INTERNAL COMBUSTION ENGINE POWERED BY SYNTHESIS GAS FROM PYROLYSED PLASTICS

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Abstract: The article discusses the application of synthesis gas from pyrolysis of plastics in petrol engine. The appropriate experimental measurements were performed on a combustion engine LGW 702 designated for micro-cogeneration unit. The power parameters, economic parameters in term of brake specific fuel consumption, and internal parameters of the engine were compared to the engine running on the reference fuel - natural gas and synthesis gas. Burning synthesis gas leads to decreased performance by about 5% and to increased mass hourly consumption by 120 %. In terms of burning, synthesis gas has similar properties as natural gas. Compared with [5] a more detailed study has been prepared on the effects of angle of spark advance on the engine torque, giving more detailed assessment of engine cycle variability and considering specification of start and end of combustion in the logarithm p-V diagram.

KEYWORDS: natural gas, synthesis gas, spark ignition engine, cogeneration unit, cycle variability

1   Introduction

Municipal waste with its quantity and quality represents a significant environmental burden. Therefore, solutions are being developed on possible most effective utilisation of the hidden energy potential of that waste. One possibility is separation of plastics from waste and subsequent conversion of this waste via pyrolytic process to gaseous fuel, which is a carrier of energy. This fuel can then be effectively utilised in the cogeneration unit. One such workplace for separation of communal waste and subsequent processing is located in Čierny Brod-Zúgov with installed separators of municipal waste. The separated plastics will be processed by pyrolysis and thus converted into a gaseous fuel. An analysis of the primary fuel from the pyrolysis process consisted of more than twenty components, which number would mean a rather complicated experimental verification on the engine. Therefore, two gas compositions were selected, which are a mix of five components each and labelled as SGE0 and SGE1. Our department has experimentally verified the gas labelled SGE1 that has the following composition: methane CH_4 10% vol., ethane C_2H_6 31% vol., carbon monoxide CO 7% vol., hydrogen H_2 12% vol., carbon dioxide, CO_2 40% vol. This gas mix best resembles the properties of real (SGE0) pyrolysed plastics gas composition. To complete the info, the SGE0 gas composition was as follows: methane CH_4 6% vol., n-butane C_4H_{10} 14% vol., propylene C_3H_6 6 vol. %, hydrogen H_2 8% vol., nitrogen N_2 66% vol.. The following table shows basic physical and chemical properties of the referential fuel entering the comparison (natural gas - 97% vol. methane), and the two gas fuel compositions, out of which the gas labelled SGE1

was experimentally referenced. Comparing natural gas with synthesis gas SGE1, the lower heating value (LHV) is lower by 62% (by weight), or lower by 30% (by volume). The volumetric heating value of the stoichiometric mixture of synthesis gas SGE1 and air is lower by 6.6 % than is the volumetric heating value of the stoichiometric mixture of natural gas and air. For this reason, we can expect somewhat lower performance parameters of the engine when operating on synthesis gas [1], [2], [8]. For prediction of performance parameters by modelling we often use the software Lotus Engine Simulation. Example is given in [7].

Tab. 1 Basic physical-chemical properties of fuel mixtures

Parameter	Unit	Natural Gas	SGE0	SGE1
LHV	$[kJ.kg^{-1}]$	48 796	18 352	18 603
LHV	$[kJ.m^{-3}]$	33 838	23 096	23 767
Density	$[kg.m^{-3}]$	0.693	1.259	1.278
Molar mass	$[g.mol^{-1}]$	16.69	30.29	30.74
A/F ratio	$[kg.kg^{-1}]$	16.98	6.17	6.57
fuel/air ratio	[% vol.]	9.3/90.7	13.4/86.6	12.3/87.7
fuel/air ratio	[% m.]	5.6/94.4	14.0/86.0	13.2/86.8
LHV mixture	$[kJ.kg^{-1}]$	2 713	2 569	2 456
LHV mixture	$[kJ.m^{-3}]$	3 130	3 095	2 923

The investigated gas fuel SGE1 was delivered as a compressed gas in a pressure container. In applications the synthesis gas, entering the engine, is pure, without any mechanical impurities and with negligible amount of tar. Because of the cost of acquisition of that fuel the experimental measurements were performed on a small combustion engine LGW 702 Lombardini suitable for micro-cogeneration unit [3].

The basic parameters of the combustion engine Lombardini LGW 702 are given in Table 2. The producer of the engine has performed adaptations as follows:
- replacement of the original head of the diesel-engine version of the combustion engine for the head with openings for installing spark plugs,
- decrease of compression ratio to 12.5:1, realised by replacement of the original pistons for pistons with a cavity in the bottom of the pistons of the requested volume and shape,
- supply of original intake manifold with common container and electronically controlled throttle.

Tab. 2 Basic parameters of combustion engine Lombardini LGW 702

Crankshaft throw angle [°]	360
Swept volume $[cm^3]$	686
Bore/ Stroke [mm]	75 / 77,6
Compression ratio [-]	12.5:1
Valve gear/ Drive	OHC/Belt drive system
Fuel	Natural gas (and alternative gasses)
Lubrication system	Pressure circuit, full-flow filtration, oil filling 1.6 l

2 Experimental Measurements and Results

2.1 Combustion engine Lombardini LGW 702

The experimental measurements were performed on Lombardini combustion engine LGW 702. It is a two-cylinder water-cooled spark ignition engine with displacement of 686 cm^3 and compression ratio of 12.5:1. The mixture was prepared using a mixer with a diffuser. The richness of the mixture was set by means of an electronic control unit VOILA. The combustion engine comprised a programmable control unit, through which it was possible to manually set the spark advance angle The primary objective was to determine the optimum angle of spark advance for the anticipated operating speed of the cogeneration unit (1500 min^{-1}) and with this optimum angle to measure all major engine characteristics. In addition to the performance and economic parameters, an analysis of the internal engine parameters was performed as well. The course of pressures in the combustion space was recorded with a piezoelectric pressure sensor from the company Kistler that was integrated in the spark plug. To correct the dynamic pressure a piezo-resistive pressure sensor was used and it was placed in the intake manifold. Both signals from the pressures and the signal from the crankshaft position were processed by the program created in Matlab.

2.2 Measurement Results

The following subsection describes the core results of the measured impact of synthesis gas on the parameters of the internal combustion engine as compared to the influence of natural gas. The following figure (Fig. 1) shows the courses of torque and effective power as depending on the engine speed at burning natural gas and synthesis gas. The comparison has shown that in the case of combusting synthesis gas, the drop in power, or torque, is by about 5%. The maximum torque (42.9 N.m) is achieved at the speed of 1600 min^{-1}. The decrease of the power parameters relates to the decrease of calorific value of the mixture, as there is a high percentage of inert gas in the synthesis gas.
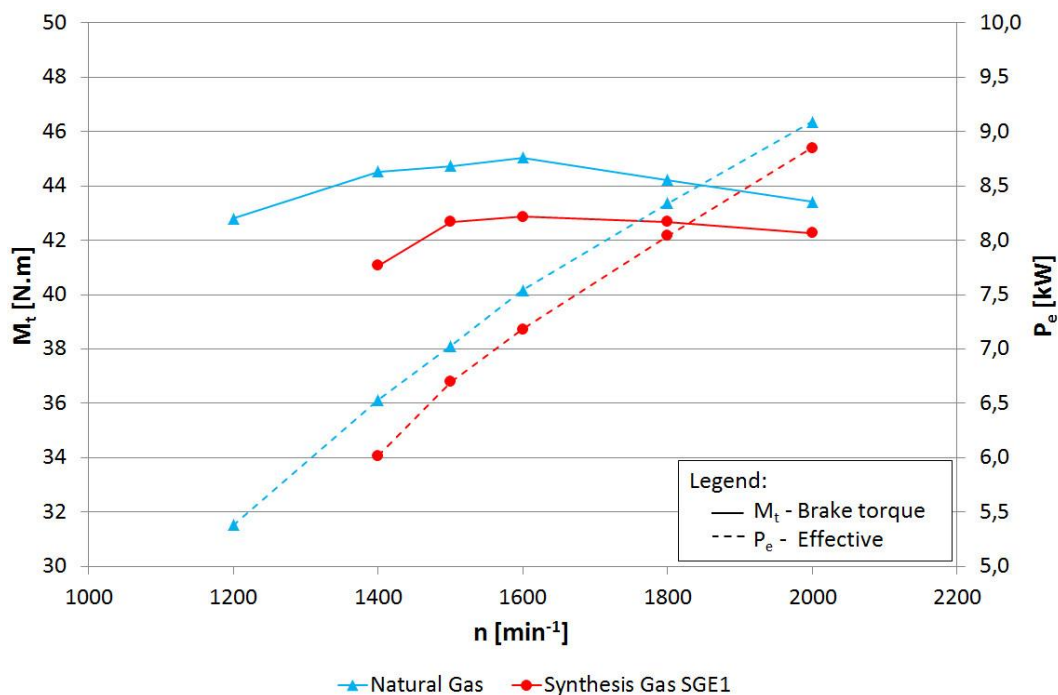

Fig. 1 Courses of torque (M_t) and effective power (P_e), as depending on the engine speed during combustion of natural gas or synthesis gas SGE1
(Conditions: spark advance $\varphi_{ign} = 25$ °CA BTDC, load $\alpha_{throttle} = 100\%$, air excess ratio $\lambda = 1$)

Fig. 2 shows the hourly fuel consumption and brake specific fuel consumption, depending on the engine speed at burning natural and synthesis gas. The hourly fuel consumption rose during combustion of synthesis gas on average by approximately 120 % within the entire speed range as compared to natural gas consumption. For the examined speed of 1 500 min^{-1} the hourly consumption of synthesis gas is higher by 2.1 kg · h^{-1} compared to the consumption of natural gas, which can be explained by a higher mass amount of synthesis gas in the stoichiometric mixture with air. Comparing the brake effective fuel consumption, the lowest value (245 $g.kW^{-1}.h^{-1}$) has been achieved while running on natural gas. When operating on synthesis gas the consumption has achieved the value 555 $g.kW^{-1}.h^{-1}$. The mean effective pressure at the operating speed (1500 min^{-1}) for the natural gas is 820 kPa, and for the synthesis gas the value is 781 kPa.
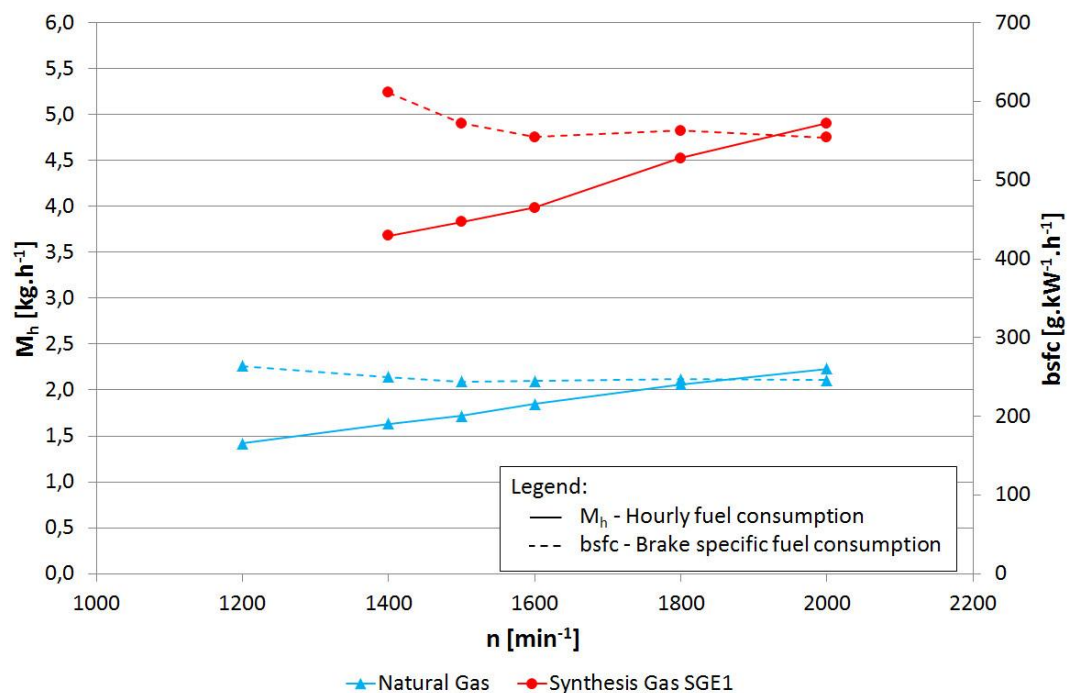

Fig. 2 Courses of hourly fuel consumption (M_h) and brake specific fuel consumption (BSFC) as dependent on the engine speed under combustion of natural and synthesis gas (Conditions: spark advance $\varphi_{ign} = 25$ °CA BTDC, full load, stoichiometric mixture)

Fig. 3 shows dependence of engine torque during on the change of the angle of advance on combustion of natural gas, or synthesis gas, at the speed of 1500 min^{-1}. As can be seen from the chart, the optimum value of the angle of advance is at 25°CA BTDC, when a maximum torque of 44.6 N.m (natural gas) was obtained, or at 42.7 N.m (syngas SGE1). The characters of the torque curves are the same for both fuels.
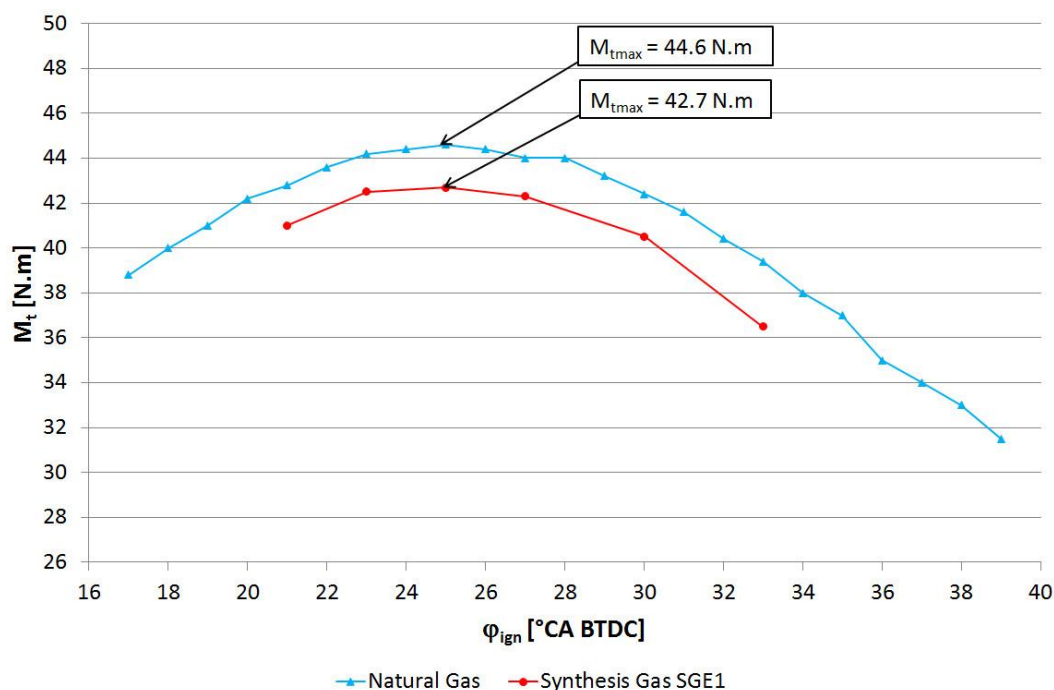
Fig. 3 Courses of engine torque (M_t) as dependent on spark advance (φ_{ign}) in the case of natural and synthesis gas SGE1
(Conditions: speed n = 1 500 min^{-1}, full load, stoichiometric mixture)

Simultaneously with measuring integral parameters an analysis of the course of pressure in the combustion chamber of the engine was performed together with the analysis of the effect of synthesis gas burning on the parameters directly relative to the actual course of burning.

The following figure (Fig. 4) shows the courses of pressures for combustion of natural and synthesis gas. It can be seen from the courses that the maximum pressure in combustion of natural gas is 6.045 MPa in the position 371.7 ° CA. When burning synthesis gas a lower value 5.820 MPa has been achieved in the position 372.4 ° CA. Thus, it is presumed that the wear of piston group will be the same.

The pressure development data that were obtained can indicate the characteristics of the operation of the internal combustion engine (variability of cycles), which is most often expressed by the so-called coefficient of variation (COV). The coefficient of variation is calculated as follows:

$$COV = \frac{\sqrt{\frac{1}{n-1}\sum_{i=1}^{n} x_i - \overline{x}}}{\overline{x}} \cdot 100 \qquad [\%] \tag{1}$$

The coefficient of variation from 195 consecutive cycles for the value of the maximum pressure is 10.9% for natural gas, and 6.6% for synthesis gas. Considering pressure rise increase (pressure increase per unit of crank angle rotation) it is more advantageous to burn natural gas (0.199 MPa / 1 ° CA) than synthesis gas (0.227 MPa / 1 ° CA), in view of the overall engine life. The ignition delay pertaining to the same operating conditions takes less time in combustion of natural gas (1.465 ms or 13.2 ° CA for the speed of 1 500min^{-1}) than in combustion of synthesis gas (1.564 ms or 14.1 ° CA for the same speed).
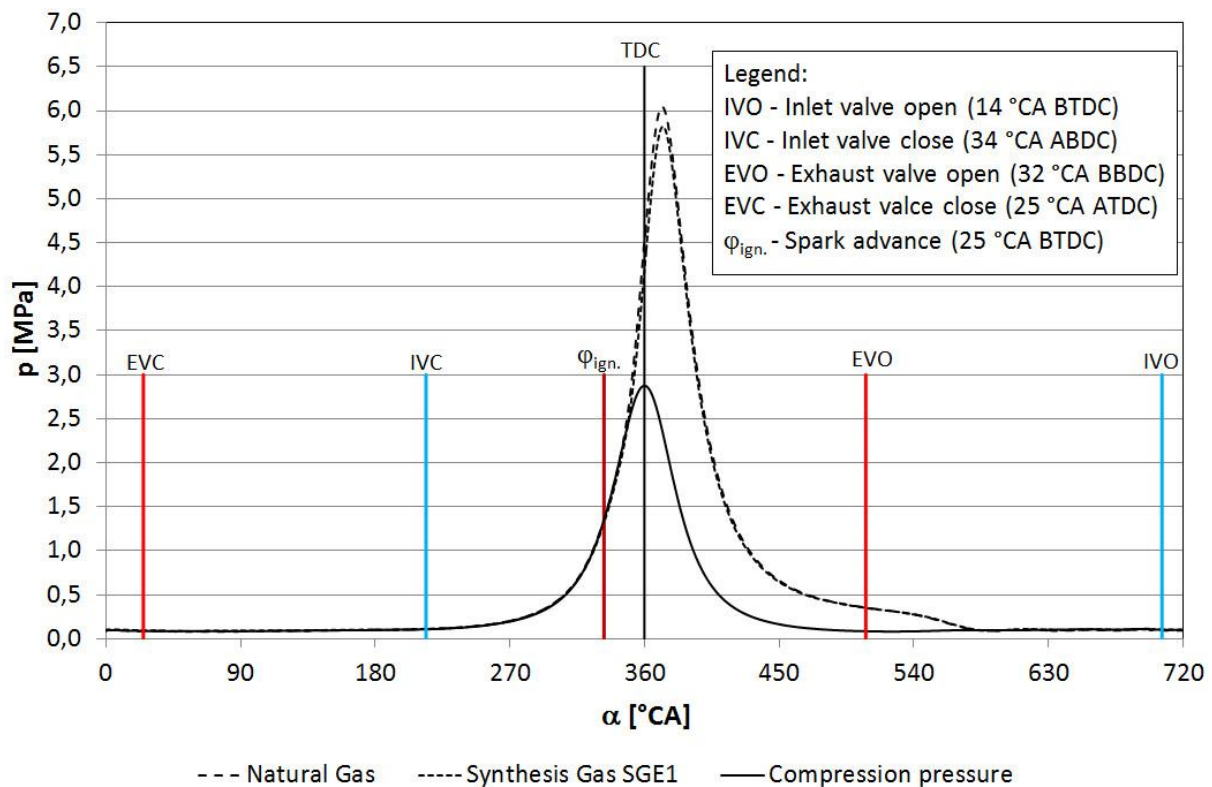
Fig. 4 Comparison of average pressure courses in the combustion chamber of the engine
during combustion of natural gas or synthesis gas
(Conditions: spark advance φ_{ign} = 25 °CA BTDC, full load, air excess ratio λ = 1)

Fig. 5 depicts the above-mentioned values of maximum pressure for 190 successive cycles, combusting synthesis gas, with an indication of the average of the maximum pressure.
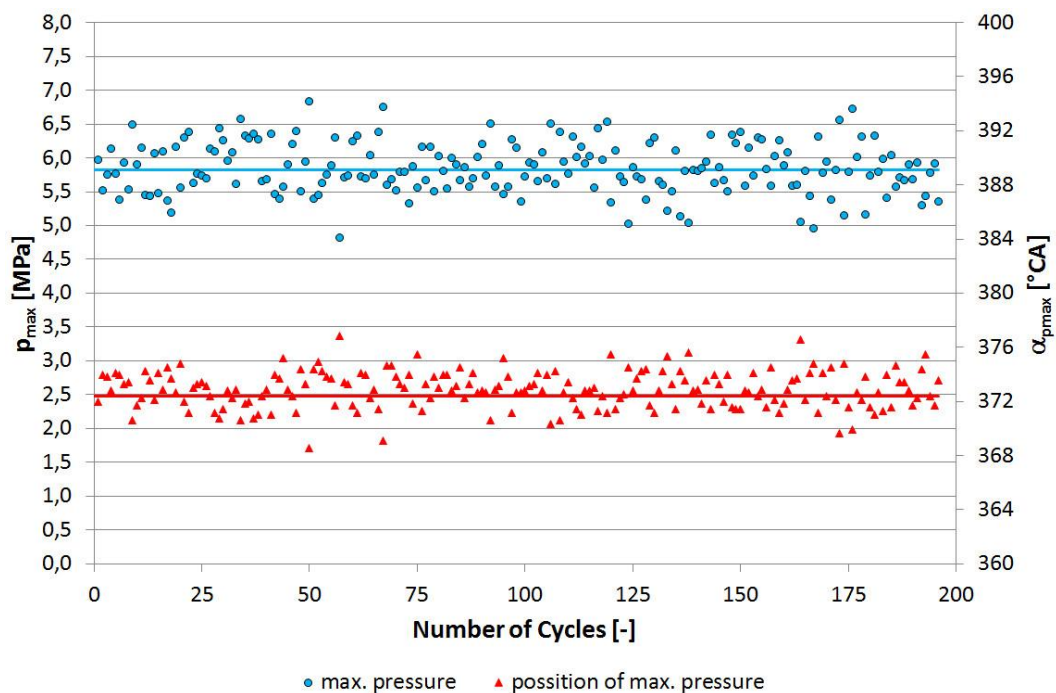

Fig. 5 The values of maximum pressure (p_{max}), and positioning of maximum cylinder pressure
(α_{pmax}) for particular successive cycles combusting synthesis gas SGE1
(Conditions: spark advance φ_{ign} = 25 °CA BTDC, full load, air excess ratio λ = 1)

The following figure (Fig 6) shows the logarithmic p - V diagram, with the help of which we determine the beginning and end of combustion. The polytrophic lines were constructed by the method of least squares. The deflection of the pressure curve from the polytrophic straight line in logarithmic coordinates determines the beginning of burning of the mixture (SOC). The value of the average polytrophic coefficient at compression is 1.33, and at expansion 1.28. The decline of the coefficient relates with changes in specific heat capacity of the combustion products and partially with blowing through the piston rings into the crankcase space [6]. The end of combustion (EOC) was determined in a similar way as the start of combustion of the mixture.
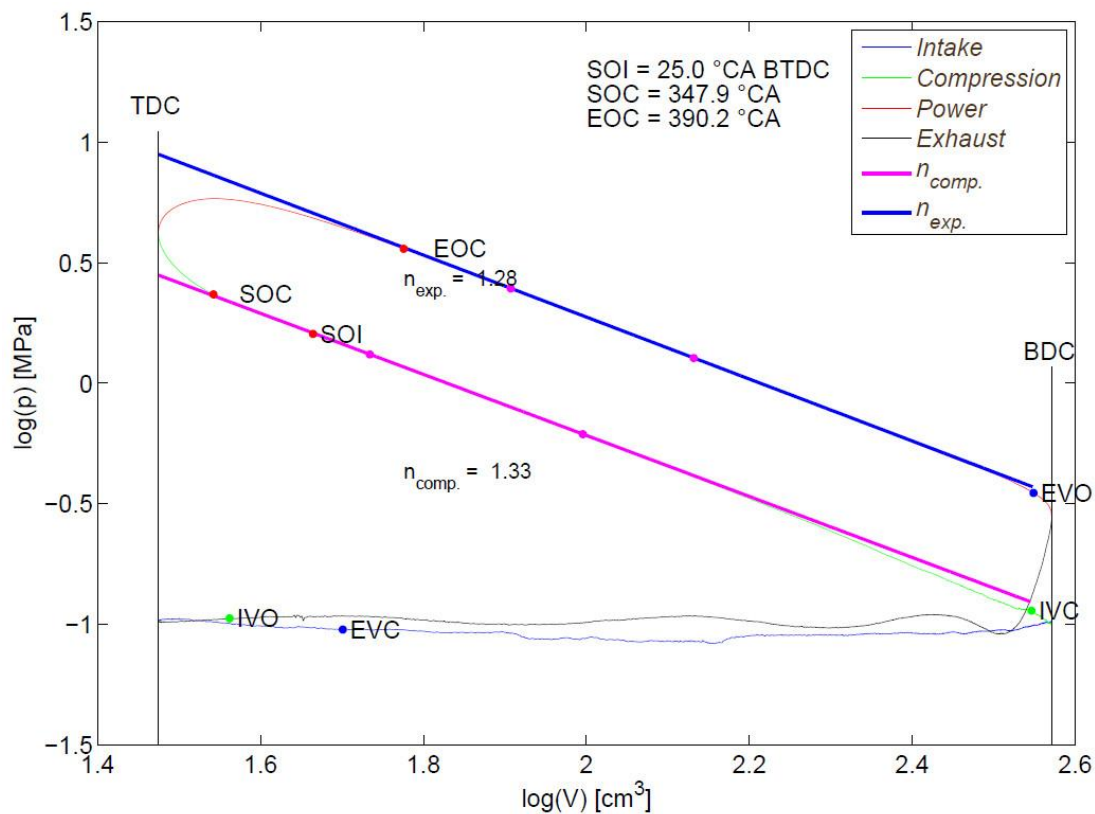

Fig. 6 Logarithmic p - V diagram of the pressure course related to the volume of the cylinder operating on synthesis gas SGE1
(Conditions: φ_{ign} = 25 °CA BTDC, full load, air excess ratio λ = 1, speed n = 1500 min^{-1})

The figure (Fig. 7) shows the dependencies of the mass fraction burned (MFB) of the fuel on the crankshaft angle α for 195 consecutive cycles, as well as the average cycle value for natural and synthesis gas. The analysis leads us to the conclusion that combustion of synthesis gas leads to more stable combustion, which is also documented by a dispersion of individual curve courses for MFB. In the early stage the burning takes place faster for combustion of synthesis gas, when the angle at which the burnt fuel is 5% is 352.6 ° CA in the synthesis gas as compared to the natural gas (354.5 ° CA). When 10% of fuel is burnt (MFB=0.1), the value of angle for the synthesis gas is 355.4 ° CA. For the natural gas the position of the crankshaft has the value of 357.5 ° CA. If higher values of MFB are present, the burning is faster in natural gas than in synthesis gas, which is reflected in closer agreement between angle positions of the crankshaft angle, when 50% of fuel is burnt. The average value of the position of the crankshaft, when MFB is 0.5 (50 % of fuel is burnt), is 366.2 ° CA for natural gas and for the synthesis gas this value is 366.5 ° CA. When burning synthesis gas, the subsequent combustion takes place much faster than when burning natural gas, which can be expressed in

numerically via the angle when 90 % of fuel is burned (MFB = 0.9). For natural gas, this value is 381.6 ° CA and for the synthesis gas the value is 377.7 ° CA. A statistical analysis of coefficients of variation for particular periods, when successively 5 %, 10 %, 50 % and 90 % of fuel was burnt, indicates that combustion of the synthesis gas leads to reduced coefficients. The greatest difference between the coefficients of variation (COV [4]) is in burning 5 % of fuel, when the COV value for synthesis gas is 0.385 % and for the natural gas is 0.744 %. Generally (both for natural and synthesis gases) with rising MFB, the coefficient of variation of the crankshaft angle also increases. The highest value is reached when burning at 90% fuel (MFB=0.9), when the coefficient of variation of the angle for natural gas has the value 0.994 % and reaches the value 0.327 % for the synthesis gas.
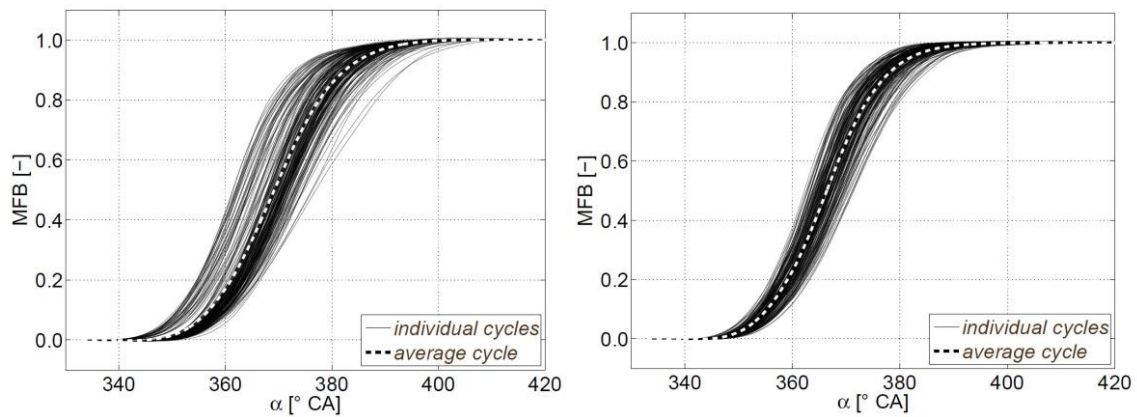

Fig. 7 Mass fraction burnt (MFB) when combusting natural (left) or synthesis (right) gas
(Conditions: spark advance φ_{ign} = 25 °CA BTDC, load $\alpha_{throttle}$ = 100 %, air excess ratio λ = 1,
speed n = 1500 min^{-1})

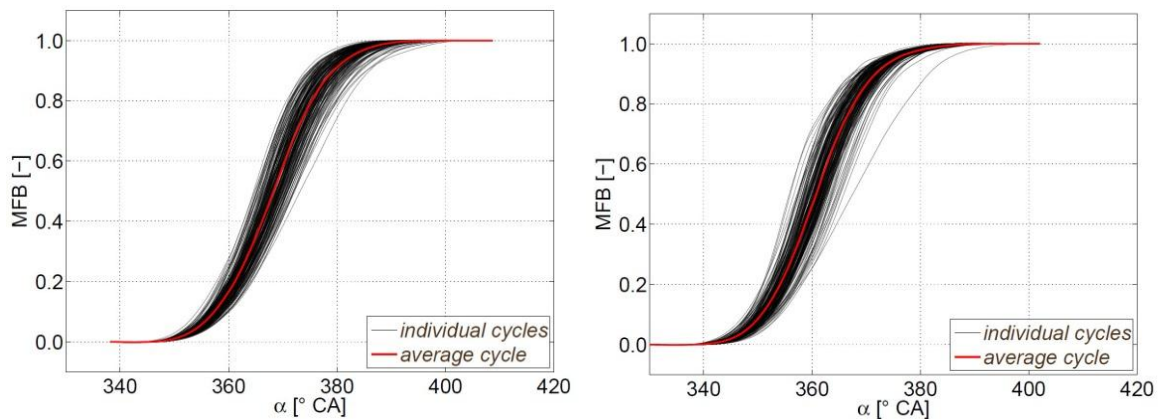The last picture (Fig. 8) shows the impact of the course of spark advance on the character of the burnt fuel (syngas).

Fig. 8 Mass fraction burnt (MFB) at the angle of spark advance φ_{ign} = 22 °CA BTDC (left)
and φ_{ign} = 30 °CA BTDC (right)
(Conditions: fuel: synthesis gas SGE1, load $\alpha_{throttle}$ = 100 %, air excess ratio λ = 1,
speed n = 1500 min^{-1})

Numeric values of the measured angles of rotation of the crankshaft, as depending on the percentage of burnt fuel, with coefficients of variation (COV), are shown in Table 3.

Tab. 3 The values of the angle of crankshaft after gradual burning 5%, 10%, 50% and 90% of fuel (MFB), and related values of the coefficient of variation (COV)

	Synthesis Gas SGE1				Natural Gas NG			
	φ_{ign} = 22 °CA BTDC		φ_{ign} = 30 °CA BTDC		φ_{ign} = 22 °CA BTDC		φ_{ign} = 30 °CA BTDC	
	α [°CA]	COV$_\alpha$ [%]	α [°CA]	COV$_\alpha$ [%]	α [°CA]	COV$_\alpha$ [%]	α [°CA]	COV$_\alpha$ [%]
5% MFB	354.5	0.27	348.0	0.32	355.9	0.30	349.3	0.36
10% MFB	357.4	0.30	350.8	0.36	359.1	0.34	352.1	0.38
50 % MFB	368.4	0.46	360.8	0.51	371.3	0.53	362.6	0.53
90 % MFB	379.0	0.61	371.3	0.50	383.4	0.79	373.3	0.63

CONCLUSION

The utilization of synthesis gas in the spark-ignition engine and the impact of this fuel on the power and economic parameters, as well as on the parameters relative to the actual incineration, when compared to natural gas data, can be summarized as follows:

- the power parameters (Fig. 1) for synthesis gas from pyrolysed plastics (SGE1), compared to the natural gas data, have been reduced by about 5-6 % across the overall operating speed range, under the optimum angle of spark advance 25° CA BTDC set at speed 1 500 min^{-1},

- characteristic curves of the engine torque (Fig. 3), or performance parameters of the engine, depending on the angle of spark advance, is the same both for natural and synthesis gas,

- the angle of the ignition delay in synthesis gas increases by about 5-7%, compared with natural gas,

- the specific fuel consumption (Fig. 2) has increased by about 127 % for synthesis gas combustion as compared to natural gas data, a rise in value from 245 g.kW^{-1}.h^{-1} to 578 g.kW^{-1}.h^{-1} at 1500 min^{-1}. This equals effective efficiency of the engine 30 % for natural gas, and 33% for synthesis gas,

- in terms of pressure courses (Fig. 4), the synthesis gas has not significantly affected the value of maximum pressure nor pressure rise increase, compared to natural gas,

- as for combusted synthesis gas, the nature of the fuel burning is more uniform (lower COV) for all successive cycles compared to the combusted natural gas (Fig.7, Tab. 3),

- synthesis gas SGE1 obtained from pyrolysis of plastics is a quality fuel fully suitable to be utilized in cogeneration units.

ACKNOWLEDGMENT

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