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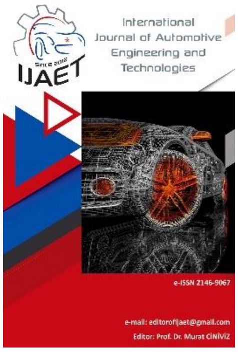

Original Research Article

Investigation of the role of methane injection timing and pressure on engine performance, emissions and environmental impact in a diesel engine

Halil Erdi Gülcan [1*]

[1,*] Mechanical Engineering Department, Faculty of Technology, Selcuk University, Konya, 42250, Türkiye.

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* Corresponding author
halil.gulcan@selcuk.edu.tr

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ABSTRACT

Methane diesel dual-fuel engines are gaining increasing interest because they offer lower emissions and higher efficiency compared to conventional single-diesel fuel engines. However, due to the low combustion efficiency and combustion stability of the methane-diesel dual-fuel application, there are still unresolved issues that need to be addressed. In this study, the effects of methane gas injection timing and pressure on engine performance and exhaust emissions are investigated in order to overcome problems related to the application of methane gas in dual-fuel engines. Additionally, the environmental and economic impacts of the exhaust emissions resulting from combustion are analyzed. The study is conducted with 5 different methane gas injection timings (25, 35, 45, 55, and 65 degrees after TDC) and 4 different methane gas injection pressures (1 bar, 1.5 bar, 2 bar, and 2.5 bar). In the experiments, the engine torque (5 Nm) and operating speed (1850 1/min) are kept constant. The results show that increasing the methane gas injection pressure (GIP) from 1 bar to 2.5 bar and delaying the methane gas injection timing (GIT) from 25° aTDC to 65° aTDC leads to an average reduction of 8.5% in SFC values and a 4% increase in thermal efficiency compared to diesel operation. Similarly, increasing GIP and delaying GIT results in an average reduction of 46% in NO emissions and an average reduction of 48% in soot emissions.

Keywords: Diesel-methane dual fuel, Engine performance, Emissions, Gas injection timing and pressure, environmental impacts

1. Introduction

The use of internal combustion engines (ICE) in transportation, agriculture, maritime sector, and industrial areas significantly contributes to the formation of greenhouse gas emissions (especially carbon dioxide) and air pollution [1-3]. The primary cause of this situation is the use of fossil-derived fuels in ICE [4]. Today, in many countries, stringent emission standards are enforced to reduce the release of harmful gases resulting from the use of fossil-derived fuels. This situation compels motor manufacturers and researchers to make various improvements in both pre-combustion and post-combustion in ICE [5, 6]. At the forefront of these improvements is the use of alternative fuels that do not involve significant structural changes for the engine [7]. Among the alternative fuels, gaseous fuels like methane [8, 9], H_2 [10, 11], and liquid fuels like biodiesel [12-15], alcohols [16-18], and wastes [19] are prominent.

Due to the potential for methane gas to be produced from biomass such as municipal waste, sludge, and trash, it is becoming increasingly important for both electricity generation and use in ICEs in the coming years [20]. Consequently, various studies are being conducted to increase the use of methane in both gasoline and diesel engines. Methane can be used directly in gasoline engines because it can be ignited by an external ignition source. However, in diesel engines, methane can either be used directly through a special injection system or, alternatively, in a dual-fuel mode without significant modifications to the engine [21]. In diesel engines, the use of methane is primarily implemented in dual-fuel mode due to its economic advantages and minimal modification requirements. In dual-fuel mode, methane is introduced into the combustion chamber during the intake phase and is injected onto the gas-air mixture at the end of the compression phase to create the diesel ignition source [22, 23].

In literature, it is possible to find various studies on the use of methane gas in diesel engines. Some of these studies are summarized as follows: Krishnan and colleagues [24] investigated the effects of different natural gas additions on performance in a diesel engine. They reported that as the natural gas ratio increased from 0% to 90% in the dual-fuel application, the engine efficiency tended to decrease. However, it was also noted that the increase in natural gas ratio led to a reduction in NO_x and smoke emissions. Papagiannakis and Hountalas [25] conducted an experimental study on the effects of natural gas ratio on performance and emissions in a natural gas-diesel dual-fuel engine. The experiments were carried out under different load conditions and natural gas energy ratios. The results showed that increasing the percentage of natural gas reduced NO and smoke emissions but significantly increased HC and CO emissions. Additionally, it was reported that the BSFC values tended to increase with a higher natural gas percentage. Di Blasio and colleagues [26] reported that increasing the methane energy ratio from 0% to 50% in a methane-diesel dual-fuel application resulted in a threefold increase in HC emissions and a tenfold increase in CO emissions. The study also highlighted those changes in the compression ratio that had a significant impact on HC and CO emissions. Chen and colleagues [27] investigated the effects of water injection on the performance and emissions of an engine operating on a diesel-methane fuel mix. The methane energy ratio in the study varied from 0% to 50% in five different configurations. The results indicated that an increase in the methane content of the mixture significantly reduced NO_x emissions, but HC and CO emissions increased. It was also emphasized that water port injections did not significantly affect HC and CO emissions. Ouchikh and colleagues [28] investigated the effects of diesel injection parameters on the performance and emissions of a diesel engine operating with a methane-diesel dual-fuel system. The results indicated that while thermal efficiency decreased with the diesel injection timing in dual-fuel operation, thermal efficiency increased with the split injection strategy. Additionally, it was noted that the split injection strategy resulted in a 20% reduction in brake specific fuel consumption (BSFC). However, HC and CO emissions showed significant increases compared to baseline diesel fuel. Tripathi and colleagues [29] found that in a diesel engine, increasing the methane energy ratio from 0% to 75% resulted in a gradual decrease in thermal efficiency. Additionally, it was observed that HC emissions increased by approximately 10 times and CO emissions by about 5 times with the rise in methane energy ratio. On the other hand, a maximum reduction of around 50% in NO_x emissions was also reported. Ahmad and colleagues [30] focused on the effects of using different proportions of ethane gas on the performance and emissions in a methane-diesel dual-fuel application. Ethane gas was used at 10% and 20% concentrations in the dual-fuel system. The results showed that the addition of ethane gas improved thermal efficiency, which had decreased with the use of methane gas. While the addition of ethane contributed to a reduction in HC emissions, it also led to an increase in NO_x emissions. Liu and colleagues [31] reported that increasing the diesel fuel ratio in a methane-diesel dual-fuel application would be beneficial for methane oxidation by raising the high-temperature regions within the cylinder. Di lorio et al. [32] reported in their study on methane-diesel dual fuel application that methane gas uses significantly reduced NO_x and smoke emissions compared to diesel-only operation. Khedkar et al. [33] focused on the effects of control parameters such as diesel injection timing, EGR control, and intake throttling on the low thermal and combustion efficiency of a methane-diesel engine. The results showed that with 55% EGR, 50% premix, and advanced diesel injection timing, thermal efficiency (TE) improved by approximately 10%, and combustion efficiency also increased. Additionally, it was emphasized that HC and NO_x emissions were significantly reduced. Zarrinkolah and Hosseini [34] utilized both the traditional methane-diesel dual-fuel mode and early and late injection RCCI modes to reduce methane emissions. The results indicated a reduction in methane emissions ranging from 12% to 33% in early and late RCCI modes compared to traditional operation. However, it was also found that the early and late RCCI modes produced soot particulate emissions approximately 417% and 67% higher, respectively. Cameretti and colleagues [35] conducted a numerical analysis of a hydrogen-methane mixture in a marine diesel engine. The results indicated that using hydrogen instead of methane reduces CO_2 emissions by 54%, while increasing NO_x emissions by 76%. Yin and colleagues [36] conducted experimental and numerical studies on a diesel/methane/hydrogen fueled engine. They reported that an increase in the hydrogen fraction within the triple mixture enhances the combustion process, reducing both ignition delay and combustion duration. It was also noted that this leads to an increase in NO_x emissions while resulting in a decrease in CO and CH_4 emissions. Zhang and colleagues [37] studied the diesel injection timing in a natural gas-diesel engine. They reported that advancing the diesel injection timing resulted in improved performance and enhanced flame development. However, it was also noted that this condition increased CH_4 emissions by approximately 4% and NO_x emissions by about 6%.

Based on the summaries of the studies presented in the literature, it is evident that the use of methane in diesel engines significantly reduces NO_x and smoke emissions, while increasing HC and CO emissions. Additionally, some studies report a deterioration in thermal efficiency and fuel consumption with methane gas usage. This indicates that engines utilizing methane-diesel fuel are still open to improvements and have unresolved issues that need to be addressed. Also, while some studies contribute to the reduction of HC emissions, this simultaneously leads to an increase in NO emissions. Therefore, it appears that research will continue to improve the operational efficiency of methane in diesel engines. In the current study, the effects of varying gas injection timing and pressure on engine performance, exhaust emissions, and environmental impact are investigated to enhance the usability of methane gas in diesel engines. When reviewing other studies, it is observed that very few focus on gas injection timing, and most of these studies primarily emphasize performance. However, the significant reduction of NO emissions resulting from the use of methane in diesel engines necessitates an examination from the perspective of environmental and economic impact analysis to enhance environmental sustainability and raise awareness. Additionally, there is a significant gap in literature regarding this area. The aim of this study is to examine performance, emissions, and environmental impact parameters under various conditions of methane gas injection timing and pressure in diesel engines.

2. Experimental Setup and Method

2.1. Experimental setup

The methane-diesel dual-fuel application conducted at different gas injection timings and pressures is carried out on a single-cylinder, air-cooled, 4-stroke diesel engine. The single-cylinder, 315 cc volume diesel engine used in the study is selected based on contemporary 4-cylinder diesel engines. When the total volume of the 4-cylinder diesel engine (1248 cc/4 cylinders = 312 cc) is divided by the number of cylinders, the resulting volume is nearly equivalent to the volume of the single-cylinder engine used in the study. Additionally, due to the CRDI systems present in modern diesel engines, the fuel system on the existing engine is similarly modified. This adaptation ensures that the findings are more realistic, as the current single-cylinder engine has been tailored to align with more modern diesel engines. Therefore, the air-cooled AD320 Anadolu engine was chosen for experimentation as the most suitable single-cylinder engine for both modification and the dual-fuel concept. A summary of the engine's technical specifications is presented in Table 1. Additionally, the experimental equipment and engine installation view are presented in Fig. 1. In the air-cooled engine, a common rail fuel system and an ECU are used to control the amount, pressure, and injection timing of the diesel fuel. The gas fuel system injected fuel into the combustion chamber through port injection. The gas pressure is adjusted to the desired level using a two-stage pressure regulator on the methane gas cylinder. Additionally, the gas injection timing was controlled by the gas ECU. Detailed stages of both the diesel fuel system and the gas fuel system have been presented in previous studies [6, 8]. For the performance and emission tests of the engine used in the experiments, the engine is first mounted on an ABB brand DC dynamometer capable of measuring up to 50 kW of power and 6000 1/min. Methane gas consumption is measured instantaneously and cumulatively using a gas flow meter. Similarly, air consumption is also measured instantaneously and cumulatively with an air flow meter. Diesel fuel consumption is calculated based on the rate set through the ECU in mg/cycle. Details of the measurement equipment used in the experiments are presented in Table 2.

Table1. Technical details of the dual-fuel engine used at variable gas injection timing and pressure variations [38].

Technical details	**Value/AD320 diesel engine**
Cylinder number	1-cylinder
Displacement	0.315 lt
Bore diameter	78 mm
Stroke diameter	66 mm
Compression ratio	17.3
Peak engine torque (at speed)	11 Nm (at 1850 1/min)
Diesel fuel system	**Common rail** Injection timing: 11° bTDC Injection pressure: 400 bar
Gas supply system	**Port injection** Injection timing: 25°, 35°, 45°, 55°, and 65° aTDC Injection pressure: 1, 1.5, 2, and 2.5 bar

Emission data measurements are conducted using a Bosch-branded emission system.

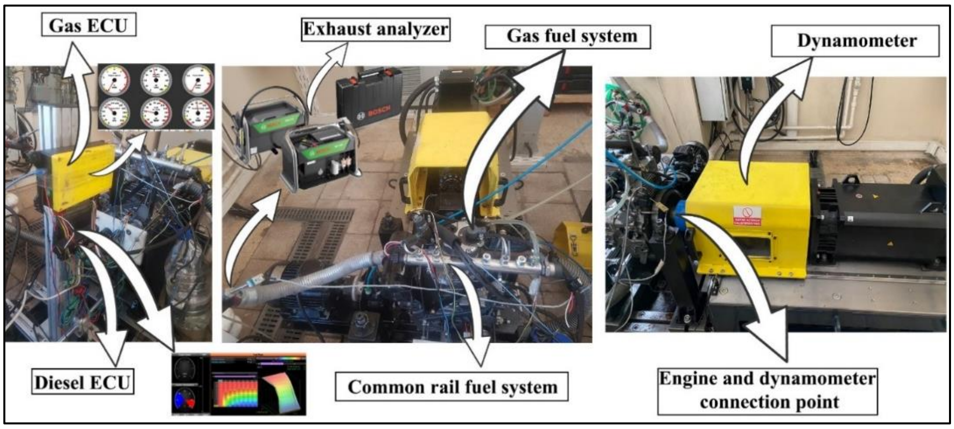

Fig. 1. View of test equipment and engine installation

Table 2. Details of the measurement equipment.

Test Instruments	**Measurement**	**Measure range**	**Sensibility**
ABB DC Dynamometer	Load/speed	0..50 kW/0..6000 rpm	± 0.01 rpm
Sierra SmartTrak 100	Methane flow meter	0..50 slpm	± 1.0%
Pietro Fiorentini series-c	Air flow meter	0..4.4 liter/s	± 1.0%
K type thermocouple	Temperature	0..850 °C	± 1.0 °C

Table 3. Technical specifications of the Bosch brand emission device utilized in the experiments [6].

Data type(method)	Measure range	Sensibility
CO (Non-dispersive infrared)	0..10% volume	±0.001%
HC (Flame ionization detector)	0..9999 ppm	±1 ppm
NO (Chemiluminescence detector)	0..5000 ppm	±1 ppm
Soot (Optical meter)	0..9.99 1/m	±0.01 1/m

Table 4. The physicochemical details of diesel and methane fuels [6].

Physicochemical details	Euro diesel	Methane (purity 99.5%)
Density, g/cm^3	0.83–0.84	0.000678
Lower calorific value, kJ/kg	42500	50000
Octane number	-	<120
Cetane number	50-55	-
Air/fuel (Stoichiometric)	14.6	17.4

Table 5. The detailed test matrix for the study.

Case	Fuel	Torque/Speed	Gas injection timing (GIT)	Gas injection pressure (GIP)
1	Euro Diesel	5 Nm/1850 rpm	-	-
2	Dual fuel	5 Nm/1850 rpm	25^o aTDC	1, 1.5, 2, and 2.5 bar
3	Dual fuel	5 Nm/1850 rpm	35^o aTDC	1, 1.5, 2, and 2.5 bar
4	Dual fuel	5 Nm/1850 rpm	45^o aTDC	1, 1.5, 2, and 2.5 bar
5	Dual fuel	5 Nm/1850 rpm	55^o aTDC	1, 1.5, 2, and 2.5 bar
6	Dual fuel	5 Nm/1850 rpm	65^o aTDC	1, 1.5, 2, and 2.5 bar

The Bosch Bea060 model is used to quantify HC, CO, and NO emissions, while the Bea070 model measures soot opacity levels. Detailed specifications and sensitivities of these devices are thoroughly presented in Table 3.
For the experimental study, diesel fuel and pure methane gas are sourced from local suppliers. The physicochemical details of these fuels are presented in Table 4.

2.2. Method

Before conducting experiments on methane injection timing and pressure, the engine is run on baseline diesel fuel at idle condition until it reaches a stable combustion. Subsequently, the load and speed for the engine tests are set using a computer-controlled dynamometer, and the experiments commence. Additionally, during the experiment process, careful attention is given to managing the engine's surface temperature, considering the possibility of excessive temperature (>200^{o}C) increases that could affect the reliability of the test results. Before the experiments, the calibration of the Bosch emission device, ABB DC dynamometer, and other measurement equipment is prepared and ensured to be ready. The experiments are initially conducted using EURO diesel fuel at 5 Nm and a constant speed of 1850 1/min. The engine speed determined in the study is also the speed at which maximum engine torque is achieved. Additionally, a vehicle equipped with a diesel engine (either passenger or light commercial) generally operates in the range of 1500 to 2200 rpm under daily traffic conditions. Considering this speed range, an engine speed of 1850 rpm aligns with the movement speed of a vehicle in daily traffic. The engine torque is defined as medium load, as most vehicles operate under low to medium loads in traffic conditions. Subsequently, methane-diesel dual-fuel experiments are carried out. In these experiments, methane gas injection timing (GIT) occurs 25^o aTDC through port injection at a pressure of 1 bar. Following this, experiments for GIT25 are completed at gas injection pressures (GIP) of 1.5 bar, 2 bar, and 2.5 bar. The same procedures are then followed for the experiments at GIT35, GIT45, GIT55, and GIT65, respectively. Previous studies [6, 8] have identified that early injection of methane causes various problems such as low volumetric efficiency and combustion temperatures. Therefore, it was decided to delay the GIT from 25^o aTDC to 65^o aTDC, considering that later injection could address the low volumetric efficiency issue. The flow diagram of the experimental process is presented in Fig. 2. Also, the detailed test matrix for the study is presented in Table 5.

In the methane-diesel dual-fuel combination,

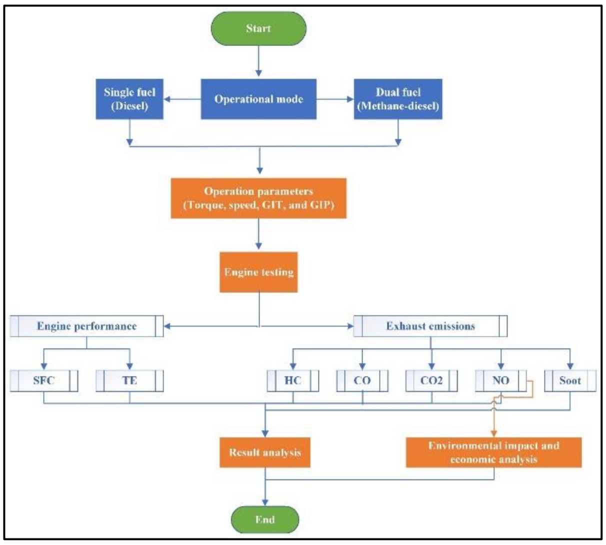

Fig. 2. The flow diagram of the experimental process

the percentage of methane gas sent to the combustion chamber, referred to as gas energy percentage (GEP), can be calculated using Eq. (1) [39]. Fig. 3 presents the contribution to total fuel energy of methane and diesel fuel corresponding to different GIT and the associated GIP.

$$GEP = \left[\frac{\dot{m}_{gas}LCV_{gas}}{\dot{m}_{gas}LCV_{gas}+\dot{m}_{diesel}LCV_{diesel}}\right]x100 \quad (1)$$

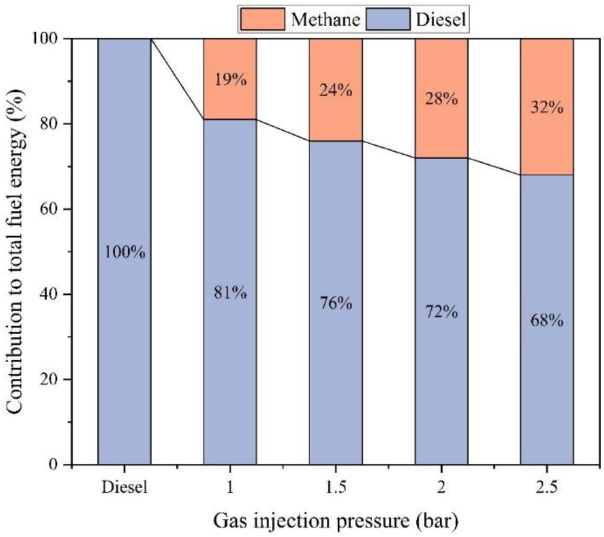

Fig. 3. The contribution to total fuel energy of methane and diesel fuel.

2.3. Environmental and economic impact analysis

A significant issue with diesel engines is the high levels of NO_x emissions they release into the environment. However, the use of methane gas as a dual fuel in diesel engines contributes to minimizing this problem. Evaluating such an important development from both environmental and economic perspectives is crucial for understanding its impacts. The analysis of these environmental and economic impacts is crucial for enhancing environmental awareness and sustainability. This study evaluates the environmental and economic effects of NO_x emissions generated from both diesel and methane-diesel combustion. To determine the environmental impact of NO_x emissions (EN_{NOx}), Eq. (2) [40] can be used. Here, $\dot{m}_{exh}$ represents the exhaust mass flow (kg/s) and en_{Nox} denotes the environmental impact coefficient.

$$EN_{NOx} = \dot{m}_{exh}\ x\ en_{Nox} \quad (2)$$

In addition, the economic impact of NO_x emissions (EC_{NOx}) can be calculated using Eq. (3) [41], where eco represents the environmental-economic impact coefficient of NO_x emissions (eco_{Nox}).

$$EC_{NOx} = \dot{m}_{exh}\ x\ eco_{Nox} \quad (3)$$

The en_{Nox} and eco_{Nox} coefficients given in Eq. (2) and Eq. (3) are used in the calculations as 2.749 mPts/g and 693.7 €/g, respectively [42].

3. Results and discussion

3.1. Impact of GIT and GIP on performance

Fig. 4 shows the variation of specific fuel consumption (SFC) corresponding to different GIT and GIP. Both methane gas injection timing (GIT) and methane gas injection pressure (GIP) positively affect SFC. The lowest SFC values are obtained in dual-fuel mode with GIP 2.5 bar operation, while the highest SFC outputs are obtained with diesel fuel. For instance, under GIP 1 bar conditions, GIT 25 reduces SFC by 1% compared to diesel fuel, while delaying GIT to 65° aTDC contributes to a 3.5% reduction in SFC. Similarly, under GIP 2.5 bar conditions, GIT 25° reduces SFC by 6% compared to diesel fuel, and delaying GIT to 65° aTDC increases the SFC reduction to 12%. The main reasons for the reduction in SFC with methane use in the dual-fuel concept are the combined energy utilization of diesel and methane gas, as well as methane's higher lower heating value compared to diesel. Overall, when GIT changes from 25° to 65°, SFC decreases by an average of 2.5%, 6%, 7%, and 8.5% compared to diesel combustion for GIP 1 bar, GIP 1.5

bar, GIP 2 bar, and GIP 2.5 bar operations, respectively.

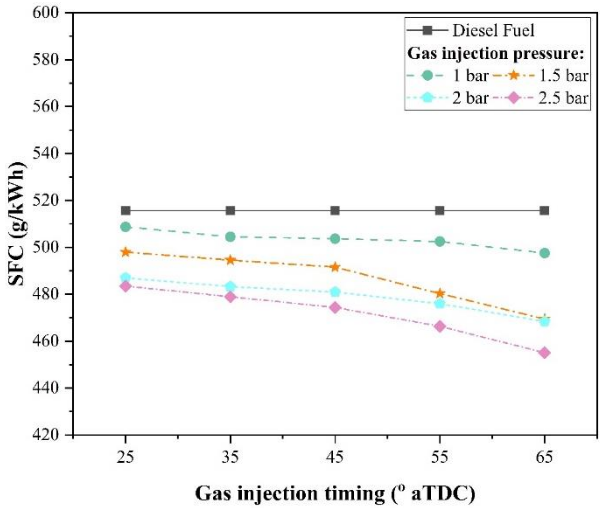

Fig. 4. The variation of SFC at different GIT versus GIP.

Fig. 5 shows the variation of thermal efficiency (TE) corresponding to different GIT and GIP. The use of methane in diesel engines contributes to an increase in TE. Additionally, the gradual delay of GIT also supports the increase in TE. The highest TE is obtained under GIP 2.5 bar conditions, and the TE outputs of GIP 2.5 bar and GIP 2 bar operations are quite similar. The lowest TE is observed under GIP 1 bar conditions (except for GIT 65). The implementation of the methane-diesel dual-fuel system based on energy ratio contributes to the reduction of SFC values. This also affects the reduction in TE. The reason for the lower TE in the GIP 1 bar operation between GIT 25 and GIT 55 compared to diesel fuel is that the fuel consumption amounts are quite similar. As the reduction rate in fuel consumption increases, TE gradually improves. Additionally, advancing the GIT causes the methane gas to block the incoming air into the cylinder, resulting in less air entering the combustion zone for the reaction. This is one of the factors that lowers TE. Delaying GIT from 25° aTDC to 65° aTDC allows more air to enter the combustion zone, leading to higher combustion efficiency and increased TE. Yuvenda and colleagues [43] reported that delaying gas injection timing in dual-fuel mode increases volumetric efficiency due to more air intake into the cylinder, resulting in lower fuel consumption and higher TE achieved. Overall, when GIT changes from 25° to 65°, TE increases by an average of 2%, 3%, and 4% compared to diesel combustion for GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively. The TE output of the GIP 1 bar operation decreases by an average of 1% compared to diesel fuel.

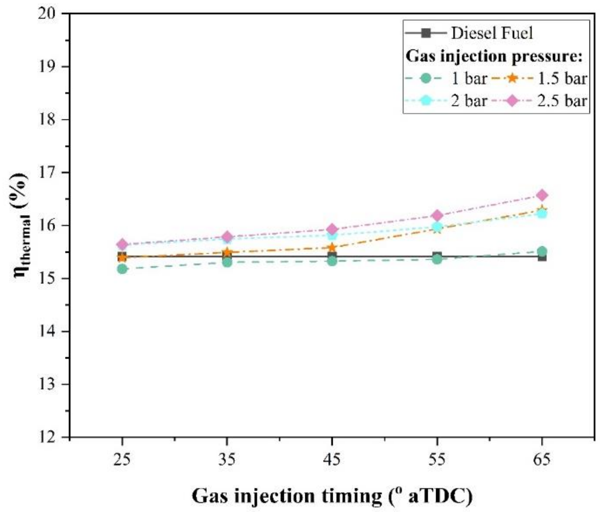

Fig. 5. The variation of thermal efficiency at different GIT versus GIP.

3.2. Impact of GIT and GIP on emissions

Fig. 6 shows the variation of HC emission corresponding to different GIT and GIP. Although the gradual increase of GIP from 1 bar to 2.5 bar significantly causes an increase in HC emissions, the gradual increase of GIT from 25° aTDC to 65° aTDC contributes to a decreasing trend in HC emissions. In experiments conducted under constant load and speed conditions, the lowest HC emissions are obtained with diesel fuel (25 ppm). Under GIP25 and GIP 1 bar conditions, HC emissions increase by 456% compared to diesel combustion, while this increase rate drops to 340% when the gas injection timing is delayed to 65° aTDC. Early injection of methane gas interrupts the mass flow rate of air taken into the cylinder and occupies some of the air volume inside the cylinder. This reduces the amount of air available for combustion and leads to the direct release of unburned CH_4 gas. Overall, when GIT changes from 25° to 65°, HC emissions increase by an average of 414%, 455%, 522%, and 562% compared to diesel combustion for GIP 1 bar, GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively. Tripathi et al. [29] reported that as the amount of methane supplied to the cylinder increases, the oxygen concentration decreases, thereby slowing down the combustion reaction and increasing HC emissions by up to 1000 times. However, in the current study, despite the increase in HC owing to increase in methane energy ratio, it is observed that the transition from 25^o aTDC to 65^o aTDC contributes to a decrease in HC emissions according to MIT.

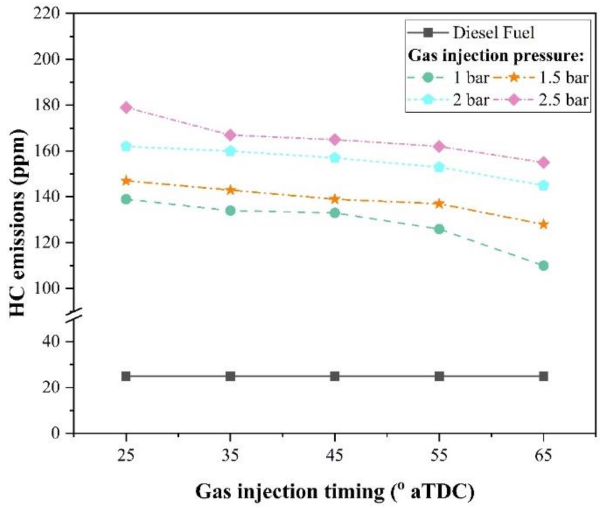

Fig. 6. The variation of HC emission at different GIT versus GIP.

Fig. 7 shows the variation of CO emission corresponding to different GIT and GIP. Similar to the results of HC emissions, CO emissions show an increasing trend with the rise in GIP. However, the rate of this increase slows down as GIT changes from 25^o aTDC to 65^o aTDC. The lowest CO emissions are obtained with diesel fuel (0.067%), while the highest CO emissions occur with early gas injection timing (GIT25) and high methane energy ratio (GIP 2.5 bar). The increase in methane energy ratio in dual-fuel mode raises the mass of methane entering the cylinder, causing the air inside the cylinder to cool down (due to methane's physicochemical properties). This chain reaction lowers combustion temperatures, stopping the oxidation of CO into CO_2, leading to higher CO emissions. Additionally, the increase in methane mass reduces the available O_2 in the combustion zone, which also contributes to the rise in CO emissions. Compared to diesel fuel, the highest increase in CO emissions is 137% under GIT 25 and GIP 2.5 bar conditions, while the lowest increase is 51% under GIT 65 and GIP 1 bar conditions. Overall, when GIT changes from 25^o to 65^o, CO emissions increase by an average of 64%, 78%, 94%, and 111% compared to diesel combustion for GIP 1 bar, GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively. Bora and colleagues [44] reported that methane gas reduces volumetric efficiency, leading to increased CO emissions in dual-fuel mode, which is consistent with the findings obtained in this article.

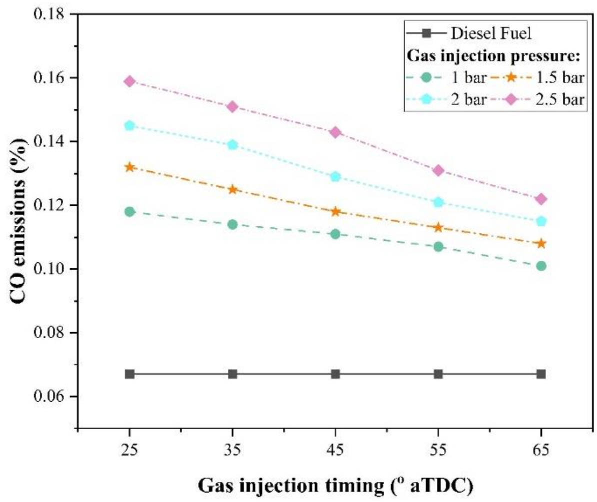

Fig. 7. The variation of CO emission at different GIT versus GIP.

Fig. 8 shows the variation of CO_2 emission corresponding to different GIT and GIP. The use of methane gas in diesel engines contributes to a reduction in CO_2 emissions. The lowest CO_2 emissions are obtained in the GIP 2.5 bar operation, while the highest CO_2 emissions are observed in diesel operation. For instance, at GIP 2.5 bar, where the lowest CO_2 emissions are recorded, the GIT 25 operation shows approximately a 13% reduction in CO_2 emissions compared to diesel operation, with an average reduction of 11.5% between GIT25 and GIT65 operation conditions. In the GIP 1 bar operation, which has a low methane energy ratio, GIT25 reduces CO_2 emissions by about 7% compared to diesel fuel, while the change in GIT from 25^o aTDC to 65^o aTDC results in a 3.5% reduction in CO_2 emissions compared to diesel combustion. As seen, an increase in the methane energy ratio contributes to a reduction in CO_2 emissions, while delaying methane injection timing tends to increase CO_2 emissions. Nevertheless, all CO_2 outputs obtained in the dual-fuel concept are lower than those produced by diesel combustion. The main reason for this is that with the increase in methane energy ratio, the carbon content of the mixture decreases. Additionally, the decrease in combustion efficiency (as HC and CO emissions rise) also contributes to the reduction in CO_2 emissions. Another significant factor is that as CO emissions increase, CO_2 emissions decrease. Prabhu and colleagues [45] reported that an increase in the methane content in biogas reduces CO_2 emissions, and their findings are consistent with the results obtained in this article. Overall, when GIT changes from 25° to 65°, CO_2 emissions decrease by an average of 5%, 6%, 10%, and 11.5% compared to diesel combustion for GIP 1 bar, GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively.

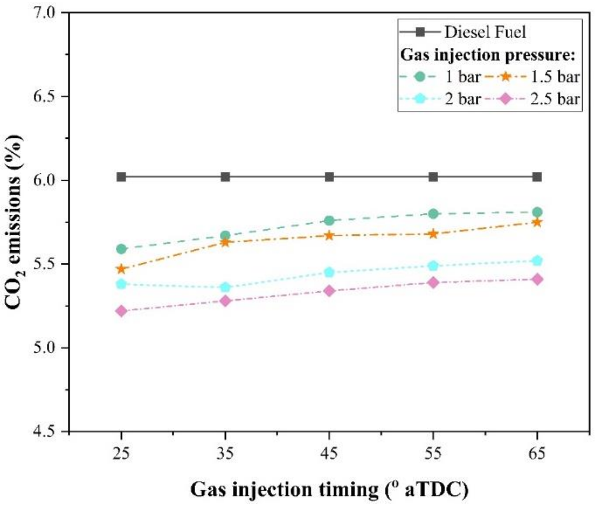

Fig. 8. The variation of CO_2 emission at different GIT versus GIP.

Fig. 9 shows the variation of NO emission corresponding to different GIT and GIP. One of the most significant problems in diesel engines is the formation of NO emissions due to the high compression ratio and high lambda value. However, the use of methane gas in diesel engines significantly contributes to the reduction of NO emissions. As seen in Fig. 9, NO emissions gradually decrease as GIP increases from 1 bar to 2.5 bar. The lowest NO emissions are obtained in the GIP 2.5 bar operation, while the highest NO emissions occur in diesel operation. For instance, in the GIP 1.5 bar operation, GIT 25 reduces NO emissions by 36% compared to diesel, while in the GIP 2.5 bar operation, GIT 25 reduces NO emissions by 47% compared to diesel. As GIT increases from 25° aTDC to 65° aTDC, NO emissions tend to increase, but they still remain lower than the NO emission outputs from diesel combustion. For example, in the GIP 1.5 bar operation, GIT 65 reduces NO emissions by 33% compared to diesel, while in the GIP 2.5 bar operation, GIT 25 reduces NO emissions by 44% compared to diesel. NO emissions are formed in conditions of high O_2 availability and high combustion temperatures. In the present study, the use of methane gas lowers the intake air temperature and also reduces the amount of air entering the cylinder. This significantly contributes to the reduction of NO emissions. Allouis and colleagues [46] reported that using methane in a diesel engine reduces the air-fuel ratio, thereby contributing to the reduction of NO_x emissions. In the current article, methane quantity increases with GIP change. Therefore, the decrease in NO emissions with an increase in GIP from 1 bar to 2.5 bar in this article is consistent. Overall, when GIT changes from 25° to 65°, NO emissions decrease by an average of 35%, 37%, 40%, and 46% compared to diesel combustion for GIP 1 bar, GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively.

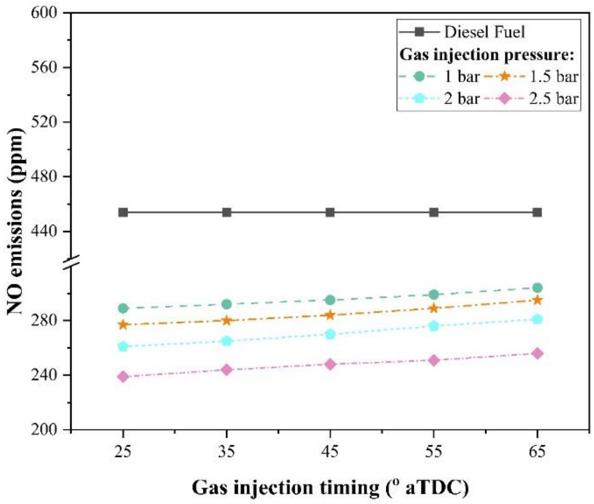

Fig. 9. The variation of NO emission at different GIT versus GIP.

Fig. 10 shows the variation of soot emission corresponding to different GIT and GIP. Soot formation from combustion is another major issue in diesel engines. In this study, the use of methane as a dual fuel significantly contributes to the reduction of soot emissions. For instance, in diesel operation, 1.21 1/m soot emissions are recorded, while in the dual-fuel mode, under GIP 2.5 bar conditions, GIT 65 reduces soot emissions by approximately 59.5% compared to diesel combustion. When GIT advances from 25° aTDC to 65° aTDC, the rate of reduction in soot emissions slows down. For example, under GIP 2.5 bar conditions, GIT 25 reduces soot emissions by approximately 39% compared to diesel combustion. The highest soot emissions in dual-fuel operation are observed under GIP 1 bar conditions. The primary reason for the reduction in soot is that methane reduces diesel fuel energy (i.e., the amount of fuel injected into the combustion chamber). Additionally, methane's low C/H ratio is another important factor contributing to the reduction in soot formation. Overall, when GIT changes from 25° to 65°, soot emissions decrease by an average of 37%, 39%, 43%, and 48% compared to diesel combustion for GIP 1 bar, GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively. Liu and colleagues [47] reported that in CNG-diesel operation, increasing the amount of diesel fuel injected into the combustion chamber increases soot emissions, but the opposite occurs with an increase in CNG energy ratio. They also attribute this to the decrease in the amount of diesel fuel injected with an increase in CNG quantity. Consequently, in the current article, the increase in GIP from 1 bar to 2.5 bar results in an increase in the CH_4 energy ratio and a decrease in the amount of diesel fuel injected. This contributes to a decrease in soot emissions, aligning the results with the literature.

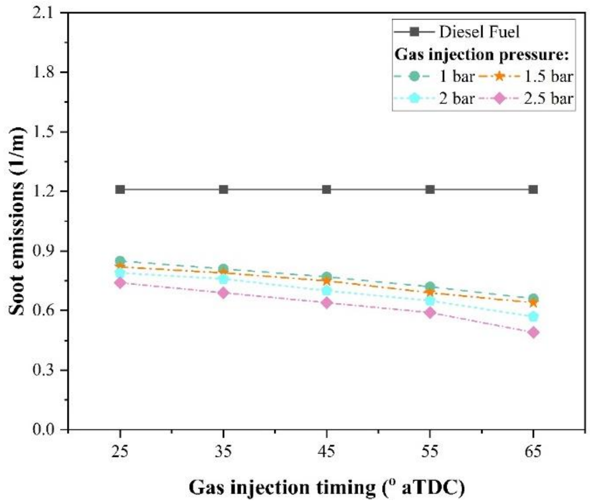

Fig. 10. The variation of soot emission at different GIT versus GIP.

3.3. Impact of GIT and GIP on environmental and economic of NO emissions

Fig. 11 shows the variation of environmental impact of NO (EN_{NOx}) corresponding to different GIT and GIP. When evaluating the EN_{NOx} outputs of both diesel and dual-fuel operations, the high NO emissions from diesel fuel result in higher EN_{NOx} compared to the dual-fuel application. The lowest EN_{NOx} is obtained with the GIP 2.5 bar operation. This is because the increase in GIP raises the GEP, which in turn reduces NO emissions. The EN_{NOx} value for diesel-only operation is recorded at 33.77 mPts/kWh. Under the GIP 2.5 bar conditions, where the lowest EN_{NOx} values are obtained, the EN_{NOx} values for GIT 25, GIT 35, GIT 45, GIT 55, and GIT 65 are 17.1 mPts/kWh, 17.5 mPts/kWh, 17.8 mPts/kWh, 17.9 mPts/kWh, and 18.4 mPts/kWh, respectively. In the dual-fuel concept, the highest EN_{NOx} values are obtained under GIP 1 bar conditions, where the EN_{NOx} values for GIT 25, GIT 35, GIT 45, GIT 55, and GIT 65 are 20.7 mPts/kWh, 20.9 mPts/kWh, 21.4 mPts/kWh, 21.4 mPts/kWh, and 21.8 mPts/kWh, respectively. Overall, when GIT changes from 25° to 65°, EN_{NOx} decreases by an average of 37%, 39.5%, 42.5%, and 47.5% compared to diesel combustion for GIP 1 bar, GIP 1.5 bar, GIP 2 bar, and GIP 2.5 bar operations, respectively.

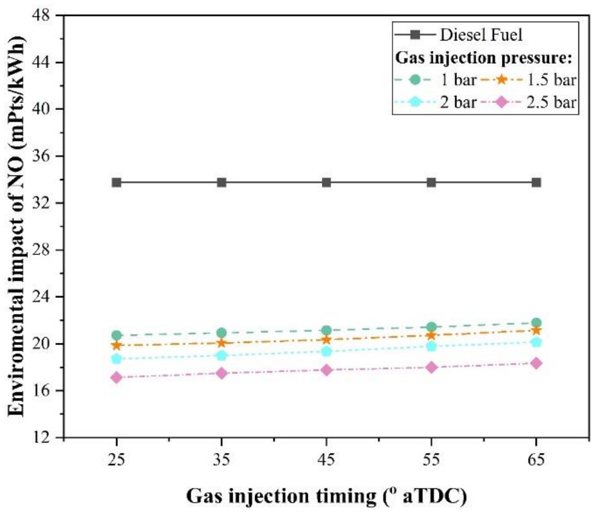

Fig. 11. The variation of environmental impact of NO at different GIT versus GIP.

Fig. 12 shows the variation of economic impact of NO (EC_{NOx}) corresponding to different GIT and GIP. As seen in the figure, the highest EC_{NOx} is obtained with diesel operation, while increasing GIP from 1 bar to 2.5 bar contributes to reducing the EC_{NOx}. The primary reason for the reduction in EC_{NOx} is the decrease in NO pollutants as the methane gas energy ratio increases. The lowest EC_{NOx} values in the GIP 2.5 bar operation for GIT 25, GIT 35, GIT 45, GIT 55, and GIT 65 are 43.3 Euro/kWh, 44.2 Euro/kWh, 44.9 Euro/kWh, 45.5 Euro/kWh, and 46.4 Euro/kWh, respectively. As the results show, changing GIT from 25 aTDC to 65 aTDC causes a slight increase in EC_{NOx}. The main reason for this is that delaying GIT improves combustion stability and efficiency, which increases NO emissions and, consequently, the EC_{NOx}. In the dual-fuel concept, the highest EC_{NOx} values are obtained under GIP 1 bar conditions, where the EC_{NOx} values for GIT 25, GIT 35, GIT 45, GIT 55, and GIT 65 are 52.4 Euro/kWh, 52.9 Euro/kWh, 53.4 Euro/kWh, 54.2 Euro/kWh, and 55.1 Euro/kWh, respectively.

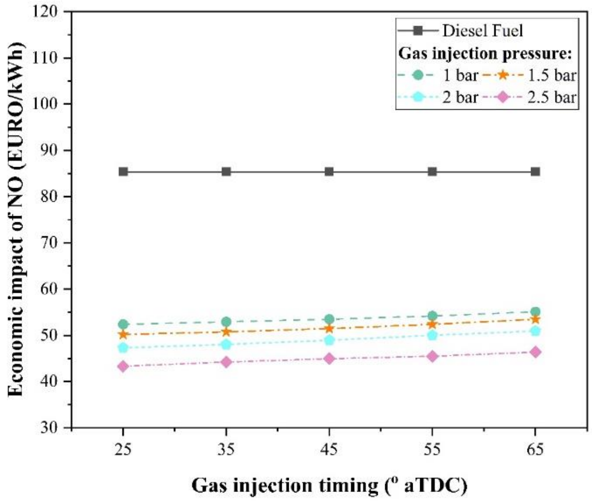

Fig. 12. The variation of economic impact of NO at different GIT versus GIP.

4. Conclusions

In this experimental study, the effects of GIT and GIP variations on performance and emissions are examined. Additionally, since one of the main problems of diesel engines is NO emissions, environmental impact and economic analysis are conducted using NO emission data from the experimental results. The outcomes obtained from both the experiments and the analysis are summarized below:

- The use of methane gas in diesel engines based on its energy fraction provides significant improvements in engine performance and exhaust emissions.
- An increase in GIP from 1 bar to 2.5 bar, along with a change in GIT from 25 aTDC to 65 aTDC, significantly contributes to the reduction of SFC. The lowest SFC is obtained under GIP 2.5 bar and GIT 65 conditions, with a maximum reduction of 12% compared to diesel. Under GIP 2.5 bar conditions, the average reduction in SFC between GIT 25 and GIT 65 is 8.5%.
- Increasing GIP and delaying GIT contribute to an increase in TE. The later injection of methane allows for more O_2 in the combustion zone and results in lower intake temperatures, leading to improved combustion efficiency and stability. Consequently, TE improves. The highest TE is obtained in the GIP 2.5 bar and GIT 65 setup, showing a 7.5% increase compared to diesel. Additionally, operating under the same GIT and GIP conditions shows an average increase of approximately 4% in TE compared to diesel operation.
- HC and CO emissions increase significantly with the rise of GIP from 1 bar to 2.5 bar. The replacement of some air by methane in the combustion zone contributes to the increase of both HC and CO. Additionally, due to the physicochemical properties of methane, its cooling effect on the intake air temperature causes the reactions of HC and CO to halt.
- However, delaying the GIT under all GIP conditions contributes to a reduction in HC and CO emissions. For example, the operation at GIP 1 bar and GIT 65 shows an approximately 21% reduction in HC emissions compared to GIT 25. Similarly, the operation at GIP 2.5 bar and GIT 65 demonstrates a 13% reduction in HC emissions compared to GIT 25.
- Also, the operation at GIP 1 bar and GIT 65 shows an approximately 14% reduction in CO emissions compared to GIT 25. Similarly, the operation at GIP 2.5 bar and GIT 65 demonstrates a 23% reduction in CO emissions compared to GIT 25.
- The methane-diesel dual-fuel combustion applied using the energy fraction reduces fuel consumption and decreases the C/H ratio as the methane content increases. This contributes to a reduction in CO_2 emissions, or in other words, the carbon footprint. For example, the lowest CO_2 emissions are achieved under GIP 2.5 bar conditions, with an average reduction rate of

11% compared to diesel operation.

- NO and soot emissions decrease significantly with the rise of GIP from 1 bar to 2.5 bar. NO emissions show a reduction of 47% at earlier GIT and higher GIP conditions. Delaying GIT allows more air to enter the combustion zone, resulting in a more homogeneous mixture. This improves combustion development, which in turn increases NO emissions.
- Soot emissions decrease by 59.5% under GIT 65 and GIP 2.5 bar conditions compared to diesel operation. The increase in the share of methane in the energy ratio leads to a reduction in the share of diesel, thereby decreasing the regions of rich mixtures. Additionally, as the gas energy fraction increases, the C/H ratio further decreases.
- The use of methane gas in an energy fraction in diesel engines significantly reduces the environmental and economic impact of NO. Under high GIP and early GIT conditions, both the environmental and economic effects decrease by approximately 48% compared to diesel.

In methane-diesel studies, while the increase in GIP leads to higher HC and CO pollutants, this issue can be mitigated by delaying GIT. Under high GIP conditions, sending heated gas fuel or heated intake air to the combustion chamber can overcome the problem of reduced intake air temperature due to the physicochemical properties of methane. Future studies could involve preheating the intake air or gas fuel before combustion. Additionally, the reduced air quantity in the combustion zone due to methane usage can lead to slower combustion reactions and a decrease in flame speed. This situation may cause the flame to extinguish before reaching the cylinder walls, resulting in the formation of high amounts of HC and CO. To address this issue, future studies could utilize nanoparticle additives that increase the surface-to-volume ratio and reaction rate within the diesel fuel.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

CRediT authorship contribution statement

Halil Erdi Gülcan: Conceptualization, Data curation, Formal analysis, Investigation, Methodology, Software, Validation, Visualization, Writing – original draft, Writing – review & editing.

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5. References

1. Van Fan Y, Perry S, Klemeš JJ, Lee CT. A review on air emissions assessment: Transportation. Journal of cleaner production, 194:673-84, 2018. Doi: https://doi.org/10.1016/j.jclepro.2018.05.151
2. Chintala V, Subramanian K. A comprehensive review on utilization of hydrogen in a compression ignition engine under dual fuel mode. Renewable and Sustainable Energy Reviews, 70:472-91, 2017. Doi: https://doi.org/10.1016/j.rser.2016.11.247
3. Altun Ş, Adin MŞ, Ilcin K. Monohydric aliphatic alcohols as liquid fuels for using in internal combustion engines: A review. Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering, 238(4):1941-75, 2024. Doi: https://doi.org/10.1177/09544089231160472
4. Erol D, Yeşilyurt MK, Yaman H, Doğan B. Evaluation of the use of diesel-biodiesel-hexanol fuel blends in diesel engines with exergy analysis and sustainability index. Fuel, 337:126892, 2023. Doi: https://doi.org/10.1016/j.fuel.2022.126892
5. Deheri C, Acharya SK, Thatoi DN, Mohanty AP. A review on performance of biogas and hydrogen on diesel engine in dual fuel mode. Fuel, 260:116337, 2020. Doi: https://doi.org/10.1016/j.fuel.2019.116337
6. Gulcan HE, Ciniviz M. Experimental study on the effect of piston bowl geometry on the combustion performance and pollutant emissions of methane-diesel common rail dual-fuel engine. Fuel, 345:128175, 2023. Doi:

https://doi.org/10.1016/j.fuel.2023.128175
7. Akcay M, Yilmaz IT, Feyzioglu A. The influence of hydrogen addition on the combustion characteristics of a common-rail CI engine fueled with waste cooking oil biodiesel/diesel blends. Fuel Processing Technology, 223:106999, 2021. Doi: https://doi.org/10.1016/j.fuproc.2021.106999
8. Gülcan HE, Ciniviz M. The effect of pure methane energy fraction on combustion performance, energy analysis and environmental-economic cost indicators in a single-cylinder common rail methane-diesel dual fuel engine. Applied Thermal Engineering, 230:120712, 2023. Doi: https://doi.org/10.1016/j.applthermaleng.2023.120712
9. Feroskhan M, Ismail S. Investigation of the effects of biogas composition on the performance of a biogas–diesel dual fuel CI engine. Biofuels, 7(6):593-601, 2016. Doi: https://doi.org/10.1080/17597269.2016.1168025
10. Demir U, Çelebi S, Özer S. Experimental investigation of the effect of fuel oil, graphene and HHO gas addition to diesel fuel on engine performance and exhaust emissions in a diesel engine. International Journal of Hydrogen Energy, 52:1434-46, 2024. Doi: https://doi.org/10.1016/j.ijhydene.2023.08.007
11. Zhang Z, Wang S, Pan M, Lv J, Lu K, Ye Y, et al. Utilization of hydrogen-diesel blends for the improvements of a dual-fuel engine based on the improved Taguchi methodology. Energy, 292:130474, 2024. Doi: https://doi.org/10.1016/j.energy.2024.130474
12. Lawrence KR, Anchupogu P, Reddygari MR, Gangula VR, Balasubramanian D, Veerasamy S. Optimization of biodiesel yield and performance investigations on diesel engine powered with hydrogen and acetylene gas injected with enriched Jojoba biodiesel blend. International Journal of Hydrogen Energy, 50:502-23, 2024. Doi: https://doi.org/10.1016/j.ijhydene.2023.09.166
13. Solmaz H, Calam A, Yılmaz E, Şahin F, Ardebili SMS, Aksoy F. Evaluation of MWCNT as fuel additive to diesel–biodiesel blend in a direct injection diesel engine. Biofuels, 14(2):147-56, 2023. Doi: https://doi.org/10.1080/17597269.2022.2122154
14. Uyaroğlu A, Uyumaz A, Çelikten İ. Comparison of the combustion, performance, and emission characteristics of inedible Crambe abyssinica biodiesel and edible hazelnut, corn, soybean, sunflower, and canola biodiesels. Environmental Progress & Sustainable Energy, 37(4):1438-47, 2018. Doi: https://doi.org/10.1002/ep.12794
15. Erol D, Yaman H, Doğan B, Yeşilyurt MK. The assessment of fusel oil in a compression-ignition engine in the perspective of the waste to energy concept: investigation of the performance, emissions, and combustion characteristics. Biofuels;13(10):1147-64, 2022. Doi: https://doi.org/10.1080/17597269.2022.2111779
16. Gürbüz H, Demirtürk S. Investigation of dual-fuel combustion by different port injection fuels (Neat Ethanol and E85) in a DE95 diesel/ethanol blend fueled compression ignition engine. Journal of Energy Resources Technology;142(12):122306, 2020. Doi: https://doi.org/10.1115/1.4047328
17. Doğan B, Erol D, Yaman H, Kodanli E. The effect of ethanol-gasoline blends on performance and exhaust emissions of a spark ignition engine through exergy analysis. Applied Thermal Engineering, 120:433-43, 2017. Doi: https://doi.org/10.1016/j.applthermaleng.2017.04.012
18. Adin MŞ, Altun Ş, Adin MŞ. Effect of using bioethanol as fuel on start-up and warm-up exhaust emissions from a diesel power generator. International Journal of Ambient Energy;43(1):5711-7, 2022. Doi: https://doi.org/10.1080/01430750.2021.19773

87
19. Uyar M, Aydın H. Production of low sulfur diesel-like fuel from crude oil wastes by pyrolytic distillation and its usage in a diesel engine. Energy, 244:122683, 2022.
Doi: https://doi.org/10.1016/j.energy.2021.12268
20. Rosha P, Dhir A, Mohapatra SK. Influence of gaseous fuel induction on the various engine characteristics of a dual fuel compression ignition engine: a review. Renewable and Sustainable Energy Reviews, 82:3333-49, 2018.
Doi: https://doi.org/10.1016/j.rser.2017.10.055
21. Wei L, Geng P. A review on natural gas/diesel dual fuel combustion, emissions and performance. Fuel Processing Technology, 142:264-78, 2016.
Doi: https://doi.org/10.1016/j.fuproc.2015.09.018
22. Park H, Shim E, Bae C. Improvement of combustion and emissions with exhaust gas recirculation in a natural gas-diesel dual-fuel premixed charge compression ignition engine at low load operations. Fuel, 235:763-74, 2019.
Doi: https://doi.org/10.1016/j.fuel.2018.08.045
23. Hernández JJ, Lapuerta M, Barba J. Separate effect of H2, CH4 and CO on diesel engine performance and emissions under partial diesel fuel replacement. Fuel, 165:173-84, 2016.
Doi: https://doi.org/10.1016/j.fuel.2015.10.054
24. Krishnan S, Biruduganti M, Mo Y, Bell S, Midkiff K. Performance and heat release analysis of a pilot-ignited natural gas engine. International Journal of Engine Research, 3(3):171-84, 2002.
Doi: https://doi.org/10.1243/14680870260189280
25. Papagiannakis R, Hountalas D. Experimental investigation concerning the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine. Applied Thermal Engineering, 23(3):353-65, 2003.
Doi: https://doi.org/10.1016/S1359-4311(02)00187-4
26. Di Blasio G, Belgiorno G, Beatrice C. Effects on performances, emissions and particle size distributions of a dual fuel (methane-diesel) light-duty engine varying the compression ratio. Applied energy, 204:726-40, 2017.
Doi: https://doi.org/10.1016/j.apenergy.2017.07.103
27. Chen Z, Wang L, Wang X, Chen H, Geng L, Gao N. Experimental study on the effect of water port injection on the combustion and emission characteristics of diesel/methane dual-fuel engines. Fuel, 312:122950, 2022.
Doi: https://doi.org/10.1016/j.fuel.2021.122950
28. Ouchikh S, Lounici M, Loubar K, Tarabet L, Tazerout M. Effect of diesel injection strategy on performance and emissions of CH4/diesel dual-fuel engine. Fuel, 308:121911, 2022.
Doi: https://doi.org/10.1016/j.fuel.2021.121911
29. Tripathi G, Nag S, Sharma P, Dhar A. Effect of methane supplementation on the performance, vibration and emissions characteristics of methane-diesel dual fuel engine. Frontiers in Thermal Engineering, 3:1101333, 2023.
Doi: https://doi.org/10.3389/fther.2023.1101333
30. Ahmad Z, Kaario O, Karimkashi S, Qiang C, Vuorinen V, Larmi M. Effects of ethane addition on diesel-methane dual-fuel combustion in a heavy-duty engine. Fuel, 289:119834, 2021.
Doi: https://doi.org/10.1016/j.fuel.2020.119834
31. Liu J, Guo Q, Guo J, Wang F. Optimization of a diesel/natural gas dual fuel engine under different diesel substitution ratios. Fuel, 305:121522, 2021.
Doi: https://doi.org/10.1016/j.fuel.2021.121522
32. Di Iorio S, Magno A, Mancaruso E, Vaglieco BM. Diesel/methane dual fuel strategy to improve environmental performance of energy power systems. International Journal of Heat and Technology, 34(2):581-8, 2016.
Doi: https://doi.org/10.18280/ijht.34S254
33. Khedkar ND, Sarangi AK, Sreedhara S. Impact of engine control variables on low load

combustion efficiency and exhaust emissions of a methane-diesel dual fuel engine. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 09544070231197613, 2023. Doi: https://doi.org/10.1177/09544070231197613

34. Zarrinkolah MT, Hosseini V. Methane slip reduction of conventional dual-fuel natural gas diesel engine using direct fuel injection management and alternative combustion modes. Fuel, 331:125775, 2023. Doi: https://doi.org/10.1016/j.fuel.2022.125775

35. Cameretti MC, De Robbio R, Palomba M. Numerical Analysis of Dual Fuel Combustion in a Medium Speed Marine Engine Supplied with Methane/Hydrogen Blends. Energies, 16(18):6651, 2023. Doi: https://doi.org/10.3390/en16186651

36. Yin C, Wang C, Shen X, Zhang Z. Experimental and simulation study of diesel/methane/hydrogen triple-fuel combustion progression in a heavy-duty optical engine. International Journal of Hydrogen Energy, 62:562-78, 2024. Doi: https://doi.org/10.1016/j.ijhydene.2023.10.198

37. Zhang X, Gao J, Fan D, Yang Q, Han F, Yu H. Impact of pilot diesel injection timing on performance and emission characteristics of marine natural gas/diesel dual-fuel engine. Scientific Reports, 14(1):10713, 2024. Doi: https://doi.org/10.1038/s41598-024-61672-5

38. Gülcan HE, Gültekin N, Ciniviz M. Experimental investigation of the effect of variable valve lift on combustion stability and exhaust emissions in a diesel/methane CRDI engine. Energy, 300:131603, 2024. Doi: https://doi.org/10.1016/j.energy.2024.131603

39. Zheng J, Wang J, Zhao Z, Wang D, Huang Z. Effect of equivalence ratio on combustion and emissions of a dual-fuel natural gas engine ignited with diesel. Applied Thermal Engineering, 146:738-51, 2019. Doi: https://doi.org/10.1016/j.applthermaleng.2018.10.045

40. Gürbüz H, Akçay İH. Evaluating the effects of boosting intake-air pressure on the performance and environmental-economic indicators in a hydrogen-fueled SI engine. International Journal of Hydrogen Energy, 46(56):28801-10, 2021. Doi: https://doi.org/10.1016/j.ijhydene.2021.06.099

41. Gürbüz H, Akçay H, Aldemir M, Akçay İH, Topalcı Ü. The effect of euro diesel-hydrogen dual fuel combustion on performance and environmental-economic indicators in a small UAV turbojet engine. Fuel, 306:121735, 2021. Doi: https://doi.org/10.1016/j.fuel.2021.121735

42. Gülcan HE. Effect of methane injection strategy on combustion, exergetic performance, and enviro-economic analyses in a diesel/methane CRDI engine. Applied Thermal Engineering, 243:122654, 2024. Doi: https://doi.org/10.1016/j.applthermaleng.2024.122654

43. Yuvenda D, Sudarmanta B, Wahjudi A, Muraza O. Improved combustion performances and lowered emissions of CNG-diesel dual fuel engine under low load by optimizing CNG injection parameters. Fuel, 269:117202, 2020. Doi: https://doi.org/10.1016/j.fuel.2020.117202

44. Bora BJ, Saha UK, Chatterjee S, Veer V. Effect of compression ratio on performance, combustion and emission characteristics of a dual fuel diesel engine run on raw biogas. Energy conversion and management, 87:1000-9, 2014. Doi: https://doi.org/10.1016/j.enconman.2014.07.080

45. Vijin Prabhu A, Manimaran R, Jeba P, Babu R. Effect of methane enrichment on the performance of a dual fuel CI engine. International Journal of Ambient Energy, 42(3):325-30, 2021. Doi: https://doi.org/10.1080/01430750.2018.1550017

46. Allouis C, De Robbio R, Mancaruso E, Vaglieco B. Effect of hydrogen and methane in dual fuel mode in light diesel engine by VIS

and IR imaging. International Journal of Hydrogen Energy, 52:1491-501, 2024. Doi: https://doi.org/10.1016/j.ijhydene.2023.08.163

47. Liu J, Yang F, Wang H, Ouyang M, Hao S. Effects of pilot fuel quantity on the emissions characteristics of a CNG/diesel dual fuel engine with optimized pilot injection timing. Applied Energy, 110:201-6, 2013. Doi: https://doi.org/10.1016/j.apenergy.2013.03.024