modified signals to the diesel ECU, which correspondingly alters the diesel fuel quantity.

Utilizing alternative gaseous fuels, such as compressed natural gas (CNG), liquefied natural gas (LNG), and liquefied petroleum gas (LPG), is a promising solution for minimizing toxic compounds in exhaust gases and reducing dependence on conventional liquid petroleum fuels [2]. LPG is widely adopted as an alternative energy source employed in households, industry, and transport worldwide. It is regarded not only as a practical substitute for liquid petroleum fuels but also as an effective solution for decreasing harmful emissions such as NO_x and particulate matter [12]. LPG is derived as a by-product from the refining of crude oil and is mainly composed of propane and butane. LPG's high octane number and ignition temperature, along with its low cetane number, inhibit auto-ignition under compression. However, when operated in dual-fuel mode with diesel injected (pilot fuel), the gas ignites effectively. Engines operating on LPG can achieve a higher energy output per unit of fuel than those using diesel, due to LPG's superior calorific value, which may result in lower overall fuel consumption. Moreover, LPG contains negligible amounts of sulfur. The use of diesel fuel in combination with gaseous fuels helps to extend engine life by reducing sulfur dioxide emissions, a primary contributor to corrosion [12, 16]. The main properties of the fuels are given in Table 1.

Despite extensive research on the combustion and emission characteristics of dual-fuel (LPG/diesel) engines, the majority of studies have concentrated on single-cylinder or

laboratory-scale configurations, making a direct transfer of the findings to passenger car applications challenging. For instance, study [16] analyzed the effects of LPG addition on the combustion characteristics of a single-cylinder, four-stroke, direct injection diesel engine under variable compression ratios. Increasing the LPG flow rate from 5 to 20 L/min gradually reduced peak in-cylinder pressure and heat-release rate compared to diesel-only operation, indicating a slight decrease in combustion intensity. Higher compression ratios increased both peak pressure and combustion temperature for all cases, with the diesel-only case achieving the highest values. Emission analysis showed that LPG addition reduced CO, HC, and NO_x emissions compared to diesel-only operation, although these emissions tended to rise at higher compression ratios.

In study [5], the effects of EGR on RCCI combustion were investigated using a single-cylinder, four-stroke diesel engine operating with diesel blended with biodiesel (from waste cooking oils) and a constant 10% LPG fraction for both EGR and non-EGR conditions. Increasing EGR from 0% to 30% resulted in a reduction of peak in-cylinder pressures and heat-release rates, delayed the start of combustion, and extended fuel-air mixing times due to oxygen dilution and lower temperatures. Emission analysis revealed a substantial decrease in NO_x with higher EGR, while CO and HC emissions increased due to slower combustion and local oxygen depletion. The biodiesel content in the fuel blend further reduced NO_x emissions. However, increased CO and HC, indicating a trade-off between emission reduction and incomplete combustion under RCCI with EGR, where the optimal biodiesel fraction depends on balancing this compromise.

Authors [22] studied a four-cylinder, four-stroke turbo-charged marine diesel engine operating in LPG/diesel dual-fuel mode to determine the maximum diesel substitution before knocking occurred. The engine ran without knock at low and medium loads, while full-load conditions caused knocking due to the low cetane number of LPG and the high compression ratio. With the addition of LPG, PM emissions were reduced, but HC and CO emissions increased due to incomplete combustion. NO_x emissions decrease at low and medium loads as the LPG fraction increases, due to ignition delay causing incomplete combustion and lower combustion temperatures. According to the authors [22], NO_x emissions may increase at full load because the higher equivalence ratio reduces air availability compared to lower loads.

The authors [21] made an effort to investigate a four-stroke, naturally aspirated, single-cylinder diesel engine modified for dual-fuel operation using ULSD (ultra-low sulfur diesel) and LPG. The results showed that higher LPG fractions increased peak in-cylinder pressure and heat-release rates due to enhanced premixed combustion. In contrast, the start of combustion was slightly delayed because of LPG's low cetane number and reduced oxygen availability. With EGR addition, combustion was further retarded, and peak pressure decreased slightly, consistent with the thermal effect of recirculated exhaust gases and charge dilution. In terms of emissions, the addition of LPG led to higher HC and CO levels, while soot and NO_x for-

mation were reduced. The application of EGR further lowered NO_x and soot by reducing in-cylinder temperatures, although it slightly increased unburned HC and CO, especially at low engine loads.

Consequently, this study investigates the effects of LPG addition, up to 30% of total fuel energy, and EGR application on the combustion and emission characteristics of a modified Volkswagen Golf IV AJM 1.9 TDI engine. Key parameters including gross: mean effective pressure, in-cylinder pressure, temperature traces, heat release rate, mass fraction burned, burn rate (MFB), crank angle timings at key MFB stages, combustion duration, and peak pressure, as well as NO_x , CO, and CO_2 emissions, were analyzed to provide insights that can support the optimization of older generation diesel engines operating in dual-fuel mode, aiming for improved efficiency and reduced emissions.

Table 1. Fuel properties

Parameter	Unit	Diesel fuel	Butane	Propane
Molecular weight	[g/mol]	170	58.13	44.11
Liquid density at 20°C	[kg/m ³]	800–840	610	501
Liquid viscosity at 25°C	[cSt]	2–4	0.2	0.2
Vapor pressure at 25°C	[bar]	–	8.4	2.1
Boiling temperature	[°C]	125–400	–0.5	–42.1
Self-ignition temperature	[°C]	254	365	470
Cetane number	[–]	40–55	10	5
Lower heating value	[MJ/kg]	36	45.74	46.36
Stoichiometric A/F	[kg/kg]	14,6	14,8	15,7
Latent heat of evaporation	[kJ/kg]	250	390	426
Content of carbon, oxygen and hydrogen	[% by mass]	86/0/14	82.8/0/17.2	75/0/25
Sulfur content	[ppm]	~250	0.01	0.01

2. Experimental setup

The experimental setup consisted of a turbocharged, water-cooled, four-cylinder, four-stroke, direct-injection diesel engine (AJM 1.9 TDI), manufactured by Volkswagen, and equipped with a pump–nozzle injection system. The engine featured a high-pressure exhaust gas recirculation (EGR) system, which directed a portion of the exhaust gases through a cooler before reintroducing them into the intake manifold to reduce nitrogen oxides (NO_x) emissions. The engine was coupled with a Schenck D450-1 hydraulic dynamometer, with the stock electronic control unit (ECU) connected to the dynamometer control unit for precise management of engine speed and torque through dedicated control software. The accelerator pedal signal was electronically emulated to adjust engine speed, while variations in engine load were obtained by applying corresponding torque settings on the dynamometer. The configuration of the test bench is illustrated in Fig. 2, while the engine's technical specifications are provided in Table 2.

The engine was modified for dual-fuel operation by integrating additional ports into the intake manifold for gase-

ous LPG injection, implemented using a standard "auto gas" conversion kit. This kit consisted of an LPG fuel tank with auxiliary parts, a gas filter, fuel supply lines, gas injectors, a gas reducer, sensors, signal emulators, and a dedicated DFECU. As part of the instrumentation setup, the stock air filter was replaced with a laminar flow meter to ensure more accurate airflow measurements. Additionally, to mitigate the impact of intake air temperature variability, the intercooler fan, located downstream of the turbocharger, was replaced with a water-based cooling system. Note that the dual-fuel conversion was implemented without modifying the diesel injection timings, injection patterns, or the engine control logic of the stock ECU. The original diesel system and all factory-installed components remained unmodified, ensuring stable operation in standard DIC mode.

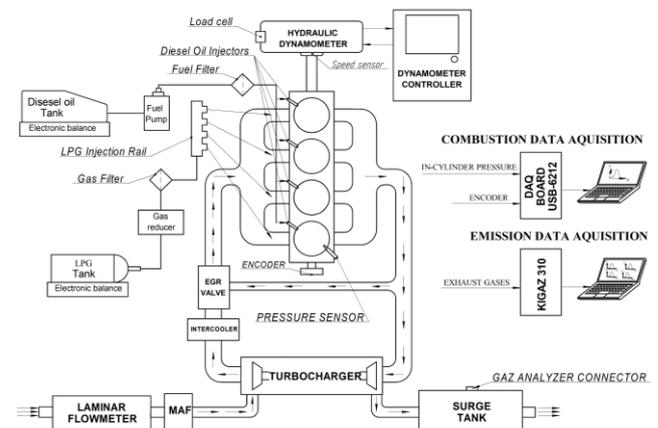


Fig. 2. The schematic diagram of the test bench

Table 2. Engine specifications [9]

Engine parameter	Unit	Specification
Model	[–]	AJM 1.9 TDI
Manufacturer	[–]	Volkswagen
General Details	[–]	Four stroke, compression ignition, water-cooled, direct injection, turbocharger, intercooler
Displacement	[cm ³]	1896
Diesel fuel injector type	[–]	Pump and nozzle unit
Gas fuel injector type	[–]	4-cylinder LPG/CNG injection rail (BARRACUDA)
No. of cylinders	[–]	4
No. of valves per cylinder	[–]	2
Bore	[mm]	79.5
Stroke	[mm]	95.5
Compression ratio	[–]	18:1
Maximum speed	[rpm]	4000
Rated power	[kW]	85
Maximum torque at 1900 rpm	[Nm]	285

In this study, combustion and emission parameters such as in-cylinder pressure and exhaust gases (NO_x , CO, CO_2 , O_2 , and λ) were measured using a computerized setup. The in-cylinder pressure was measured with the AutoPSI-S fiber-optic sensor from Optrand Inc., while crank angle data were obtained via a CKQH-58 sensor manufactured by LIKA. Data acquisition was performed using the USB-6212 module (National Instruments), with sampling rates set

according to the Nyquist–Shannon sampling theorem to ensure accurate signal reconstruction (Table 3). Fuel consumption was monitored with an AWO electronic balance, while exhaust emissions were measured and logged with a KIGAZ-310 gas analyzer from KIMO.

3. Test scenarios

To assess the effects of LPG addition and EGR application on combustion and emission characteristics, two steady-state operating points (OP) from the dataset described in the previous study [9] were selected. These operating points correspond to the engine’s behavior during the Worldwide Harmonized Light Vehicles Test Procedure (WLTP), specifically within the WLTC 3b cycle, for an unmodified Volkswagen Golf IV equipped with the AJM 1.9 TDI engine. The WLTP is divided into four distinct driving stages: low (urban), medium (suburban), high (extra-urban), and extra-high (highway). Each stage is defined by specific vehicle speed versus time profiles. To replicate these real-world driving conditions on a test bench, the vehicle speed profiles were converted into corresponding engine speed and torque profiles. This conversion was performed using computer simulations based on a simplified longitudinal dynamics model, combined with WLTP gear-shifting logic and a density-based grid clustering approach to identify steady-state operating points that accurately reflect real driving scenarios.

It should be noted that the experiments on the test engine were conducted in the following manner: initial tests were carried out in single-fuel mode, using diesel only, followed by dual-fuel operation with LPG and diesel at the same operating points.

At each OP, experiments were conducted under two conditions: EGR-on and EGR-off. For both cases, four diesel-to-LPG substitution ratios were tested. The experiments are marked OP.1.1 to OP.2.4, where the first digit refers to the operating point and the second to the trial number (see Table 3). It should be noted that during EGR-on test cases, the EGR rate was controlled automatically by the stock ECU. This approach allows the evaluation of engine behavior under dual-fuel conditions with minimal modifications and without any recalibration for LPG usage.

In the dual-fuel setup used in the present study, the quantities of LPG and diesel were inversely adjusted according to engine operating conditions: an increase in LPG was accompanied by a proportional reduction in diesel, and

vice versa, ensuring that the total fuel energy supplied to the engine remained constant.

Substitution ratio (SR) was used as an indicator to quantify the energy share of LPG in the total fuel energy input:

$$SR = \frac{\dot{m}_G \times LHV_G}{\dot{m}_D \times LHV_D + \dot{m}_G \times LHV_G} \times 100\% \quad (1)$$

As previously mentioned, increasing the LPG share in the dual-fuel working concept results in a proportional reduction of diesel injection, which alters the air–fuel ratio. Throughout the experiments, to ensure measurement consistency, the air–fuel ratio (λ) was first measured during diesel-only operation with EGR-on. In dual-fuel mode, λ was maintained at the same level as measured for diesel-only operation for EGR-on. At the same time, for EGR-off, λ was adjusted to match the value used for EGR-on (Fig. 10 and Fig. 11). These adjustments were implemented through modifications in the DFECU control algorithms. It should be noted that, these adjustments were made considering the trade-off between diesel reduction, LPG addition and the limits of λ adjustment via turbocharger operation.

4. Experimental data processing

The in-cylinder combustion dynamics govern the performance and emissions of internal combustion engines. The co-introduction of gaseous fuels with diesel alters combustion dynamics due to differences in the molecular composition, chemical reactivity, and thermophysical properties of the fuels. To evaluate combustion behavior and emission formation, approximately 100 consecutive cycles were sampled and recorded for each experimental trial. These datasets were then averaged and smoothed to minimize noise and reduce sensor uncertainty. The combustion metrics analyzed in this study include (see Fig. 3–Fig. 6, Table 4):

- mean in-cylinder pressure (p) traces
- net heat release rate (NHRR) profiles
- mean in-cylinder temperature (T) traces
- mass fraction burn (MFB) profiles
- burn rate (BR) dynamics
- crank angle timings at key MFB stages (ϕ_{MFB10} , ϕ_{MFB50} , ϕ_{MFB90})
- combustion duration (CD)
- maximum in-cylinder pressure (p_{max}) and its corresponding crank angle ($\phi_{p,max}$)
- gross mean effective pressure (GMEP).

Table 3. Test scenarios

Parameter	Unit	Information							
		1				2			
Operation point	[-]								
Trial	[-]	1	2	3	4	1	2	3	4
Engine speed	[rpm]	1690				2280			
Engine torque	[N·m]	71				145			
Diesel rate EGR-on	[mg/cycle]	19.85	16.96	16.69	15.56	30.98	28.60	25.49	22.85
Diesel rate EGR-off	[mg/cycle]	19.17	17.75	16.65	15.40	31.69	29.49	26.10	22.18
LPG rate EGR-on	[mg/cycle]	0	2.69	2.92	3.96	0	2.20	5.07	7.51
LPG rate EGR-off	[mg/cycle]	0	1.31	2.33	3.49	0	2.04	5.17	8.79
SR EGR-on	[%]	0	14.55	15.92	21.61	0	7.70	17.72	26.25
SR EGR-off	[%]	0	7.40	13.17	19.68	0	6.98	17.67	30.02
Sampling frequency	[kHz]	92				124			

4.1. Computational approach and governing equations

In this study, NHRR was estimated using the Krieger and Borman model [10], derived from the first law of thermodynamics for closed systems and the ideal gas law:

$$\text{NHRR} = \frac{dQ_{\text{net}}}{d\phi} = \frac{\gamma}{\gamma-1} \cdot p \frac{dV}{d\phi} + \frac{1}{\gamma-1} \cdot V \frac{dp}{d\phi} \left[\frac{\text{J}}{\text{deg}} \right] \quad (2)$$

where specific heat ratio (γ), is expressed as follows:

$$\gamma = \frac{c_p}{c_v} \quad (3)$$

Based on the JANAF tables and NASA's polynomial formulation, the specific heat at constant pressure (c_p) is expressed as a function of temperature as follows [15]:

$$c_{p_i}(T) = R \cdot (a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4) \left[\frac{\text{J}}{\text{K-mol}} \right] \quad (4)$$

By rearranging the thermodynamic relation between specific heats and the universal gas constant (R), the expression for specific heat at constant volume (c_v) can be expressed as:

$$c_v = c_p - R \left[\frac{\text{J}}{\text{K-mol}} \right] \quad (5)$$

According to the ideal gas equation, the in-cylinder mean gas temperature (T) was calculated in this manner:

$$T = \frac{p \cdot V}{m \cdot R} [\text{K}] \quad (6)$$

The instantaneous cylinder volume is governed by the crankshaft kinematics and can be expressed by the following general equation:

$$V(\phi) = \frac{V}{\varepsilon_s - 1} + \frac{\pi B^2}{4} \cdot (1 + a - s(\phi)) [\text{m}^3] \quad (7)$$

The relationship between piston displacement (s) and crank angle (ϕ) is described by the following expression:

$$s(\phi) = a \cdot \cos(\phi) + \sqrt{(1 - a \cdot \sin(\phi))^2} [\text{m}] \quad (8)$$

The progression of combustion is commonly characterized by the mass fraction burned (MFB). Assuming that the mass of fuel burned is directly proportional to the energy released during combustion (cumulative heat release), MFB was calculated using the following expression:

$$\text{MFB} = \frac{Q_c(\phi)}{\max(Q_c(\phi))} \quad (9)$$

The cumulative heat release, $Q_c(\phi)$, as a function of crank angle, was calculated by integrating the net heat release rate (Q_{net}) over the combustion phase, from intake valve closing (IVC) to exhaust valve opening (EVO):

$$Q_c(\phi) = \int_{\text{IVC}}^{\text{EVO}} \left(\frac{dQ_{\text{net}}}{d\phi} \right) d\phi [\text{J}] \quad (10)$$

The burn rate (BR) is computed by differentiating the mass fraction burned (MFB) with respect to the crank angle (ϕ), as given by the following expression:

$$\text{BR}(\phi) = \frac{d(\text{MFB})}{d\phi} \quad (11)$$

Combustion duration is more often defined as the crank angle interval between 10% and 90% mass fraction burned

(MFB₁₀ to MFB₉₀). This approach excludes the initial and final combustion stages, where the pressure data traces may be unreliable due to signal noise. Additionally, complete combustion (MFB₁₀₀) is rarely achieved as some fuel remains unburned at the end [23]. Therefore, in this study, combustion duration is calculated as follows:

$$\text{CD}_{\text{deg}} = \phi_{\text{MFB90}} - \phi_{\text{MFB10}} [\text{deg}] \quad (12)$$

Hence, combustion duration in milliseconds was obtained by applying the following equation:

$$\text{CD}_{\text{ms}} = \frac{\phi_{\text{MFB90}} - \phi_{\text{MFB10}}}{n \times 0.006} [\text{ms}] \quad (13)$$

The Gross Mean Effective Pressure (GMEP) represents the average effective pressure acting on the piston during the compression and expansion strokes, excluding pumping losses. It provides a useful metric for comparing the performance of an engine under different operating conditions. Mathematically, GMEP can be expressed as:

$$\text{GMEP} = \frac{\int_{\phi_i=-180\text{CA}}^{\phi_i=180\text{CA}} \frac{(p_1 + p_2)}{2} dV}{V_d} [\text{bar}] \quad (14)$$

5. Combustion: results and discussion

The variation of mean in-cylinder pressure and the net heat release rate as functions of crank angle for each test condition (EGR-on and EGR-off) is illustrated in Fig. 3. It can be observed that, for EGR-off scenarios, for each operating point (OP), an increase in the LPG fraction in the fuel mixture results in slightly higher in-cylinder pressure and a more pronounced heat release rate.

Possible reasons explaining this behavior are:

1. In the dual-fuel combustion mode, LPG in gaseous form is introduced into the intake manifold and mixes with air to create a homogenous, combustible charge. A higher LPG fraction intensifies the premixed combustion phase, resulting in enhanced energy output and elevated in-cylinder pressure [21].
2. LPG is composed primarily of propane and butane, which have higher flame propagation than conventional diesel.
3. It is well known that the introduction of recirculated exhaust gases into the intake manifold leads to charge dilution, thereby reducing the oxygen concentration, which directly affects the upcoming combustion event [7]. Therefore, in EGR-off test cases, an increasing LPG ratio enhances the BR dynamics (Fig. 6a and Fig. 6c), which results in higher mean in-cylinder pressures and temperatures (Fig. 4a and Fig. 4c).

Under EGR-on conditions, a higher proportion of LPG results in reduced mean in-cylinder pressure and temperature compared to diesel-only operation (see Fig. 3 and Fig. 4). This is likely due to a combination of factors that leads to an incomplete and unstable combustion event, such as:

1. Limited oxygen availability from EGR, and the displacement of intake air by LPG, lead to decreased volumetric efficiency and poorer mixing quality of the charge.
2. As it was stated earlier, the author attempted to maintain λ at the same (Fig. 10 and Fig. 11) in both single- and dual-fuel modes through turbocharger system adjust-

- ments. As a result of this approach, the initial cylinder pressure before (BTDC) was altered.
- As gaseous LPG is inducted into the intake manifold, its high latent heat of vaporization absorbs energy, cooling the intake charge and lowering the in-cylinder temperature.
 - Under dual-fuel (LPG/diesel) conditions, the reduction in liquid diesel quantity due to fuel substitution leads to poorer spray atomization and slower droplet development, thereby affecting combustion quality [21]. This effect occurs under both EGR-on and EGR-off conditions. However, this effect is more pronounced during EGR-on operation due to the combined influences of oxygen dilution, charge cooling, and the changes in initial in-cylinder pressure described above.

It should be noted that most of the phenomena described above occur under both EGR-on and EGR-off conditions, although their intensity differs: effects such as oxygen dilution, intake charge cooling, and poorer spray atomization are more pronounced under EGR-on operation, while enhanced premixed combustion and higher burn rates due to increased LPG fractions are more evident under EGR-off conditions.

As illustrated in Fig. 3b (OP.1.4) and Fig. 3c (OP.2.4), the net heat release (NHRR) profiles exhibit a more complex shape. This behavior was observed at high proportions of inducted LPG in the intake charge (approximately 21.61% for OP.1.4-EGR-on and 30.02% for OP.2.4-EGR-off, see Table 3), which were associated with combustion instabilities such as knocking and misfiring. These can be

attributed to a combination of factors, including the physicochemical properties of the fuels, the quality of fuel-air mixture formation, the influence of EGR, and overall combustion dynamics.

Figure 4 shows that the calculated mean temperature follows a trend similar to that of the in-cylinder pressure. Under EGR-off conditions, an increase in LPG fraction leads to higher mean temperatures for both operating points, with OP.1 (Fig. 4a) rising from 1580 K (Trial 1) to 1700 K (Trial 4) and OP.2 (Fig. 4c) increasing from 1680 K (Trial 1) to 1915 K (Trial 4). In contrast, under EGR-on conditions, the mean temperature decreases with the increase LPG content. For OP.1 (Fig. 4b), it drops from 1580 K (Trial 1) to 1420 K (Trial 4), while for OP.2 (Fig. 4d), it fluctuates due to unstable combustion, ranging from 1650 K (Trial 1) to 1695 K (Trial 4), with intermediate values of 1470 K (Trial 2) and 1530 K (Trial 3).

Figure 5 and Figure 6 illustrate the variation in mass fraction burned (MFB) and burn rate (BR) dynamics as functions of crank angle. Variations in combustion behavior can be attributed to previously discussed factors such as fuel properties, poor mixture homogeneity, EGR application, and the lack of optimal engine calibration for better LPG utilization. Compared to DICI mode, increasing the proportion of premixed LPG in dual-fuel mode alters the progression of the burn. It shifts the positions of MFB indicators (see Table 4), thereby modifying both the mass fraction burn profile and burn rate. These changes influence the NHRR, which in turn affects in-cylinder temperature and emission formation.

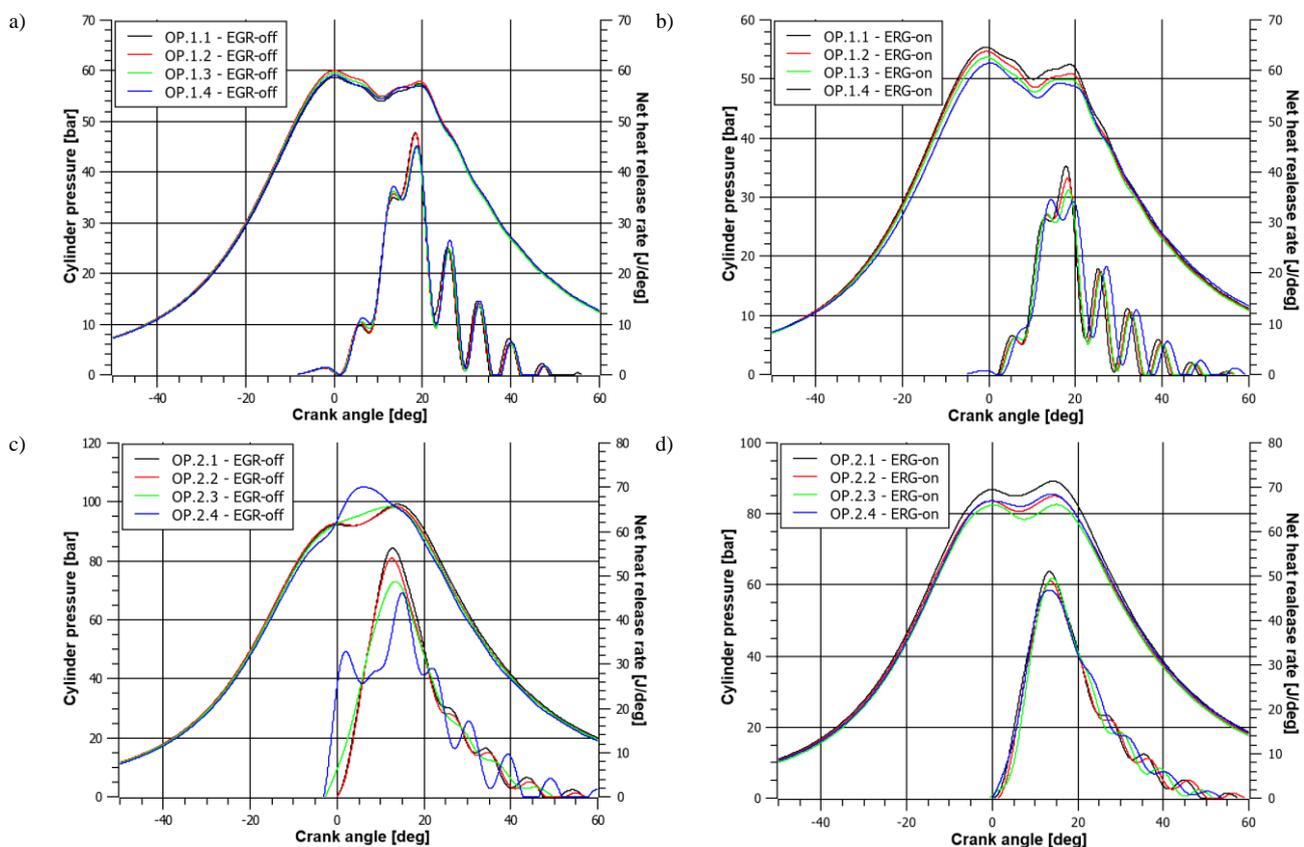


Fig. 3. In-cylinder pressure and NHRR variations

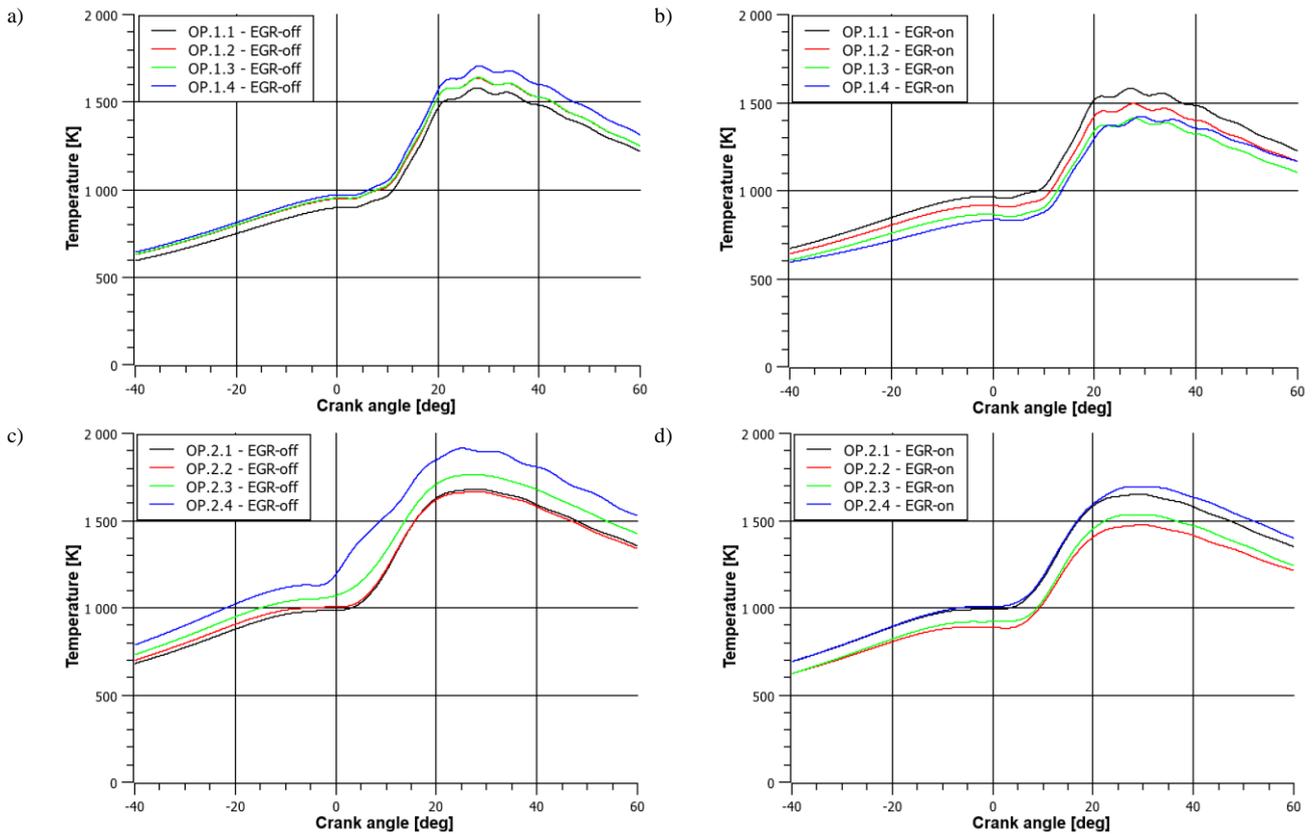


Fig. 4. In-cylinder mean temperature variations

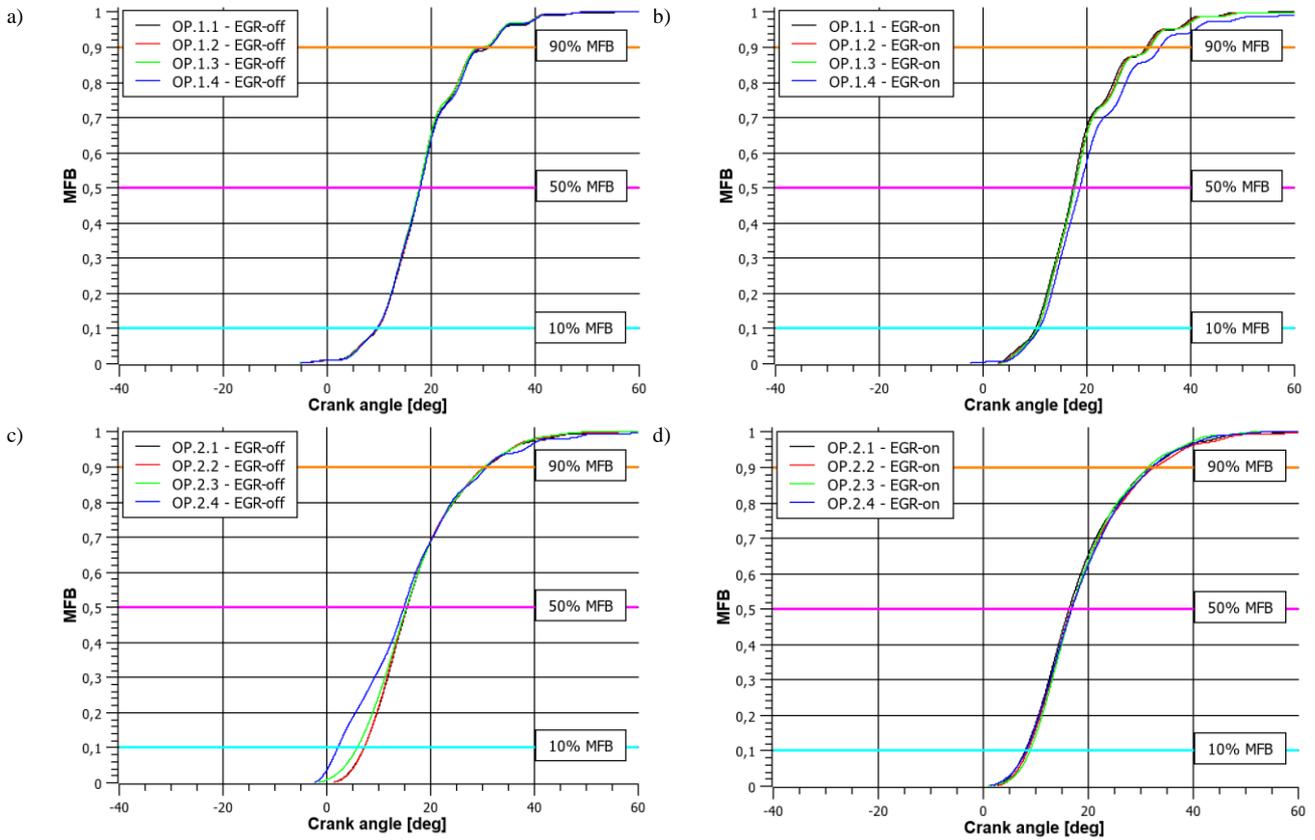


Fig. 5. The mass fraction burn variations

6. Emission: results and discussion

The nitrogen oxides (NO_x) primarily consist of nitric oxide (NO) and a minor amount of nitrogen dioxide (NO₂). These compounds are primarily formed through the high-temperature oxidation of nitrogen in the intake air within the combustion chamber. In particular, NO formation occurs in the post-flame region during the combustion of gaseous fuels [11].

NO_x formation depends on several factors, including in-cylinder temperature and pressure, oxygen content, and the duration of oxidation under high-temperature conditions [16].

Figure 7 indicates that EGR-off test conditions yield higher NO_x emissions as compared to EGR-on. The reintroduction of exhaust gases into the intake charge via the EGR system increases the mixture's specific heat capacity. It dilutes the intake air with inert gases (reducing O₂ availability) and thus negatively affects combustion kinetics [5, 7, 20, 22]. Consequently, in-cylinder pressure and temperature

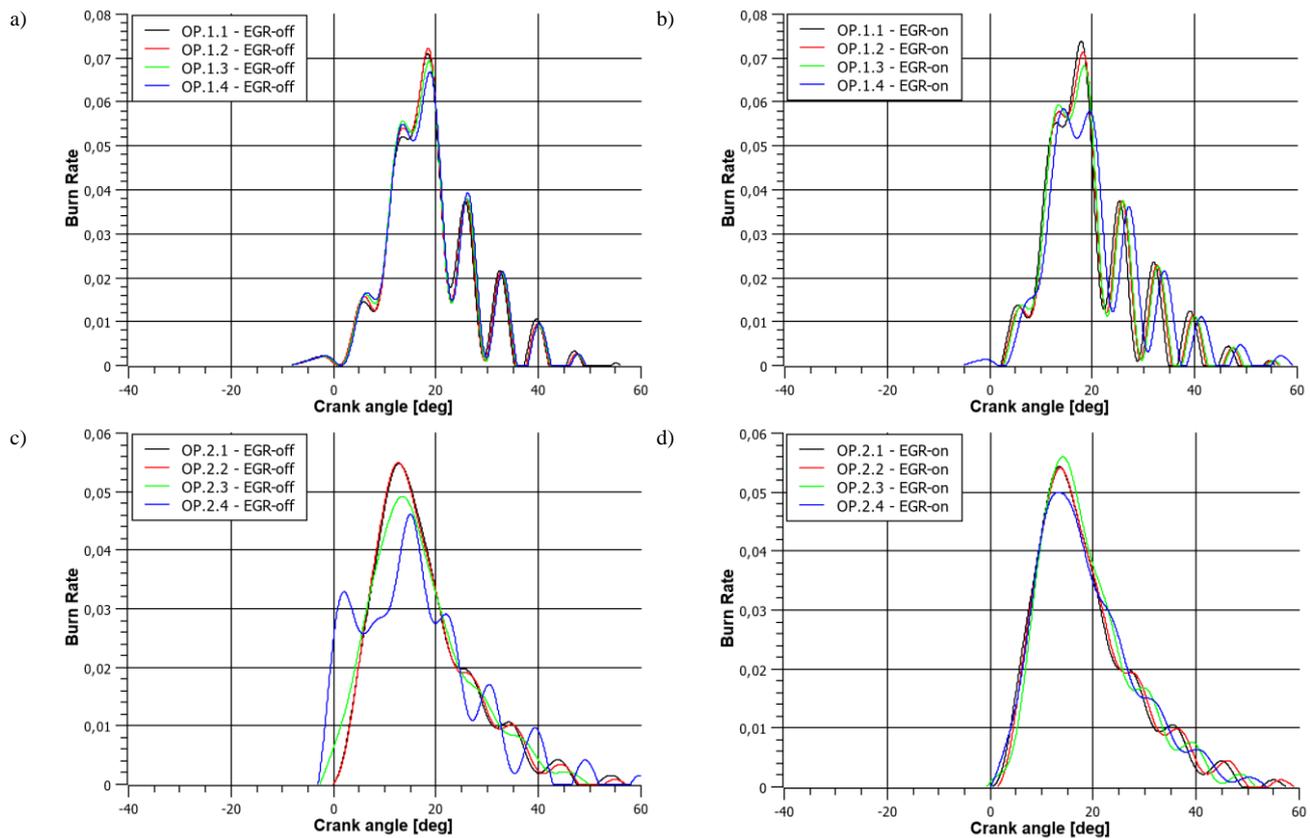


Fig. 6. The burn rate variations

Table 4. Combustion metrics

OP.	Trial number	Test condition	GMEP	p _{max}	φ _{p,max}	φ _{MFB10}	φ _{MFB50}	φ _{MFB90}	CD _{deg}	CD _{ms}
			[bar]	[bar]	[deg]	[deg]	[deg]	[deg]	[deg]	[ms]
1	1	EGR-off	7.01	58.74	0.20	9.72	17.79	31.05	21.33	2.10
	1	EGR-on	5.65	55.39	-0.70	10.06	17.35	31.46	21.40	2.11
	2	EGR-off	6.88	60.06	0.10	9.66	17.71	30.47	20.81	2.05
	2	EGR-on	5.50	54.66	-0.55	10.46	17.63	31.85	21.39	2.11
	3	EGR-off	6.79	59.58	0.20	9.51	17.66	29.22	19.70	1.94
	3	EGR-on	5.42	53.66	-0.50	10.40	17.66	32.11	21.71	2.14
	4	EGR-off	7.07	59.08	0.25	9.52	17.87	30.85	21.34	2.10
	4	EGR-on	5.82	52.70	0.30	10.81	18.66	34.13	23.32	2.30
2	1	EGR-off	11.30	99.21	13.95	7.16	15.46	30.52	23.36	1.71
	1	EGR-on	10.17	89.18	14.25	7.96	16.31	31.78	23.82	1.74
	2	EGR-off	10.77	98.36	13.55	7.17	15.37	30.32	23.16	1.69
	2	EGR-on	9.78	85.02	14.55	8.42	16.71	32.58	24.16	1.77
	3	EGR-off	10.93	98.32	12.10	5.86	15.33	30.38	24.53	1.79
	3	EGR-on	9.85	82.61	15.05	8.88	16.82	31.36	22.48	1.64
	4	EGR-off	10.88	105.03	6.10	2.09	14.84	30.79	28.69	2.10
	4	EGR-on	10.23	85.54	13.80	8.13	16.84	32.12	23.99	1.75

are lowered, and NO_x formation under EGR-on test conditions is reduced. Experimental results confirm this trend (Fig. 7). At Operating Point 1 (OP.1), NO_x emissions decreased from around 340 ppm under EGR-off conditions to approximately 210 ppm with EGR-on. For OP.2, which represents a higher engine speed and load, NO_x levels decline from about 670 ppm (EGR-off) to 510 ppm (EGR-on).

As depicted in Fig. 7, employing LPG in dual-fuel mode further alters NO_x emissions in both EGR-on and EGR-off modes. Based on the findings previously described in this paper, along with insights from the scientific literature, this phenomenon can be explained by the interplay of several factors:

1. In dual-fuel operation mode, increasing the proportion of LPG results in a corresponding decrease in the amount of diesel fed.
2. Due to the gaseous state of LPG, it tends to fill small gaps and crevices inside the cylinder more effectively than liquid diesel.
3. Maintaining a constant level of air–fuel equivalence ratio (λ), along with the factors mentioned in 1 and 2 above, may lead to locally non-uniform or inhomogeneous mixture distribution within the cylinder, which can be considered suboptimal in terms of combustion efficiency and emissions.
4. When LPG is introduced into the intake manifold, its higher latent heat results in greater heat absorption during vaporization, which in turn cools the intake charge. As the LPG proportion increases, this cooling effect becomes more pronounced, resulting in lower in-cylinder temperatures and reduced thermal NO_x formation.
5. LPG, which is primarily a mixture of propane and butane, has a higher laminar flame speed than liquid diesel. These characteristics alter the rate of pressure rise during combustion.
6. In single-fuel mode, the energy is mainly released during the diffusion combustion stage, whereas dual-fuel combustion shifts the energy release toward the pre-mixed combustion stage.
7. The lack of optimization in injection strategy and ECU control algorithms for dual-fuel operation with LPG leads to unstable engine behavior and jerking, thereby negatively affecting mixture formation and combustion quality.

Under EGR-on operation, the addition of premixed LPG further influenced NO_x formation. At Operating Point 1 (OP.1), NO_x concentrations ranged from 210 ppm (Trial 1, no LPG) to 265 ppm (Trial 4, the highest LPG share). In contrast, at OP.2, NO_x initially rose from 510 ppm (Trial 1) to 528 ppm (Trial 2) before decreasing markedly to 382 ppm (Trial 4).

Under EGR-off conditions, increasing the LPG fraction also led to a reduction in NO_x emissions, despite higher in-cylinder pressures and temperatures. At OP.1, NO_x decreased from 340 ppm (Trial 1) to 290 ppm (Trial 4), while at OP.2, it declined from 670 ppm (Trial 1) to 436 ppm (Trial 4).

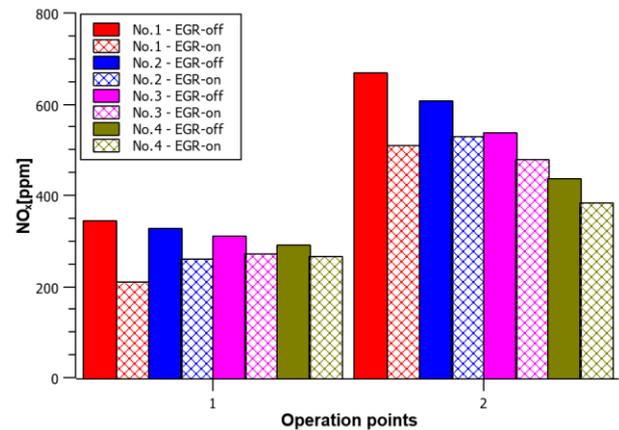


Fig. 7. Variation in nitrogen oxide (NO_x) concentrations corresponding to a specific engine operation point

Carbon monoxide (CO) is a hazardous air pollutant that contributes to environmental degradation and poses significant risks to human health. It is produced as a byproduct of the incomplete combustion of hydrocarbon-based fuels. Figure 10 shows the variation of CO concentrations corresponding to different engine operation test points. Dual-fuel mode yields higher CO emissions compared to single-fuel mode. The formation of CO is facilitated by conditions such as limited oxygen availability, non-optimal air–fuel ratios, poor charge mixing quality and the presence of excess exhaust gases in the combustion chamber, all of which lead to poor combustion [6, 8, 13, 22]. Under EGR-off conditions, CO concentrations increased from 280 ppm to 1874 ppm in OP.1, and similarly, from 354 ppm to 1930 ppm in OP. 2. Under EGR-on operation, CO levels rose from 272 ppm to 1787 ppm in OP.1 and from 106 ppm to 1900 ppm in OP.2. These results indicate that substituting diesel with gaseous LPG promotes incomplete CO oxidation. According to the findings in [6, 8, 13], and other sources, higher CO emissions in exhaust gases can be controlled by adjusting factors such as pilot fuel quantity, injection timing, and EGR rate. Advancing the injection timing improves the oxidation rate of CO, while adjusting the EGR rate enables control over the air–fuel mixture in the intake charge, both of which may contribute to reduced CO emissions.

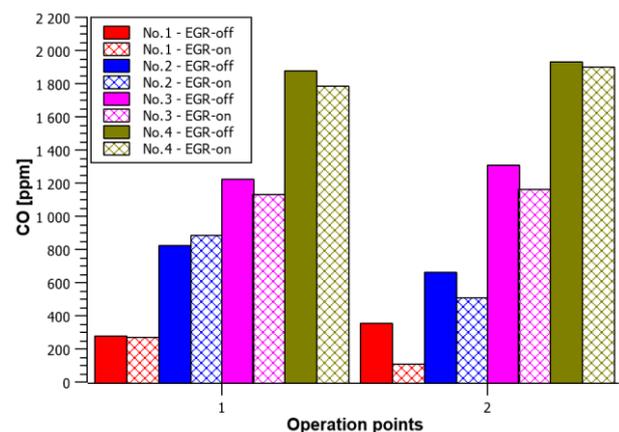


Fig. 8. Variation in carbon monoxide (CO) concentrations corresponding to specific engine operation point

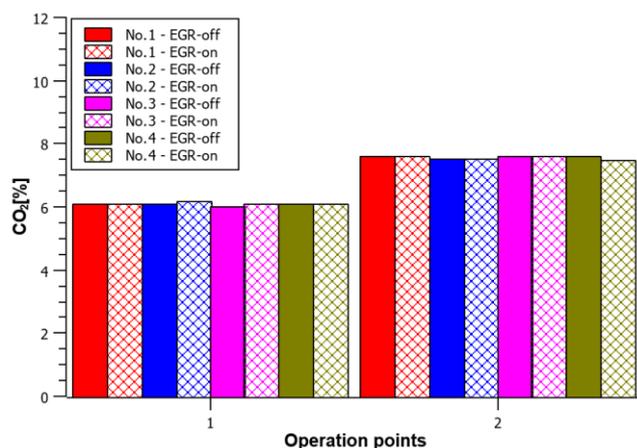


Fig. 9. Variation in carbon dioxide (CO₂) concentrations corresponding to specific engine operation point

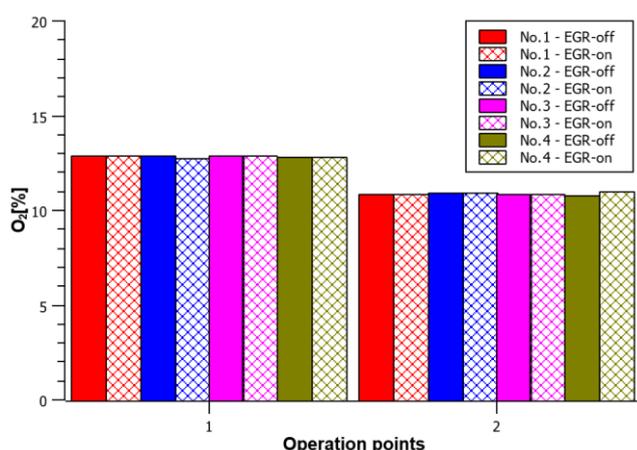


Fig. 10. Variation in oxygen (O₂) content corresponding to specific engine operation point

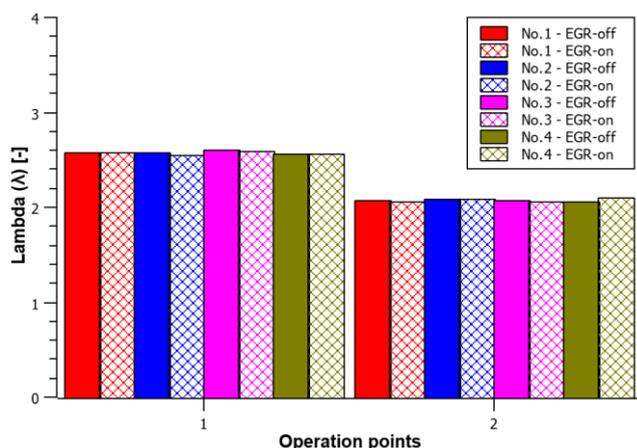


Fig. 11. Variation in air–fuel equivalence ratio (λ) corresponding to specific engine operation point

Carbon dioxide (CO₂) is a major component of exhaust gas recirculation, and its high heat capacity makes it an effective heat absorber during combustion, thereby reducing the maximum combustion temperature [25]. As depicted in Fig. 9, CO₂ levels remain consistent, ranging from 6–6.18% for OP.1 and 7.47–7.6% for OP.2, regardless of the

LPG ratio or EGR condition. This stability is most likely due to the maintenance of a constant air–fuel ratio (AFR) throughout all test trials (Fig. 10 and Fig. 11). Furthermore, analysis of Fig. 9 and Fig. 10 indicates an inverse relationship between CO₂ and O₂ levels in the exhaust gases.

7. Conclusions

The present study experimentally investigates the effects of LPG addition and EGR application on the combustion and emission characteristics of a modified four-cylinder, turbocharged diesel engine (AJM 1.9 TDI) operating in dual-fuel mode. The key conclusions of this investigation are summarized below:

1. The introduction of premixed LPG under EGR-off conditions enhances the premixed combustion phase, resulting in elevated in-cylinder pressures and temperatures, increased burn rates, and a more pronounced net heat release rate. This effect is primarily attributed to the higher laminar flame propagation of LPG. For instance, the peak in-cylinder pressure rose from 58.74 bar to 59.08 bar in OP.1, and from 99.21 bar to 105.03 bar in OP.2. The corresponding mean temperatures increased from 1580 K to 1700 K in OP.1, and from 1680 K to 1915 K in OP.2. In OP.1, the gross mean effective pressure (GMEP) showed minimal variation, ranging from 7.01 bar in Trial 1 to 7.07 bar in Trial 4, with a slight decrease to 6.79 bar in Trial 3, reflecting minor fluctuations in cylinder work as the LPG fraction increased. In OP.2, the corresponding GMEP decreased from 11.3 bar in Trial 1 to 10.88 bar in Trial 4, indicating that although the combustion process intensified and produced higher peak pressures, the overall cycle work was reduced due to changes in combustion timing and mixture distribution. Under EGR-on conditions, combustion is hindered due to charge dilution, reduced oxygen availability, and the cooling effect caused by the vaporization of LPG. These effects hinder fuel atomization and suppress flame propagation, especially at elevated LPG levels. As a result, the maximum pressure decreased from 55.39 bar to 52.70 bar in OP.1 and from 89.18 bar to 85.54 bar in OP.2, accompanied by a reduction in mean temperature from 1580 K to 1420 K in OP.1. For OP.2, the temperature fluctuates due to unstable combustion, ranging from 1650 K to 1695 K, with intermediate values of 1470 K and 1530 K. Correspondingly, the GMEP for OP.1 varied between 5.42 and 5.82 bar. In contrast, for OP.2, GMEP ranged from 10.17 bar to 10.23 bar, with intermediate values of 9.78 bar and 9.85 bar, indicating combustion instability under EGR-on conditions.

2. The formation of NO_x emissions is strongly influenced by in-cylinder temperature, in-cylinder pressure, oxygen availability, and the residence time required for thermal reactions to occur at elevated temperatures. Experimental results confirm that in EGR-off conditions, the engine produces significantly higher NO_x emissions compared to EGR-on conditions. Under EGR-on conditions, the introduction of premixed LPG shows potential for reducing NO_x emissions. At OP.1, NO_x concentrations ranged from 210 ppm (Trial 1) to 265 ppm (Trial 4). In contrast, at OP.2, NO_x initially increased from 510 ppm (Trial 1) to 528 ppm (Trial 2) before decreasing significantly to 382 ppm (Trial 4). This behavior is primarily attributed to the combined

effects of charge dilution, lower in-cylinder temperatures due to LPG vaporization, and reduced diesel fuel input, all of which act to suppress thermal NO_x formation. Notably that, under EGR-off conditions, increasing the LPG fraction leads to a reduction in NO_x emissions despite higher in-cylinder pressures and temperatures. At OP.1, NO_x decreased from 340 ppm (Trial 1) to 290 ppm (Trial 4), while at OP.2, it declined from 670 ppm (Trial 1) to 436 ppm (Trial 4). This phenomenon can be explained by several factors, including a more uniform temperature distribution resulting from the premixed combustion of LPG, which minimizes localized hot spots responsible for NO_x formation. Furthermore, the absence of optimized engine calibration for LPG use and the maintenance of a constant air-fuel equivalence ratio (λ) cause suboptimal fuel-air mixtures, promoting incomplete and unstable combustion.

3. It was observed that carbon monoxide (CO) emissions are higher in dual-fuel mode compared to single-fuel operation, primarily due to the excess of exhaust gases, incomplete combustion caused by limited oxygen availabil-

ity, suboptimal air-fuel ratios, and poor charge mixing quality. Under EGR-off conditions, CO concentrations increased from 280 ppm to 1874 ppm in OP.1, and from 354 ppm to 1930 ppm in OP.2. Similarly, under EGR-on conditions, CO levels rose from 272 ppm to 1787 ppm in OP.1, and from 106 ppm to 1900 ppm in OP.2.

4. Maintaining a constant air-fuel ratio (λ) across all tests resulted in consistent CO_2 levels, regardless of the LPG ratio. CO_2 concentrations ranged from 6 to 6.18% for OP.1 and from 7.47 to 7.6% for OP.2. Additionally, an inverse relationship between CO_2 and O_2 concentrations in the exhaust gases was observed.

The LPG dual-fuel combustion mode may be a promising approach for mitigating toxic emissions in older diesel engine generations. Modifying conventional diesel engines for dual-fuel mode is cost-effective and involves minimal alterations. However, achieving improved engine performance and reduced emissions requires careful adaptation of engine operating conditions, including optimization of the injection strategy and ECU algorithms.

Nomenclature

a_1 – a_7	JANAF polynomial coefficients.	MAP	manifold absolute pressure sensor
APP	accelerator pedal position sensor	MFB	mass fraction burn
B	cylinder bore	m_G, m_D	mass flow rates of gaseous and diesel fuel
BR	burn rate	n	engine revolutions per minute
BTDC	before top dead centre	NHRR	net heat release rate
CD	combustion duration	NO_x	nitrogen oxides
CI	compression ignition	OP	operating point of the engine
CNG	compressed natural gas	ϕ	crank angle
c_p	specific heat at constant pressure	P	mean in-cylinder pressure
c_v	specific heat at constant volume	PM	particulate matter
DFECU	dual-fuel electronic control unit	Q_c	cumulative heat release
DI	direct injection	Q_{net}	total amount of in-cylinder net heat release
DICI	direct injection compression ignition	R	universal gas constant
EGR	exhaust gas recirculation	s	piston displacement
EVO	exhaust valve opening	SR	substitution ratio
GMEP	gross mean effective pressure	T	mean in-cylinder temperature
ICE	internal combustion engine	V	in-cylinder volume
IVC	intake valve closing	WLTP	Worldwide Harmonized Light Vehicles Test Procedure
l	connecting rod length	γ	specific heat ratio
$L_{\text{HVG}}, L_{\text{HVD}}$	lower heating values of gaseous and diesel fuel	ε_s	engine compression ratio.
LPG	liquified petroleum gas	λ	air-fuel equivalence ratio (lambda)
MAF	mass air flow sensor		

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