SSRG International Journal of Mechanical Engineering
ISSN: 2348-8360/ https://doi.org/10.14445/23488360/IJME-V11I3P103

Volume 11 Issue 3, 17-26, March 2024

Original Article

Evaluation of Combustion and Emission Characteristics of CI Engines Operating Dual-Fuel with BioCNG/Diesel and BioCNG/HVO

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Received: 13 January 2024 Revised: 15 February 2024 Accepted: 10 March 2024 Published: 31 March 2024

Abstract - *This paper presents experimental research results on dual-fuel engines using diesel, HVO, bioCNG/diesel, and bioCNG/HVO fuels. The dual-fuel engine was converted from Cummins' diesel engine 4.5L. The purpose is to analyze each fuel type's combustion process and emission characteristics. The experiments were carried out at a fixed speed of 1900 rpm, and the load increased gradually from low to maximum load, with a jump of 100 Nm. The experimental results show that the dual-fuel engine (bioCNG/diesel or bioCNG/HVO) was retarded premixed combustion of diesel, leading to a prolonged ignition delay compared to the operation with liquid fuel (diesel or HVO). As a result, there is more time for a more homogenized dual-fuel mixture, accelerated diffusion combustion, and shorter combustion times at this stage. The disadvantage of dual-fuel engines is that there is detonation in some cycles. In this study, the Knock peak method was used to determine the detonation limit, and the cycles with detonation in a set of 200 cycles were measured. In addition, BTE increases when part of the energy of liquid fuel is replaced by bioCNG fuel. At low loads, the largest percentage ratio of bioCNG can replace 74% in dual-fuel engines using bioCNG/diesel and 66% in dual-fuel engines using bioCNG/HVO. Dual-fuel engines have significantly reduced CO_2 and PM emissions concentrations compared to engines using liquid fuel. Especially when operating with bioCNG/HVO dual-fuel, the concentration emission is reduced by 20% CO_2 and 50% PM compared to diesel; and vice versa, the concentration of CO, HC, and NO_x emissions increased many times.*

Keywords - *Bio-fuel, Biomass, Bio-liquids, Dual-fuel, CNG, BioCNG, HVO, BioCNG/diesel, BioCNG/HVO, Dual-fuel engine.*

1. Introduction

The use of biogas in internal combustion engines in recent decades is an important turning point in the transition from the traditional oil industry to new renewable energy sources [1]. The target is that by 2030, renewable energy sources can replace traditional energies by 32%. The European Council has issued EU Directive 2018/2001 on promoting and using energy from renewable sources in transport. It also sets the criteria for future sustainability and reduction of greenhouse emissions for biogas, bio-liquids, and biomass fuels.

Some new renewable energy sources have the potential to be applied in diesel engines, such as methanol, ethanol, butanol, methyl ester and Fatty Acid Methyl Ester (FAME), Hydrogen Vegetable Oil (HVO), Dimethyl Ether (DME), biomethane (bioCNG), bioLPG fuel [1, 2, 3]. The production of biogas has contributed to diversifying energy sources, leading to a reduction in dependence on fossil energy sources. They are produced from waste sources from plants or animals. Biogas is an abundant, cheap, clean, renewable energy source and reduces pollution.

Biomethane gas (bioCNG) is being considered a more attractive new renewable energy source than other biogas fuels. The study of internal combustion engines using biomethane to partially replace traditional fuels is a suitable and feasible solution to reality. Biomethane is already used on public buses in several European countries. Most popular in Sweden, accounting for 17% of vehicles using bioCNG gas in traffic. In the production process, depending on the source of the waste and the anaerobic decomposition process, the composition of biomethane gas is different. Therefore, biomethane gas is obtained by biogas purification using membrane separation technology for high quality and purity.

Many studies have shown that biomethane from new renewable sources has physical and chemical properties similar to natural gas derived from fossils, presented in Table 1 [1, 4]. Therefore, when meeting automotive fuel standards, biomethane gas can be injected directly into vehicles using natural gas. Biomethane fuel must meet high-quality standards with methane content greater than 96% and CO_2 content lower than 6%.

Biomethane gas is used in the diesel engine in dual-fuel mode, resulting in increased thermal efficiency compared to original diesel engines at high loads [1, 4-6]. In this case, a part of diesel fuel is replaced by biomethane gas, while the diesel pilot acts as the ignition source to ignite the fuel-air mixture. BioCNG gas can be considered the main energy source to operate dual-fuel engines.

The disadvantage of the engine when operating dual fuel is that knocking occurs in some operating modes. The knock occurs partly due to the physicochemical properties of bioCNG caused during combustion at pressure and high temperature. Continuous knocking will lead to incomplete combustion, reducing the thermal efficiency of the engine and increasing emissions [1]. In addition, strong local knocks can cause damage to the piston-cylinder assembly structure.

In addition, when increasing the biomethane replacement ratio, the ignition delay decreases, and the speed of combustion fuel is improved, thus increasing the flame propagation rate. Therefore, an engine operating with dual fuel has a higher peak cylinder pressure (Pmax) and Heat Release Rate (HRR) than one operating with diesel fuel.

The bioCNG/diesel dual-fuel engine has better environmental performance than the diesel engine. When the compression ratio of the dual-fuel engine is increased, the engine runs smoothly and reduces emissions significantly. The more homogeneous the fuel mixture burns, the more CO_2 and particulate/smoke emissions are reduced. CO and HC emissions concentration increase because the combustion temperature is higher and lasts longer than in diesel engines. Dual-fuel engines have higher peak cylinder pressure, which stops the pre-combustion of a homogeneous mixture, and HC can form in the narrow gap between the piston and the cylinder wall. NO_x emissions concentration also increases, especially at low engine loads [9].

HVO is evaluated as a potential fuel for transportation, with enormous resources and the easiest renewable energy source. HVO is obtained from the hydrogenation of vegetable or animal fats, fish, or by-products of petroleum refining. In addition, it can be processed from palm oil or a large amount of other plants with high oil content. HVO has characteristics similar to diesel and is easily compatible with the fuel system of diesel engines without any modification, as presented in Table 1.

HVO with a high cetane coefficient of about 70 - 95 is highly flammable and produces little CO_{2qe}, as assessed in the W_tW test cycle. The bioCNG/HVO dual-fuel engines improve brake thermal efficiency and environment, which positively impacts emissions reduction more than bioCNG/diesel dual-fuel engines. However, it is necessary to study further the frictional performance of HVO affecting the combustion process and the mechanical properties of the cylinder piston assembly.

The main purpose of this study is to compare and evaluate combustion processes and emission characteristics when operating dual fuel with bioCNG/diesel and bioCNG/HVO. From there, it is suggested that the operating parameters of the dual-fuel engine are suitable for improving brake thermal efficiency and reducing emissions.

Table 1. Basic properties of fuels

Parameter	Unit	Diesel	HVO	CNG	BioCNG
C	%	86.2	84.2	74.7	74.7
H	%	13.5	15.1	24.9	24.9
Other Elements	%	0.3	0.7	0.4	0.4
The Boiling Point	°C	180 - 360	200 - 350	-162	-162
Density at 20^0C	kg/m^3	835	780	0.68	0.68
Calorific Value	kWh/kg	11.94	12.22	13.9	13.9
Calorific Value	kWh/l	9.97	9.55	5.8	5.8
Cetan Number	-	> 51	> 70	-	-
Octane Number	-	-	-	125	125
WtW	kg CO_{2qe} /kg	3.84	0.9 - 2.23	3.28	0.36 - 0.83
WtW	g CO_{2qe} /kWh	322	78 - 183	236	26 - 60

2. Experimental

2.1. Experimental Setup

All the tests are carried out on the diesel engine model operating with dual fuel. A dual-fuel engine converted from Cummins ISBe4 diesel engine and combined with a dual fuel injection system. The engine specifications are presented in Table 2.

The original common rail fuel system remains the same. It is controlled by the ECM 850 control unit, which is capable of injecting diesel into the cylinders in three different stages with a small pre-injection dose at approx. 30ºC before TDC; the main injection dose at approx.10ºC before TDC, and a small amount of pre-injection dose at approx. 30 - 50ºC after TDC.

Ac electronically controls the dual fuel system (STAG/diesel, Poland). Included accessories include a control unit ECU, an air pressure regulator, air mixer gas, an air release valve, and sensors for various operating parameters (such as a knock sensor, exhaust temperature sensor, air pressure, and lambda probe to detect the oxygen content of the exhaust gas). The working principle diagram of the dual-fuel engine is shown in Figure 1 and Figure 2.

The entire model was installed on a test bed, and an eddy current dynamometer and other external measuring devices were utilized to measure the engine's torque, speed, temperature, pressure, and emission. The AVL X-ion measuring instrument measured the engine cylinder pressure, and the high-pressure indicator was analyzed using the ALV-Indicom software.

Table 2. Base diesel engine specifications

Description	Value
Type	Cummins ISBE 4.5L/ CI/ EURO 4/ 4 cylinder inline/ turbocharged/ after cooled
Engine Displacement	4.5 dm^3
Stroke/Bore	123.7/107.61 mm
Compression Ratio	17.3:1
Power	152 kW
Torque	720 Nm

Horiba's Mexa-one instrument measures and analyzes emissions (CO_2, CO, NOx, HC). The Horiba SPCS counter, Horiba MDLT-One, and TSI EEPS machine are used to measure particle size. Gas and liquid fuel consumption is measured with a highly accurate Coriolis Flow electronic meter. The crankshaft speed measurement signal is set to an accuracy of 0.5°C.

In addition, all measurements in this study were performed in a simulated environment with steady-state conditions. To avoid the influence of temperature changes and external factors interfering with the measurement results. The engine coolant temperature control system is maintained at 82 ± 1°C. The intake air temperature is controlled from 20 - 40oC during test measurement. The basic parameters of measuring equipment are shown in Table 3.

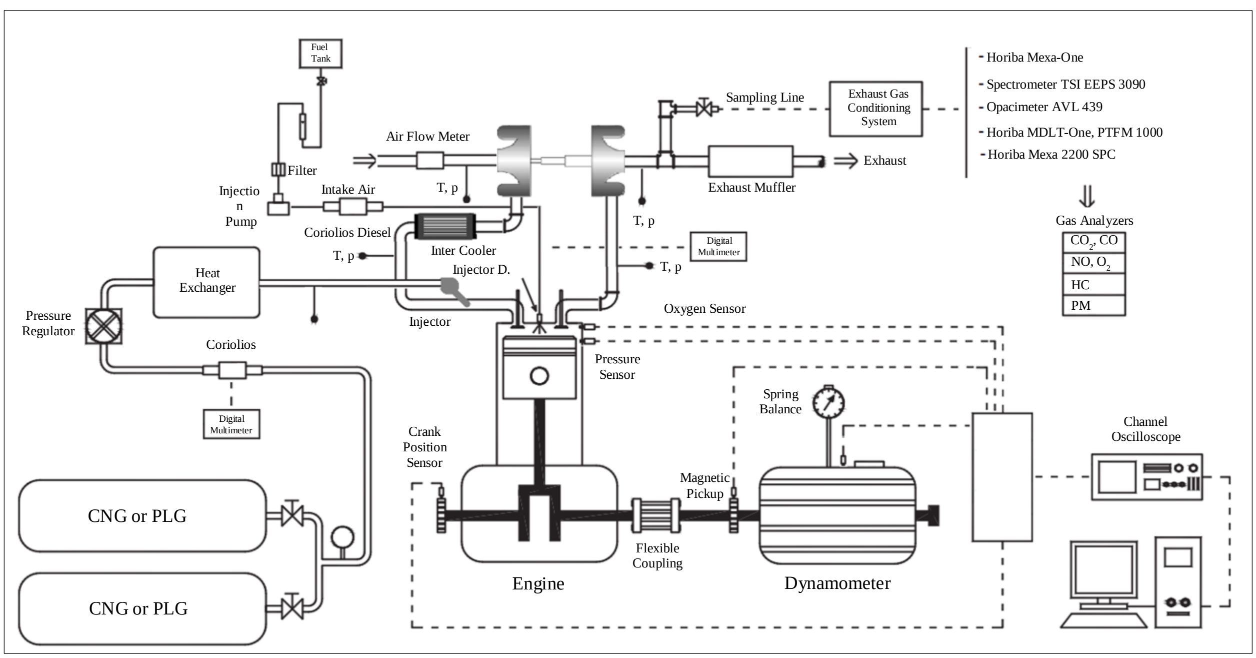

Fig. 1 The working principle diagram of the dual-fuel supply system and engine test installation diagram

Fig. 2 Model of a dual-fuel engine in the laboratory of KVM TUL

Table 3. Basic parameters of measuring equipment

Parameter	Device	Model	Range	Accuracy
Speed	Dynamometer	D-MPO Z1710665.001	0 - 10000 rpm	± 1 rpm
Torque			0 - 2500 Nm	± 0.22%
Diesel/HVO/bioCNG Consumption	Coriolis Flow	Meter/CMF0	0 - 108 $kg.h^{-1}$	± 0.02 $kg.h^{-1}$
CO_2	Horiba Mexa One D1 EGR	AIA-32	0 - 20%	± 0.50%
CO		AIA-11	0 - 5.000 ppm	± 0.01 ppm
HC		FIA-02O-ND	0 - 60.000 ppm	± 1 ppm
NO_x		CLA-02OV-3	0 - 10.000 ppm	± 1 ppm
Exhaust Gas Flow Meter	Horiba PTFM	PTFM-1000	0 - 10 m3. min^{-1}	± 0.5%
Air Flow Meter	Mass Flow Meter	620S	0 - 1000 $kg.h^{-1}$	± 0.3%
PM	Opacimetr	AVL 439	0 - 10 m^{-1}	±1%
	Spectrometer	TSI 3090	1 - 107 $\#/cm^3$	± 0.01$\#/cm^3$
	PM	HORIBA MEXA-2200SPCS	> 23nm	± 0.5%
Pressure	AVL X-ion/AVL Indimeter	619	0 - 25 MPa	± 0.5%

According to the first law of thermodynamics, the Heat Release Rate (HRR) formula can be established.

$$\frac{dQ_B}{d\theta} = \frac{1}{k-1}\left(k.P.\frac{dV}{d\theta} + V.\frac{dP}{d\theta}\right) + \frac{dQ_w}{d\theta} \qquad (1)$$

Where,
P : In-cylinder pressure,
V : Cylinder volume,
K : Specific heat ratio, and
Dθ : Variation of the crank angle, respectively.

According to Newton's law of heat transfer, the equation for heat transmitted through the cylinder wall is calculated as follows:

$$Q_w = \alpha . S . (T - T_w) \quad (2)$$

Where
T_W : Temperature of the cylinder wall,
T : The instantaneous gas temperature,
S : The actual area of the cylinder wall, and
α : Is the heat transfer coefficient, respectively.

Woschnio's law determines the formula for calculating the heat transfer coefficient:

$$\alpha = 794 . d^{-0.2}.p^{0.8}.T^{-0.546}.\left[C_1.c_s + C_2.V_z.\frac{T_1}{p_1 V_1}.(p - p_0)\right]^{-0.8} \quad (3)$$

Liquid fuel (diesel or HVO) and bioCNG are two types of fuel that are not in the same phase. The amount of fuel replaced is calculated using the Energy Substitution Ratio (ESR). It is understood as the percentage of energy released by bioCNG compared to the total energy released in a dual-fuel engine.

$$ESR = \frac{\dot{m}_{NG} . H_{uNG}}{\dot{m}_{NG} . H_{uNG} + \dot{m}_L . H_{uL}}.100 \quad (4)$$

Where, $\dot{m}_L$ and $\dot{m}_{NG}$ represent the liquid fuel (diesel or HVO) and bioCNG consumption per cycle (mg/cyc), while H_{uNG}, H_{uL} (MJ/kg) denotes the lower heating value.

- Brake-Thermal Efficiency (BTE):

$$BTE = \frac{3.6 \ x \ 10^3 P_e}{\dot{m}_{NG}.H_{uNG}+\dot{m}_L.H_{uL}} \quad (5)$$

- Brake-Specific Energy Consumption (BSEC):

$$BSEC = \frac{\left(\frac{H_{uNG}}{H_{uL}}\right).\dot{m}_{NG}+\dot{m}_L}{P_e}.10^3 \quad (6)$$

Where, P_e is the engine power output.

Use the Knock Peak (KNK_PK) method to determine the knock. The high-pressure indicator in the engine cylinder analyzes the knock. The combustion pressure in the engine is measured with an AVL Indicom instrument. By filtering out the low-pass pressure in the measured all-pass pressure, the high-pass pressure is obtained.

Maximum high-pass pressure is compared with the knocking limit value determined by the mean filter. For dual-fuel engines, the filter averaging is taken in a range of 10 values (5 values before and 5 values after), and sampling from 0.1°CA is recommended.

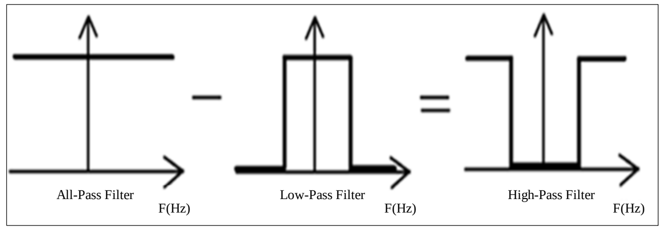

Fig. 3 High-pass pressure filter in the engine cylinder

The following relationships determine the filtration process:

$$Y_{FIR}{}^i = \sum_{j=i-n}^{j+n} k_j.Y_j \quad (7)$$

$$k_j = h_j.C_j \quad (8)$$

$$h_j = \frac{1}{2}\left(1 + \cos\left(\frac{2.\pi.j}{2.n+1}\right)\right) \quad (9)$$

Where,
n : Number of elements,
i, j : Order of elements, and
y : General values.

2.1.1. Low-Pass

$$C_j = \frac{\sin\left(2.\pi.\frac{f_c}{f_s}.j\right)}{\pi.j} \qquad C_0 = 2.\frac{f_c}{f_s} \quad (10)$$

2.1.2. High-Pass

$$C_j = -\frac{\sin\left(2.\pi.\frac{f_c}{f_s}.j\right)}{\pi.j} \qquad C_0 = 1 - 2.\frac{f_c}{f_s} \quad (11)$$

Where,
f_c : Selected frequency, and
f_s : Sampling frequency.

2.2. Experimental Procedures

In the first step, the diesel engine runs on diesel; in the second step, the diesel engine operates entirely on HVO fuel; in the third step, the diesel engine is used with bioCNG/diesel dual-fuel mode; and finally, the diesel engine is used with bioCNG/HVO dual fuel mode. All tests should be retried three times under the same engine operating conditions to increase accurate results and eliminate random errors. With the condition that the experiment is carried out at 1900 rpm, the torque increases gradually to the maximum. The measurement is carried out so that the load increases gradually, from the initial load of 100 Nm with a jump of 100 Nm up to the maximum load.

3. Results and Discussions

Figure 4 shows the combustion process of all four fuels (diesel, HVO, bioCNG/diesel, and bioCNG/HVO) under the same engine operating conditions (1900 rpm, full load). For dual-fuel engines (bioCNG/diesel and bioCNG/HVO), the energy replacement rate of bioCNG in the engine is 30%.

The fuel combustion process is represented by a graph of dual-fuel engine cylinder pressure (P) and Heat Release Rate (HRR), as shown in Figure 4. With the heat release rate of fuel, there are usually two main stages: the first stage is premixed combustion, and the second stage is diffusion combustion. In the first stage of the combustion process (premix combustion), it is easy to see that when replacing a part of bioCNG, the intensity of this combustion stage is enhanced more than when using only liquid fuel. This stage of dual-fuel combustion can be divided into two parts: part one is the pre-combustion of a small amount of diesel and a small amount of entrained bioCNG; part two is the combustion of the bioCNG-air mixture fuel.

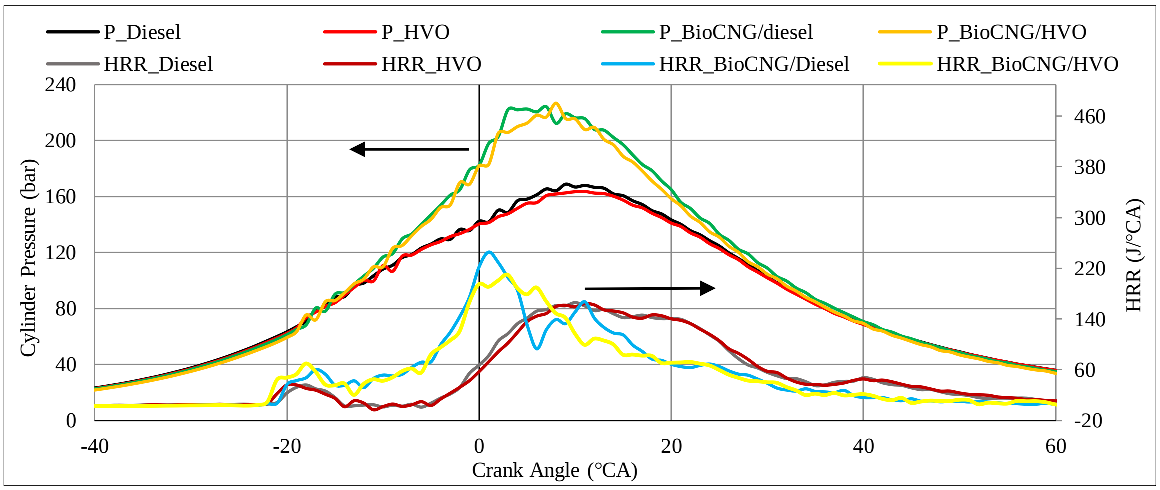

Fig. 4 P and HRR evolutions of the dual-fuel engine, 1900 rpm, full load with diesel, HVO, bioCNG/diesel, and bioCNG/HVO

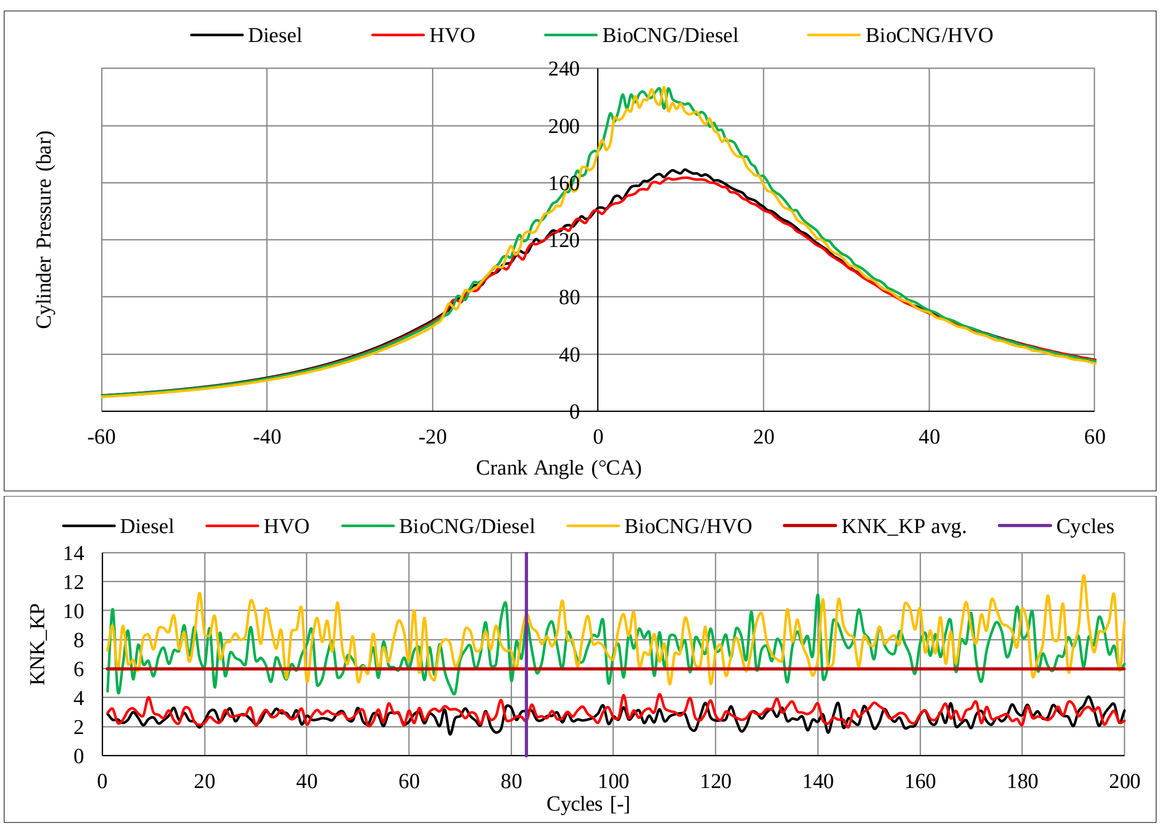

Fig. 5 The cylinder pressure curve (PCYL) and knock peak (KP_PK) values for 200 cycles

When liquid fuel (diesel or HVO) is partially replaced by bioCNG, the bioCNG fuel evaporates and absorbs some of the heat from the engine compression. As a result, dual-fuel combustion has a prolonged ignition delay compared to operating entirely on liquid fuel.

In addition, partial replacement of bioCNG leads to a lower cetane number and an increase in dual fuel octane number. Therefore, the self-ignition of liquid fuel is inhibited after the partial replacement of bioCNG. As a result, this dual-fuel combustion stage is slowed down, but the cylinder pressure and HRR are still greater than when using diesel or HVO alone.

In the second stage of the combustion process (diffuse combustion), the engine using dual fuel (bioCNG/diesel and even bioCNG/HVO) has a faster combustion rate when using it entirely with liquid fuel. This could be due to the delayed pre-combustion and more time for vaporization of fuel and mixing of the dual-fuel mixture, resulting in a more homogeneous fuel mixture. On the other hand, bioCNG's combustion rate is fast, making dual fuel combustion more efficient, resulting in peak pressure and maximum HRR closer to TDC.

The KNK_PK safety knock limit for dual-fuel engines is defined as 6 bar. It is easy to see that for dual-fuel modes, the KNK_PK value is higher than when operating with liquid-fuel modes. Part of the reason is the large replacement rate of ESR at high speed and large load leading to unburnt bioCNG combined with high pressure and temperature conditions, which easily causes the knocking phenomenon presented in Figure 5.

Figure 6 shows the brake thermal energy input (PE) of the engine and the Energy Substitution Ratio (ESR). It is easy to see that the ESR of bioCNG is almost continuous from low load to maximum load. The ESR of dual-fuel operation modes is relatively high at low loads and tends to decrease as the load increases.

The ESR reached the maximum of 74% when using bioCNG/diesel mode, and the ESR reached the maximum of 66% when using bioCNG/HVO mode at 1900 rpm with a load of 100 Nm. In addition, the bioCNG/HVO operating mode has a higher ESR than the bioCNG/diesel operating mode at medium loads.

The hypothesis is that the engine operating conditions are the same fuels with the same efficiency. However, heat input in dual-fuel operation mode is higher at low loads and conversely lower at high loads than in liquid fuel operation mode. The reason may be that bioCNG replaces liquid fuel at a high ratio at low loads, leading to incomplete combustion of the dual-fuel mixture and higher energy consumption.

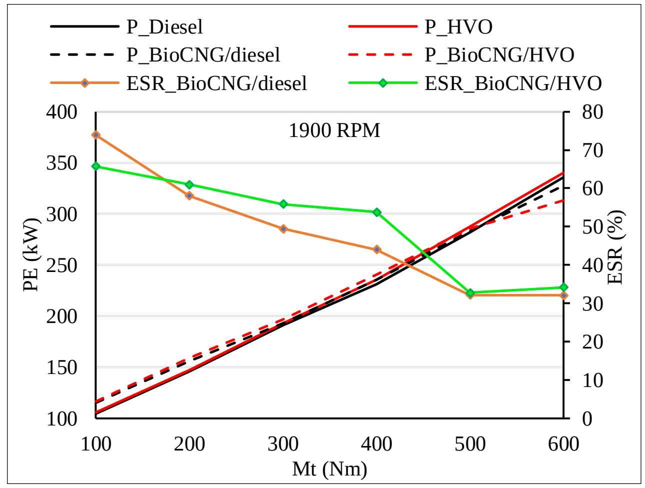

Fig. 6 Heat input of type fuels and bioCNG replacement energy ratio in dual fuel engines

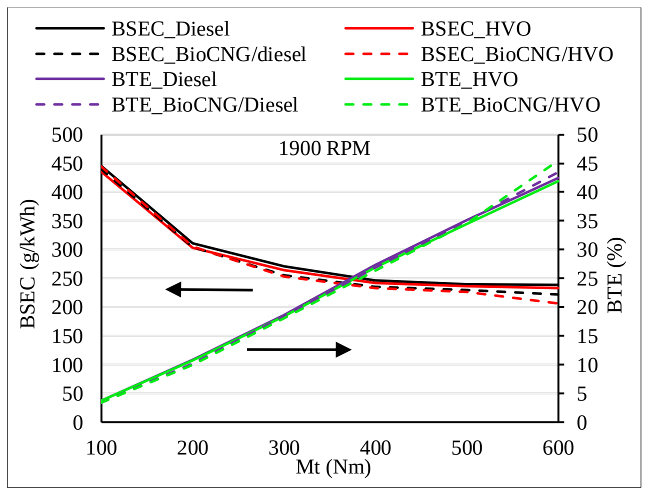

Fig. 7 Comparison of BTE and BSEC in dual fuel engines using diesel, HVO, bioCNG/diesel, and bioCNG/HVO

Similarly, a low ESR results in better and more efficient fuel mixture combustion at high loads, and a good mechanical successful energy conversion requires lower energy input. Figure 7 shows a comparative study on the effects of type fuels on the BTE of dual-fuel engines. The BTE in dual-fuel mode is larger than in liquid-fuel mode.

For instance, when operating bioCNG/HVO dual fuel, BTE increases from 41.8% to 45.5%, and BTE increases from 42% to 44.1% when operating bioCNG/diesel at 600Nm load. When operating dual fuel modes, the BSEC is lower than when working entirely on liquid fuel modes. In particular, the bioCNG/HVO dual-fuel engine has the lowest BSEC value compared to when operating other fuels.

Figure 8 shows that at low loads, the exhaust gas temperature of the engine when operating dual-fuel and liquid fuel mode is similar, but at high loads, the exhaust temperature tends to be lower.

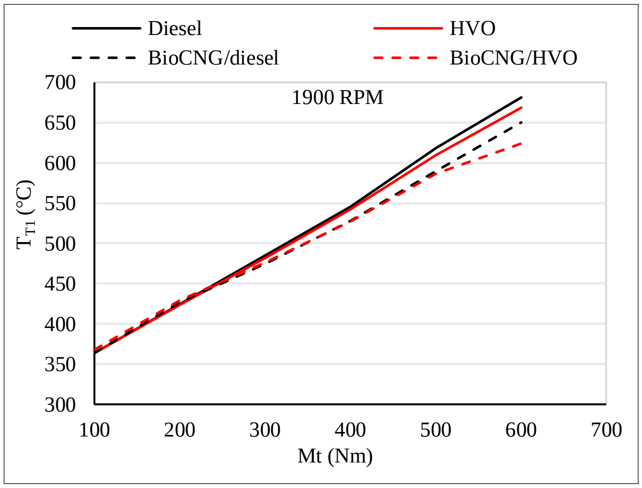

Fig. 8 Exhaust gas temperature evolutions of the dual-fuel engine using diesel, HVO, bioCNG/diesel, and bioCNG/HVO

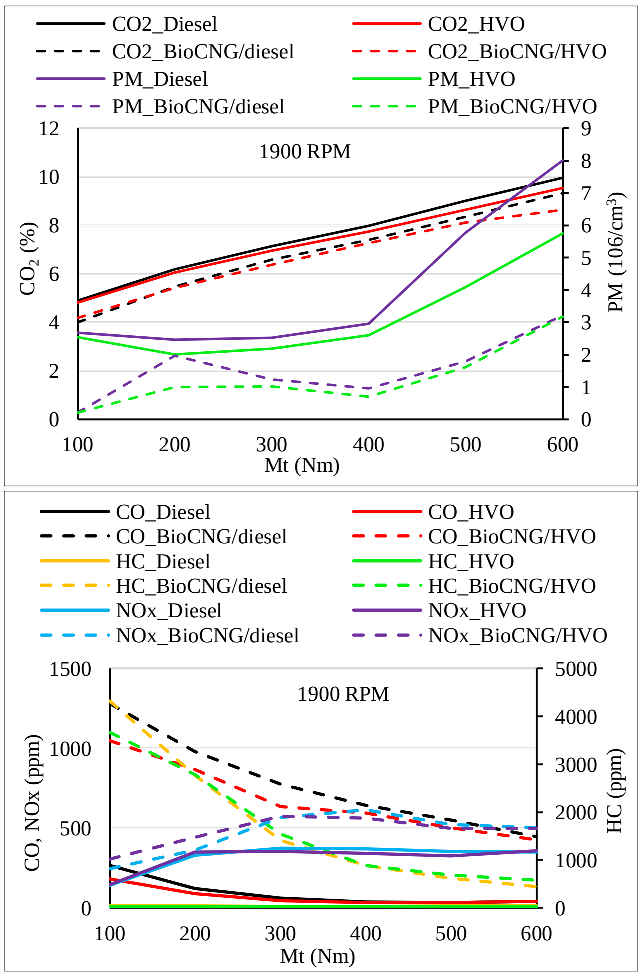

Fig. 9 Emissions of dual-fuel engines operating with diesel, HVO, bioCNG/diesel, and bioCNG/HVO

This demonstrates that when operating in dual fuel mode, the conversion from thermal energy to mechanical work is more efficient. On the other hand, when bioCNG is added to the cylinder, part of the heat is absorbed during the vaporization process, reducing the exhaust gas temperature.

Figure 9 shows that in both cases, the operating bioCNG/diesel and bioCNG/HVO modes have a lower concentration of PM emissions than when operating with diesel or HVO modes, especially at high loads. Exhaust particles are often formed in some areas with high combustion temperatures and rich-fuels areas; most of the particles are produced during diffuse combustion.

Therefore, adding bioCNG fuel prolonged the ignition delay time, resulting in more time to create a homogeneous combustion mixture. More accelerated combustion and better quality at the diffusion stage, as well as burned soot, reduce the formation of particulate emissions. In dual-fuel operation mode, there is a lower concentration of CO_2 emissions than when the engine runs entirely on liquid fuel mode at all loads.

This results from the lower C/H content ratio in bioCNG than in diesel and HVO. Replacing conventional diesel fuel with renewable HVO fuel in the bioCNG/HVO dual fuel mode has reduced 20% CO_2 and 48% PM concentration emissions compared to diesel-only mode.

Diesel engines, when using dual fuel modes, have an increased concentration of NO_x emissions compared to using only liquid fuel modes. Because the prolonged high-temperature residence time and the sharp increase in the peak pressure after adding bioCNG caused the NO_x concentration to increase. In particular, the NO_x emissions of dual-fuel engines when using bioCNG/HVO mode are slightly lower than when using bioCNG/diesel mode.

It can be seen that CO and HC concentration emissions are many times higher when operating dual fuel modes than when using only diesel or HVO modes. The cause of the increase in HC and CO content is largely due to the higher ignition temperature of the gaseous fuel, which contributes to the cessation of the pre-combustion of the homogeneous mixture burnt, and HC formation usually takes place in the narrow gap between the piston and the cylinder wall. In addition, the HC and CO concentrations emission result from incomplete oxidation of intermediate products during the combustion of hydrocarbon molecules.

4. Conclusion

After studying the conversion of cummins diesel engines to dual-fuel engines operating with diesel, HVO, bioCNG/diesel, and bioCNG/HVO, the following conclusions can be drawn:

- Partially replacing liquid fuel (diesel or HVO) with bioCNG gaseous fuel, the premix combustion stage is enhanced, and prolonged ignition delay time. As a result, more time is allowed to mix the more homogeneous mixture burnt, leading to faster and better diffuse combustion. The resulting peak pressure and HRR of dual-fuel mode are higher when operating liquid fuel mode.

- The knock peak method was used to determine the knock phenomenon and knock limit of dual-fuel engine KNK_PK = 6 bar. The knock frequency of the motor increases with increasing bioCNG replacement ratio at high loads.
- The energy replacement ratio of bioCNG can be as high as 74% when operating with bioCNG/diesel mode and 66% when operating with bioCNG/HVO mode, but the engine still runs smoothly.
- The brake-thermal efficiency of dual-fuel modes increases, and fuel consumption is reduced compared to engines operating entirely on liquid fuel modes.
- The engine exhaust temperature in dual-fuel operation modes is lower than in liquid-fuel operation modes. In which the exhaust temperature of bioCNG/HVO mode is lower than other fuel modes.
- When using dual fuel modes, the engine has lower CO_2 and PM concentration emissions than when operating entirely on liquid fuel mode at 20% and 48%, respectively. In contrast, CO, HC, and NO_x concentrations emissions were much higher. However, the bioCNG/HVO operation mode helps to improve CO, HC, and NO_x more when operating bioCNG/diesel mode.

In short, bioCNG/HVO dual-fuel mode has many advantages over all other fuels, from brake thermal efficiency to environmental emissions.

Acknowledgments

The authors gratefully acknowledge the support in publishing costs and providing the laboratory resources of the Department of Vehicles and Engines, Technical University of Liberec under No. 2370/117.

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Nomenclature

CNG	:	Natural gas
bioCNG	:	Biomethane gas
HVO	:	Hydrogen vegetable oil
ESR	:	Energy substitution ratio
Pmax	:	Peak cylinder pressure
^{0}CA	:	Crank angle
BMEP	:	Brake means effective pressure
BSEC	:	Brake-specific energy consumption
HRR	:	Heat release rate

BTE	:	Brake thermal efficiency
SOI	:	Start of injection
ECU	:	Electronic control unit
TDC	:	Top dead center
ERG	:	Exhaust gas recirculation
PM	:	Particulate matter
NO_X	:	Nitrogen oxide
CO	:	Carbon monoxide
HC	:	Hydrocarbon
CO_2	:	Carbon dioxide