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A 0/1-Dimensional Numerical Analysis of Performance and Emission Characteristics of the Conversion of Heavy-Duty Diesel Engine to Spark-Ignition Natural Gas Engine

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Abstract

Increasing air pollution has brought about the search for alternative fuels instead of conventional fuels. It is aimed to make existing internal combustion engines work with alternative fuels with the least structural changes. Natural gas (NG) is one of the most recent alternative fuel studies because it is both cheaper and more environmentally friendly. In this study, it was aimed to minimize the dependence on petroleum-based fuels by enabling an existing compression ignition (CI) engine to operate with spark ignition with NG. For this reason, in heavy-duty diesel engine; it was modeled as 0/1-dimensional with spark plug assembly instead of diesel injector and low-pressure NG fuel injector mounted on the intake manifold. Afterwards, the performance, combustion characteristics, and emission values of the engine, which were converted to NG, were compared with the experimentally validated diesel model. In addition to the comparisons made under similar conditions, the effects of start of combustion (SOC) time and Air/Fuel (A/F) ratio changes in NG use were performed parametrically. In the same conditions, it was observed that the power, fuel consumption, and efficiency of the engine increased in NG fuel use compared to diesel fuel use. However, with the parametric studies in NG use, an improvement of 84.5% was achieved in NO_X emission without any performance loss compared to diesel use.

Keywords: Compression ignition; Natural gas; Numerical simulation; Spark ignition

Research Article

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1. Introduction

Studies predict that internal combustion engines (ICE) will continue to be used as the main power source in the transportation industry soon [1,2]. Energy consumption sources are given in Figure 1 and approximately 70% of these resources are consumed by internal combustion engines [3]. However, the use of alternative fuels such as NG, LPG [4], hydrogen, etc. in both road and off-road vehicles can reduce the dependence on petroleum-based fuels by reducing engine emissions [5]. Among these alternative fuels, NG is one of the most remarkable fuels due to its high availability, cheaper price than diesel or gasoline, and less environmental damage [6]. It is predicted that the proportion of NG-powered vehicles in the United States may be approximately 20% of the total heavy-duty vehicles by 2025 [7].

In Figure 1, it is seen that NG use has increased recently compared to petroleum-based fuel usage. Compared to diesel fuel, NG creates a homogeneous air-fuel mixture, avoiding the formation of fuel-rich zones during combustion, thereby reducing soot emissions [8]. However, NG's high resistance to auto-ignition can cause difficulties in starting the combustion process in CI applications [9, 10]. Replacing the original diesel fuel injector with a high-energy spark plug can provide a repeatable and reliable ignition source.

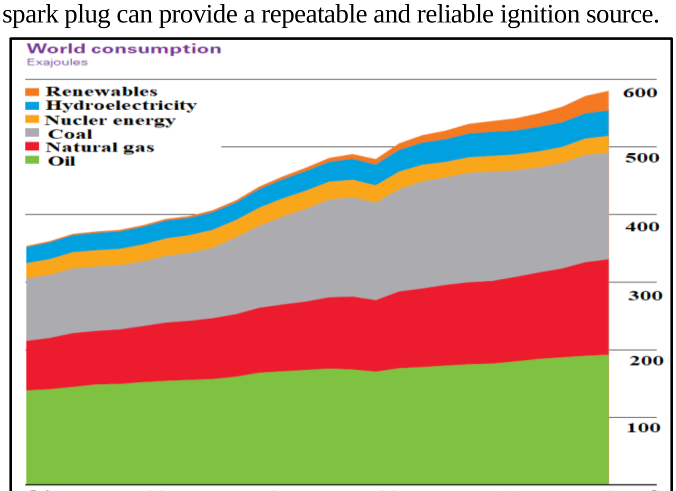


Fig. 1. World energy consumption (millions of petroleum equivalent) in the last 25 years [3]

Meanwhile, NG fuel can be injected into the air inlet using a low-pressure injector [12]. With its high compression ratio, lean combustion feature, and special combustion chamber design of a diesel engine, it can provide a higher turbulence formation, increase thermal efficiency and reduce emission compared to the combustion chamber of a gasoline engine [13].

When the literature is examined, it is possible to come across a limited number of studies in which diesel engines are converted to work with NG only with the help of injector and spark plug replacement [6, 12, 14-16]. These studies were mostly carried out experimentally, and the combustion characteristics of the cylinder that occurred in the use of NG in the diesel engine converted without any changes in the combustion chamber were investigated. In addition, the effect of methane, propane, and butane ratios in NG's content on combustion and performance has also been investigated by parametric studies [14, 17]. Studies that have been converted to work with an existing diesel engine with NG were generally performed at low and medium loads for a compression ratio of 13.3: 1 or less. In this study, it is seen that the desired examinations can be made in a high compression ratio engine such as 17.5:1 by saving time and cost with the use of a 0/1-dimensional program. As a result, this study aims to convert an existing diesel engine with a high compression ratio so that it can operate with NG at high speed and full load, and numerically examine the effects of combustion start time and A/F ratio on performance, combustion characteristics, and emission values.

2. Engine simulation model with AVL Boost

In this study, the diesel injector was removed and replaced with a spark plug in a three-cylinder diesel engine and modeled as 0/1-Dimensional using AVL Boost commercial software. In addition to the injector-spark plug replacement, holes were opened on the intake ports for natural gas injection, natural gas injectors were mounted, and natural gas (pure CH_4) was sent premixed instead of diesel fuel. The flow diagram where all the components and connections of the modeled engine could be seen was shown in Figure 2. Here, the connection points between each element (1-28), plenums representing intake and exhaust manifolds (PL1-PL2), measuring points (MP1-MP10) for obtaining the desired temperature, pressure, emission, etc., intercoolers (CO1-CO2) for cooling the hot gas from the compressor and exhaust manifold, air filter (CL1) for cleaning air entering the engine, the turbocharger (TC1), which provides more air intake into the cylinder, with using exhaust gas waste heat, cylinders (C1-C3), system boundary conditions (SB1-SB2) and restriction (R1) which determines how much of the exhaust gas goes to the intake manifold. "General species" was selected in the model, and emission values were obtained.

The flow through the pipes and manifolds was simulated one-dimensional with the flow coefficients that must be defined as the initial data. The flow was defined by continuity, momentum, and energy conservation equations; the friction coefficient and the heat transfer to the walls vary throughout the pipes [18, 19].

The cylinder block is the part of the program which contains the most sub-models and requires the most information. In this section, engine construction information, initial temperature, and pressure values, combustion model, fuel quantity, the start of the combustion information, emission calibration factors, heat transfer model, heat transfer, surface area information, as well as intake and exhaust valve information should be introduced to the program correctly. The engine characteristics used in the model were given in Table 1. Vibe 2 Zone model was used for the combustion model, and Woschni 1978 heat transfer model was used for heat transfer. Detailed information about the modeling stages and the models used were given in Aktas's studies [20,21].

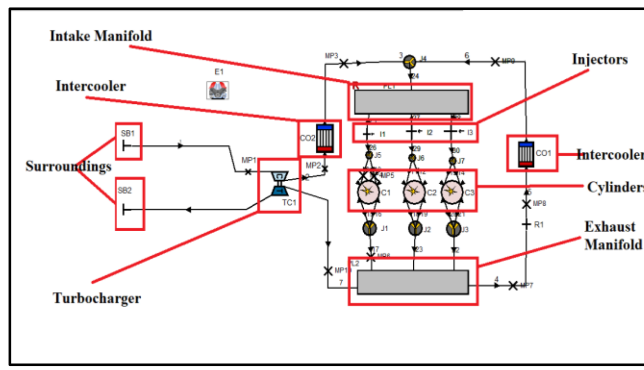


Fig. 2. AVL Boost image of the three-cylinder converted engine modeled.

Table 1. Engine Specifications

Specification	Value
Displaced volume - total (cm^3)	2930
Number of cylinders	3
Bore / Stroke (mm)	104 / 115
Connection rod (mm)	182
Compression ratio (-)	17.5:1
Injection system	Port fuel injection
Compressor pressure ratio	1.40
Valve Time (° CA)	IVO: 28 BTDC
	IVC: 60 ABDC
	EVO: 65 BBDC
	EVC: 33 ATDC

In Table 2, the speed at which the simulation takes place, the combustion start time, the A/F ratio, and fuel mass were given. While underlined, bold, and italicized were the main operating parameters, the others were parametric operating values. Methane was used as an NG surrogate and its lower calorific value was given in Table 2. The natural gas injection was carried out in a continuous regime with premixed 64 mg per cylinder (for 26.8 A/F ratio). Unlike the diesel injector, the spray angle, duration, or time were not defined.

Table 2. Simulation parameters

Parameter	Value
Engine speed (m/s)	2300 @ Full load
Start of Combustion (°CA ATDC)	-30, -20, -15, -10, -5, -0.5, ***0***, +2, +5, +15
Air/Fuel ratio (-)	22, …***26.8***,…30
Fuel mass (mg/st) (each cylinder)	79.8,…***64,***….58.9
Lower Heating Value (MJ/kg) (CH_4-Diesel)	50-44

3. Results and discussion

In this section, the performance, in-cylinder combustion characteristics, and emission values obtained by converting the diesel combustion model, which has been verified with experimental data under full load at 2300 rpm, into a spark ignition model operating with methane, have been investigated. First, the effects of the SOC time were examined, then it was aimed to reduce the NO_X emission without sacrificing power, and fuel consumption, and the effects of the A/F ratio were investigated by selecting the appropriate SOC time.

3.1 Start of combustion (SOC) time effects.

In this section, the effect of the start of combustion (SOC) time between -30 °CA ATDC and +15 °CA ATDC on the performance, combustion characteristics, and emission behavior were investigated for the ratio of A/F = 26.8.

One of the most important issues to be considered when converting from a diesel engine to a spark-ignition engine operating with gas fuels such as NG, LPG, Syngas is the maximum pressure rise rate. By controlling this ratio, necessary engine calibrations should be made to prevent knocking, especially at medium and high loads. Although the knock limit is determined as 1 MPa/degree [22] for CI engines in the literature, this limit is generally accepted as 0.4 MPa/degree [9] with the conversion from CI engines to SI engines. In Figure 3, it was seen that SOC time can cause knock by exceeding the MPRR limit before -2 °CA ATDC, and it was defined as "over the limit". To overcome this, it was ensured that flame spread was dominant during the combustion event with delayed SOC timing. Thus, by preventing knock, safe margin design requirements were met [9, 23]. In Figure 3, it was seen that BMEP and BTE values increased by 19.3% and 2.9%, respectively, compared to the reference diesel combustion regime (+ 0 °CA ATDC), depending on the increased power in NG use. While the optimum SOC time for BMEP was -0.5 °CA ATDC, this value was -1 °CA ATDC for BTE. Combustion starting before these values causes the power to decrease by doing work against the piston throughout the compression stroke.

In Figure 4, the changing brake power and brake specific fuel consumption values were given depending on the SOC time. It was seen that brake power increase by 21.14% and BSFC values decrease by 16.93%, compared to the reference diesel combustion regime (+0 °CA ATDC), depending on NG use. Thanks to its high lower heating value, NG provided better performance for the same SOC time compared to the diesel combustion regime. The optimum combustion start time was determined as -0.5 °CA ATDC. In addition, considering the knock limit, it was seen that the combustion start time was more applicable for -2 °CA ATDC and after.

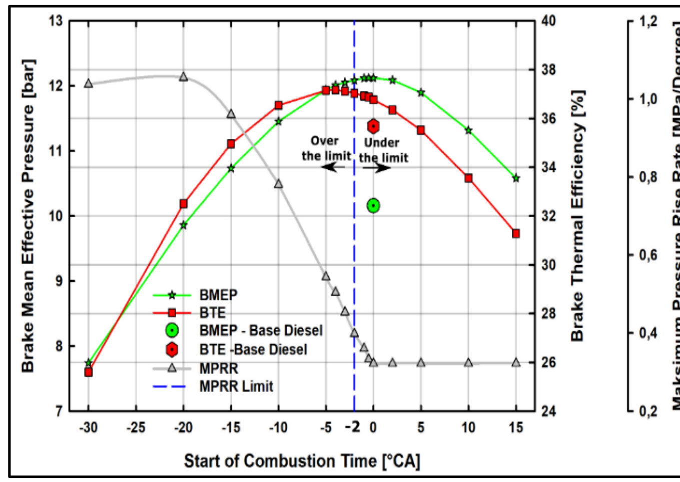


Fig. 3. The effect of SOC time on BMEP, BTE, and MPRR

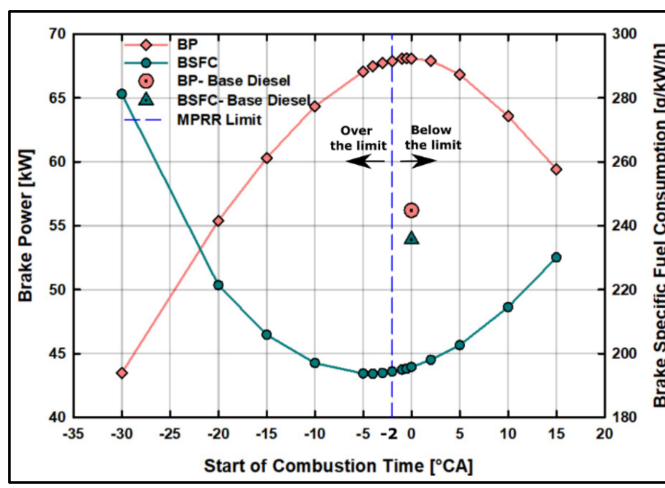


Fig. 4. The effect of SOC time on brake power and specific fuel consumption

In Figure 5, the changing maximum mean pressure and maximum mean temperature values were given depending on the SOC time. In general, it was seen that the in-cylinder pressure decreases with the retarded SOC time and the combustion phase going into the expansion stroke, so the MPRR decreases. Similarly, taking the SOC time retarded caused a decrease in the average maximum temperature inside the cylinder. Due to the high lower heating value of NG, both in-cylinder pressure and temperature values were obtained higher at + 0 °CA ATDC compared to the reference diesel combustion regime.

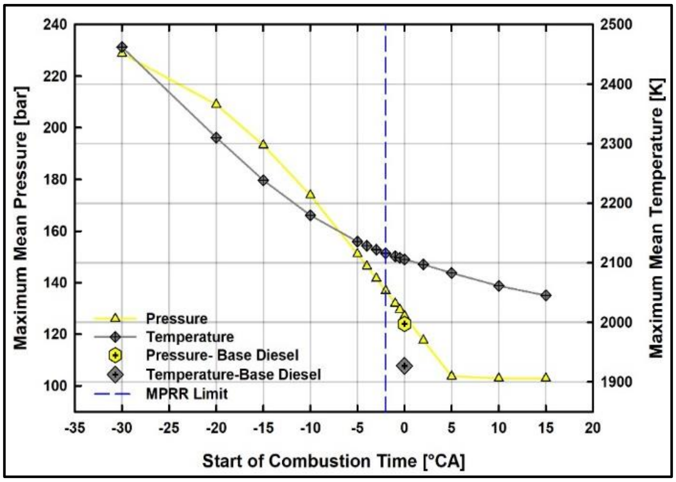
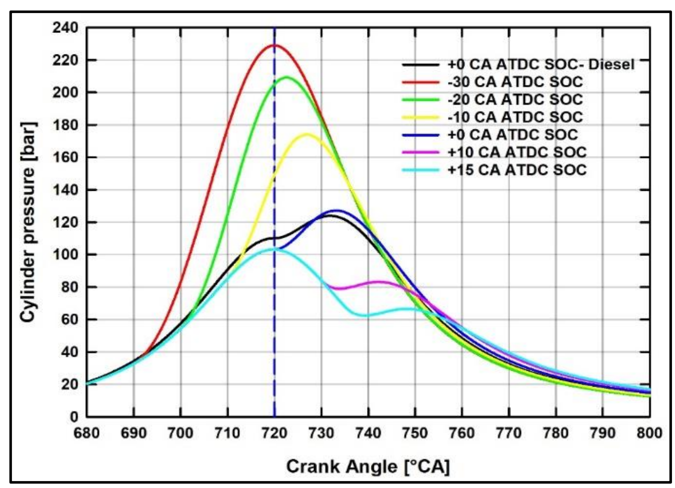


Fig. 5. The effect of SOC time on maximum mean pressure and temperature values

In Figure 6(a), the changing effect of SOC time on in-cylinder pressure values was given. In the figure, as expected [8, 15] with the advanced SOC time, the point where the maximum pressure was formed in the advanced and the MPRR rate increased. The advance of the SOC time caused the in-cylinder pressure to rise before the top dead center and worked against the upward movement of the piston. This counterwork caused the power obtained from the crankshaft to decrease. On the contrary, SOC retarding the time too much causes the maximum cylinder pressure to be formed in the expansion stroke, reducing both the maximum pressure and the work done on the piston [23]. In Figure 6(b), the changing effect of SOC time on in-cylinder HRR values was given. Generally, retarding the SOC time increases the in-cylinder heat release rate, but it was observed that it occurs well after the top dead center and does not achieve the expected increase in power generation due to the cooling in the expansion stroke. At the time of +0 °CA ATDC SOC, it was seen that NG released more heat compared to the conventional diesel combustion regime due to its lower heating value.

In Figure 7(a), the changing effect of SOC time on in-cylinder temperature values was given. With the advance of the SOC time, the values of the temperature and pressure inside the cylinder increased with the effect of the compression stroke. On the contrary, with the delay of the SOC time, the in-cylinder temperatures decreased with the transition to the expansion stroke and the cooling due to the length of the stroke length of the diesel engine. In the time of +0 ° CA ATDC SOC, it was seen that the temperature inside the cylinder was higher than the conventional diesel combustion regime due to the higher heat release of NG. In Figure 7(b), the changing effect of SOC time on emission values was given. The emission values given in the graph were normalized according to the +0 °CA ATDC diesel combustion regime. NO_X emission formation is observed in the stoichiometric region around the diffusion flame and when local temperatures are high. While it was expected that the NO_X emissions would decrease compared to the diesel regime due to the delivery of NG in + 0 °CA ATDC as premixed, it has been observed to increase. The reason for this can be shown that the in-cylinder temperatures dominated the homogeneous diffusion flame. According to the results, with the retard of the SOC time, the in-cylinder temperature values decreased;therefore, NO_X emission values decreased. With the SOC time taken from +0 °CA ATDC to +15 °CA ATDC, a 68% reduction in NO_X emission was observed. THC and CO emissions appear as incomplete combustion products. With the delay of the SOC time, the CO emission increased by 15%, while the HC emission was reduced by 80%.

As a result, the effects of different SOC times on performance, combustion, and emission characteristics were examined in this section. First, it was decided that the SOC time should be after -2 °CA ATDC according to the knock limit, and then it was determined that the optimum SOC time should be +15 °CA ATDC, without compromising power and fuel consumption, especially by concentrating on NO_X emission. In the next section, the effects of different Air/ Fuel ratios for + 15 °CA ATDC SOC time were investigated.



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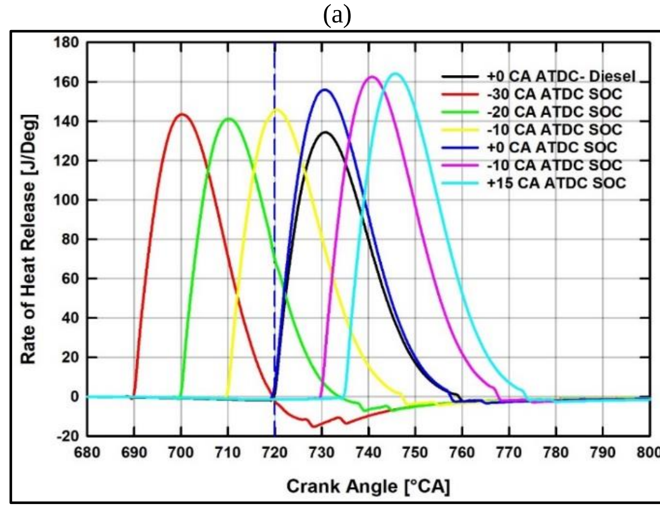
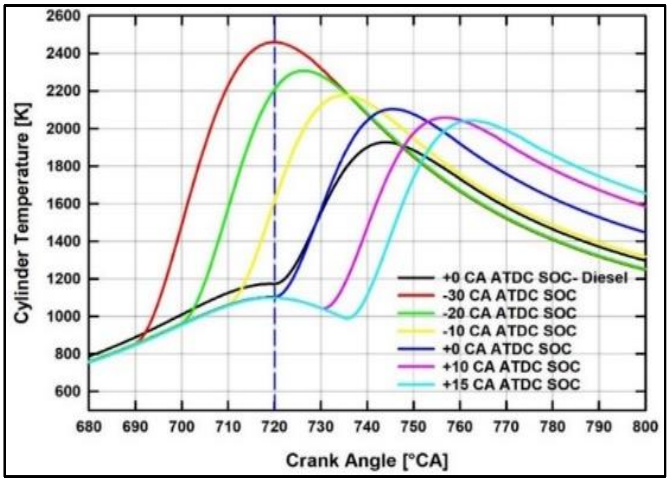


(b)

Fig. 6. The effect of SOC time on (a) in-cylinder pressure and (b) rate of heat release



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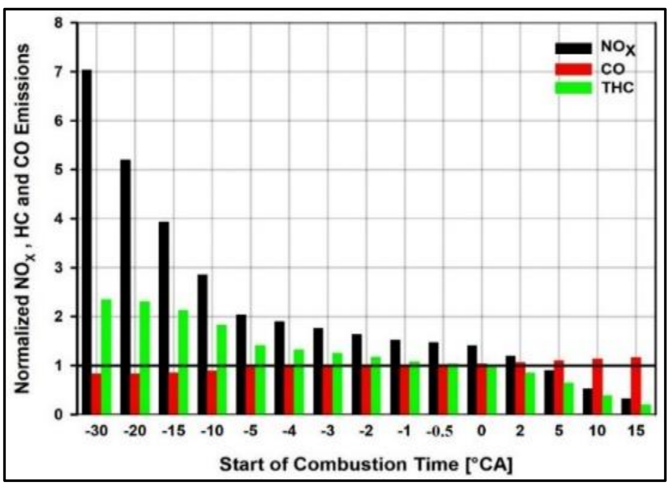


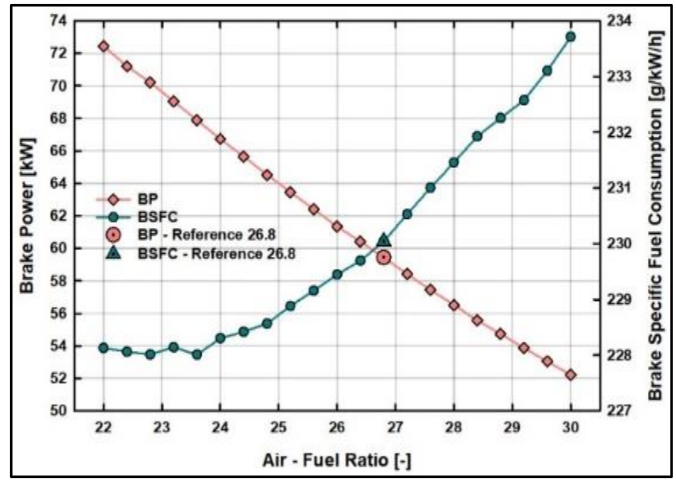
(b)

Fig. 7. The effect of SOC time on (a) in-cylinder temperature and (b)emissions

3.2 Air/Fuel (A/F) ratio results

A/F ratio is one of the basic engine operation variables. In this section, engine performance, combustion, and emission characteristics have been investigated for different values with an A/F ratio between 22 and 30 at +15 °CA ATDC SOC time. In the figures, the "Reference 26.8" value is the A/F ratio in NG use at SOC time + 15 °CA ATDC.

In Figure 8(a), the effect of the A/F ratio on brake power and brake-specific fuel consumption values were given. It was observed that due to the worsening of the combustion with increasing A/F ratio, there was a decrease in brake power and an increase in fuel consumption. Compared to the reference diesel fuel use, the A/F ratio provided a better brake power and fuel consumption performance between 22-28. However, as the mixture gets leaner, the brake power decreases, and the fuel consumption increases. In Figure 8(b), it was seen that BMEP and BTE values generally decrease with the increasing A/ F ratio. Compared to the use of reference diesel fuel, a better BMEP value was achieved in the range of 22-28 A/F ratio, while BTE values for all A/F ratios remained below diesel values.



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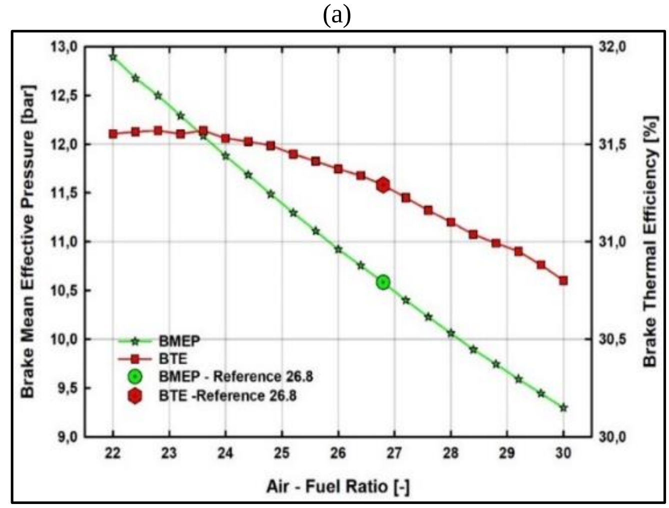


(b)

Fig. 8. The effect of the A/F ratio on (a) brake power and brake specific fuel consumption

In Figure 9, the effect of the A/F ratio on maximum mean pressure and maximum mean temperature values were given. With increasing the A/F ratio, the temperature inside the cylinder decreased considerably, while the average maximum pressure in the cylinder changed very little. The reason for this was that as seen in Figure 10(a) since the SOC was moved from +0 °CA ATDC to +15 °CA ATDC, the ignition cannot start before the top dead center, and the increased A/F ratio increases the in-cylinder pressure. However, when looking at the 2nd peak (at about + 30 °CA ATDC) where the maximum pressure was formed, the increasing A/F ratio caused a decrease in the cylinder pressure as a result of the decrease in flame velocity [17] and the worsening of the combustion.

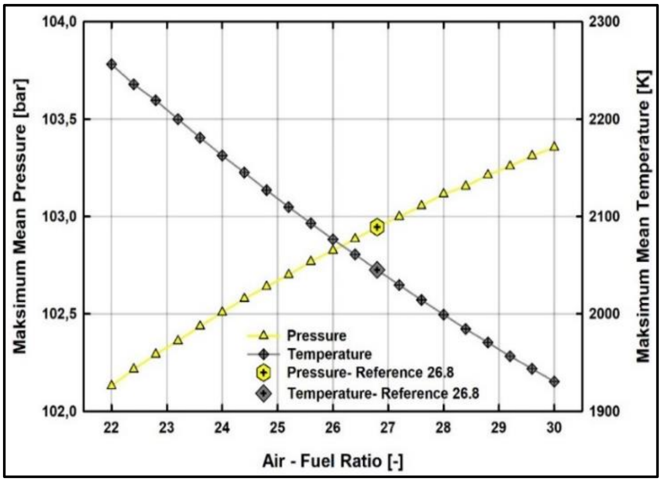
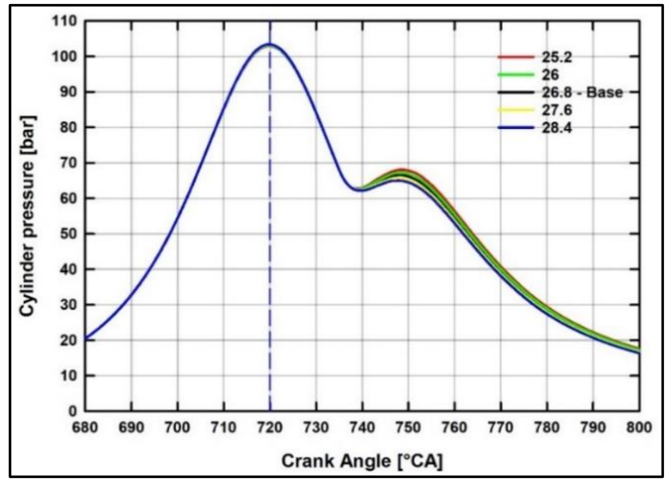


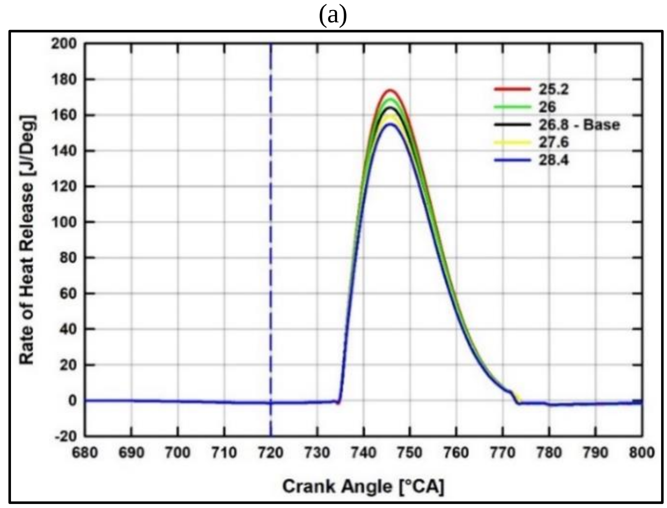
Fig. 9. The effect of the A/F ratio on maximum mean pressure and temperature

In Figure 10(b), the effect of the A/F ratio on the in-cylinder rate of heat release was given. The increased amount of fuel both increased the available chemical energy per cycle and increased the flame speed, leading to the development of in-cylinder combustion and increased the rate of heat release. This increase in the heat release rate also had a positive effect on the performance values as mentioned before.

In Figure 11(a), the effect of the A/F ratio on the in-cylinder temperature was given. The increased A/F ratio worsened the combustion in the cylinder, reducing the amount of heat released. This decreasing amount of heat also caused the resulting average maximum temperature values to decrease. In Figure 11(b), the effect of the A/F ratio on the emission values was given. All values in the graph were given by normalizing with 26.8 A/F ratio in NG usage at + 15 °CA ATDC SOC. In general, the increase in the A/F ratio caused leaner combustion and caused the temperature to decrease with the heat release in the cylinder. The falling in-cylinder temperature, on the other hand, directly reduced the formation of NO_X. It was previously discussed that the A/F ratio could be 28 in terms of performance. In this case, NO_X emission was reduced by 46%. However, due to the influence of the A/F ratio on local temperature, CO emissions increased with a higher acceleration, which in turn affected the reaction rates associated with the formation of CO in post-flame products [17]. Due to the increasing amount of O_2 in the cylinder, CO emission decreased, but it was observed that there was an increase in HC emissions with more reactions of O_2 molecules with CO emissions. While the decrease in CO emissions was 24%, the increase in HC emissions was 34%.



(a)



(b)

Fig. 10. The effect of the A/F ratio on the (a)in-cylinder pressure and (b) the rate of heat release

As a result, when both SOC time and A/F ratio effects were examined, it was seen that there was an 87.5% reduction in NO_X emissions, 73.4% in HC emissions, and 15% in CO emissions. Under normal use conditions, it is known that the SCR conversion efficiency is between 70% and 95% [24]. As it is understood, it has been concluded that there was no need to use SCR to reduce NO_X emissions in the relevant cycle and load conditions of the engine. In addition, since the formation of soot emission remains in the range of 10^{-6} -10^{-7} g / kWh, there was no need to use DPF.

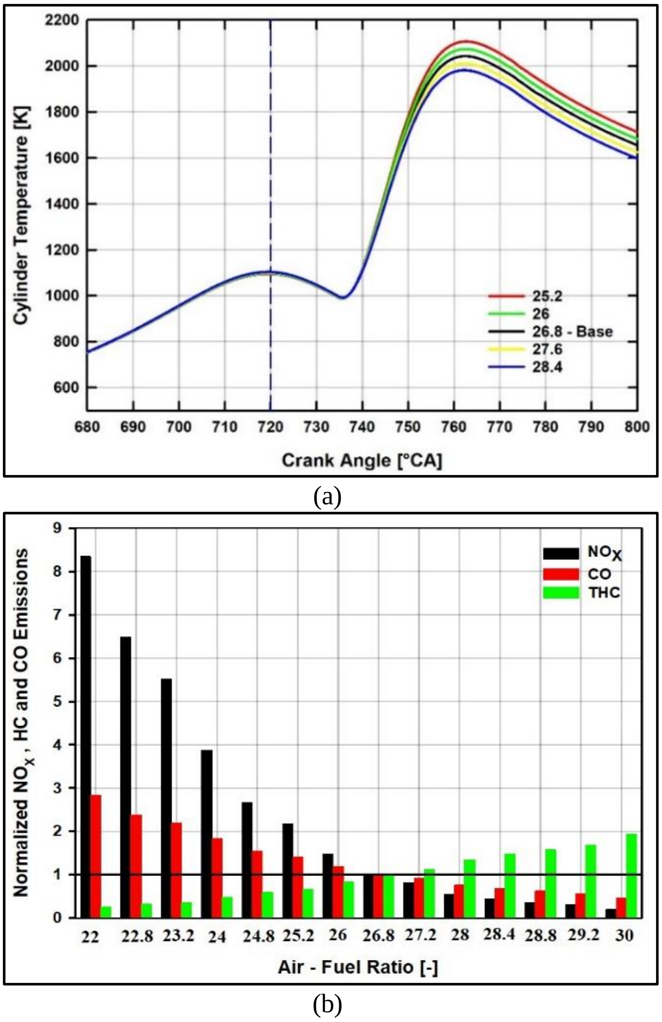


Fig. 11. The effect of the A/F ratio on (a) in-cylinder temperature and (b) emissions

4. Conclusion

For diesel engines that have exceeded a certain mileage, it is an economical approach to replace the diesel injector with a spark plug and mount a low-pressure NG fuel injector on the intake manifold. With the advantage of working with a leaner mixture compared to conventional SI engines, high compression ratios, and the shape of the combustion chamber, engine efficiency generally increases, and emission values decrease. In this study, the effects of SOC time and A/F ratio in 0/1 dimensions were investigated with the conversion of a modern diesel engine with a high compression ratio to operate with natural gas, contrary to the studies in the literature. The main results obtained from this study can be summarized as follows:

1- NG fuel use for similar SOC time provided more power and less fuel consumption compared to diesel fuel usage.

2- For similar SOC time, there was a slight increase in NO_X emissions due to the increase in cylinder temperatures in NG fuel use. While HC and CO emissions remained unchanged, soot emissions remained below the norm values.

3- Although the increased A/F ratio in NG use causes a decrease in performance, it has maintained its advantage up to A/F = 28 compared to diesel fuel use.

4- It has been observed that for the proper SOC time and A/F ratio at maximum speed and full load, after treatment equipment such as SCR and DPF will not be required.

As a result, it has been observed that instead of a diesel injector with control complexity (injection time, duration, angle, profile, etc.), a spark plug assembly and a low-pressure NG fuel injector mounted on the intake manifold can provide an environmentally friendly engine with more performance and low emissions.

Acknowledgment

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Nomenclature

LPG:	liquefied petroleum gas
BTDC:	before the top dead center
ABDC:	after bottom dead center
BBDC:	before the bottom dead center
ATDC:	after the top dead center
CA:	crank angle
HRR:	heat release rate

Conflict of Interest Statement

The author declares that there is no conflict of interest in the study.

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