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A Study on Performance and Emission Characteristics of Direct Injection Natural Gas Engine

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Abstract

Natural gas is well known as a promising alternative fuel for internal combustion engine. In this study, a numerical model was built based on the experimental CNG converted engine. Direct injection and port fuel injection method was operated in the model on the same conditions such as design parameter, engine speed, compression ratio. The effect of injection method on engine performance and emission characteristics were compared. The mixing and burning properties were also investigated to have further insight about the fuel injection effects. The results have illustrated that the injection strategy had strongly effects on the mixing and combustion process. Due to the better mixture, the direct injection engine has showed the obviously improvement on the emission as well as engine performance. All these results have provided the data and theory to improve the engine performance for the development of the direct injection system for the engine using CNG.

Keywords: Compressed natural gas (CNG), converted engine, direct injection, engine emissions.

1. Introduction

In recent decades, natural gas in general or methane (CH4) in particular is widely used in the world as an alternative fuel gasoline and diesel fuel in internal combustion engines, with the expectation of reducing harmful emissions into the environment [1]. Generally, natural gas is available in the form of a gas, the air-fuel mixing process is easier than formal liquid fuel, therefore, the combustion process is better with a proper fuel mixture [2]. The excellent properties of natural gas are decisive in the use of natural gas as an alternative fuel for engines. In addition, the octane number of natural gas is higher than that of gasoline, which favours the use of a high compression ratio in order to improve thermal efficiency [3]. In the view of fact that the raw component in natural gas is methane, which have a low carbon/hydrogen (C/H) ratio, the COx emissions is reduced in the combustion products [4]. The natural gas vehicles are demonstrated to reduce the greenhouse gas emissions compared to gasoline or diesel fuel transportations [5, 6].

Furthermore, as a new alternative fuel CNG-powered engines have attracted a lot of attention from researchers to improve the engine performance as well as reduce emissions, in which, fuel intake process plays a key role in CNG engines [7, 8]. Recently, direct injection engines are widely used in either spark ignition or compression ignition engines due to their superiority in engine efficiency and emission performance [9]. The CNG direct injection engine has a higher working volume than the indirect injection engine because natural gas requires a larger mixture volume in the gas mixture formation process compared to the mixture on the intake manifold and indirect injection [5,10]. However, the major disadvantage of natural gas fuel is the slow flame speed compared to gasoline and diesel [8], which needs to be solved towards better combustion performance. DK Srivastava *et al*. [11] illustrated that higher combustion ratio enhanced the flame speed issue of an compressed natural gas engine. Many scientific works have demonstrated that the direct injection and injection strategies can enhance the engine performance and overcome the low flame speed disadvantage of CNG engines [12-14].

In this study, two fuel injection methods are developed on the same engine model to estimate the performance and emission characteristics of a CNG-fuelled engine. The heat release rate and fuel burned fraction are considered to evaluate the fuel mixture combustion performance. The difference between direct and port injection is analysed under many conditions throughout the study.

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2. Theoretical Framework

To have an overview of the two direct injection and indirect injection systems, it is necessary to understand the fuel supply system as well as the fuel injection method of each type of fuel injection. Fig. 1 and Fig. 2 illustrate the fuel system of a port injection and direct injection engine, respectively. CNG supply system consists of a fuel tank, a solenoid valve, pressure regulator and injection nozzle. Due to the specific characteristics of CNG fuel, the pressure required to supply fuel to the injector is 5 bar to avoid vaporization on the fuel pipeline. Otherwise, because of the low density, CNG fuel is compressed to over 200 bars and storage in a high-pressure tank.

When engine is started, the current through the coil generates a magnetism to open the solenoid valve for the CNG to be compressed from the tank to the pressure reducer. Therein, the fuel pressure is reduced to the working value, then the fuel passes through the low-pressure filter before leading to the injector. The nozzle is automatically controlled by the microprocessor, the injection time is controlled proportionally to the percentage of the throttle position through the throttle position sensor. This processor receives most of the necessary signals from the CNG fuel supply system. The control system consists of sensors that record the information of the engine working conditions, the engine ECU processes the information received from the sensors and sends a control signal to the CNG injectors to vary the nozzle opening time. The control signals to the injector are timing pulses of corresponding length proportional to the amount of CNG to be injected into the intake manifold. Fuel system sensors includes a throttle position sensor, an engine speed sensor, intake air temperature and oxygen concentration sensor. At the same time, the processor has an additional pressure sensor. CNG fuel tank, from which the signal is processed by the ECU and sends a control signal to the injector.

On the other hand, the CNG port injection system allows to improve the performance of the engine and the level of pollution emission. Unlike the mixer, this system injects fuel under a pressure of about 5 bar. This allows to provide an exact amount of fuel according to the engine's operating mode. On the other hand, since there is no venturi throat, the loading coefficient is significantly improved. CNG fuel injection is carried out in a separate way, so it reduces the possibility of flame return to the intake manifold and improves the uniformity of fuel supply to the cylinders of the engine. A microprocessor is used to control the flow of CNG loaded into the cylinder.

The direct injection method has many advantages because it allows simultaneously reducing the level of pollution and increasing the economy of the engine. Direct injection of CNG into the combustion chamber allows to combine the advantages of natural gas and the combustion of a poor mixture of layers. Moreover, the CNG injection system also inherits the advantage of the original compressed fuel, so there is no need to pump high pressure fuel. The engine can operate without loss of charge factor and in poor mixed conditions. The main disadvantage of this system is that it requires precise fabrication and adjustment of the injection system.

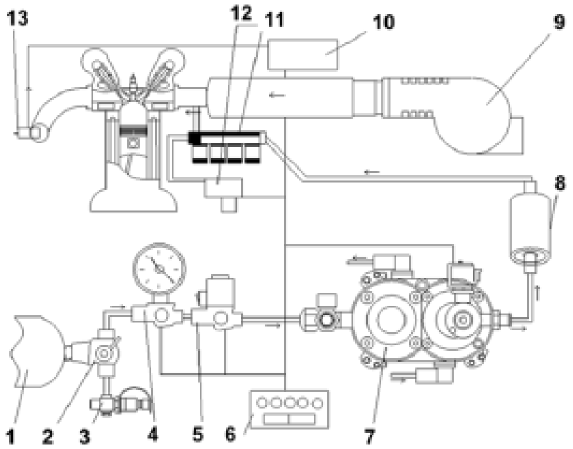

Fig 1. Schematic of fuel system on a port injection engine.
1. CNG container; 2. Check valve; 3. CNG filling valve; 4. Pressure gauge; 5. Solenoid valve; 6. Electric switch; 7. Pressure reducer; 8. CNG controller; 9. Air filtration; 10. Engine ECU; 11. CNG nozzles; 12. Pressure sensor; 13. Oxygen sensor.

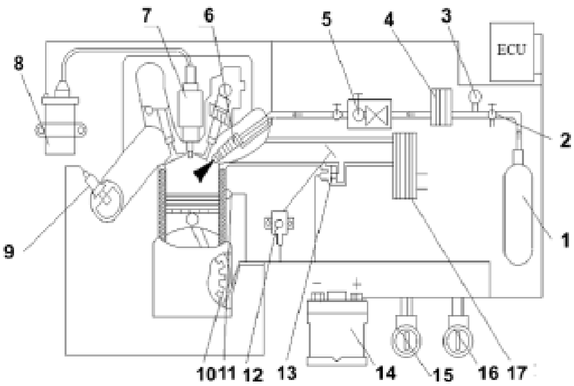

Fig 2. Schematic of fuel system on a direct injection engine.
1. CNG container; 2. Check valve; 3. Pressure gauge; 4. CNG filter; 5. Solenoid valve; 6. CNG nozzles; 7. Spark plugs; 8. Bobbin; 9. Oxygen sensor; Speed sensor; 11. Coolant water sensor; 12. Throttle sensor; 13. Intake air pressure sensor; 14. Ignition switch; 15. Electric lock; 16. Batteries; 17. Air filter.

In this work, engine performance as well as emission composition of this CNG natural gas engine are investigated. The designed converted engine QTC2015 is developed for evaluation of fuel direct injection using in CNG engine. Basic parameters of engine is presented in Table 1. Detailed information about QTC2015 engine can be found in our previous study [15].

3. Engine Simulating, Calibration and Controlling Model

3.1. Simulation Setup

Based on the QTC2015 engine structural parameters, CNG test fuel and AVL Boost software manual, the schematic of one-dimensional model of the engine is shown on Fig. 3. Table 2 lists the simulation elements in the dynamic model.

The Fractal Combustion Model is selected as the research model to consider the bowl-shape piston for and the fuel injection strategy. The fractal geometry model is suitable for investigation of turbulent combustion in spark ignition engines [16].

In the combustion model, ignition timing is considered to be the starting time of the simulation. Flame film formation is a parameter to correct the combustion delay time (C_{ign}). Flame spread rate is a parameter to correct burning delay time ($r_{f,ref}$). The mass of fuel burned in a unit of time is calculated by the formula:

$$\frac{dm_b}{dt} = \rho_u \left(\frac{L_1}{l_k}\left(\frac{\rho_{soc}}{\rho_{uz}}\right)^m\right)^{D_3-2} .A_L.S_L \qquad (1)$$

where m is the calibration parameter of turbulence model; ρ_{soc} is the unburn density at the start of combustion; ρ_{uz} is the unburn density. The small amount of burned mass at the start of wall combustion determined in wall combustion process was $\left(\frac{m_b}{m}\right)_{tr}$, where the transition time t_{tr} has been determined when a small amount of mass was burned. The laminar burning speed $S_L = c_{lfs}\, S_{L,RG=0}\left(1 - m_{fRG}\right)^d$ has been determined at the start of wall combustion (d), and allowed to adjust more S_L depending on residual gas mass coefficiency.

Table 1. Basic parameters of QTC2015 engine

Parameter	Symbol	Value	Dimensional
Cylinder bore	D	103	*mm*
Stroke	S	115	*mm*
Displacement	V_{tp}	1,03	*Liter*
Compression ratio	ε	10	-

Table 2. Elements in the dynamic simulation model

Element name	Symbols	Amount
Engine	E	1
Cylinder	C	1
Air filter	CL	1
Throttle	TH	1
Injector	I	1
Plenum	PL	1
Measuring points	MP	7
Restriction	R	3
System Boundary	SB	2
Pipes	→	8

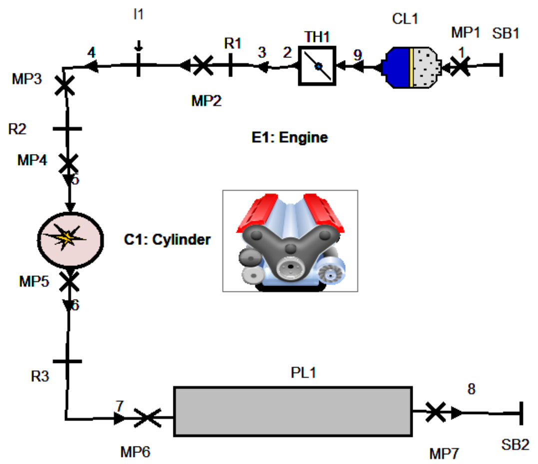

Fig 3. The simulation model of QTC2015 engine

3.2. Simulating Theoretical Framework

Fig. 4 shows the Required Octane Number (ON) results between direct injection and indirect injection, respectively. It can be seen that the required octane number decreases as the engine speed increase for every compression ratio. Conversely, as the compression ratio increases, the ON value increases at each engine speed. This result illustrates that when converting a diesel engine to a natural gas engine, the compression ratio should be reduced to avoid the abnormal combustion phenomenon at low engine speeds. Because the number of times fresh mixtures enter the engine cylinder at the same time increases as the speed increases, this fresh mixture has a significantly lower temperature than the temperature inside the combustion chamber, and it will absorb and decrease the risk of autoignition sources.

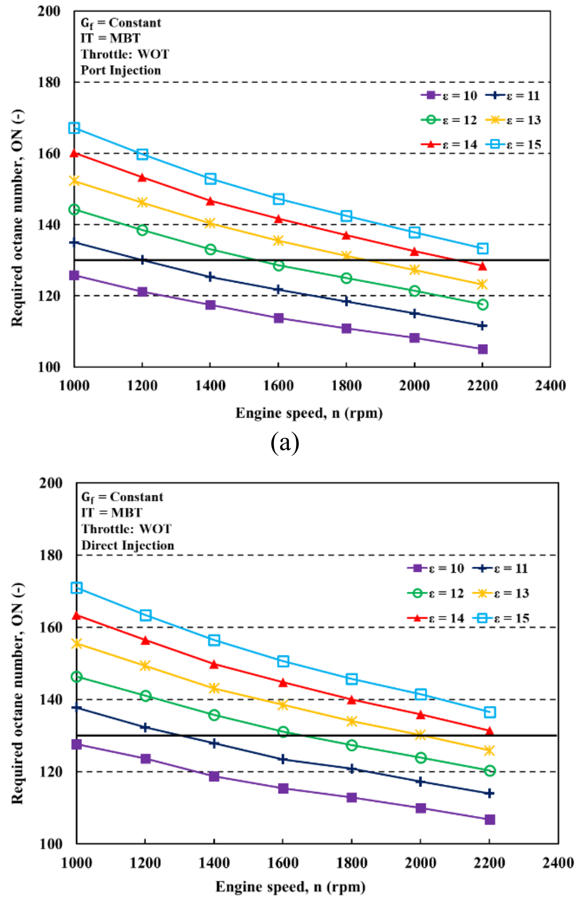

Fig. 4. Required octane number of port injection and direct injection engine

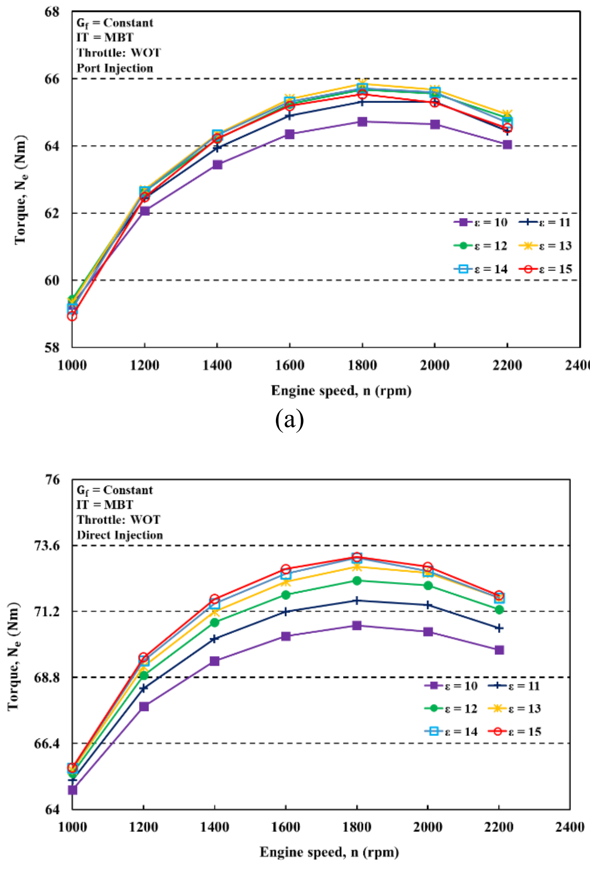

Fig. 5. The effective torque of port injection and direct injection engine

The engine test conditions on the test bench are mentioned as fully open throttle (WOT), the structure of the piston and the engine cover will be kept fixed in the form of a flat top, the ignition angle is adjusted to start the ignition before top dead center (IT:BTDC) and the compression ratio is set at ε = 10 because this compression ratio will not cause knocking in the engine. Considering the whole test area (n = 1000 - 2000 rpm) will get the torque result (M_e) between the two types of direct injection and indirect injection as shown in Fig. 5.

From the two figures above, we can see that the speed n = 1800 rpm will give the maximum torque in each compression ratio. Therefore, this speed will be selected for comparative study on the effect of fuel injection on the performance characteristics as well as emission characteristics of the CH4 conversion engine.

3.3. Controlling the Model

The operating conditions in this study were carried out specifically as following. The air intake is fully open with both direct injection and indirect injection (Throttle: WOT) to reduce drag on the intake manifold. The injection pressure on the intake line is kept constant with P_f = 1 for the case of indirect injection. The amount of fuel is kept constant during the simulation for each case of direct injection and indirect injection. Structural parameters are kept fixed in the condition that the piston and the top cover are flat. Working parameters include engine speed varying from n = 1000 rpm to n = 2200 rpm with Δn = 200. Compression ratio is changed from ε = 10 to ε = 15 and taking the fields The simulation case gives ON value less than or equal to 130.

During the simulation operation, the ignition timing is adjusted so that the model reaches the maximum torque (IT = MBT).

4. Results and Disscussion

Fig. 6 shows the change of torque according to engine speed for two cases of direct injection and indirect injection with ON less than 130. Since the geometric size of the piston top does not change, when changing the method, the fuel supply will not change the shape of the combustion chamber, but only the pressure on the top of the piston.

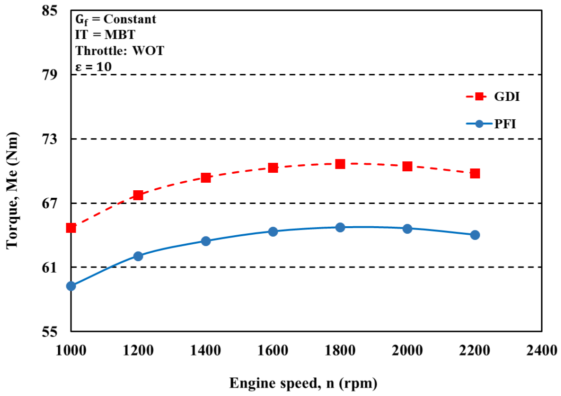

Fig 6. Effect of injection method on engine torque

Considering the engine speed in the range from $n = 1000$ (rpm) to $n = 2200$ (rpm), the torque of the two cases of fuel injection tends to change relatively similar. When increasing the engine speed, the torque also increases and the torque reaches the maximum value at n = 1800 (rpm), if the engine speed is higher, the motor torque tends to decrease. The switch from indirect injection to direct injection will increase the efficiency of the engine and increase the work done in the compression process, in addition to increasing the pressure of the gas mixture on the top of the piston, and this also cause an increase in the phenomenon of knocking (knocking) in the engine. Previous studies have demonstrated that when the engine is working at low speed with a high compression ratio, it is easier to explode than in the high-speed region. The results in the figure can show that at compression ratio ε = 10, the direct injection engine gives a larger torque than the indirect injection engine. This result shows the influence of the fuel injection method on the torque not only in the working speed range but also on the magnitude of the engine torque.

Fig. 7 shows the pressure lines in the cylinder according to the crankshaft rotation angle for each injection method under simulated conditions. The pressure lines have a marked change at the end of the compression process and at the combustion-expansion process. When changing the injection method from indirect injection to direct injection, the pressure in the engine cylinder reaches the maximum value after the dead center about 15 degrees of crankshaft rotation angle, the maximum pressure value in the cylinder increases when changing from indirect to direct injection. From the results obtained in Fig. 7, under the same research conditions, it can be confirmed that the quality of the combustion process has improved significantly due to the change in the fuel injection method. The reason for this change in pressure in the cylinder is that with the direct injection method, the fuel flow velocity is greater than that of indirect injection, so the entanglement intensity of the elements inside the cylinder is increased when the piston is close to the cylinder. The top dead center is also larger. This results in more pressure on the piston top in the case of direct injection, which tends to produce a greater value of torque and power than in the case of indirect injection.

Fig. 8 shows the effect of engine speed on the amount of air supplied into the cylinder. As it is known with constant simulation conditions, the engine cylinder volume does not change during the simulation, the amount of air supplied to the engine in the case of direct injection is larger than the case of indirect injection. This shows that the amount of fuel supplied in the case of direct injection is less than in the case of indirect injection. On the other hand, in the direct injection, the amount of fuel supplied is less, but produces a larger torque than indirect injection, so direct injection is more optimal. From the speed of 1000 (rpm) to 2200 (rpm), the amount of air supplied into the engine increases steadily in both cases of injection. This can save fuel supplied to the engine, but as the engine speed increases, more air enters the engine and has a significantly lower temperature than the temperature in the combustion chamber, this will absorb heat and reduce the self-ignition of the fuel. This phenomenon also explains why the engine torque increases to a certain value and then gradually decreases.

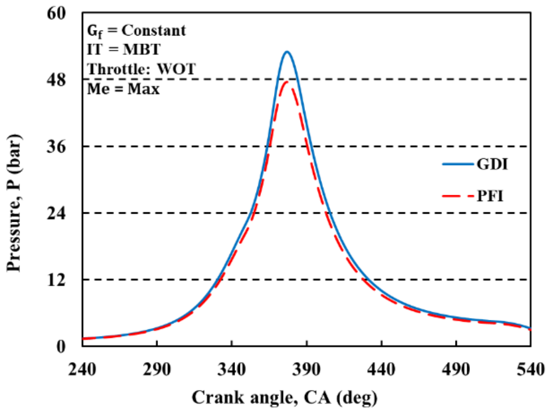

Fig. 7. The pressure in the cylinder changes with the crankshaft rotation angle

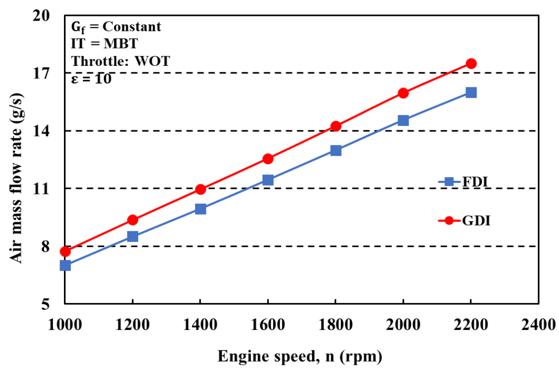

Fig. 8. Effects of engine speed on the air mass flow rate

The mass faction burned (MFB) in each engine duty cycle is normalized with a scale of 0 to 1, this amount of burned fuel is used to describe the chemical energy release process of fuel according to the crankshaft rotation angle. Fig. 9 shows the change in the percentage of fuel burned according to the crankshaft rotation angle for two injection methods, direct injection and indirect injection. Combustion process in the forced engine is divided into three main stages as follows: the flame film development stage, the main combustion stage and the late combustion stage. The first stage is the development of the flame film, which begins with the appearance of a spark at the spark plug electrode, then the formation of a flash point and the beginning of the development of the flame film, the mass factor of burned fuel MFB equal to 0 - 10%. This stage is equivalent to crankshaft rotation CA equal to 345-355 (deg). The main combustion stage corresponds to the crankshaft rotation angle from CA equal to 355-370 (deg) and the MFB reaches 90% of the volume contained in the cylinder, this is the largest heat release stage. At this stage, if the turbulent kinetic energy of the gas flow inside the engine cylinder can be controlled, the greatest amount of heat is released. The later combustion stage, also known as the end combustion, only about 10% of the remaining fuel volume continues to be burned, at this time because the piston moves down to the bottom dead center, the volume of the combustion chamber increases, the kinetics of the air flow inside the engine cylinder is reduced. The results show that the ratio of the mixture burned inside the engine cylinder according to the crankshaft rotation angle when changing the injection method is different. The results in Fig. 9 show that when converting the method from indirect injection to direct injection, the amount of fuel burned in each working cycle of the engine increases and creates a larger torque than the injection method. Indirect. This can be shown that with the direct injection method, the mixture quality and mixture preparation before and during combustion are better than the indirect method.

At the same time, the direct injection method creates better turbulent kinetic energy of the gas flow inside the cylinder, resulting in greater torque and power.

While supplying a certain amount of fuel in the simulation model, the direct injection method requires a lower amount of fuel than indirect injection but still produces power and larger torque. In addition, it is possible to optimize the ratio of burned fuel in the cylinder by changing the shape of the combustion chamber or other methods.

Fig. 10 shows the heat release rate that varies with crankshaft rotation. Obviously, the trend of heat release rate with crankshaft rotation is the same for both fuel injection methods. The speed of heat release inside the engine cylinder increases gradually from the time the spark plug turns on and reaches the maximum value after the top dead center, immediately after that, the heat release rate inside the cylinder decreases very quickly. From the results, it can be seen that the heat release rate of the indirect spray method is lower than that of the direct spray method. In both injection methods, after reaching the maximum value after the top dead center, the heat release rate gradually decreases and tends to decrease similarly. This result illustrated that the rate of heat release in the engine cylinder does not depend entirely on the amount of fuel supplied or the intake air in the engine cylinder, but also depends on the conditions to carry out the combustion reaction easily.

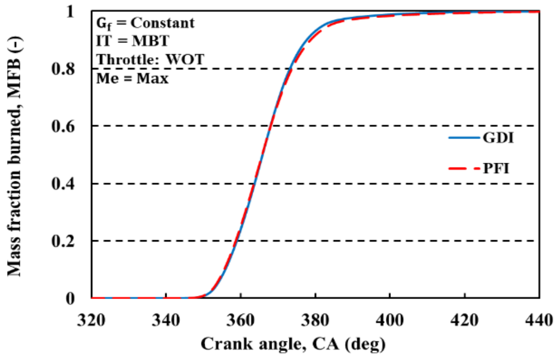

Fig. 9. The mass fraction burned according to crankshaft angle

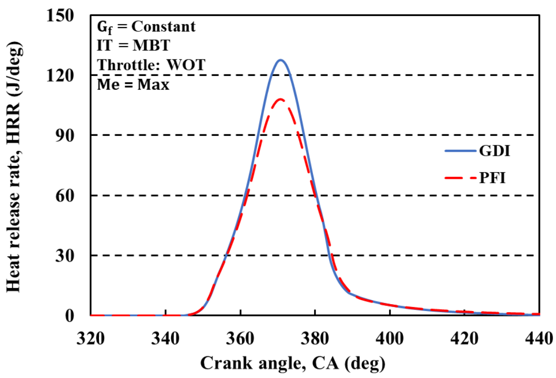

Fig. 10. Heat release rate varies with crankshaft rotation angle

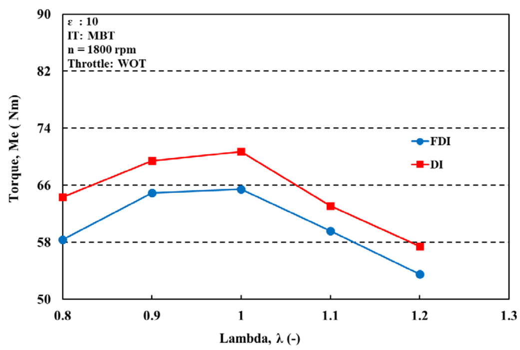

Fig. 11. Effect of injection method on engine torque

Fig. 11 shows the effect of the fuel injection method on the engine torque in the case of varying the air residue coefficient. The λ changes from the value $\lambda = 0.8$ to $\lambda = 1.2$ with a step of $\Delta\lambda = 0.1$. When the compression ratio is fixed at $\varepsilon = 10$ and the engine speed is $n = 1800$ (rpm), the change of torque when the λ increases is the same with each injection method and reaches the maximum at $\lambda = 1$, but if the λ is greater than 1, the moment immediately tends to decrease. The decreasing in engine torque is mainly due to the decrease in the temperature in the combustion chamber due to the decrease in the temperature in the combustion chamber. However, the source of temperature reduction is due to the lack of oxygen needed for the fuel's combustion reaction (when $\lambda < 1$) or lack of fuel when the engine is working (when $\lambda > 1$). On the other hand, it can be seen that the torque generated from the engine using the direct injection method is relatively much larger than that from the engine using the indirect injection method. This can be concluded that the fuel mixture formed from the direct injection method is better mixed than the indirect injection method. From the engine torque results obtained, it can be confirmed that, in order to optimize the engine's power, it is necessary to supply the amount of fuel so that the air residue coefficient of the engine is approximately 1.

Similarly to the case of fixing the amount of fuel during the simulation, when providing an appropriate amount of fuel so that $\lambda = 1$, the result is obtained about the ratio of fuel burned according to the crankshaft rotation angle.

The proportion of fuel burned according to the crankshaft rotation angle is shown in Fig. 12. It can be obtained that with the direct injection method, the entire volume of fuel supplied for a working cycle is burned faster than with the indirect injection method. For the case of direct injection, the time required to ignite the amount of fuel loaded into the engine cylinder is 95 degrees of crankshaft angle (from CA equal 345 deg to CA equal 438 deg) and in the case of indirect injection then the corresponding time is 118 degrees of crankshaft rotation (from CA equal 345 deg to CA equal 463 deg). This result shows that the time to burn all the fuel inside the engine cylinder depends very much on the degree of mixing of fuel and air in the engine. From there, it can be affirmed that the direct injection method mixes the fuel better than the indirect injection method. To clarify, it is necessary to consider the heat release ability of the two above-mentioned spraying methods.

Fig. 13 shows the heat release rate of DI and PFI engine on the same conditions. It can be seen that the trends of heat release rate with crankshaft rotation are the same for both fuel injection methods. Due to the difference in the degree of mixing of the fuel mixture in the engine cylinders, the heat released from the combustion of the fuel mixture is different. Since the simulation conditions in both injection methods are $\lambda = 1$, it can be understood that the amount of oxygen required in the mixture is just enough to easily carry out the fuel oxidation reaction, so the amount of heat released is different. It can be confirmed again due to the degree of fuel mixing in the engine. This result demonstrates that the direct injection method can be considered as a better method to maintain the kinetics of the gas flow inside the engine cylinder than the indirect injection method to improve the slow combustion of gaseous fuels.

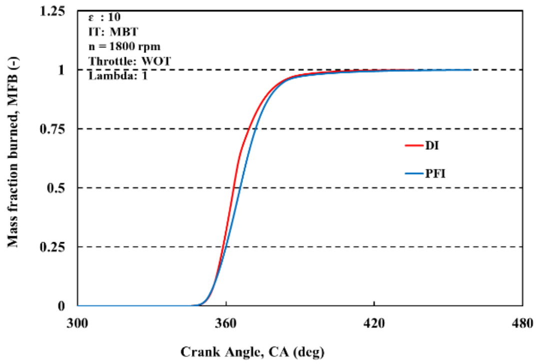

Fig 12. The mass fraction burned according to crankshaft angle

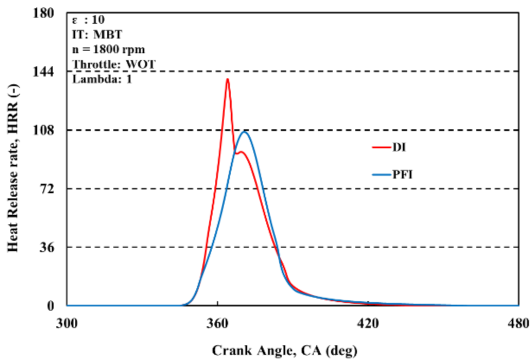

Fig 13. Heat release rate varies with crankshaft rotation angle

5. Conclusion

The engine performance as well as the combustion of a CNG converted engine are studied in this work for comparison between the direct injection and port fuel injection methods. Some conclusions are obtained as follows:

- The engine torque tends to increase when changing the injection method from indirect injection to direct injection of a conversion engine using natural gas. However, the detonation phenomenon tends to increase more strongly than the generated torque. Therefore, to avoid detonation and for the engine to operate safely and stably in the speed range from 1000 to 2200 rpm, it is necessary to reduce the compression ratio to $\varepsilon = 10$.

- The amount of fuel required to produce a unit of power of direct injection is lower than that of indirect injection.
- The combustion of direct injection engines can be improved by various methods, such as changing the shape of the combustion chamber.
- The direct injection method has many advantages over the indirect injection method and tends to replace the current indirect injection engines due to simple and quick conversion.

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