ARCHIWUM MOTORYZACJI
5 pp. 103–111 (2000)

Performance and emissions of a spark-ignition engine fuelled with methane

HAKAN SERHAD SOYHAN, HALIT YASAR, CEM SORUSBAY[1]

Sakarya University, Engineering Faculty, Mechanical Engineering, Dept., Turkiye
[1]Istanbul Technical University, Mechanical Engineering Faculty, Turkiye

In this study, the engine in-cylinder flow and its effect on performance and exhaust emissions of a spark ignition engine fueled with methane were evaluated. Two-dimensional computations were investigated by a mathematical model employing a global kinetic reaction mechanism for the oxidation of methane. The following results have been achieved; methame is superior to other fuels in terms of its lean burn limit and combustion characteristics at low temperatures although the combustion speed is slower. The NO_x emissions are also lower than the conventional fuels.

1. Introduction

Stricter requirements have been imposed on engines in recent years with respect to environmental preservation and energy conservation. For this reason, in recent years, a great deal of researches is being done on natural gas engines. They have been attracting attention because of their superior emission characteristics compared to conventional engines [1, 2, 3].

Natural gas can be used in internal combustion engines either by converting an existing diesel engine for dual-fuel operation or by converting the base engine to a dedicated natural gas engine operating with a spark ignition system. Spark ignition engines only require the installation of additional fuel supply equipment, and there is no need for a major change to the engine construction. Because of its superior antiknock property, a diesel engine can easily be converted to a high compression spark ignition engine. Dedicated natural gas engines, which use only natural gas, are far superior in terms of exhaust emissions [4].

Natural gas is an excellent fuel for Otto-cycle (spark-ignition engines). As a gas under normal conditions, it mixes readily with air in any proportion. Unlike liquid fuels, it does not need to vaporize before it will burn. Thus, cold engine starting is easier (especially at low temperatures) and cold-start enrichment is not required. Cold-start enrichment is a major source of CO emissions and emissions-related problems in gasoline-fueled vehicles [5].

This study is concerned especially with the parameters such as compression ration, ignition timing, swirl ratio, engine speed and mixture strength. The effects of these parameters on performance and emissions are investigated in this paper.

2. The engine specifications and calculation parameters

The considered engine was a spark-ignition engine. The engine specifications are given in Table 1. Natural gas is a fuel many constituents, but methane is the main constituent. The natural gas has about 96% methane. For this reason, in the study, pure methane is considered as fuel. The engine parameters used in the study are given in Table 2.

Table 1. The engine specifications

Bore, mm	108.0
Stroke, mm	95.5
Connecting rod length, mm	155.7
Bowl diameter, mm	62.9
Squish clearance, mm	2.9
Fuel type	Methane

Table 2. Calculation parameters

Excess air ratio	1.0, 1.1, 1.2
Swirl ratio	0 – 1 – 2
Compression ratio	8, 10.58, 13
Spark advance, CA	15, 20, 30
Engine speed, rpm	2000, 3000, 4000

3. The combustion model

In cylinder flow conditions are determined for an axially-symmetrical combustion chamber geometry in two-dimensions [6].

The conservation of chemical species k is given by,

$$\frac{\partial \rho k}{\partial t} + \frac{1}{r}\nabla(r\rho_k u) = \frac{1}{r}\nabla\left[r\rho D\nabla\left(\frac{\rho k}{\rho}\right)\right] + \rho_{kc}, \quad (1)$$

where D is the species diffusivity and ρ_{kc} is the rate of change of ρ_k due to chemical reactions.

The continuity equation is given by,

$$\frac{\partial \rho}{\partial t}+\frac{1}{r}\nabla(r\rho u)=\rho_s. \tag{2}$$

Conservation of momentum is given by,

$$\frac{\partial}{\partial t}(\rho u)+\frac{1}{r}\nabla(r\rho uu)=-\nabla p+\frac{1}{r}\nabla(r\sigma)-\frac{\sigma_o-\sigma_w{}^2}{r}\nabla r+\rho G, \tag{3}$$

where p is pressure, σ is the two-dimensional stress tensor, G is the external force per unit mass, σ_o is the cylindrical viscous stress and w is the swirl velocity.

Angular momentum equation that determines the swirl velocity w is,

$$\frac{\partial}{\partial t}(r\rho w)+\frac{1}{r}\nabla(r^2\rho wu)=\frac{1}{r}\nabla(r\tau), \tag{4}$$

where τ is the swirl stress vector.

The internal energy equation is,

$$\frac{\partial}{\partial t}(\rho I)+\frac{1}{r}\nabla(r\rho Iu)=-\frac{p}{r}\nabla(ru)+\sigma\nabla u+\tau\nabla(w/r)+\frac{\sigma_0}{r}u\nabla r-\frac{1}{r}\nabla(rJ)+Q_c, \tag{5}$$

where I is the specific internal energy, J is the heat flux vector, Q_c is the rate of chemical heat release.

The kinetic and equilibrium reactions considered in the combustion model are specified in Table 3.

Table 3. Chemical reactions of the combustion model

Kinetic reactions	$2\,CH_4+4O_2\rightarrow 2\,CO_2+4\,H_2O$ $O_2+2\,N_2 \Leftrightarrow 2\,N+2\,NO$ $2\,O_2+N_2 \Leftrightarrow 2\,O+2\,NO$ $N_2+2\,OH \Leftrightarrow 2\,H+2\,NO$
Equilibrium reactions	$H_2 \Leftrightarrow 2\,H$ $O_2 \Leftrightarrow 2\,O$ $N_2 \Leftrightarrow 2\,N$ $O_2+H_2 \Leftrightarrow 2\,OH$ $O_2+2\,H_2O \Leftrightarrow 4\,OH$ $O_2+2\,CO \Leftrightarrow 2\,CO_2$

4. Computations and discussions

The model has been used to determine the effects of the parameters such as compression ratio, ignition timing, swirl ratio, engine speed and mixture strength on engine performance and exhaust emissions. The calculation results are given as diagrams.

4.1. Effects of ignition timing

Ignition timing has significant influence on cylinder pressure history depending on the combustion rate of the mixture. In SI engines, increase of spark advance increases the tendency of combustion knock, while the decrease of spark advance reduces maximum cylinder pressure, and thus maximum engine power output, of the engine. The effect of ignition timing on cylinder pressure history is shown in Fig. 1 for different spark advances (15°, 20° and 30° BTDC). The maximum cylinder pressure is 4.58 MPa at 6 CAD ATDC for 30° spark advance value. In the case of spark advance of 20°, the maximum cylinder pressure is 4.08 MPa at 8.98 CAD ATDC. For 15°, these values are 3.79 MPa at 12.16 CAD ATDC. The optimum spark timing is obtained by ignition at 30 CAD BTDC.

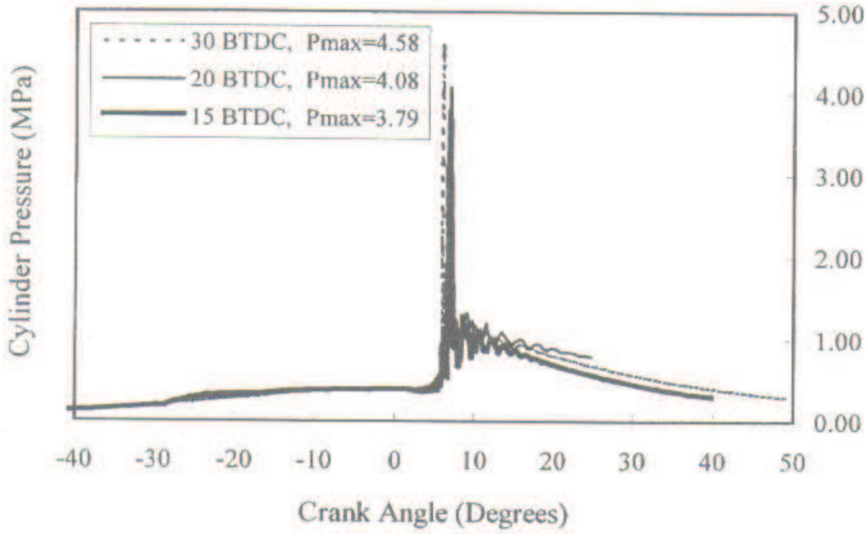

Fig. 1. The effect of ignition timing on P-α history

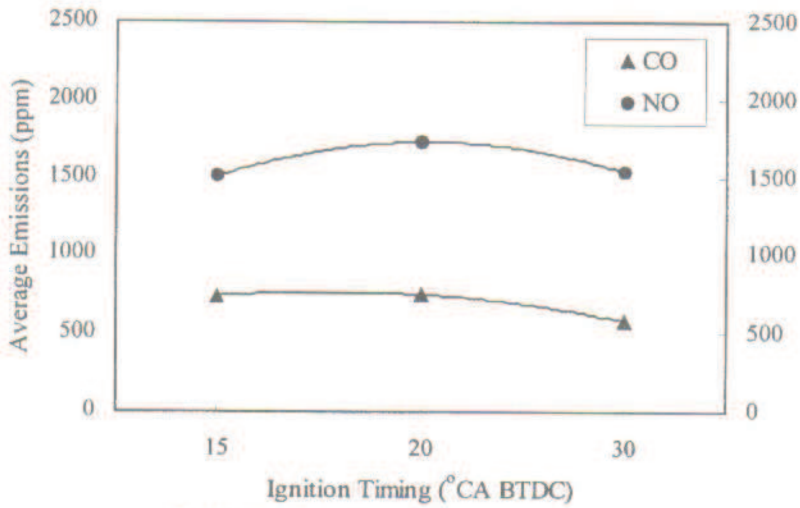

Fig. 2. The effect of ignition timing on emissions

Figure 2 shows the CO and NO emissions as a function of ignition timing. In this study, the all emissions were determined for three different spark advance values (15–20 and 30 crank angle degrees), and all of the emission data are average values at 20 CAD ATDC. It can be seen that CO emissions were lower with increasing spark advance. This is due to the increase of transformation of CO to CO_2 because of earlier combustion starting time. NO emissions decreases with increasing spark advance from 20 to 30° CA due to the decreasing gas temperatures.

4.2. Effects of compression ratio

In spark ignition engines increase of the compression ratio increases the thermal efficiency of the cycle and thus increases the maximum power output of the engine, while reducing the specific fuel consumption. However, this increase is limited due to knocking tendency of the engine [7].

In Figure 3, the effect of compression ratio on cylinder pressure history is shown for the compression ratios of 8, 10.8 and 13. At the compression ratio of 8, the maximum cylinder pressure is 0.29 MPa and the crank angle reached this pressure is 0.38 CAD BTDC. In the case of compression ratio of 10.8, the maximum cylinder pressure is 3.79 MPa and the crank angle is 12.16 CAD ATDC. For compression ratio of 13, these values are 4.65 MPa and 180°. It can be seen that the engine can operate with higher compression ratios without knocking tendency, compared to gasoline engines, obtaining higher power output because of the high octane number of methane.

The variation of CO and NO emissions are given as a function of compression ratio in Fig. 4. It may be seen from the figure that both CO and NO emissions increased with increasing compression ratio. The emissions were lower due to

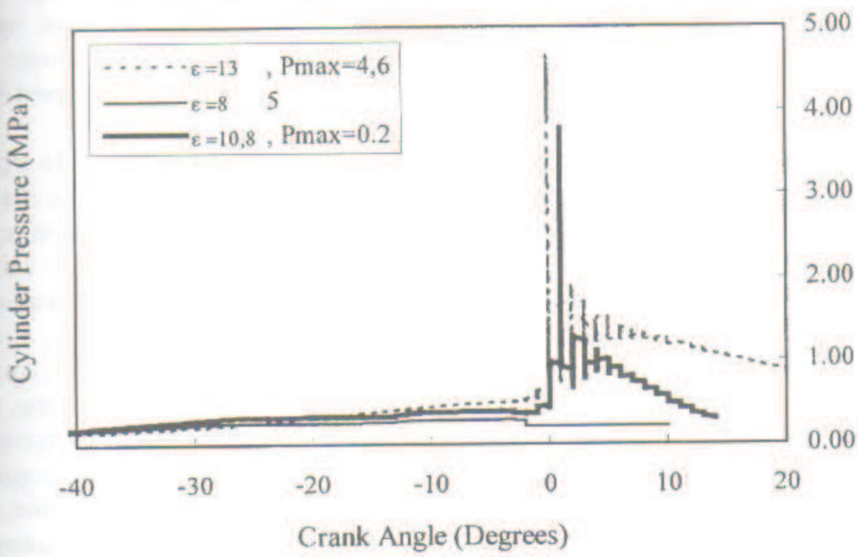

Fig. 3. The effect of compression ration on P-α history

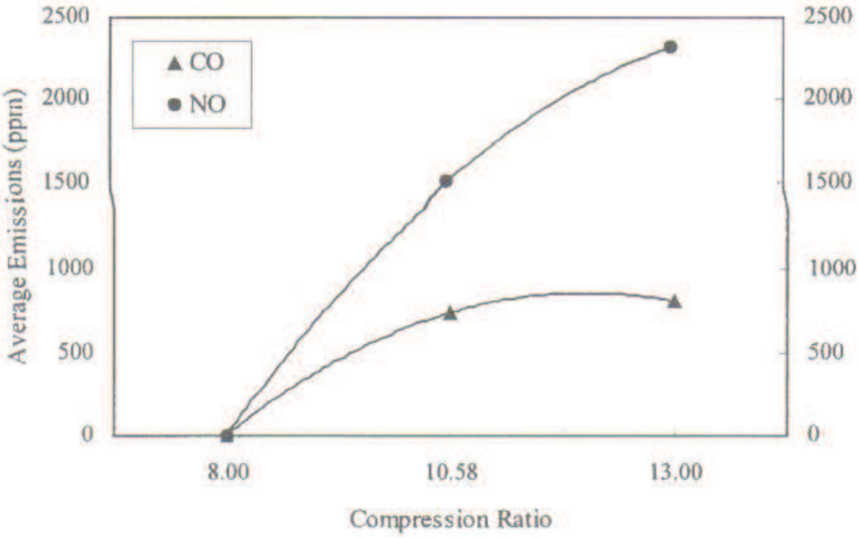

Fig. 4. The effect of compression ratio on emissions

insufficient combustion for the compression ratio of 8. The NO emissions were increased with increasing compression ratio due to the increasing gas temperatures. Decrease in transformation of CO to CO_2 caused an increase in CO emissions.

4.3. Effects of mixture strength

Lean operations of SI engines gives better specific fuel consumption and lower exhaust emissions in general. Leaning the mixture increase cycle-to--cycle variations and combustion stability is reduced as it is got closer to the ignition limits of the mixture. Fuels with wide ignition limits can be used for this problem.

The effect of excess air ratio on cylinder pressure history is shown in Fig. 5 for different excess air ratios (1.0, 1.1, 1.2). In the stoichiometric mixture, the maximum cylinder pressure is 4.04 MPa at 9.29 CAD ATDC. The maximum cylinder pressure is 3.79 MPa at 12.16 CAD ATDC in the case of $\lambda = 1.1$. These values are 1.58 MPa at 18.12 CAD ATDC for $\lambda = 1.2$. It can be seen from these values that, the leaner the mixture, the lower maximum cylinder pressure.

Figure 6 shows the effects of mixture strength on CO and NO emissions. It is well known that CO emissions are primarily a function of excess air ratio, and decreases with increasing excess air ratio. CO emissions were lower at lean mixtures. The ON emissions were also lower with increasing excess air ratio due to decreasing gas temperatures. In a spark ignition engine fueled with methane, it is possible to work with lean mixtures to lower emissions.

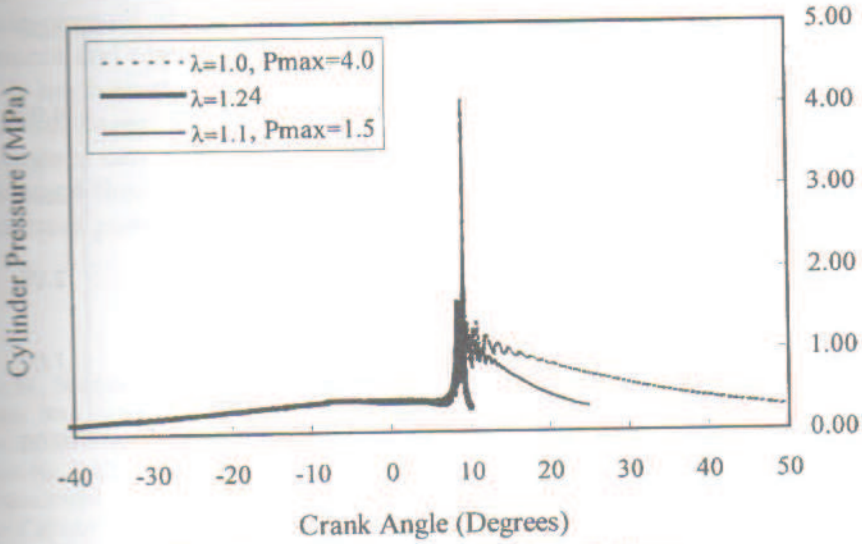

Fig. 5. The effect of mixture strength on *P*-α history

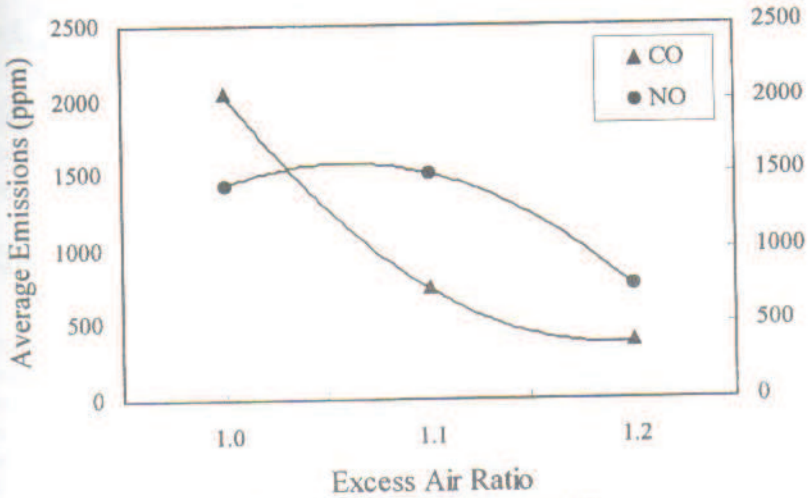

Fig. 6. The effect of mixture strength on emissions

4.4. Effect of swirl ratio

Swirl is defined as an organized rotation of charge about the cylinder axis, to promote a rapid mixing in diesel engines and a speed up the combustion process in SI engines, especially for lean mixtures [8].

The effect of swirl ratio on cylinder pressure history is shown in Fig. 7 for different swirl ratios (SR = 0, SR = 1, SR = 2). The maximum cylinder pressure is 0.4 MPa and the crank angle reached this pressure is at 1° BTDC. In the case of swirl ratio SR = 1, the maximum cylinder pressure is 3.79 MPa and the crank angle is 12.16° ATDC. For SR = 2, these values are 3.12 MPa and 6.37° ATDC.

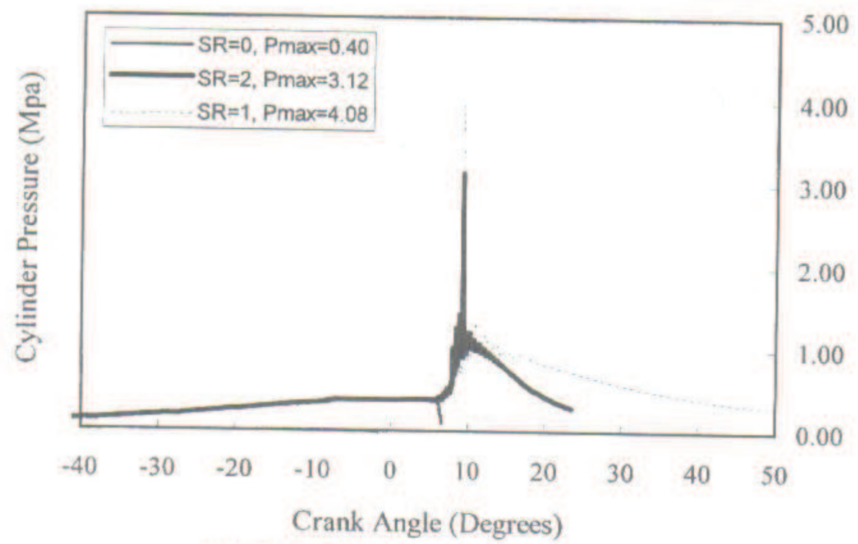

Fig. 7. The effect of swirl ratio on P-α history

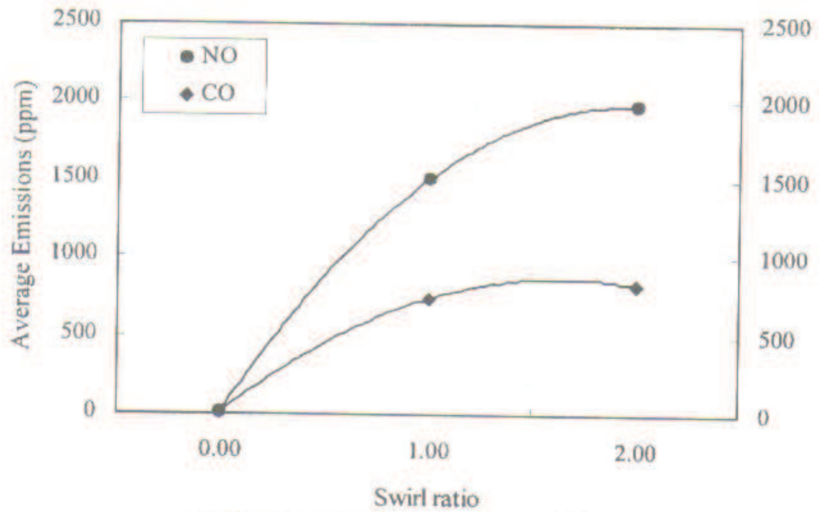

Fig. 8. The effect of swirl ratio on emissions

Figure 8 shows the effects of swirl ratio on CO and NO emissions. The combustion speed increases. Thus, the gas temperatures and NO emissions increase. The combustion takes place very slowly at the swirl ratio of 1. For this reason, the emissions were very low in this case.

5. Conclusions

In this study, the effects of certain engine parameters on combustion performance and exhaust emissions of a SI engine fueled with methane were investigated.

The main purpose of the study was to investigate the parameters, which influence performance and emissions, rather than to optimize these parameters for a certain engine. It has been shown from the results that the methane fueled SI engines could operate with leaner mixture than gasoline engines with very low exhaust emissions. These engines can also operate with high compression ratios without knocking tendency, and thus it is possible to increase the peak cylinder pressure and a result the maximum power output of the engine.

References

[1] ISHII M., ISHIZAWA S., INADA E., IDOGUCHI R., and SEKIBA T., *Experimental Studies on a Natural Gas Vehicle*, SAE Paper No. 942005, 1994.

[2] HAN Z., TSAO K., ABDALLA M., *Computation of the In-Cylinder Process of a Natural Gas Engine*, SAE Paper No. 940213, 1994.

[3] JAASKELAINEN H., WALLACE J., *Performance and Emissions of Natural Gas-Fueled 16 Valve DOCH Four-Cylinder Engine*, SAE Paper No. 930380, 1993.

[4] SOYHAN H. S., SORUSBAY C., *Combustion Modeling in Dedicated Natural Gas Engine*, Mathematical & Computational Applications Journal, Vol. 1, No. 1, 1996.

[5] WEAVER C., *Natural Gas Vehicles-A Review of the State of the Art*, SAE Paper No. 892133, 1989.

[6] CLOUTMAN L., DUKOWICZ J., RAUSHAW J., AMSDEN A., *Conchas Spray, A Code for Reactive Flows with Fuel Sprays*, Los Alamos National Laboratory, Los Alamos, U.S.A., 1982.

[7] HEYWOOD J. B., *Internal Combustion Engine Fundamentals*, McGraw-Hill Book Co., 1988.

[8] TSAO K., HAN Z., *An Exploratory Study on Combustion Modeling and Chamber Design of Natural Gas Engines*, SAE Paper No. 930312, 1993.

Osiągi i emisja z silnika o zapłonie iskrowym zasilanego metanem

Streszczenie

Przedstawiono ocenę wpływu przepływów w cylindrze na osiągi i emisję silnika o zapłonie iskrowym zasilanego metanolem przy wykorzystaniu symulacji matematycznej w oparciu o dwuwymiarowy model przepływu i mechanizm globalnej reakcji utleniania metanu. W wyniku stwierdzono, że metan jest lepszy od innych paliw jeśli chodzi o zasilanie ubogą mieszanką oraz charakterystyki spalania przy niskich temperaturach, chociaż szybkość jego spalania jest mniejsza. Także emisja NO_x jest niższa niż przy zasilaniu paliwami konwencjonalnymi.