Marek FLEKIEWICZ
Silesian University of Technology
Krasińskiego Street, 8, 40-019 Katowice, Poland
e-mail: marek.flekiewicz@polsl.pl

IGNITION TIMING ADVANCE IN THE BI-FUEL ENGINE

Summary. The influence of ignition timing on CNG combustion process has been presented in this paper. A 1.6 liter SI engine has been tested in the special program. For selected engine operating conditions, following data were acquired: in cylinder pressure, crank angle, fuel mass consumption and exhaust gases temperatures. For the timing advance correction varying between 0 to 15 deg crank angle, the internal temperature of combustion chamber, as well as the charge combustion ratio and ratio of heat release has been estimated. With the help of the mathematical model, emissions of NO, CO and CO_2 were additionally estimated. Obtained results made it possible to compare the influence of ignition timing advance on natural gas combustion in the SI engine. The engine torque and in-cylinder pressure were used for determination of the optimum engine timing advance.

KĄT WYPRZEDZENIA ZAPŁONU W SILNIKACH DWUPALIWOWYCH

Streszczenie. W artykule przedstawiono wpływ kąta wyprzedzenia zapłonu na przebieg spalania mieszaniny powietrza i gazu ziemnego. Program badań przeprowadzono na stanowisku wyposażonym w silnik ZI, o pojemności skokowej 1.6 dm^3. Dla wybranych warunków pracy silnika rejestrowano między innymi takie parametry jak: ciśnienie wewnątrz cylindra, położenie wału korbowego, zużycie paliwa. Korygując kąt wyprzedzenia zapłonu w zakresie od 0 do 15 stopni w stosunku do nastaw ustalonych przez producenta dla paliwa benzynowego wyznaczono stopień wypalenia ładunku, temperaturę ładunku oraz współczynnik wydzielania ciepła. Za pomocą modelu matematycznego opisującego proces spalania wyznaczono również stężenia produktów spalania, tj.: NO, CO i CO_2. Uzyskane wyniki umożliwiły porównanie wpływu kąta wyprzedzenia zapłonu na przebieg spalania gazu ziemnego w silniku ZI. Na podstawie analizy położenia maksymalnej wartości ciśnienia oraz maksymalnej wartości momentu obrotowego wyznaczono najkorzystniejsze dla gazu ziemnego wartości zmian kąta wyprzedzenia.

1. INTRODUCTION

A significant reduction of CO_2 emission in the road transport is a major challenge for next years to come. In the combination with efficient powertrain technologies, the potential of natural gas is excellent for comparably light and cost effective reduction of CO_2 and toxic emission in the future [1-2]. As the CNG fuelling stations in Poland are not so far widespread enough to make a dedicated natural gas vehicle practical, it results necessary to start with proposing alternatively CNG powered engines. Such a bi-fuel automotive engines are necessary to bridge the gap between petrol and natural gas. As a sample of modern engine design an Opel 1.6 liter 4-cylinder engine has been selected as a base of powertrain for the development of bifuel passenger car engine.

Many previous natural gas engine conversions have made compromise in engine control strategies, including mapped open-loop methods or resorting to translating the signals to or from the original controller. The engine control system prepared for tests however employs adaptive closed-loop control, optimizing fuel delivery and spark timing for both fuels. Each fuel is metered by injectors and fuel injection control maintains a present air-fuel ratio thanks to exhaust gas oxygen sensor (UEGO).

Spark timing is also controlled to maintain the appearance of in-cylinder pressure peak at optimum value for best torque, which has been determined experimentally to be 22 degree after TDC for tested engine case.

For the tested engine, a significant reduction of hydrocarbons, carbon dioxide and nitrogen oxides emissions as compared with stock operation were observed when using the controller with petrol. Further reduction in emission was possible with natural gas operation, due to fuel properties. An improvement in engine stability has also been obtained with the use of the controller.
In this paper we present results obtained at wide open throttle, for selected ignition timing setups and without EGR.

2. MEASUREMENT SET-UP

The tested engine was an Opel Astra naturally aspirated four cylinder petrol engines with displacement of 1.6 l with power output of 55 kW at 5200 rpm and torque of 128 Nm at 2600 rpm. This engine was modified in a way allowing its CNG propulsion without compression ratio variations. The engine was operated on strictly stoichiometric ratios and used one TW catalyst. Test procedure provided analysis on idle and for selected higher RPM's at the wide range of loads. Studies provided in-cylinder pressure registration in the crank angle domain for two different series. First series featured engine running on petrol, while the second one was registered for natural gas stoichiometric operation. Experimental setup included pressure transducer type 6121, 2613B charge amplifier, crankshaft speed and position sensor DPA type by Kistler. Data were acquired through an eight channel NI board of the PCI-6143 type, driven by an application compiled in the LabView environment.

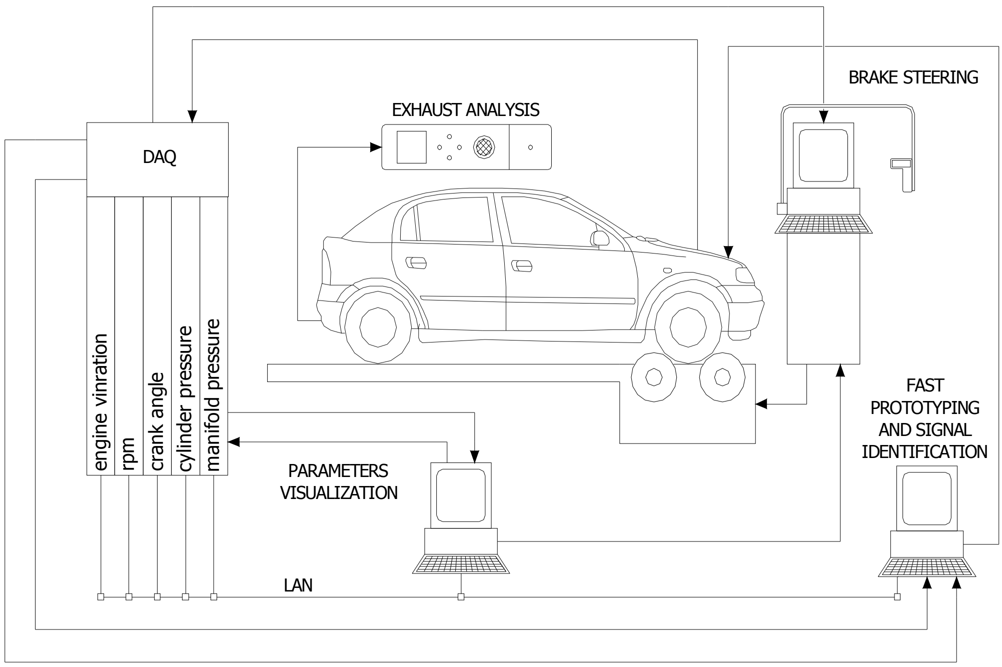


Fig. 1. Schematic diagram of experimental setup
Rys. 1. Schemat stanowiska pomiarowego

Engine load variation was realized with the help of the BOSCH FLA 203 roller bench. Exhaust gases were registered by a fast response Pierburg HGA 400 5GR gas analyzer, while fuel consumption was measured respectively for petrol with the use of precise Pierburg PLU 401 device, while gaseous fuel consumption was registered by a tensometric balance. Experimental setup diagram has been presented on figure 1.

Registered for the tested engine in-cylinder pressure traces in the crank angle domain were the bases for further model calculations.

3. EXPERIMENTAL RESULTS AND DISCUSSION

All measurements were made for petrol and natural gas, at the rpm range from 500 to 5300 rpm for different engine loads, starting from 0 to 100%. All tests completed on the Opel engine were done for stochiometric mixtures and with spark timing provided for natural gas by engine ECU for every 1 deg in the range varying from optimal for petrol to +15 deg of advance. Comparisons of operating characteristics of tested engine powered by petrol and natural gas with chosen timing advance were presented on fig. 4.

Fig. 2. Engine compartment of tested vehicle
Rys. 2. Widok komory silnika badanego samochodu

A B

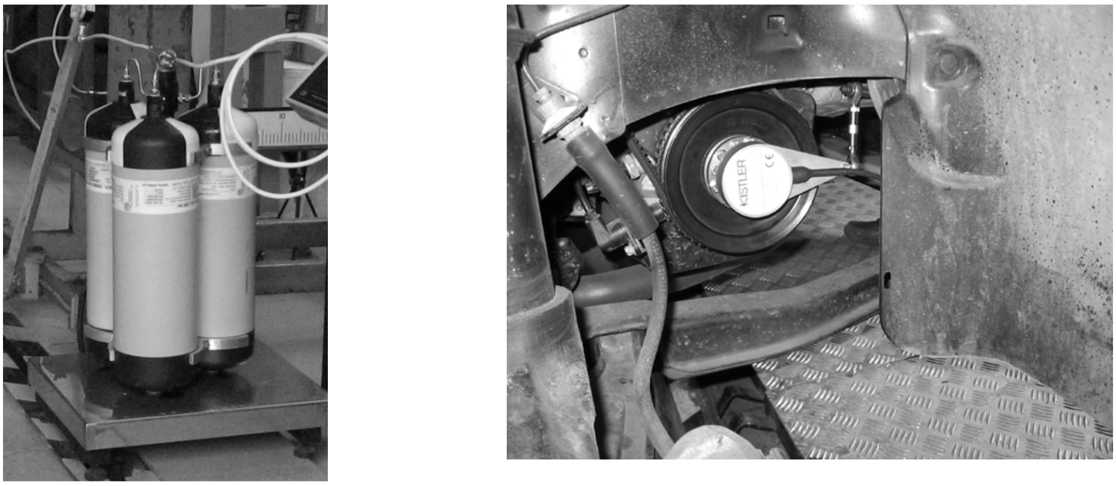


Fig. 3. Special container for gaseous fuel (A) and engine crank angle encoder (B)
Rys. 3. Zbiornik specjalny do magazynowania paliw gazowych (A) i znacznik położenia wału korbowego (B)

During engine tests following main parameters were registered: cylinder pressure, TDC recognition, rpm, manifold pressure, mass fuel consumption, air mass flow rate and engine power and torque.

Numerical calculations carried on the basis of a mathematical model [7, 8, 9] made it possible to estimate and compare:

- in-cylinder pressure and mass fraction burned increase for the engine running on petrol and natural gas in the function of the crank angle,
- mass fraction burned for engine running on petrol and alternatively on natural gas, in the function of crank angle,
- maximum in-cylinder temperature, exhaust gases temperatures for the engine fed with petrol and gaseous fuel, in the function of engine crank angle,
- combustion process products both in the function of crank angle (in their formation process), as well as a summary values in the entire cycle.

Obtained results, were later applied to optimize the engine ignition timing for the CNG operation.

Table 1

Main characteristics of the tested engine

Type	Four cylinder in-line
Displacement	1600 dm^3
Bore	79,0 mm
Stroke	81,5 mm
Compression ratio	9,6
Exhaust valve opening	41^o BTDC
Exhaust valve closing	11^o ATDC
Inlet valve opening	11^o BTDC
Inlet valve closing	41^o ATDC
EGR ratio	0 %

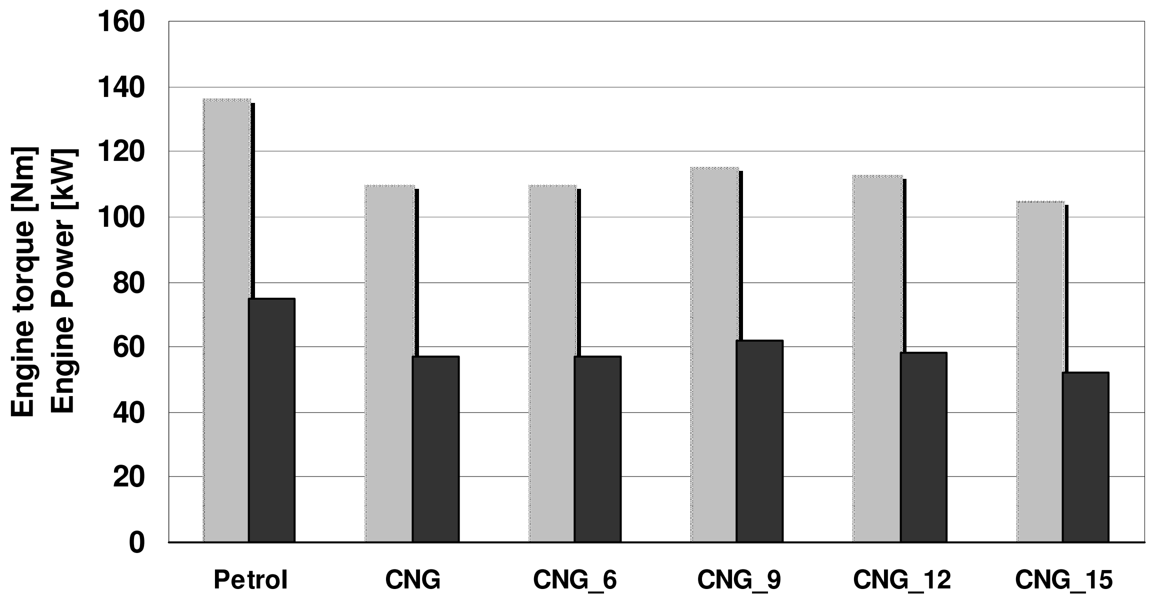


Fig. 4. Engine power and torque by wide open throttle for petrol and natural gas for chosen ignition advance
Rys. 4. Moc i moment badanego silnika przy całkowicie otwartej przepustnicy dla wybranych wartości kąta wyprzedzenia zapłonu

In modern systems the ignition timing is controlled using open-loop schemes that rely on look-up tables. The look-up tables are determined through extensive calibration experiments in either an engine or chassis dynamometer. According to Heywood [3] a calibration procedure usually follows these guidelines:

First the torque at MBT is determined. Then the ignition timing is retarded towards TDC until the torque is reduced by approximately 1% below the maximum and that value is then used. There are three reasons for this: First, it is easier to determine this position, since the torque as a function of ignition timing is at the optimum. Second, with a slightly retarded schedule the margin to knocking conditions is increased. Third, the NO_x formation is reduced. The calibrated schedule is stored in a look-up table, covering the engine operating range, and compensation factors are added and used during e.g. cold start and idle conditions. Optimal ignition timing depends on:

- how the flame propagates through the combustion chamber and the losses such as heat transfer to the walls and piston, flows into and out of crevices, and piston blowby,
- many engine parameters. (Some of the parameters that are measured and accounted for, in today's systems are: engine speed, engine load, coolant temperature, and intake air temperature).

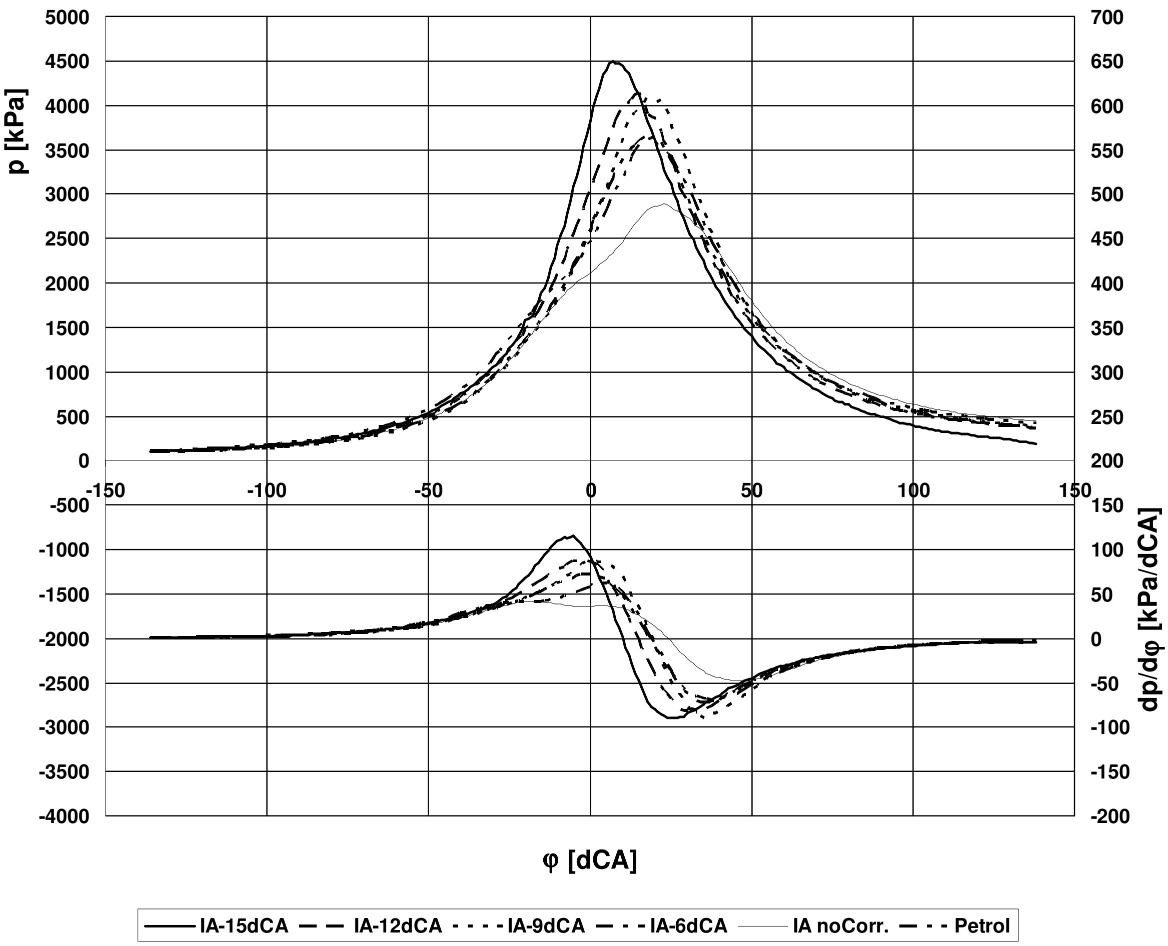


Fig. 5. Cylinder pressure and rate of pressure rise as a function of crank angle for tested engine (for rpm=2500 and full load)

Rys. 5. Ciśnienie w komorze spalania i jego przyrost, dla n=2500obr/min i pełnego otwarcia przepustnicy

A calibrated scheme has to guarantee good performance over the range of the non measured parameters and is often chosen to be conservative; it is thus not optimal when the non measured parameters change. A feedback scheme on the other hand, that measures the result of the ignition instead of measuring and accounting for things that affect it, has the potential to guarantee good performance over the entire range of non-measured parameters, improve the efficiency, and additionally reduce the calibration effort and requirements.

The spark advance positions the combustion and cylinder pressure development in relation to the crank shaft rotation. Under normal driving conditions the mixture is ignited around 15 - 30 before the piston has reached top dead center (TDC), and the pressure reaches its maximum around 15 - 20 degree after TDC. The figure 5 shows different pressure traces resulting from different spark timings.

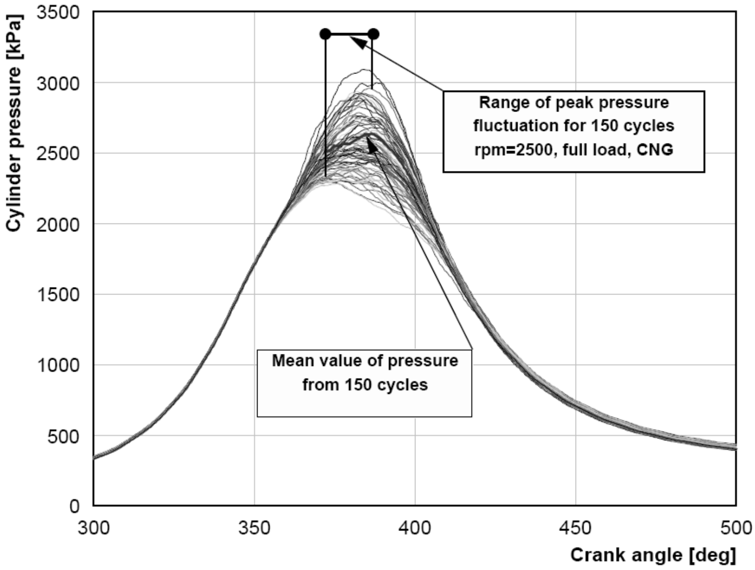


Fig. 6. Peak pressure position for 150 cycles of CNG powered engine at 6 deg timing advance
Rys. 6. Położenie maksymalnej wartości ciśnienia dla 1500 kolejnych cykli roboczych silnika zasilanego CNG, przy kącie wyprzedzenia zapłonu zwiększonym o 6 stopni

An earlier spark advance normally gives a higher maximum pressure and temperature, which occurs at earlier crank angles. The spark advance for maximum brake torque (MBT) for the conditions shown in the figure, and the resulting pressure peak lies around 17 degree after TDC. With too early ignition timing the pressure rise starts too early and counteracts the piston movement. This can be seen in the same figure, where the pressure rise starts 20 degree before TDC. With early ignition timing there are also increased losses due to heat transfer to the walls and flows into and out of crevices. The temperature will rise earlier and more energy will be dissipated during the cycle. Similarly, will the earlier combustion, which results in a maximum pressure, force more of the gases into the crevices. Too late ignition produces a pressure increase that comes too late so that energy is lost during the expansion phase. In Figure 5, the pressure increase for timing advance starts as late as at TDC. But work is also gained, partially due to the later start of the effects mentioned above, which can be seen on the same figure. The pressure trace from the spark advance with correction +15 deg, is higher than the others. However, this gain in produced energy cannot fully compensate for the loss early in the expansion phase, and work is lost compared to the optimal spark advance. Another possibility for describing the position of the combustion is to use the mass fraction burned profile -x. Heywood [3] states that with optimal spark timing half of the charge is burned (50% mass fraction burned) about 10 deg after TDC. This has been further investigated and supported by Bargende [4]. Other possible measures of good combustion could be the positions for 30% or 90% mass fraction burned. A mass fraction burned profile is shown in Figure 7.

Selected results obtained for 2500 rpm were presented in figure 6. Cylinder pressures and their rise rate for increasing ignition advance were presented on figure 5.

Advanced ignition timing for CNG does respectively:

- accelerate the pressure peak appearance, increasing as well the engine running hardness, $dp/d\alpha$,
- causes charge temperature.

Temperature registered for the highest ignition timing advance – 15deg CA is about 500K higher compared to one measured for petrol.
For the cases of 10 and 50% of mass fraction burned the differences between results for natural gas and petrol do reach 18 degCA, while for the 90% case – 22 deg CA. Differences are also presented in table 2, defining the charge complete burning time for three ranges: 0-10, 10-90%. Obtained results do point on a significant influence of engine speed and its load on charge burning time for selected time windows. The biggest difference in the charge burning time, up to 30% when compared to petrol was noticed for the range 10 do 90%. Increasing the timing advance up to 15 deg does cause the charge burning curve in function of deg of CA to present the similar smooth raise, to the x obtained for petrol, especially in the range of 0 to 50%.
Increasing in-cylinder temperature does significantly raise the NO_x emissions, compared to emissions registered for petrol, fig. 8 and 9. For the case of engine running under the load, the lowest NO emissions were obtained for the engine with the timing advance around 12 deg. That ignition timing allowed also it to get the lowest CO_2 emissions, reaching just 40% of value registered for petrol.

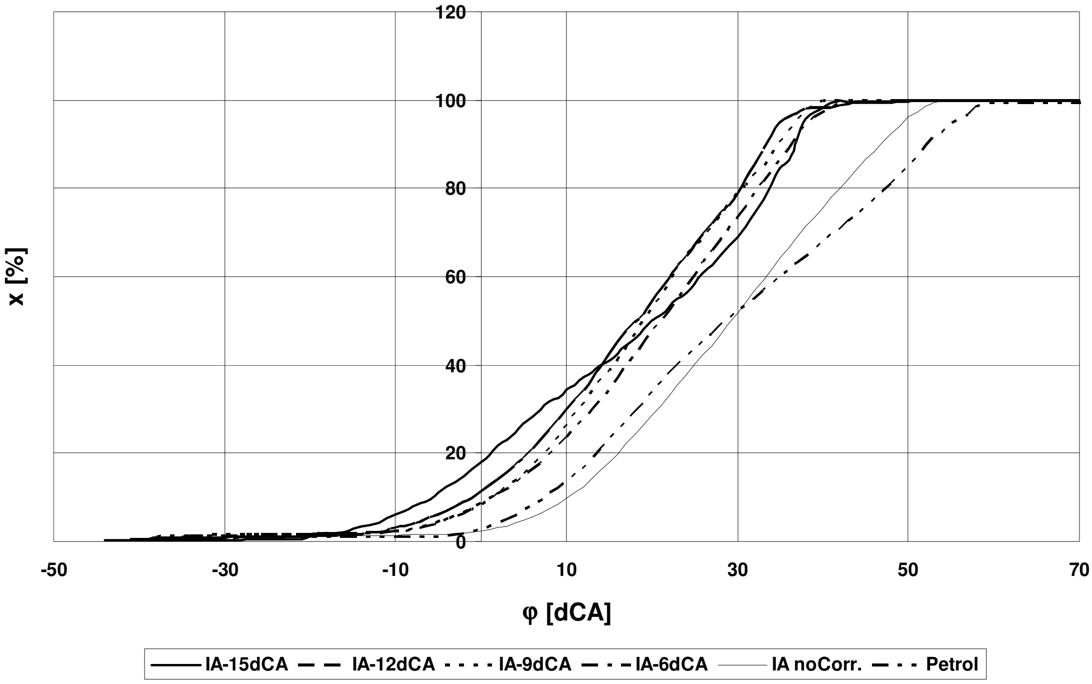


Fig. 7. Mass fraction burned as a function of CA for rpm=2500 and full load
Rys. 7. Współczynnik wypalenia ładunku w zależności od kata obrotu wału korbowego dla n=2500 obr/min przy pełnym otwarciu przepustnicy

Table 2 shows that it is the duration of the main burn period that influences the efficiency and output, and that a long delay or early burning period not matter, as long as the combustion is correctly phased through the use of the MTB ignition timing.
In this study for peak pressure position determination Erikson's algorithm was used [3]. The principles of this method were presented on figures 10 and 11. This figure illustrates that when the mean peak pressure position is at optimum variations in the output torque are minimal. At range A the mean peak pressure position is at optimum which give small variation of output torque. At range B the mean peak pressure position is some degrees off from optimum and resulting variations are larger.

After comparison of two main parameters, i.e. location of peak pressure and the mass fraction burned profile as very good combustion descriptors, spark advance map for tested engine has been designed.

To define the position of the in-cylinder pressure relative to TDC, the peak pressure position is used, Fig. 11. The peak pressure position is the position in crank angle where the in-cylinder pressure reaches its maximal value. There are also other ways of describing the positioning of the combustion relative to crank angle, e.g. based on the mass fraction burned curve. In Figures 10 and 11, mean values, over 200 cycles, of the peak pressure are plotted together with the mean value of the produced torque as a sample of operating points of full load. The peak pressure position for maximum output torque in the figure is around 18^o after TDC for all operating points.

The engine load and speed are changed over large intervals, and the pressure peak position for maximum output torque at the different operating points does not differ too much. The peak pressure versus torque curve results flat around the position for the maximum. Therefore a spark timing that maintains a constant pressure peak position at 18^o is close to optimum.

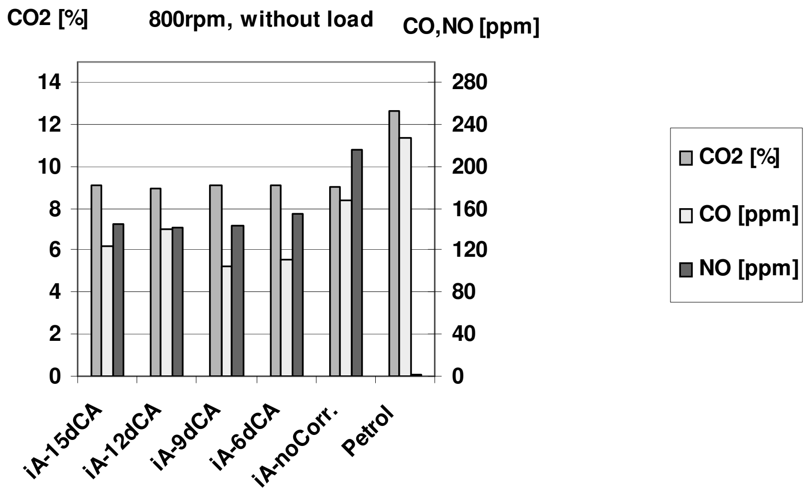
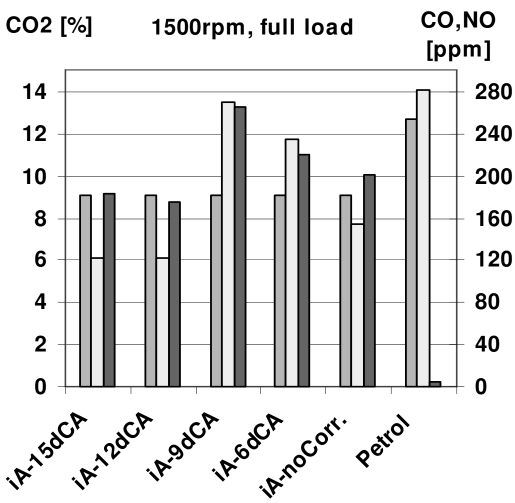


Fig. 8. Calculated CO_2, NO and CO emission at idle
Rys. 8. Obliczone stężenia CO_2, NO i CO w spalinach, bieg jałowy



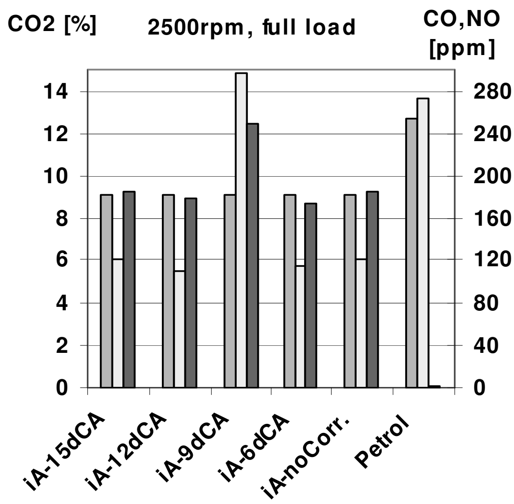


Fig. 9. Calculated CO_2, NO and CO emission at rpm=1500 and 2500
Rys. 9. Obliczone stężenia CO_2, NO i CO w spalinach dla prędkości obrotowych 1500 i 2500 obr/min

Considering only the work produced, this motivates that optimal spark timing maintains almost the same position for the peak pressure [3]. However, the optimal pressure peak position changes slightly with the operating points. The efficiency can thus be improved a little bit further by mapping the optimal pressure peak position for each operating point, and providing these values as reference signal to the spark timing controller. The peak pressure positioning principle can also be used for meeting emission standards. In [4] this question is addressed by rephrasing the emission regulations on the spark advance to desired peak pressure positions.

Table 2

The effect of the burn duration on the MBT

0-10% burn. ^{o}CA		10-90%burn. ^{o}CA		MBT timing ^{o}CA BTDC		p_{max} MPa		Angle of p_{max} ^{o}CA ATDC	
Petrol	CNG	Petrol	CNG	Petrol	CNG	Petrol	CNG	Petrol	CNG
35	46	46	42	19,9	19,9	3,61	2,76	18,16	22,7
x	47	x	34	x	26,00	x	3,61	18,16	18,2
x	49	x	34	x	29,00	x	4,00	18,16	18,2
x	47	x	31	x	31,00	x	4,14	18,16	13,62
x	43	x	34	x	34,00	x	4,46	18,16	6,81

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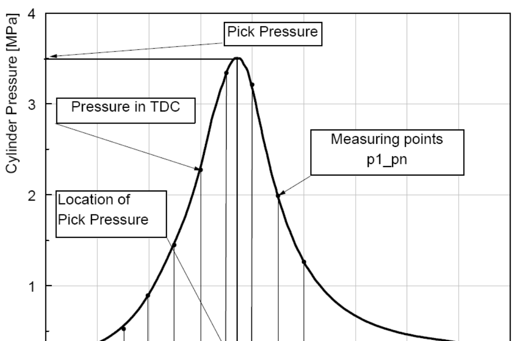

Fig. 10. Combustion pressure measuring points for peak pressure position estimation

Rys. 10. Punkty pomiaru ciśnienia dla określenia maksymalnej jego wartości

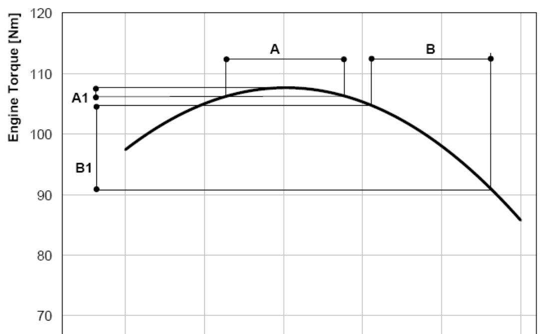

Fig. 11. Ranges of peak pressure positions with small (A) and larger (B) variation of engine torque

Rys. 11. Zakres zmian kątowych położenia maksymalnej wartości ciśnienia z małymi (A) i dużymi zmianami momentu obrotowego

4. CONCLUSIONS

From the environmental and technical perspectives, natural gas can significantly improve emission characteristics of the IC engine. However to obtain a proper combustion process for the CNG powered bi-fuel engine it is necessary to prepare a dedicated ignition timing map, allowing a proper use of the fuel energy. Presented results do allow only to evaluate the influence of ignition timing advance on the stoichiometric air-CNG mixture burning quality.

Increased charge burning temperature does also require proper EGR operating strategies, taking into consideration wider valve openings. In the analyzed bi-fuel engine case, optimized for petrol operation, modifications of EGR regulation strategies are the most important tool allowing the in-cylinder temperature drop.

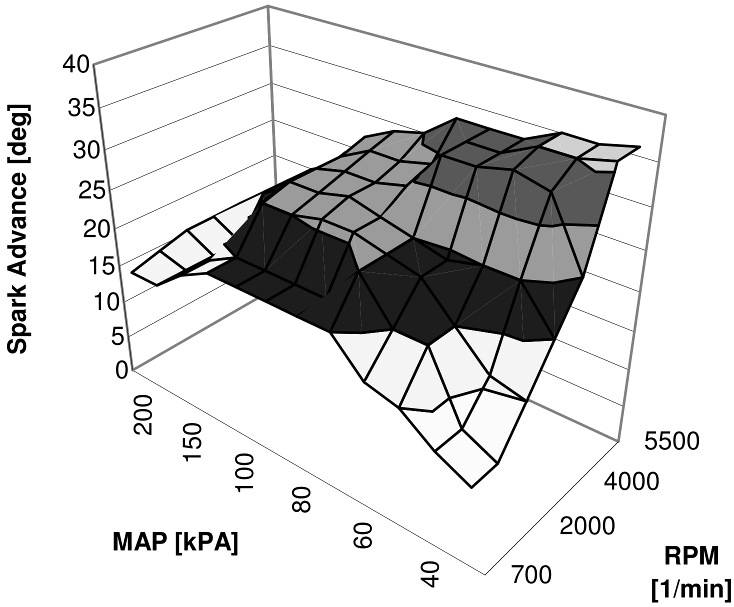

Fig. 12. Ignition advance for tested engine
Rys. 12. Kąt wyprzedzenia zapłonu dla badanego silnika

The best combinations of EGR ratios, and spark timing for optimal in-cylinder pressure characteristics providing moderate combustion temperatures and low expansion cylinder temperatures are being currently tested on a bifuel engine.

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