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Experimental Investigation into the Impact of Natural Gas-Diesel Mixture on Exhaust Emissions and Engine Performance in a Heavy-Duty Diesel Engine with Six Cylinders

Volkan Sabri Kül[1*], S. Orhan Akansu[2]

0000-0002-6412-6062, 0000-0002-0085-7915

[1*] *Mechanical Eng. Dep. Engineering Faculty, Erciyes University, 38039, Kayseri, Turkey*

[2] *Mechanical Eng. Dep. Engineering Faculty, Erciyes University, 38039, Kayseri, Turkey*

Abstract

In this study, experiments were conducted with a mixture of pure diesel and natural gas. In the experiments, a 6-cylinder heavy-duty diesel engine with an engine displacement of 11,670 cc was used and the engine speed was kept constant at 660 rpm. The 386 Nm torque value was accepted as 100% and experiments were performed at torque ratios of 25, 50, 75 and 100%. In the experiments conducted with natural gas -diesel dual fuel, natural gas was injected into the intake manifold with 1.5 bar pressure and 1.29 g/sec mass flow. The present study aimed to investigate the diesel-natural gas dual fuel combustion characteristics in heavy-duty diesel engines with high emission values, such as maritime transportation. In addition, the feasibility of natural gas as a secondary fuel in existing heavy-duty diesel engines without making significant structural changes was investigated. The findings obtained in this study showed that the BTE value of a mixture of natural gas-diesel fuel decreased by 157, 89, 53, 53 and 28% at 25, 50, 75, and 100 torque values, respectively, compared to pure diesel. At maximum load, CO and UHC emissions were 4.42 and 4.5 g/kWh for pure diesel and 19.9 and 11.9 g/kWh for a mixture of natural gas, respectively.

Keywords: Diesel; Natural gas; Dual fuel; Emission; Performance

Research Article

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* Corresponding author
Volkan Sabri Kül
volkanskul@gmail.com
Adress: Mechanical Eng. Dep. Engineering Faculty, Erciyes University, 38039, Kayseri, Turkey
Tel: +905340763976

1. Introduction

Today, reducing the consumption of fossil fuels and their polluting effects on the environment is a critical issue [1]. Diesel engines are internal combustion engines widely used, especially in transportation [1, 2]. Diesel engines are more efficient than other internal combustion engines but emit harmful emissions [3, 24]. According to the report of the US Energy Information Administration (EIA), a 50% increase in world energy demand is projected until 2050. It is stated that Asia has the largest share of this increase in energy demand [4]. The increasing need for energy and the desire to reduce harmful emissions push researchers to search for alternative fuels. The world's primary energy sources are oil, coal and natural gas [5]. Among these fuels, natural gas stands out as a suitable alternative fuel that can be used to reduce emissions due to its low C/H ratio [6, 7, 43]. Natural gas can be used as a good alternative fuel in diesel engines when proper combustion conditions are provided. It can be mixed with diesel and applied in dual fuel systems [8]. Christopher J. Ulishney et al. studied a dual-fuel engine running on natural gas and diesel at low load conditions. In their study, part of the diesel fuel was substituted with ethane, methane and propane and no EGR was used. It was observed that pure methane significantly affected the ignition delay. They reported that the best efficiency was obtained in propane-containing mixed, while the opposite was true for methane [9]. Wojciech Tutak et al. used hydrogen-enriched natural gas in combination with diesel fuel in a dual-fuel engine. Natural gas was supplied by a CNG (Compressed Natural Gas) storage system. In the experiments, diesel fuel was substituted with hydrogen, keeping the natural gas ratio constant. The experiments were performed at a constant engine speed of 1500 rpm. It was reported that the combustion time of the hydrogen-enriched natural gas-diesel mixture provided better results than non- hydrogen-enriched natural gas. In addition, no increase in UHC and NO_x emissions was observed [10]. Subrata Bhowmik et al. used ethanol addition to diesel fuel as a pilot fuel. The engine performance and emission values of this ethanol-blended diesel pilot fuel and natural gas mixture at different rates were investigated. It was reported that the efficiency increased with a 5% ethanol blend in the pilot fuel and 0.02 kg/h natural gas flow from the intake manifold. NO_x and CO emissions decreased under the same conditions [11].

Pedrozu et al. [33] investigated dual-fuel combustion in a single-cylinder compression ignition heavy-duty diesel engine at 0.6, 1.2 and 1.8 Mpa engine load (IMEP) with natural gas containing 80.9, 87.6 and 94.1% methane. In their study, they reported that as the methane content in natural gas decreased from 94.1% to 80.9%, NO_x emissions increased, but UHC decreased. Additionally, they stated that they achieved higher efficiency at lower methane rates. Felayati et al. [34] stated that diesel-natural gas dual-fuel combustion is more environmentally friendly than monodiesel fuel in terms of NO_x and soot emissions, but has negative effects concerning efficiency, carbon monoxide and unburned hydrocarbon emissions. They aimed to stratify the air and natural gas mixture with a split natural gas injection system to improve engine efficiency and emissions in dual fuel combustion. They stated that keeping the natural gas injection advance longer in the split injection strategy increases engine performance but negatively affects emissions. They stated that this strategy has the potential to improve engine efficiency and emissions if the split injection ratio is optimized. Armin and Gholinia [35] investigated the engine performance at different compression ratios, staged injection systems and different injection times in a heavy-duty diesel engine with Reactivity Controlled Compression Ignition. They reported that as the compression ratio increases, NO_x emissions worsen, while HC and CO emissions decrease. They stated that delayed combustion occurs as the injection start time approaches the top dead center (from -50 CA to -30 CA). They reported that as the injection start time approaches the top dead center, the in-cylinder pressure decreases and the HRR decreases accordingly. Park et al. [36] operated a heavy-duty natural gas engine with an engine volume of 11,000 cc, using natural gas fuels with different calorific values, without restricting the air intake. In their studies, they carried out full load tests using fuels with calorific values of 9,800, 10,200 and 11,000 kcal/Nm3 and evaluated the engine performance. They reported that the engine torque increased depending on the increase in calorific value. They stated that while the least thermal efficiency was seen in the fuel with the lowest calorific value, the fuel with 10,200 kcal/Nm3 calorific value (excluding 2000 rpm engine speed) reached the highest efficiency. They also reported that as the calorific value increases, CO_2 and CO emissions increase, while HC emissions decrease. Zhang et al. [37] aimed to improve engine efficiency and emissions by using different amounts of natural gas in a diesel-fuelled marine engine. They stated that as the ratio of natural gas and biodiesel increases, the start of combustion is delayed and natural gas has a greater effect on this. They also reported that as the proportion of natural gas increases, NO_x emissions decrease, while unburned methane emissions increase. Liu et al. [38] aimed to optimize the diesel-natural gas fuel ratio and combustion data in a dual-fuel diesel engine. According to their results, as the diesel pilot fuel ratio increased, the unburned methane ratio decreased. Additionally, NO_x emissions were stated as 6.14, 16.71, 3.24 and 8.38 g/kWh at the injection time -31.6, -25, -19.4 and -19 CA, respectively. Although there is an unstable variability in NO_x emissions depending on the diesel-natural gas fuel ratio, it can be said that NO_x emissions increase as the pilot fuel ratio increases. Zheng et al. [39] evaluated the emissions of heavy-duty vehicle engines running on diesel, CNG and LNG using a portable emission meter. They observed that the CO and NO_x emissions of heavy-duty engines running on CNG are higher than those of both LNG and diesel engines. They stated that the HC emissions of heavy-duty engines running on LNG are higher than both CNG and diesel engines. Thus, they stated that heavy-duty vehicles running on LNG and CNG are not suitable to replace diesel vehicles and cannot be described as more environmentally friendly than diesel vehicles. Shim et al. [41] compared conventional homogeneous-charged compression combustion, premixed-charged combustion, and dual-fuel premixed-charged combustion to simultaneously reduce NO_x and soot emissions. They stated that these three combustion technologies can reduce NO_x and soot emissions equally. They reported that since single-fuel advanced combustion technologies cannot provide complete combustion, they have HC and CO emissions that are 38 times higher than conventional diesel combustion. Accordingly, they reported that the thermal efficiency decreased. They observed that HC emissions were 7.8 times higher in dual-fuel premix technology compared to conventional diesel combustion. They emphasized that dual-fuel premix-filled technology has efficiency problems and that this issue should be emphasized in future studies. Deng et al. [42] investigated different emission reduction technologies and the effects of using LNG in these engines to reduce the emissions of ship engines used in maritime transportation. They stated that emission reduction methods (e.g., exhaust after-treatment system) could significantly reduce NO_x and SO_x emissions. However, these methods increase fuel consumption, take up a lot of space on the ship and cost high costs. It is thought that using LNG as fuel in ships is a good solution to reduce the harmful emissions produced by ship engines. However, they emphasize that LNG's storage difficulties, consuming a lot of space and the low thermal efficiency of LNG-fueled engines are challenges that must be overcome. They recommend that studies be conducted in the future that will contribute to overcoming this challenge. Liu et al. [43] designed a heavy-duty compression ignition diesel engine with spark ignition to test it under stoichiometric and lean combustion conditions. In their study, they examined engine performance and emissions in two different combustion conditions without using EGR. While the in-cylinder pressure decreased by 10-15% under poor operating conditions, the volumetric efficiency increased from 72% to 74%. The lean mixture also increased ignition delay. Therefore, they reported that CO emissions decreased and thermal efficiency increased from 35% to 37%. They stated that the equivalence ratio significantly affects the emission rates, and that as the equivalence ratio increases, emissions increase. They reported that spark plug ignition provided positive results under natural gas operating conditions, regardless of the equivalence ratio. Ulishney and Dumitrescu [44] argue that positive results in terms of efficiency can be achieved by using a natural gas-diesel dual fuel system in vehicles operating in natural gas production areas. Therefore, they examined the performance and emission changes of a natural gas-diesel dual-fuel engine with different gas components under low load conditions. They replaced some diesel fuel with 100% methane, 90% methane - 10% ethane, 90% methane - 5% Propane - 5% ethane and 90% methane - 10% propane gas fuels as secondary fuel in a diesel engine. They did not use EGR in their experiments. 100% methane substitution caused the longest ignition delay. It was observed that the in-cylinder pressure was highest with 90% methane - 10% propane substitution and lowest with 100% methane substitution. They reported that 100% methane replacement had the lowest performance. They suggested adding propane and ethane to methane to achieve better efficiency in dual-fuel engines. They reported that while CO_2 and NO_x emissions decreased in dual-fuel combustion, CO and HC emissions increased. Yousefi et al. [45] investigated the improvement of engine knocking and thermal efficiency with split and single diesel injection in a dual-fuel engine. They reported that in a single diesel injection, when the injection start time (SOI) was moved away from TDC (-10.5, -14 and -15 ATDC), the pressure in the cylinder increased, but the knocking worsened. They reported that flame sparks in split diesel injection are slower and less homogeneous than in single diesel injection. They stated that as the injection start time in single diesel injection was increased from -10.5 to -15 ATDC, efficiency increased, but NO_x emissions worsened. Jamrozik et al. [46] examined the effects of using natural gas-diesel dual fuel in a compression ignition engine on engine performance and emissions. In their studies, the natural gas substitution rate varies from 0 to 90%. They reported that increasing the replacement rate of diesel fuel with natural gas (especially at rates higher than the 45% natural gas replacement rate) increased the ignition delay and shortened the combustion time. As a result, they stated that the maximum in-cylinder pressure and heat release rate decreased. They reported that increasing the natural gas ratio from 0% to 90% caused the NO_x ratio to increase. Zhou et al. [14] examined the change in soot emissions in different ejection strategies and different natural gas substitution rates. They used a single-cylinder diesel engine in their study. They conducted tests at 0 – 40 - 70% natural gas replacement rates. They observed that soot emissions decreased as the natural gas substitution rate increased in both single and split injection strategies. Appswamy et al. [40] examined the performance changes at different natural gas ratios in an engine running on natural gas and diesel dual fuel. Engine thermal efficiency reached the highest values when the natural gas replacement rate was lowest. They reported that NO_x and soot emissions decreased at higher natural gas substitution rates.

In many studies in the literature, it has been stated that natural gas substitution reduces NO_x and soot emissions while causing an increase in HC and CO emissions. In addition, the thermal efficiency of natural gas-fueled combustion has generally remained low, and it is recommended that future studies be carried out to increase the efficiency of engines running on premixed natural gas fuel [41, 42]. It is thought that using natural gas as a secondary fuel in heavy-duty diesel engines without making major structural changes can provide positive results [44]. In this study, some of the diesel fuel was replaced with natural gas in a heavy-duty diesel engine and the engine performance and emission characteristics of natural gas substitution were investigated without making too many structural changes in the engine. In the present study, experiments were carried out with pure diesel and natural gas mixtures at a constant engine speed of 660 rpm and torque ratios of 25, 50, 75 and 100% (maximum torque 386 Nm). In the experiments with the natural gas mixture, natural gas was delivered to the cylinder premixed with air from the engine intake manifold at a constant flow rate of 1.29 g/sec. The natural gas supplied through the intake manifold replaces part of the air and slightly reduces the intake-air ratio. During the experiments, the air intake of the engine was not restricted, and maximum air intake was allowed. Experiments were conducted with pure diesel and natural gas mixtures at different torque values, and aimed to investigate the engine performance and emission values of a heavy-duty diesel engine.

2. Procedure of This Study

2.1 The Test Setup and Materials

The experiments in this study were conducted at Erciyes University motors laboratory. The experimental setup for the experiments was set up as shown in Figure 1. As seen in the experimental setup diagram, the natural gas used in the natural gas mixture experiments was supplied from the CNG tank. The natural gas in CNG form at 200 bar pressure was reduced to 1.5 bar pressure with the help of a regulator. It was then passed through a flow meter and delivered to the engine intake manifold. The diesel container was positioned on a precision scale, and the consumption was measured with a precision scale. Test data were obtained with an in-cylinder pressure measurement system, dynamometer and exhaust analyzers.

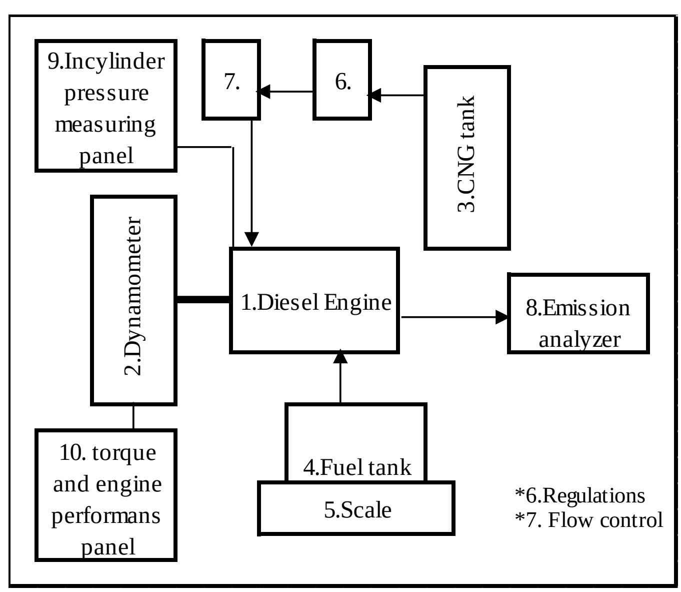

Fig. 1. The experimental setup

Figure 2 shows the diesel engine used as the test engine. The test engine has the same technical specifications as the engine in [17]. The test motor transmission shaft and eddy current are connected to the dynamometer. The highest torque, power and speed of the dynamometer are 600 Nm, 150 kW and 12.000 rpm, respectively.


Fig. 2. Test engine

For the experiments with natural gas-diesel mixture fuel, the natural gas in the CNG tank at 200 bar pressure was reduced to 1.5 bar with a regulator. For this pressure reduction process, the “ALKAN 840” brand and model regulator, whose inlet pressure is max. 230 bar and outlet pressure can be adjusted in the range of 0-10, was used. A heat exchanger was used to prevent overcooling during the drop from 200 bar pressure to 1.5 bar pressure. By circulating the water heated by the 7.5 kW resistance with a circulation motor, the gas temperature was tried to be kept at 15ºC - 20ºC. This whole setup is shown in Figure 3.


Fig. 3. CNG storage, regulation and heat exchanger system

At 1.5 bar pressure, the “ALICAT” brand flow controller and precision control valve shown in Figure 4 were used to control the flow rate of natural gas delivered to the engine intake manifold. Natural gas was injected into the intake manifold at a constant rate of 1.29 g/s.


Fig. 4. Natural gas flow control unit

The Desis-KF-H2W precision balance in Figure 5 was used to measure diesel fuel consumption. The sensitivity of the scale is 0.5 grams. Every 5 minutes, diesel consumption was monitored. The properties of natural gas and pure diesel used as fuel in the experiments [18] are given in Table 1.

Table 1. Fuels Properties

Contents	Natural Gas	Diesel	Units
Methane	94.3054	-	%
Ethane	3.2621	-	%
Propane	1.2214	-	%
I-Butane	0.2451	-	%
N-Butane	0.3326	-	%
I-Pentane	0.1005	-	%
N-Pentane	0.0785	-	%
Hexane	0.0662	-	%
N_2	0.2042	-	%
CO_2	0.1840	-	%
Higher Heating Value (HHV)	54.5	-	MJ/kg
Lower Heating Value (LHV)	49.2	42.98	MJ/kg
Density	0.734	820	kg/sm^3, kg/m^3
Wobbe Index	63.57	-	MJ/kg
Cetane Number	-	52	-
Octane Number	120-125	-	-
Flash point	-188.5	56	°C
Auto-ignition Temperature	539-640	186–230	°C
Viscosity	-	2.2	mm^2 /s

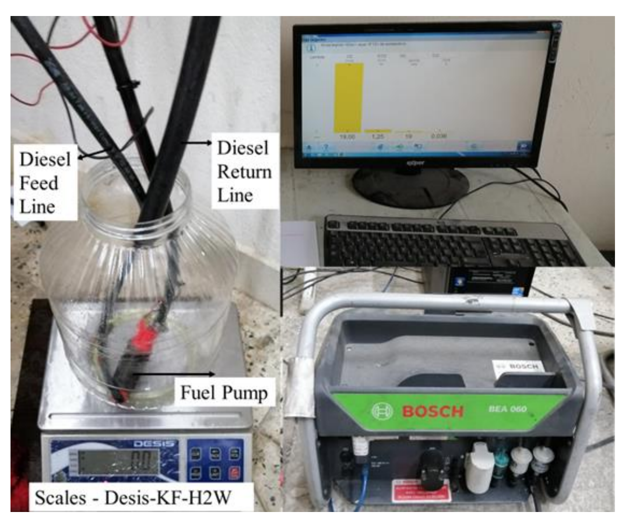

Fig. 5. Precision balance and fuel tank (left). Exhaust analyzer (right)

Bosch BEA 060 brand emission analyzer was used to measure exhaust emission values (Figure 5). A piezoelectric sensor was used for in-cylinder pressure measurement and the data was recorded on the control panel.

2.2 Methods

The experiments were carried out with pure diesel and natural gas mixtures at four different torque ratios. First, the dynamometer load value was gradually increased to determine the maximum torque value at a constant engine speed of 660 rpm. At 660 rpm, a maximum torque value of 386 Nm was reached and this torque value was taken as 100%. Then, experiments were carried out using torque ratios of 25, 50, 75, and 100%. After the tests with pure diesel were completed, the tests were started at constant natural gas flow. Natural gas was supplied to the cylinder using the intake manifold, pre-mixed with air, at a constant rate of 1.29 g/s. As the torque value increased, the engine speed was kept constant by changing/increasing the diesel fuel ratio. At 25, 50, 75, and 100% torque ratios, diesel was substituted with natural gas at 87, 81, 73, and 65%, respectively. Due to the increased torque, the proportion of diesel fuel in the mixture increased, and the natural gas substitution decreased proportionally. However, the natural gas flow in terms of mass flow rate was kept constant at 1.29 g/sec. During the experiments, the engine air intake was not restricted, and maximum air intake was allowed. Experiments were conducted with pure diesel, natural gas-diesel mixture and engine performance and exhaust emission values were analyzed. The coding of different torques and fuel mixtures used in the experiments are shown in Table 2.

Table 2. Naming the fuel type and torque values used in the experiments.

Fuel code	Explanation
Diesel 25%	pure diesel and 25% torque
Diesel 50%	pure diesel and 50% torque
Diesel 75%	pure diesel and 75% torque
Diesel 100%	pure diesel and 100% torque
NGDiesel 25%	diesel+1.29 g/sec CNG 25% torque
NGDiesel 50%	diesel+1.29 g/sec CNG 50% torque
NGDiesel 75%	diesel+1.29 g/sec CNG 75% torque
NGDiesel 100%	diesel+1.29 g/sec CNG 100% torque

To measure exhaust emissions, the pipe coming from the emission analyzer was connected to the exhaust pipe, allowing the exhaust gas to reach the analyzer. Measurements were made and recorded with the BEA 060 Bosch brand emission analyzer and the BEA 070 turbidity analyzer, as shown in Figure 6.

Fig. 6. Emissions measurement

3. Results and Discussion

3.1 Break Thermal Efficiency

Brake thermal efficiency (BTE) is a very important performance criterion in internal combustion engines [12]. It is an important indicator of the conversion rate of the fuel mass injected into the cylinder into work [13]. The ratio of the power produced by the engine to the energy of the fuel gives the thermal efficiency of the brake. The equation used in the calculations is shown in equation (1).

$$\boldsymbol{BTE} = \frac{\boldsymbol{P_b}}{\dot{m}_f \boldsymbol{xLHV}} \quad (\%) \tag{1}$$

Equation (2) was used to calculate the lower heat value of fuel mixtures.

$$LHV_k = \frac{m_{ng}xLHV_{ng} + m_d xLHV_d}{m_{ng} + m_d} \tag{2}$$

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} xPx\frac{dV}{d\theta} + \frac{1}{\gamma - 1} xVx\frac{dP}{d\theta} \tag{3}$$

Heat release rates were computed according to equation 3.

The BTE values and graphs obtained from the experiments are shown in Table 3 and Figure 7, respectively. The BTE values of natural gas mixture fuels were 157, 89, 53 and 28% lower than the BTE values obtained from pure diesel fuel at 25, 50, 75, and 100% torque ratios, respectively. Auto-ignition temperature is one of the most important factors affecting BTE. As the methane content in the in-cylinder fuel mixing increased, the auto-ignition temperature of the mixture increased and the combustion time shortened. For this reason, in experiments conducted with a natural gas mixture, it was observed that BTE decreased compared to pure diesel. As the engine torque value increased, the pilot diesel fuel ratio increased and the natural gas substitution ratio in the mixture decreased. An increase in BTE was observed due to the decrease in natural gas replacement rate with increasing torque. A better combustion occurred because the auto-ignition temperature of the filling in the cylinder decreased. These results are consistent with the results of Deng et al. [42], Ulishney and Dumitrecu [44], and Jamrozik et al. [46]. As a result of the increase in the natural gas ratio, the auto-ignition temperature increased and led to a short combustion time. Due to this situation, the decrease in HRR, along with the decreasing in-cylinder pressure, was effective in the decrease in BTE. This situation is mentioned in detail in the following paragraph.

Table 3. Brake Thermal Efficiency values (%)

	Diesel BTE values	NGDiesel BTE values	Diesel-NGDiesel BTE differences	Efficiency ratio of Diesel compared to NGDiesel
Torque 25%	24,11	9,37	14,74	157
Torque 50%	32,51	17,21	15,3	89
Torque 75%	36,56	23,86	12,7	53
Torque 100%	36,93	28,77	8,16	28

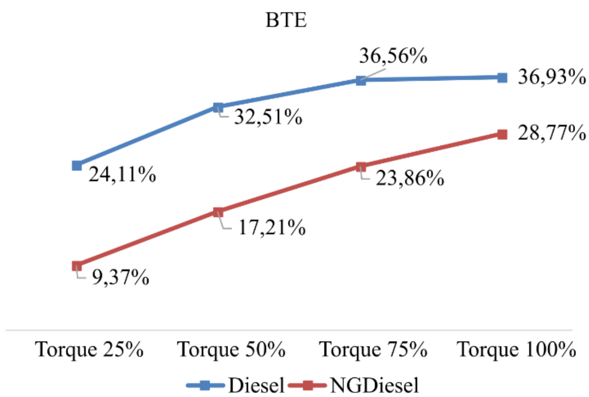

Fig. 7. Brake Thermal Efficiency (BTE)

In-cylinder pressure is the most important parameter for thermal efficiency, engine performance measurements and combustion analysis [26, 27]. The maximum in-cylinder pressure reached when the test engine is operated with a mixture of natural gas at all torque values (Figure 8) is lower than the maximum in-cylinder pressure reached when using pure diesel fuel. The maximum in-cylinder pressure values obtained in the experiments with a mixture of natural gas were 2.61, 3.75, 5.32, and 3.33 bar lower than the in-cylinder pressures obtained in the experiments with pure diesel fuel at 25, 50, 75, and 100% torque ratios, respectively. The decrease in in-cylinder pressure affected the BTE and the heat release rate (HRR), as shown in Figure 8. The maximum HRR values reached in the experiments with pure diesel are 72.99, 84.19, 106.27, and 121.88 J/ºCA, while the maximum HRR values reached in the diesel-natural gas mixture are 64.47, 65.19, 71.35, 82.7 J/ºCA, respectively. Since the auto-ignition temperature of natural gas is higher than that of pure diesel fuel as seen in Table 1, the auto-ignition temperature of the mixture increased. This situation made ignition difficult and caused retarded combustion and slow combustion. In addition, natural gas is more resistant to combustion than diesel owing to its high-octane number [8, 15]. In the experiments conducted with a natural gas mixture, the high-octane number of natural gas made combustion difficult, resulting in delayed and slow combustion [14-16]. In the experiments conducted with a mixture of natural gas-diesel, the fact that the HRR curve between 360 - 380 CA is flatter and has more than one peak compared to pure diesel indicates that there is slow combustion; combustion is not complete. Even flame extinction may have occurred in places. As can be seen in Figure 14, the fact that the UHC emission of a mixture of natural gas-diesel fuels is higher than that of pure diesel at all load conditions confirms that there is no complete combustion, and therefore, unburned fuel is discharged from the exhaust. In addition, CO emission is higher than pure diesel at all load conditions in natural gas-diesel mixture experiments (Figure 12), indicating that not whole fuel injected into the cylinder burns and causes CO emission to increase. In addition, since the air intake was not restricted in the experiments and maximum air intake was allowed, it is predicted that an excessively lean mixture was formed in the cylinder. O_2 emissions are 20.4%, 20.05, 19.5, 18.8 for pure diesel and 20.4%, 19.9, 19.3, and 18.6 for dual fuel at 25, 50, 75, and 100% torque ratios, respectively. The high O_2 emissions support the prediction of an extremely lean mixture in the cylinder. Especially in the experiments with natural gas, it is thought that the combination of two negative parameters, such as the high-octane number of natural gas making combustion difficult and excessive air intake into the cylinder caused flame extinction in the cylinder. These conditions explain the decrease in the BTE value of the experiments with natural gas. Jinwen You et al. [31] reported that the highest flame spread rate of methane was realized at λ=0.9. In this study, it is thought that if the air concentration in the cylinder is reduced and the lambda is brought to these levels, there will be an improvement in BTE.

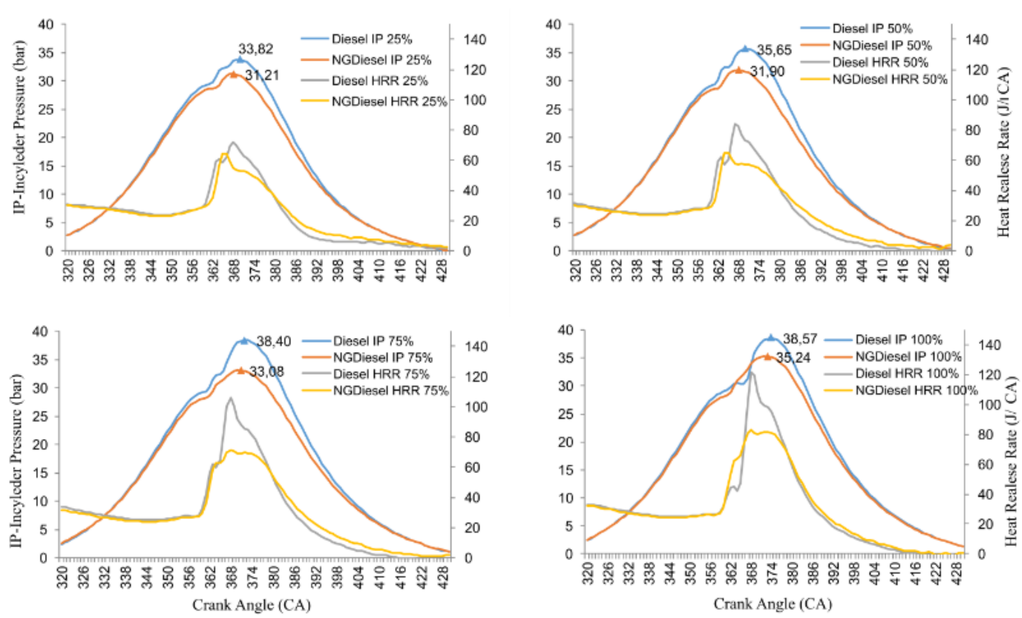

Fig. 8. In-cylinder Pressure and Heat Release Rate

In the previous paragraph, it is stated that when natural gas is added to diesel fuel, the in-cylinder pressure decreases, and consequently, both the BTE and the heat release rate decrease. Important parameters affecting the change in in-cylinder pressure are ignition delay, pilot fuel injection time and the amount of oxygen in the combustion chamber (air/fuel ratio) [25]. In the experiments with dual fuel in the present study, the flow of natural gas premixed through the intake manifold was kept constant at 1.29 g/s. The fuel energy required by the engine due to the increasing torque ratio was met by increasing the diesel fuel ratio. When the test engine was operated with dual fuel, diesel fuel was substituted with 87%, 81%, 73%, and 65% natural gas at 25, 50, 75, and 100% torque ratios, respectively. Natural gas is more combustion-resistant than diesel fuel because it has a high auto-ignition temperature and high octane number. Therefore, SOI (start of injection) should be earlier. In this study, since SOI was kept constant at 16 CA BTDC, which was set for pure diesel fuel, delayed and slow combustion occurred in a mixture of natural gas. As a result, the thermal efficiency decreased compared to pure diesel. This situation is parallel to the studies of Felayati et al. [34], Armin & Gholinia [35], and Yousefi et al. [45]. The region indicated by the red ring in Figure 10 is the after (late) combustion zone. In this region, the HRR value of the mixture of natural gas is higher than diesel fuel. This confirms the delayed and slow combustion. To understand how the delayed combustion changes with decreasing natural gas substitution rate, the crank angle values at which pure diesel and a mixture of natural gas-diesel fuel reach the HRR of 40 J/ºCA shown in Figure 9 were analyzed. As indicated in Figure 9, pure diesel fuel reached 40 J/ºCA HRR at 360.1919 CA at 25% torque - 87% natural gas substitution ratio. It decreased to 360.4971 crank angle at 50% torque - 81% natural gas substitution ratio. It then increased to 360.7146 and 361.2889 CA at 75% torque-73% natural gas substitution and 100% torque - 65% natural gas substitution, respectively. It can be said that the crank angle curve of pure diesel fuel reaching 40 J/ºCA HRR shows a horizontal trend. However, the crank angle to reach the same HRR value of the mixture of natural gas-diesel fuel decreased continuously with increasing torque and decreasing natural gas substitution ratio. The crank angle to reach 40 J/ºCA HRR was obtained as 362.5503, 361.7788, 361.3231, 360.9436 CA at 25, 50, 75, and 100% torque and 87, 81, 73 and 65% natural gas substitution ratios, respectively.

The ignition delay is the time from the start of injection to the ignition point. During the ignition delay, starting with injection, the fuel vaporizes, mixes with air and reaches the ignition point [28]. Natural gas is more resistant to combustion than diesel owing to its higher octane number. Therefore, it needs a longer ignition delay. To extend the ignition delay, the injection start time should be earlier [28]. Zhongshu Wang et al. [29] conducted experiments with a mixture of natural gas-diesel by varying the injection start times between -5º and 55º CA BTDC. They showed that the ignition delay increased when the injection start time was increased from -5º CA BTDC to 55º CA BTDC in dual fuel experiments. They stated that the thermal efficiency increased with increasing ignition delay and the most ideal injection start time was 42.5º CA BTDC. In another study, it was stated that as the injection start time is advanced (i.e., earlier than TDC), in-cylinder pressure increases and thermal efficiency increases [30]. The injection start time of the test engine used in this study is 16 CA BTDC. As in similar studies, it is thought that if the injection advance is increased as the natural gas substitution rate in the mixture increases, i.e., if it is moved earlier, an improvement in the BTE ratio will be achieved. In addition, if additives with high vapor pressure, such as acetone, are added to the diesel pilot fuel, faster vaporization of the pilot fuel can be achieved [32]. It is thought that the doped pilot fuel, which has a faster evaporation rate than pure diesel, can contribute to the ignition of natural gas more quickly and increase thermal efficiency.

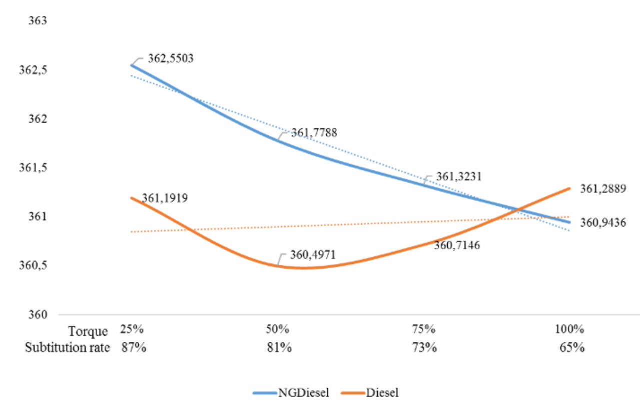

Fig. 9. Crank angle at 40 J/ºCA HRR

In addition, as shown in Figure 10, the distances a, b, c, d between pure diesel and dual fuel HRR curves calculated according to the points at 40 J/ºCA were 1.3584, 1.2817, 0.6085, and 0.3553 CA, respectively. Since delayed combustion decreases steadily in dual fuel due to increasing torque ratio and decreasing amount of natural gas in the mixture, the heat release rate (HRR) curves of the mixture of natural gas-diesel fuel approach those of pure diesel fuel. Based on these analyses, it was concluded that the delayed combustion decreased with decreasing natural gas substitution rate in the mixture and increasing torque value. Especially in the experiments with dual fuel, slow combustion and delayed combustion decreased, combustion timing improved, in-cylinder pressure increased and BTE and HRR increased accordingly.

3.2 Brake-Specific Fuel Consumption

Brake-specific fuel consumption (BSFC) is an important parameter that indicates the fuel efficiency of an engine. Brake-specific fuel consumption refers to the amount of fuel consumed per power produced in internal combustion engines [8]. In this study, calculations were made according to equation 4: Pb represents brake power, ṁf represents fuel flow rate.

$$bsfc = \frac{\dot{m}f}{P_b} \quad (g/kW.h) \qquad (4)$$

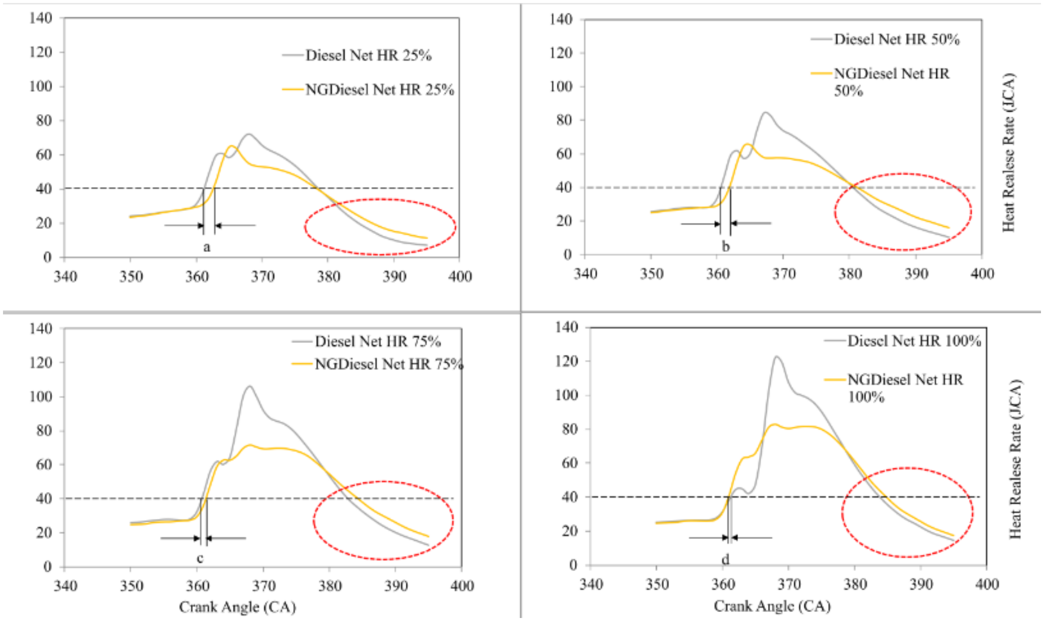

Fig. 10. Heat Release Rate

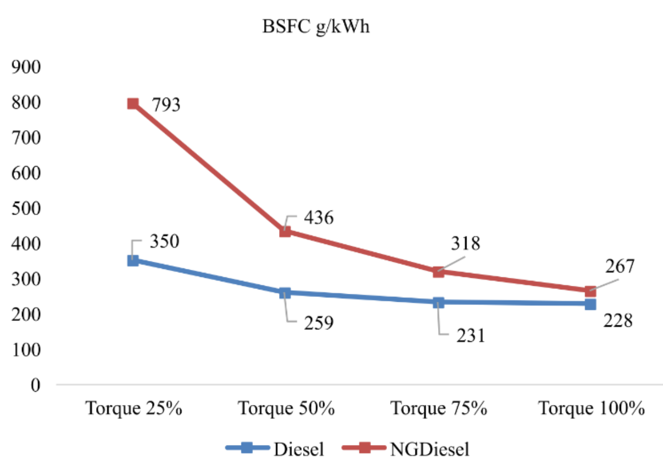

Fig. 11. Brake-Specific Fuel Consumption (BSFC)

Table 4. Brake-specific fuel consumption values (g/kWh)

Torque	Diesel	NGDiesel	Less fuel consumption rate of diesel compared to NG diesel (%)
25%	350	793	127
50%	259	436	68
75%	231	318	38
100%	228	267	17

The results obtained in the experiments are presented in Table 4 and Figure 11. The BSFC value of the mixture of natural gas is 127, 68, 38, and 17% higher than pure diesel fuel at 25, 50, 75, and 100% torque ratios, respectively. In other words, it is seen that the fuel economy with a mixture of natural gas deteriorates by 127, 68, 38, and 17% at 25, 50, 75, and 100% torque ratios, respectively, compared to pure diesel fuel. One of the reasons for the low BTE values of fuels, with a mixture of natural gas is the high fuel energy consumption per unit of power. The start of injection time is an important element affecting the specific fuel consumption of the brake. If the fuel injection (SOI-start of injection) is not at the optimum value, engine efficiency and fuel consumption will deteriorate. Thus, not all fuels injected into the cylinder can burn and BSFC increases [17]. As mentioned in detail in the previous section, it is thought that if the pilot SOI is moved earlier than 16 CA BTDC and the ignition delay is increased, it is possible to increase the efficiency in dual fuel experiments.

3.3 Brake-Specific Fuel Consumption

The combustion of hydrocarbon fuels releases harmful emissions, such as CO, CO_2, NO_x, PM, and SO_x, to the environment and human health. These emissions are caused by the lack of ideal combustion in the combustion chamber, pollutants in the fuels and the decomposition of nitrogen due to high temperatures [19, 20]. Researchers are constantly working to reduce these emissions that threaten human and environmental health. In this study, the effect of the mixture of natural gas on exhaust emissions was experimentally investigated.

CO emission occurs when there is incomplete combustion. CO molecules cannot be converted to CO_2 due to the lack of oxygen or low-temperature combustion and are discharged from the exhaust as CO [19,21]. The carbon monoxide emission amounts obtained in the experiments are given in Figure 12. Diesel CO amounts were lower than fuels with a mixture of natural gas. Since the BTE values of pure diesel are higher, better combustion was realized in pure diesel operation and CO rates decreased. A decrease in CO values was observed due to the increase in thermal efficiency. The decrease in CO emission indicates a decrease in incomplete combustion. The higher BTE value of pure diesel fuel compared to fuels which is a mixture of natural gas, explains this situation.

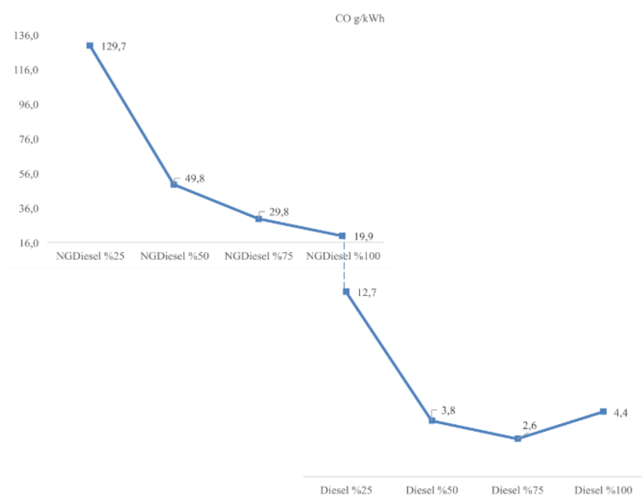

Fig. 12. Carbon monoxide (CO)

The combustion temperature in the combustion chamber and the length of the combustion time are effective in forming NO_x emission. At high temperatures, N_2 and O_2 molecules react by decomposing into atoms, and NO_x emission occurs as a result of this reaction [22, 23]. NO_x values obtained in the experiments are given in Figure 13. While no significant difference was observed in all torque values of pure diesel fuel, NO_x values of fuels with a mixture of natural gas increased with the increase in torque value. With increasing torque and decreasing natural gas substitution ratio in fuels with a mixture of natural gas, in-cylinder pressure increased, combustion tended to improve and incomplete combustion decreased. Increased in-cylinder pressure increased the in-cylinder temperature and increased the NO_x content. At low loads in fuels with a mixture of natural gas, incomplete combustion due to a high natural gas substitution ratio resulted in low NO_x emissions. As the load increased and the natural gas substitution ratio in the mixture decreased, the combustion improved. These results are parallel to the studies of Pedrozo et al. [33] and Zhang et al. [37]. Therefore, it is considered that the in-cylinder temperature increased and NO_x emission increased due to the increased in-cylinder temperature.

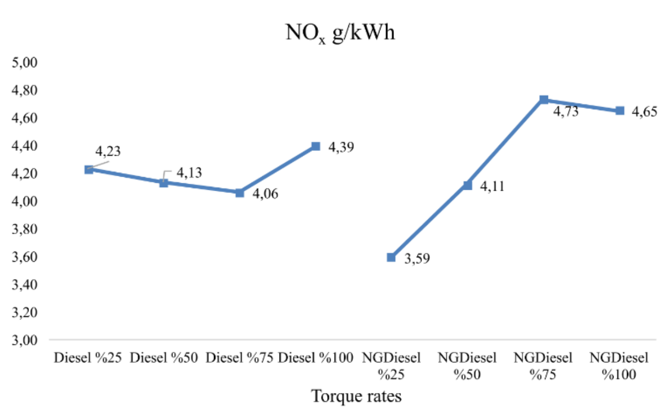

Fig. 13. Nitrogen oxides (NO_x)

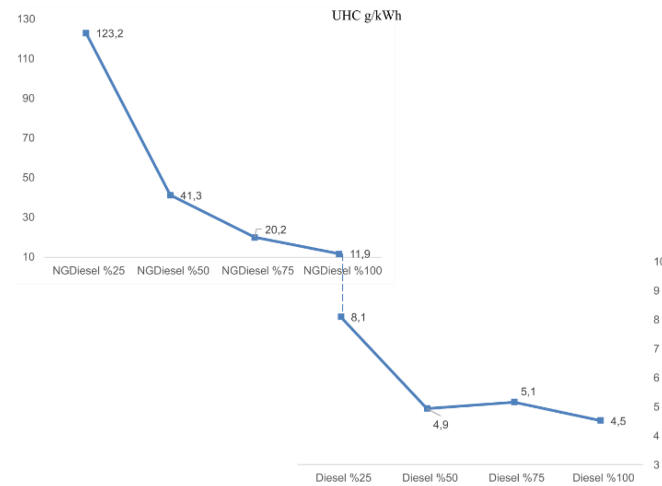

Fig. 14. Unburned hydrocarbon (UHC)

The most effective factor in the formation of unburned hydrocarbon emissions is the air-fuel mixture ratio. Since there is not enough oxygen in rich mixtures, not all fuel can be burned and UHC emissions occur. In lean mixtures, high levels of UHC emissions occur due to the deterioration of combustion and flame extinction [19]. The UHC emissions obtained in the experiments are presented in Figure 14. As can be seen from the graph, the highest UHC emission was realized at low load and the highest natural gas substitution rate. Then, with increasing torque value and decreasing natural gas substitution ratio in the mixture, UHC emission tends to decrease. This is because with increasing torque and decreasing natural gas substitution ratio, the combustion in the cylinder improved compared to low loads and more fuel participated in the combustion in the cylinder, increasing the in-cylinder pressure. Accordingly, the BTE value increased. The decrease in UHC emission with increasing torque and decreasing natural gas substitution ratio confirms each other with the BTE curve. Moreover, the results are consistent with Liu et al. [38], Zheng et al. [39] and Shim et al. [41].

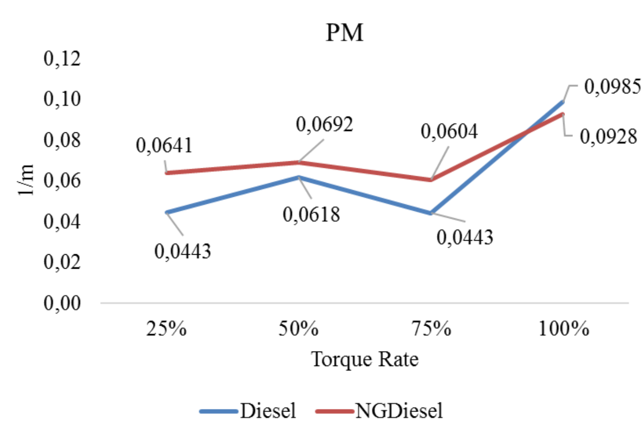

Fig. 15. Particulate Matter

In this study, at 25, 50, 75, and 100% torque ratios, diesel was substituted with natural gas at 87, 81, 73, and 65%, respectively.

In experiments conducted with natural gas mixed fuels, it was observed that PM was at a relatively higher level than diesel fuel at 25, 50, and 75% torque values (Fig. 15). The reason for this is that at high natural gas replacement rate, combustion deteriorates and complete combustion cannot occur. At 100% torque, where natural gas substitution is minimal, the amount of PM was relatively lower than pure diesel. Results Zhou et al. It is compatible with the studies of [14].

4. Conclusion

The present study aimed to investigate the diesel-natural gas dual fuel combustion characteristics in heavy-duty diesel engines with high emission values, such as maritime transportation. In addition, the feasibility of natural gas as a secondary fuel in existing heavy-duty diesel engines without making significant structural changes was investigated. In this study, experiments were conducted at 25, 50, 75, and 100% torque values using natural gas-diesel mixtures and pure diesel fuel. The diesel engine, which has six cylinders, was used as the test engine, and engine performance and emission values were analyzed.

Break-specific thermal efficiency: In experiments conducted with pure diesel fuel, thermal efficiencies of 24.11, 32.51, 36.56 and 36.93% were obtained at 25, 50, 75 and 100% torque values, respectively. However, in natural gas mixed experiments, thermal efficiencies of 9.37, 17.21, 23.86 and 28.72% were observed at the same torque values, respectively. In this study, at 25, 50, 75, and 100% torque ratios, diesel was substituted with natural gas at 87, 81, 73, and 65%, respectively. It has been observed that increasing the amount of natural gas in the fuel mixture causes a decrease in thermal efficiency. The high auto-ignition temperature of natural gas increased the auto-ignition temperature of the mixture in the cylinder. This situation caused delayed combustion and incomplete combustion. For this reason, thermal efficiency decreased in experiments conducted with natural gas additives. The current study is compatible with the findings in the literature that efficiency decreases at natural gas contribution rates above 45%. In many studies in the literature, it is stated that thermal efficiency decreases if the natural gas ratio is high in natural gas-diesel dual-fuel combustion. To solve this problem, further studies should be conducted on lower natural gas contribution rates and a gradual injection strategy. After the current study, new studies are planned to be conducted in this vision.

Brake-specific fuel consumption: Thermal efficiency and BSFC are two interconnected motor performance factors. Increasing BSFC directly affects the worsening of thermal efficiency. In this study, in the experiments conducted with pure diesel fuel, 350, 259, 231 and 228 g/kWh BSFC were obtained at 25, 50, 75 and 100% torque values, respectively. However, in natural gas mixed experiments, BSFC values of 793, 436, 318 and 267 g/kWh were observed at the same torque values, respectively.

Exhaust emission changes: In experiments with natural gas mixing, it was observed that the amount of CO emissions was at higher levels compared to experiments using pure diesel fuel. It has been observed that the amount of CO in natural gas mixed fuels tends to decrease with increasing torque and decreasing natural gas replacement rates. However, it has a higher CO emission amount than pure diesel fuel in all conditions. This is because the natural gas substitution rate in the current study is high. The fact that natural gas is more resistant to combustion than diesel fuel has increased its auto-ignition temperature. Thus, in the experiments with a natural gas mixture, not all fuel taken into the cylinder could participate in combustion. As a result, the amount of CO emissions increased. For the same reasons, UHC emissions have also worsened compared to pure diesel fuel. It is thought that a more efficient combustion will occur if the dual fuel system is operated with less substitution rates at high torque values. It is also thought that CO and UHC emissions can be improved. It is planned to work with this strategy in further studies.

While the amount of NO_x was at its lowest levels at low loads where natural gas substitution was highest, it increased with the increase in torque value and decrease in natural gas substitution rate. It is thought that since complete combustion does not occur at high natural gas replacement rates, a relatively colder environment is formed in the cylinder and this causes a decrease in the amount of NO_x emissions.

A higher natural gas substitution rate at low loads disrupted combustion and complete combustion could not occur. It is expected that the amount of PM will increase as a result of incomplete combustion. The decrease in thermal efficiency already supports this situation. On the other hand, with the decrease in natural gas substitution at high load (100% torque), it has been observed that the PM emission amount of natural gas mixed fuel is relatively lower than that of pure diesel. It is thought that some natural gas additive can reduce PM emissions of diesel engines at high loads.

5. Outlook

Heavy-duty diesel engines are often used in maritime transportation. The most crucial handicaps of these engines are other emissions, especially soot and NO_x emissions. When the studies in the literature are examined, the use of natural gas in diesel engines has the potential to solve this problem. However, it is stated in the literature that there are some issues that require further study. In the light of this information, it is recommended to investigate them, such as the natural gas substitution rate, other gases contained in natural gas and the ability to store natural gas on ships.

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Nomenclature

CNG	:	Compressed Natural Gas
BTE	:	Brake Thermal Efficiency
BSFC	:	Brake-Specific Fuel Consumption
CO	:	Carbon Monoxide
CO_2	:	Carbon Dioxide
NO_x	:	Nitric Oxides
UHC	:	Unburned Hydrocarbons

BTDC	:	Before Top Dead Center
λ	:	Lambda
HDV	:	Heavy-Duty Vehicle
$\dot{m}f$	:	Fuel flow rate
γ	:	Heat capacity ratio
V	:	In-cylinder volume
m	:	mass
g	:	gram
rpm	:	revolutions per minute
CA	:	Crank Angle
HHV	:	Higher Heating Value
LHV	:	Lower Heating Value
Nm	:	Newton meter
HRR	:	Heat Release Rate
TDC	:	Top Dead Center
P_b	:	Braking Power
$dQn/d\Theta$	:	Net heat release rate
IP	:	In-cylinder Pressure
Θ	:	Crankshaft angle

Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Volkan Sabri Kül: Conceptualization, Supervision, Conceptualization, Writing-original draft, Validation, Data curation, Formal analysis
Selahaddin Orhan Akansu: Conceptualization, Writing-original draft, Data curation, Formal analysis

References

[1] Reşitoğlu İA, Altinişik K, & Keskin A. The pollutant emissions from diesel-engine vehicles and exhaust aftertreatment systems. Clean Technologies and Environmental Policy. 2015; 17;1; 15-27. https://doi.org/10.1007/s10098-014-0793-9

[2] Hoang AT. Waste heat recovery from diesel engines based on Organic Rankine Cycle. Applied energy. 2018; 231; 138-166. https://doi.org/10.1016/j.apenergy.2018.09.022

[3] Zhou JH, Cheung CS, Leung CW. Combustion, performance and emissions of a diesel engine with H_2, CH_4 and H_2–CH_4 addition. International journal of hydrogen energy. 2014; 39;9; 4611-4621. https://doi.org/10.1016/j.ijhydene.2013.12.194

[4] Kober T, Schiffer HW, Densing M, Panos E. Global energy perspectives to 2060–WEC's World Energy Scenarios 2019. Energy Strategy Reviews. 2020; 31; 100523. https://doi.org/10.1016/j.esr.2020.100523

[5] Dale S. BP statistical review of world energy. BP Plc: London, UK. 2021; 14-16.

[6] Huang H, Zhu Z, Chen Y, Chen Y, Lv D, Zhu J, Ouyang T. Experimental and numerical study of multiple injection effects on combustion and emission characteristics of natural gas–diesel dual-fuel engine. Energy Conversion and Management. 2019; 183;84–96. https://doi.org/10.1016/j.enconman.2018.12.110

[7] Tree DR, Svensson KI. Soot processes in compression ignition engines. Progress in Energy and Combustion Science. 2007; 33;3; 272–309.https://doi.org/10.1016/j.pecs.2006.03.002

[8] 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. 2013; 110; 201-206. https://doi.org/10.1016/j.apenergy.2013.03.024

[9] Ulishney CJ, Dumitrescu CE. Effect of gas composition on the performance and emissions of a dual-fuel diesel-natural gas engine at low load conditions. Fuel. 2022; 324; 124531. https://doi.org/10.1016/j.fuel.2022.124531

[10]Tutak W, Jamrozik A, Grab-Rogaliński K. Effect of natural gas enrichment with hydrogen on combustion process and emission characteristic of a dual fuel diesel engine. International Journal of Hydrogen Energy. 2020; 45;15; 9088-9097. https://doi.org/10.1016/j.ijhydene.2020.01.080

[11]Bhowmik S, Paul A, Panua R. Performance, exhaust emission and combustion of ethanol–diesel–compressed natural gas dual-fuel compression-ignition engine: a trade-off study. Clean Technologies and Environmental Policy. 2022; 1-20. https://doi.org/10.1007/s10098-021-02256-z

[12]Thangavel V, Subramanian B, Ponnusamy VK. Investigations on the effect of H_2 and HHO gas induction on brake thermal efficiency of dual-fuel CI engine. Fuel. 2022; 126888. https://doi.org/10.1016/j.fuel.2022.126888

[13]Ramalingam S, Rajendran S. Assessment of performance, combustion, and emission behavior of novel annona biodiesel-operated diesel engine. In Advances in eco-fuels for a sustainable environment. 2019; 391-405. https://doi.org/10.1016/B978-0-08-102728-8.00014-0

[14]Zhou H, Li X, Lee CFF. Investigation on soot emissions from diesel-CNG dual-fuel. International Journal of Hydrogen Energy. 2019; 44;18; 9438–9449. https://doi.org/10.1016/j.ijhydene.2019.02.012

[15]Meng X, Tian H, Long W, Zhou Y, Bi M, Tian J, Lee CFF. Experimental study of using additive in the pilot fuel on the performance and emission trade-offs in the diesel/CNG (methane emulated) dual-fuel combustion mode. Applied Thermal Engineering. 2019; 157; 113718. https://doi.org/10.1016/j.applthermaleng.2019.113718

[16]Demirci OK, Çınar C. HCCI-DI bir motorda doğal gaz kullanımının performans ve egzoz emisyonlarına etkisinin incelenmesi. Gazi University Journal of Science Part C: Design and Technology. 2019; 7;2; 317-330. https://doi.org/10.29109/gujsc.521668

[17]Kül VS, Akansu SO. Experimental Investigation of the impact of boron nanoparticles and CNG on performance and emissions of Heavy-Duty diesel engines. Fuel. 2022; 324; 124470. https://doi.org/10.1016/j.fuel.2022.124470

[18]Hwang J, Hirner FS, Bae C, Patel C, Gupta T, Agarwal AK. HRTEM evaluation of primary soot particles originated in a small-bore biofuel compression-ignition engine. Applied Thermal Engineering. 2019; 159; 113899. https://doi.org/10.1016/j.applthermaleng.2019.113899

[19]Pulkrabek WW. Engineering Fundamentals of the Internal Combustion Engines Novi Bios. 2021.

[20]Mohammad NKJ, Mansor WNW, Abdullah S, Wan C, Othman AAB,

Sheikh AA, Jalaludin J. Effects of exhaust emissions from diesel engine applications on environment and health: a review. Journal of Sustainability Science and Management. 2022; 17;1; 281-301. http://doi.org/10.46754/jssm.2022.01.019

[21]Turns SR. An introduction to combustion: concepts and applications. Boston; WCB/McGraw-Hill. 2000.

[22]Albayrak ÇB, Yıldız M, Akansu SO, Kahraman N. Performance and emission characteristics of an IC engine under SI, SI-CAI and CAI combustion modes. Energy. 2017; 136; 72-79. https://doi.org/10.1016/j.energy.2016.08.038

[23]Yıldız M, Çeper BA. Experimental Study on an SI Engine Mapping Considering Performance, Emissions, and Cyclic Variability. Energy, Environment and Storage. 2021; 01;01; 42-49. https://doi.org/10.52924/BBNP1133

[24]Mamedov I, Huseynova S, Javadova O, Azimova N, Huseynova R, Gasimova S. Testing of pine oil and glycerol ketal as components of B10 fuel blend. Energy, Environment and Storage. 2010; 11; 11-14. https://doi.org/10.52924/SMQS5726

[25]Ceviz MA, Öner IV, Kaya F, Karacali T. Analysis of the thermal efficiency and cyclic variations in a SI engine under lean combustion conditions. Journal of Thermal Science and Technology. 2011 31;1; 121-127.

[26]Martyr AJ, Plint MA. The combustion process and combustion analysis. In Engine Testing. 2012; 15; 375-406. https://doi.org/10.1016/B978-0-08-096949-7.00015-7

[27]Bizon K, Continillo G, Lombardi S, Mancaruso E, Vaglieco BM. ANN-based virtual sensor for on-line prediction of in-cylinder pressure in a diesel engine. In Computer Aided Chemical Engineering. 2014; 33; 763-768 https://doi.org/10.1016/B978-0-444-63456-6.50128-9

[28]Borat O, Balcı M, Sürmen A. İçten yanmalı motorlar. Teknik Eğitim Vakfı. 1994.

[29]Wang Z, Zhao Z, Wang D, Tan M, Han Y, Liu Z, Dou H. Impact of pilot diesel ignition mode on combustion and emissions characteristics of a diesel/natural gas dual fuel heavy-duty engine. Fuel. 2016; 167; 248-256. https://doi.org/10.1016/j.fuel.2015.11.077

[30]Uyumaz A, Solmaz H. Rcci Bir Motorda Enjeksiyon Zamanlamasi Ve Lamdanin Yanma Ve Performans Karakteristikleri Üzerindeki Etkilerinin Deneysel İncelenmesi. Gazi University Journal of Science. 2016; Part C: Design and Technology, 4;4; 299-308. https://dergipark.org.tr/en/pub/gujsc/issue/45198/565997

[31]You J, Liu Z, Wang ., Wang D, Xu Y, Du G, Fu X. The exhausted gas recirculation improved brake thermal efficiency and combustion characteristics under different intake throttling conditions of a diesel/natural gas dual fuel engine at low loads. Fuel. 2020; 266; 117035. https://doi.org/10.1016/j.fuel.2020.117035

[32]Coşkun AD, Kül VS, Akansu SO. Experimental Investigation of the Effect of Acetone Additive to Diesel Fuel on Engine Performance and Exhaust Emissions at Partial Loads. Energy, Environment and Storage. 2023; 03;01; 12-18 https://doi.org/10.52924/WQJR9374

[33]Pedrozo VB, Wang X, Guan W, Zhao H. The effects of natural gas composition on conventional dual-fuel and reactivity-controlled compression ignition combustion in a heavy-duty diesel engine. International Journal of Engine Research. 2021; 23;3; 397–415. https://doi.org/10.1177/1468087420984044

[34]Felayati FM, Cahyono B, Bakar RA, Birouk M. Performance and emissions of natural gas/diesel dual-fuel engine at low load conditions: Effect of natural gas split injection strategy. Fuel. 2021; 300; 121012. https://doi.org/10.1016/j.fuel.2021.121012

[35]Armin M, Gholinia M. Comparative evaluation of energy, performance, and emission characteristics in dual-fuel (CH_4/Diesel) heavy-duty engine with RCCI combustion mode. Results in Engineering. 2022; 16; 100766. https://doi.org/10.1016/j.rineng.2022.100766

[36]Park C, Kim C, Lee S, Lee S, Lee J. Comparative evaluation of performance and emissions of CNG engine for heavy-duty vehicles fueled with various caloric natural gases. Energy. 2019; 174; 1–9. https://doi.org/10.1016/j.energy.2019.02.120

[37]Zhang Z, Lv J, L, W, Long J, Wang S, Tan D, Yin Z. Performance and emission evaluation of a marine diesel engine fueled with natural gas ignited by biodiesel-diesel blended fuel. Energy. 2022; 256; 124662. https://doi.org/10.1016/j.energy.2022.124662

[38]Liu J, Guo Q, Guo J, Wang F. Optimization of a diesel/natural gas dual fuel engine under different diesel substitution ratios. Fuel. 2021; 305; 121522. https://doi.org/10.1016/j.fuel.2021.121522

[39]Zheng F, Zhang H, Yin H, Fu M, Jiang H, Li J, Ding Y. Evaluation of real-world emissions of China V heavy-duty vehicles fueled by diesel, CNG and LNG on various road types. Chemosphere. 2022; 303; 135137. https://doi.org/10.1016/j.chemosphere.2022.135137

[40]Appaswamy ST, Kumar A, Kumar S, Reddy VHV, Nashthrasan D, Buranpur SS, Mudhol LV. Analysis of (CNG + Diesel) in double fuel mode in a diesel engine for emissions and performance characteristics. AIP Conference Proceedings. 2023. https://doi.org/10.1063/5.0154919

[41]Shim E, Park H, Bae C. Comparisons of advanced combustion technologies (HCCI, PCCI, and dual-fuel PCCI) on engine performance and emission characteristics in a heavy-duty diesel engine. Fuel. 2020; 262; 116436. https://doi.org/10.1016/j.fuel.2019.116436

[42]Deng J, Wang X, Wei Z, Wang L, Wang C, Chen Z. A review of NO_x and SO_x emission reduction technologies for marine diesel engines and the potential evaluation of liquefied natural gas fuelled vessels. Science of The Total Environment. 2021; 766; 144319. https://doi.org/10.1016/j.scitotenv.2020.144319

[43]Liu J, Ulishney CJ, Dumitrescu CE. Experimental investigation of a heavy-duty natural gas engine performance operated at stoichiometric and lean operations. Energy Conversion and Management. 2021; 243; 114401. https://doi.org/10.1016/j.enconman.2021.114401

[44]Ulishney CJ, Dumitrescu CE. Effect of gas composition on the performance and emissions of a dual-fuel diesel-natural gas engine at low load conditions. Fuel. 2022; 324; 124531. https://doi.org/10.1016/j.fuel.2022.124531

[45]Yousefi A, Guo H, Birouk M. Split diesel injection effect on knocking of natural gas/diesel dual-fuel engine at high load conditions. Applied Energy. 2020; 279; 115828. https://doi.org/10.1016/j.apenergy.2020.115828

[46]Jamrozik A, Tutak W, Grab-Rogaliński K. An Experimental Study on the Performance and Emission of the diesel/CNG Dual-Fuel Combustion Mode in a Stationary CI Engine. Energies. 2019; 12;20; 38-57. https://doi.org/10.3390/en12203857