

CO₂ concentration from turbocharged common rail diesel engine dually fueled with compressed biomethane gas controlled at optimum ratio

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Abstract. The objectives of this study are to investigate the carbon dioxide (CO₂) concentration from the compressed biomethane gas (CBG) and diesel dual-fueled diesel engine and to compare the CO₂ concentration produced from the dual-fueled and the diesel-fueled engines. The duration of CBG injection was controlled by following the optimum ratio of the CBG obtained from the previous study. During the test, the engine speed was varied from 1,000 to 4,000 rpm and the engine torque was maintained to be 25, 50, 75 and 100% of the maximum engine torque. Experiment was divided into two parts consisting of the dual-fueled and the diesel-fueled modes. From the dual-fueled mode, when the engine speed increased, the CO₂ concentration decreased. Because the optimum ratio of the CBG and the volumetric efficiency decrease during the high engine speed range, the proportion of the diesel increases, the incomplete combustion occurs. The unburned carbon oxidizes to be the CO in higher proportion than the CO₂, thus, the CO₂ consequently decreases. From the CO₂ comparison, the dual-fuel mode produced the CO₂ nearly the same as that of the diesel-fuel mode during the low engine torque. On contrary, the dual-fuel mode had higher CO₂ concentration during the high engine torque.

1 Introduction

A diesel engine is one type of an internal combustion engine that is classified into a compression-ignition engine or CI engine. Advantages of diesel engines over gasoline engines are: (i) Diesel engines have higher compression ratio, the thermal efficiency is higher than that of gasoline engines. (ii) Combustion in diesel engines can be directly controlled by adjusting of injected fuel amount. And (iii) overall cost of operation of diesel engines is lower than that of gasoline engines [1-2]. These are reasons causing number of diesel powered vehicles to increase continuously, especially, in a case of the light commercial vehicles that ratio of the vehicles installed with diesel engine is obviously higher than that of with gasoline engine [3].

According to the oil's price crisis and rapid decreasing in amount of reserved crude oil, people and researchers involved with diesel-powered vehicles are interested to find alternative fuel for used in diesel engines to minimize the fuel cost. Various past studies have reported that diesel engines can run on a mixture of alternative fuel and diesel. The alternative fuel, which is in liquid or gas phase, is injected into intake air and they will completely mix as homogeneous air and alternative fuel mixture. Diesel is consequently injected into the combustion chamber and become the pilot ignition that ignites the surrounding alternative fuel [4]. The engine that utilizes two types of fuel simultaneously is defined

as a "dual-fueled engine" and so called the mixture of fuels as "dual fuel".

Biogas was become well-known alternative fuel for a few decades. Biogas is the raw product obtained from the biological process in many diversities of resource, such as, sewage treatment plants, industrial site, landfills, digestion plants of agricultural organic waste, etc. Raw biogas primarily consists of methane (CH₄, 40-75%) and carbon dioxide (CO₂, 15-60%) [5, 6]. Since proportion of the methane being the main composition in the biogas is lower than that of in the CNG, the biogas is taken into an upgrading process to enhance proportion of the methane and purity and the upgraded biogas is called as "biomethane" [6]. And the pressurized biomethane is named as "compressed biomethane gas" (CBG).

It has been clearly known that if commercial carriers need to decrease the fuel cost of dual-fueled engines, the dual fuel mixing ratio must be adjusted to have the maximum proportion of the CBG as possible while engines can produce efficient performance. However, the acceptable increase in the ratio of the CBG is not only an increase the ratio of the CBG to be the maximum, it is important to concern with safety condition of engines and country regulations. In the light of this reason, the previous studies on the optimum ratio of CBG and diesel dual fuel used in diesel engine have defined the limitation factors for increasing the ratio of the CBG that consisted of: (i) The exhaust gas temperature did not exceed 580 °C. (ii) The voltage measured from a knock

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sensor did not exceed 1 V. And (iii) the opacity of the smoke in the exhaust gas from the diesel engine must not exceed 35% measured by a partial-flow opacity-type smoke tester, which has been defined in the national regulation of The Pollution Control Department, Ministry of Natural Resources and Environment, Thailand. Since these limitation factors were defined from the experimental investigation on the diesel-fueled engine, if the optimum ratio of the CBG was adjusted by following these factors, the dual-fueled engine could operate without failure and the emission did not exceed the maximum value defined by the regulation. And the engine torque obtained from the dual-fueled engine with the optimum ratio of the CBG was identical to a case of the diesel-fueled engine [7, 8].

Although the dual fuel has been adjusted to be in line with the optimum ratio of the CBG obtained from the previous studies and the soot in the exhaust gas has already corresponded to the national regulation, concentration of other emissions occurring from combustion in the CBG and diesel dual-fueled engines is interesting issue. These became background and significance of this study with objectives as: (i) To investigate the carbon dioxide (CO₂) concentration in the exhaust gas from the CBG and diesel dual-fueled turbocharged common rail diesel engine (TCRD engine) with the optimum ratio of the CBG. And (ii) to compare the CO₂ concentration in the exhaust gas from the dual-fueled and the diesel-fueled engines at the identical operating condition and engine torque. Knowledge obtained from this study will be usefulness for researchers and automotive engineers involved in adjusting the optimum ratio of the CBG and diesel dual fuel. Obtained data can be used as a guideline to adjust the ratio for effective decreasing in the emission from the combustion.

2 Experimental setup and procedure

The engine used in this study was the common rail, direct injection diesel engine (Toyota, model 2KD-FTV, four-cylinder, four-stroke, total displacement 2,494 cm³) that was directly taken from the market-sold vehicle. The engine had the maximum power of 75 kW at 3,600 rpm and the maximum torque of 200 N·m at 1,400–3,200 rpm. The compression ratio was 18.5:1. The bore and the stroke was 92.0 and 93.8 mm, respectively. The turbocharger was installed onto the engine without intercooler as default arrangement from manufacturer. It should be noted that the primary hardware of the engine was not modified except the installation of gas electronic control unit (Gas ECU) and the gas supplying system. The CBG with the CH₄ proportion of 80.0% by volume and the diesel with the Cetane index of 55 were used. The diesel injection timing and duration were controlled by a factory engine's electronic control unit (ECU) in which none of the modification on the programmed values. The factory sensors located on the engine were completely wired to the ECU as the manufacturer default. The solenoid diesel injectors (Denso, type 415F, 6 holes) were located on the factory fuel common rail.

The fuel pressure was 160 MPa. The injected timing and duration of the CBG was controlled by the gas ECU (Autogaz, model Stag Diesel-4, for 4 injectors) which could be programmed via a computer. The CBG supplying equipments were installed on the engine to activate the dual-fuel mode, as shown in Figure 1. The configuration of the gas and the diesel systems are described as follows. The pressure of the CBG was reduced from 200 bar_g in the tanks to 2 bar_g. The inlet gas pressure and the inlet water temperature probes were attached on the gas reducer and wired to the gas ECU. Another set of the gas pressure and temperature sensors (Autogaz, PS-02, maximum temperature 125 °C, maximum pressure 6.75 bar) were installed on the outlet tube of the gas reducer for measuring the pressure and the temperature of the CBG supplied to the gas injectors. The gas injectors (Hana, maximum gas flow rate 0.13±0.0015 m³/min, maximum temperature 120 °C, maximum pressure 3.5 bar), which were directly controlled by the gas ECU, were installed on the gas rail to inject the CBG into the intake runners of the engine. In addition, the knock sensor (Bosch, 3-wire, frequency range 1–20 kHz) was installed on the outer surface of the cylinder block. The exhaust gas temperature sensor (Termo-Precyzja, Type-K, temperature range -100–1,300 °C) was installed on an outlet pipe of the turbocharger's turbine side.

A hydraulic engine dynamometer (HPA) was applied to load the engine and to maintain its speed. The CO₂ concentration was measured by the gas analyzer (Testo, model 340) with an accuracy of ±0.2%v/v for CO₂. In addition, the smoke optical absorption coefficient was measured by the opacimeter-type smoke tester (Hamann, DO 285, range 0–9.99 m⁻¹). The smoke opacity could be obtained in real-time as an advantage of using the opacimeter. Probes of the gas analyzer and the smoke meter were attached into the exhaust pipe behind the turbine side of the turbocharger.

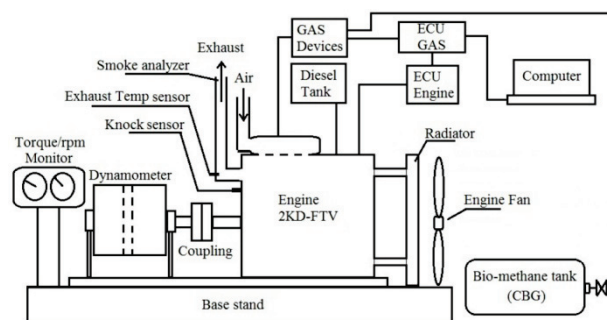


Fig. 1. Schematic diagram of experimental setup.

The experimental procedure was divided into two main sections as follows:

(i) First section, the test on the diesel-fueled engine: The engine speeds in the test were defined to be varied from 1,000 to 4,000 rpm in an increment of 200 rpm. At each defined engine speed, the load from the dynamometer applying on the engine and the throttling valve of the engine were simultaneously adjusted to maintain the engine torque to correspond with the reference engine torque curves at 25, 50, 75 and 100% of

the maximum engine torque obtained from the previous study [8] as depicted in Figure 2. This procedure was conducted in order to control the engine torque to be identical in each of the engine speed to simulate the same operating condition of the engine. When the engine speed was steady, the engine speed, the engine torque, emissions in the exhaust gas, and data from all sensors were monitored and recorded by the interval of 180 s. Then, the engine speed and the engine torque were increased step-by-step until they reached the maximum values as scoped in this study.

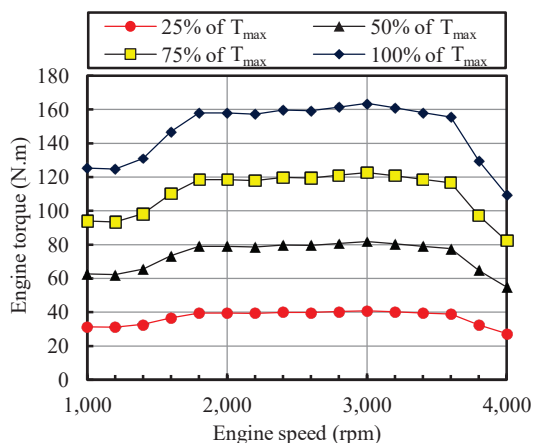


Fig. 2. Reference engine torque curve obtained from [8].

(ii) Second section, the test on the dual-fueled engine: The gas injection duration in each engine speed and torque was initially programmed onto the gas ECU, thus, the CBG was injected within the optimum ratio corresponding to the results obtained from the previous study [8]. Due to the optimum ratio of the CBG, the engine did not operate in the condition that exceeded the limitation factors as described in the introduction. The experimental procedure in this section was identical to the previous section except the CBG and diesel dual fuel was used instead of the diesel.

Finally, the measured CO₂ concentrations were plotted against the engine speed and the variations were analyzed further in the next section.

3 Results and discussions

Carbon dioxide (CO₂) is the emission occurring in the complete and incomplete combustions. For the complete combustion, the CO₂ is produced without the CO but the CO₂ and CO are both produced in the incomplete combustion. It was found that when the engine speed increased from 1,000 to 4,000 rpm, the CO₂ concentration was nearly constant. After it reached the high engine speed range, the CO₂ concentration obviously decreased. The trend of the variation in the CO₂ concentration corresponded in all reference engine torques. Moreover, It could be noticed that when the engine torque increased, the engine speed, where the CO₂ concentration began to decrease, continuously decreased as shown in Figure 3.

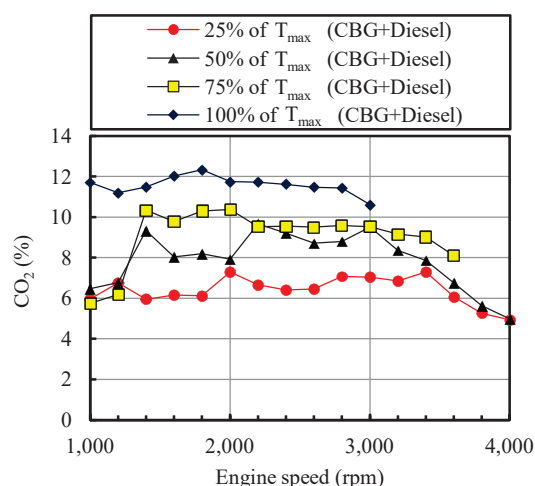
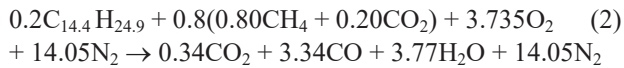
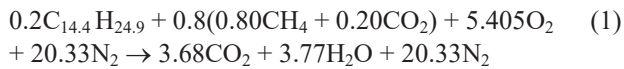


Fig. 3. CO₂ concentration from dual-fueled engine.

The variations in the CO₂ concentration were investigated in these trends because the CBG and diesel dual fuel are completely burnt during the low-to-medium engine speed. The combustion is rather complete. Only the CO₂ or the CO₂ with small proportion of the CO was produced from the combustion. Therefore, the CO₂ concentration during the low-to-medium engine speed range is higher than that of the high engine speed range. In a case of the high engine speed range, the CBG ratio is decreasingly supplied due to the limitation factors of the exhaust gas temperature and the engine knock on the optimum ratio of the CBG as obtained from the previous study [8]. This causes the ratio of the diesel, which has higher carbon atom than the CBG, to increase. The possibility that the carbon atom in diesel will not be burned consequently increases. The incomplete combustion subsequently occurs. In addition, the volumetric efficiency of the reciprocating engines decreases due to an increase in the engine speed. Mass of the air flowing into cylinders decreases. This promotes the occurrence of the incomplete combustion. The exhaust gas, thus, contains both of the CO₂ and the CO. In the light of this reason, when a proportion of the unburned carbon oxidizing to be the CO increases, the remaining proportion that will change into the CO₂ consequently decreases. These cause the CO₂ concentration during high engine speed range to decrease and vice versa for the CO concentration. This corresponds to the results obtained from the past study [9] that the complete combustion produced higher CO₂ and lower CO concentrations than that of the incomplete one. Relation between the CO₂ and the CO concentrations can be described by examples of the chemical combustion equations of the CBG and diesel dual fuel as follow. The reactions of the CBG and diesel dual fuel are considered in a case that the mass ratio between the CBG and the diesel is 80:20. The total mass of the dual fuel is 1 kg. In a case of the complete combustion with 100% of the air, the combustion equation can be found as in Equation (1). For a case of the incomplete combustion with 70% of the air, the reaction is expressed as in Equation (2). It could be seen that, in a case of the complete combustion, the mole

fraction of the CO₂ is 2.85 without the CO. After the incomplete combustion takes place, in turn, during the high engine speed range, the mole fraction of the CO₂ decreases to be 0.34 while the CO increases to be 3.34.



The CO₂ concentration releasing from the CBG and diesel dual fuel and the diesel fuel could be compared as shown in Figure 4. It could be seen that the dual fuel and the diesel had similar CO₂ concentration during the engine torque of 25% of the maximum torque. Since the optimum ratio of the CBG is relatively high during the low engine torque, the CBG being in the gas phase replaces the volume of the intake air flowing into the cylinders. The O₂ in the combustion process decreases and then the rich-fuel mixing ratio tends to occur. The unburned carbon oxidizes to be the CO in higher proportion. The CO₂ concentration consequently decreases as above mentioned relation between the CO₂ and the CO. However, during the engine torque of 100% of the maximum torque, the dual fuel produced higher CO₂ concentration than the diesel. Since the optimum ratio of the CBG is quite low during the high engine torque, the unburned carbon changes to be the CO₂ in higher proportion and vice versa for the CO as previous described. These discussions were well agreed with the one in the past study [10]. It was discussed that during the part engine load, the CO concentration sharply increases as an increase in the amount of gaseous fuel. This could be implied to be the reason why the CO₂ concentration was low as a case of the diesel fuel. In addition, it was reported for the high engine load that the CO concentration increased with an increase in gas mass ratio. Since the optimum ratio of the CBG was relatively low during the high load, the CO concentration decreased and the CO₂ concentration consequently increased to be higher than that of the diesel fuel.

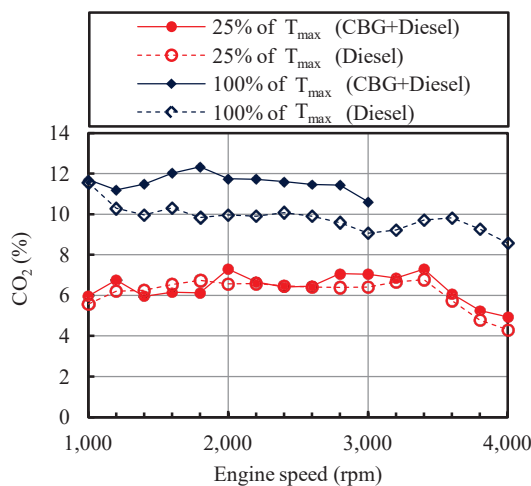


Fig. 4. Comparison on CO₂ concentration between dual-fueled and diesel-fueled engines.

4 Conclusions

The CO₂ concentration in the exhaust gas from the CBG and diesel dual-fueled turbocharged common rail diesel engine has been thoroughly investigated following the torque curves and the optimum ratio of the CBG obtained from the previous study [8]. It could be concluded in a case of the CBG and diesel dual fuel that when the engine speed increased, the CO₂ concentration decreased. Because the optimum ratio of the CBG decreases during the high engine speed range, the proportion of the diesel increases. In conjunction with a decrease in the mass of the air flowing into the cylinders due to a decrease in the volumetric efficiency, the incomplete combustion occurs. The unburned carbon oxidizes to be the CO in higher proportion than the CO₂, thus, the CO₂ consequently decreases. In addition, it was found that the CBG and diesel dual fuel produced the CO₂ nearly the same as of the diesel fuel for a case of the low engine torque. On contrary, the dual fuel had higher CO₂ concentration during the high engine torque. These variations depend on the relation between the CO₂ and the CO concentrations following the chemical combustion equation. Experimental studies on other emissions, such as, CO, NO_x, SO_x, unburned hydrocarbon, particulate matter, etc., from turbocharged common rail diesel engine dually fueled with CBG are necessary and suggested to be conducted in the future.

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