INTERNATIONAL JOURNAL OF INFORMATION TECHNOLOGIES, ENGINEERING AND MANAGEMENT SCIENCE

Analysis of Internal Parameters of the Low-energy Synthesis Gases in the Combustion Engine

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Abstract

This article is focused on the application of the low-energy synthesis gas generated from municipal waste in the combustion engine. Influence of the gas composition on the internal parameters of the combustion engine used in the cogeneration unit was analysed. These analyses show that decrease in the power output in the form of the IMEP is in the region of 10% - 40% compared to the methane. With an increasing proportion of hydrogen in the fuel the heat release speed was increased together with pressure rise rate. Presented results are useful for deeper understanding of the processes that are happening in the combustion engine, and they also show the importance of synthesis gas composition. This can be useful for knowing how to set up syngas production processes to achieve the highest energetic and economic outputs possible.

Keywords: pressure analysis, cogeneration unit, syngas

Introduction

European energy crisis in the last two years showed how dependent is whole European region on the traditional energy sources and how big of an issue can arise from this. This and other factors lead to the higher diversification of the energy sources and also higher use of alternative energy sources and their economic support. Furthermore, European regulation [1] directs as low landfilling as possible and increase in the use of the renewable energy sources. This results in the increased interest in the waste-to-energy (WtE) processes as well as combined heat and power generation (CHP) in the cogeneration unit where main fuel source can be waste generated synthesis gas.

Increase in the municipal waste production and predicted further increase in the global scale from 2.01 bil. tons in the 2016 to 2.59 bil. tons in the 2030 and 3.4 bil. tons in 2050 comes with the many negatives. These are further deepened by its composition, mismanagement and not enough recycling and utilization after the primary use [2]. This subsequently results in the negative impact on many local and global ecosystems [3]. As in European union, also in Slovakia the production of municipal waste over the years is growing. In the year 2012 one citizen of the European union produced on average 488kg of waste per year. This grew to 530kg in 2021. In Slovakia these values were 306kg in 2012 and 496kg in 2021 [4]. On the other hand, municipal waste treatment in Slovakia is developing towards more sustainable ways as can be seen in the graph bellow (Fig.1). The amount of the recycled waste is steadily growing with the slight decrease in landfilling [5].

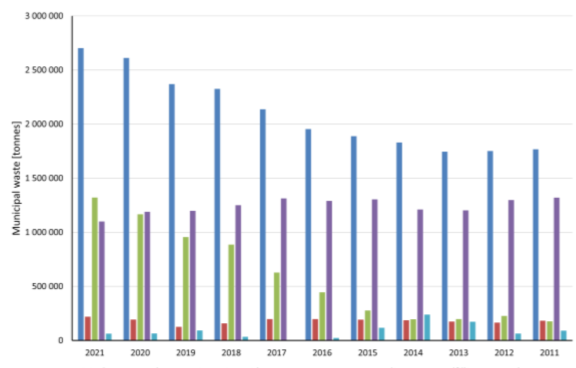

Fig. 1 Trend of municipal waste production and treatment (in Tonnes) in Slovak Republic

Ways to the decrease in the landfilling include primarily methods of reducing the waste amount by the increase in the number of the useful cycles, recycling and other primarily energy recovery utilizing. Naturally not everything can be used multiple times or effectively recycled due to the material properties or contamination resulting in the large quantity of waste that is needed to be differently processed to decrease the landfilling. One of the possible methods is above mentioned energy utilization of the waste, when energy stored in the chemical bonds of the waste is used to provide electricity and heat. This can be achieved by direct combustion or by gasification of the waste and its subsequent combustion on the cogeneration CHP unit. These processes are commonly called "Waste-to-Fuel" [6, 7, 8].

Combustion, or obtaining electricity and heat from the fuels generated in this way can be carried out with gas turbines, steam turbines or piston combustion engines. Each method has positives and drawbacks for different use scenarios. However, it is true that when using combustion engines and gas turbines, it is possible and usually necessary to mechanically and chemically clean waste fuels before combustion. Thanks to this, when compared to direct burning of waste, it is possible to additionally reduce the amount of emitted harmful gas emissions. Likewise, when using cogeneration methods (combined production of heat and electricity), it is possible to achieve a total effective production efficiency of more than 90% [9, 10].

Thanks to the use of fuels obtained in this way, we can not only significantly reduce the volume and weight of municipal waste, but also reduce the need to use fossil resources, which bring with them additional environmental risks [11].

The impact of the combustion of gases produced by the thermal decomposition of organic substrates (where we can also include municipal waste), the so-called synthesis gases ("syngas") on the parameters of reciprocating combustion engines have already been studied in several studies. However, this is a very extensive issue, as due to the chemical composition of the substrate, as well as the processing process used and their boundary conditions, it is possible to achieve high variability in the composition of the obtained gases. However, obtained gases are, after mechanical and chemical purification, composed primarily of hydrogen, carbon monoxide, methane and higher hydrocarbons and depending on the process used also of a small (up to 5% vol.) or large amount (more than 50% vol.) of inert components (carbon dioxide and nitrogen). All the mentioned components and their presence in the produced gas not only have a significant impact on the calorific value of such gas, but also on its properties during combustion. This can have a significant impact not only on the economic and performance parameters of the used engine, but also on its life expectancy [10, 12, 13, 14, 15].

This work deals with the experimental research on the effect of burning six low-energy synthesis gases, which are supposed to simulate certain real compositions of synthesis gases produced during the gasification of municipal waste. These are gases with a constant content of inert components N_2 (50% vol.) and CO_2 (10% vol.). The contents of combustible components vary: CH_4 (0 – 10% vol.), H_2 (10 – 30% vol.) and CO (10 – 30% vol.). Similar compositions can be obtained with the methods of gasification of municipal waste with access to air, as reported by e.g., Mohammad A. [15] and Santos S.M. [16]. Synthesis gases analysed by us are chemically pure, which means that they are mainly cleaned of tars and water vapor. Tars would cause clogging of narrow openings as well as filters in the combustion engine [17, 18]. The amount of tar in the gaseous fuel should not exceed the value of 0.01% vol., which corresponds to the value of 10 mg/Nm^3 for combustion engines. At values above 30 mg/Nm^3, the operation of the combustion engine begins to be problematic [19].

In the following figure (Fig. 2) a ternary diagram is shown, in which the individual compositions of synthesis gases investigated are shown in such a way as to cover the entire area of the possible composition of synthesis gas, which is created by the gasification of municipal waste. Combustible components are in the following ranges: CH_4 0 – 20% vol., H_2 10 – 30% vol., CO 10 – 30% vol. The amount of inert gases is constant for each synthesis gas (CO_2 10% vol., N_2 50% vol.).

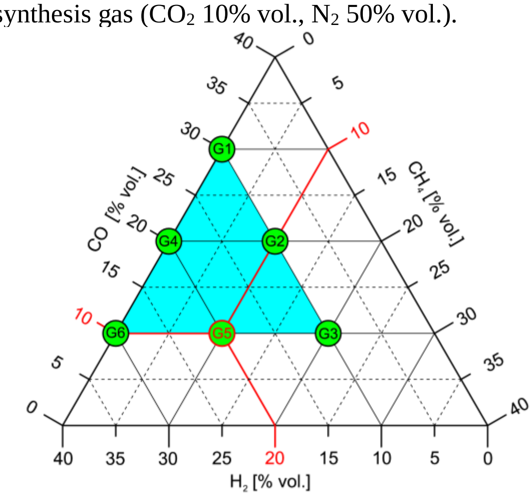

Fig. 2 Ternary diagram of synthesis gas composition with the constant inert gas proportion (10 % vol. CO_2, 50 %vol. N_2), together with the highlighted

composition of the G5 (10 %vol. CO, 10%vol. CH_4, 20%vol. H_2)

The following table (Tab. 1) shows the basic physical-chemical properties of the above selected synthesis gases, compared to the reference fuel, which is methane. The selected synthesis gases are included in the category of low-energy fuels, which have a lower calorific value in the range from 4 MJ/kg to 9 MJ/kg. Based on these properties, both the performance and economic parameters of the combustion engine can be predicted. From the preliminary analysis, the expected performance parameters can be determined in advance based on the volumetric calorific value of the stoichiometric mixture of fuel and air. This has the largest share in the output performance parameters of the engine. In addition to this parameter, the output performance parameters are also influenced by the quality of filling the cylinder with fresh mixture and, last but not least, the rate of heat release and volume formation of exhaust gases. From the point of view of the fuel consumption, the largest influence is the proportion of fuel in the mixture with air, where this parameter ranges from 29.6% vol. up to 51.5% vol. Compared to the combustion of methane (9.5% vol.), this is a three to five and a half increase in the amount of fuel in a stoichiometric mixture with air.

Tab. 1: Physical and chemical properties of methane (CH_4) and synthesis gases (G), (LHV – lower volumetric heating value of fuel, A/F – air to fuel ratio, $\rho_{NTP\ fuel}$ – density of fuel at NTP, $LHV_{mixture}$ – lower volumetric heating value of stoichiometric mixture, NTP = 20°C, 101 325 Pa

Parameter	*Unit*	*CH_4*	*G1*	*G2*	*G3*	*G4*	*G5*	*G6*
CH_4	% vol	100	0	10	20	0	10	0
H_2	% vol	0	10	10	10	20	20	30
CO	% vol	0	30	20	10	20	10	10
CO_2	% vol	0	10	10	10	10	10	10
N_2	% vol	0	50	50	50	50	50	50
LHV	MJ/kg	**50.012**	**4.043**	**6.234**	**8.650**	**4.298**	**6.762**	**4.620**
LHV	MJ/m^3	33.353	4.540	6.690	8.852	4.362	6.526	4.190
A/F ratio	kg/kg	17.1	1.0	1.9	2.8	1.1	2.1	1.3
fuel in air	% vol	9.5	51.5	37.6	29.6	51.4	37.5	51.2
$\rho_{NTP\ fuel}$	kg/m^3	0.667	1.123	1.073	1.024	1.015	0.965	0.907
$LHV_{mixture}$	MJ/kg	2.761	2.001	2.180	2.283	2.027	2.202	2.044
$LHV_{mixture}$	MJ/m^3	**3.172**	**2.323**	**2.512**	**2.619**	**2.240**	**2.449**	**2.147**

Materials and methods

Experimental measurements were carried out on a Lombardini LGW 702 spark-ignition combustion engine designed to drive a micro-cogeneration unit [10, 20]. A two-cylinder naturally aspirated engine with a displacement of 686 cm^3 was used to reduce operating costs for experimental measurements (Fig. 3). Synthesis gases (provided by supplier) were mixed in pressure bottles and with the help of a two-stage regulator, the gas was supplied to the mixer with the diffuser.

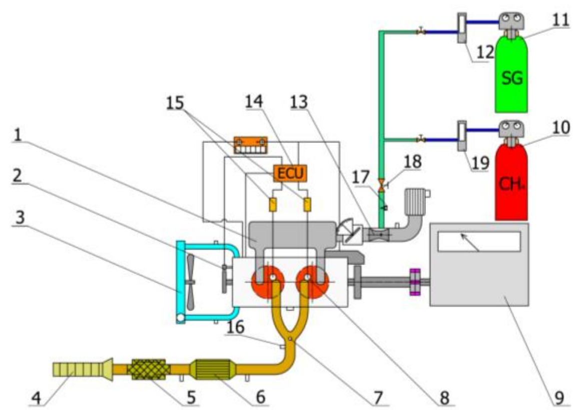

Fig. 3: The basic scheme of the internal combustion engine Lombardini LGW 702
(1 - intake manifold, 2 – crankshaft position sensor, 3 - water radiator, 4 - exhaust gas removal system, 5 - silencer, 6 – catalytic converter, 7 - exhaust temperature and pressure sensors, 8 - spark plug with an integrated pressure sensor, 9 - dynamometer, 10 - pressure bottle with methane, 11 - pressure bottle with syngas, 12,19 – fuel mass-flow meter, 13 - mixer with diffuser, 14 - engine control unit, 15 - ignition coil, 16 - broadband lambda probe, 17 - stepper engine,18 – manual mixture richness regulation)

The internal combustion engine was connected to the MEZ Vsetín electric induction brake (1DS 736V) via an elastic clutch. All measurements were realized at a full load and engine speed of 1 500 min^{-1} due to the considered operation of a micro-cogeneration unit with a four-pole electric power generator. The exhaust system was equipped with a three-way catalyst, which works most efficiently with a stoichiometric composition of the mixture. This composition of the mixture formed in the mixer is regulated by the engine control unit (ECU), which uses feedback control algorithm utilising the broadband lambda probe located in front of the catalyst and subsequently controls the flow of fuel via a stepper motor (located in the fuel line) regulating the amount of fuel to the mixer with the diffuser. A piezoelectric pressure sensor integrated in the Kistler 6118CC-4CQ02-4-1 spark plug was used to analyse the combustion process of the mixture. In order to correct the dynamic course of the pressure in the cylinder, a Kistler 4075A10 piezoresistive pressure sensor was placed in the intake manifold, which measured the absolute value of the pressure. In the area of bottom dead centre at the time of the end of the intake stroke, the same pressure is assumed in the cylinder and in the intake manifold, and in this way the value of the dynamic course of the pressure in the cylinder was adjusted. A Kistler 2613B1 encoder was mounted on the crankshaft timing pulley, which provided information on the current position of the crankshaft as well as the top dead centre position. In addition to the pressure curves (in the cylinder and in the intake manifold) and the position of the crankshaft, the current value of the ignition timing angle was also recorded in the bus module using a sensor with an optocoupler, which was connected in parallel with the combined BOSCH P65-T ignition coil, which supplies the ignition system with spark energy of up to 65 mJ. The processing and evaluation of the measured data itself was done in an in-house developed program in the Matlab programming environment from MathWorks.

The analysis of the course of the fuel combustion was carried out based on the theory of a one-zone dimensionless thermodynamic model [21]. The model was considered for a closed system (when zero enthalpy flow across the system boundaries is assumed) at the time of compression and expansion, assuming the validity of the law of conservation of energy. The Rassweiler-Withrow method was used to analyse the course of the fuel combustion (directly proportional to the release of heat) based on the knowledge that the total pressure in the cylinder consists of a partial increase in the pressure from the movement of the piston and a partial increase in pressure from the combustion itself. The calculation is based on the following relationship:

$$dU = dQ - dW + \sum_i h_i . dm_i \quad (1)$$

Where:

dU - differential of internal energy of matter in the system

dQ - differential of heat delivered to the system

dW -differential of the work produced by the system

$h_i.dm_i$ - *i*-th component of enthalpy of mass flow across system boundaries (during combustion, this term is assumed to be zero).

The start and the end of a combustion was determined based on two methods. The first method was based on the entropy change in a closed system, and the second method was based on the logarithmic *p-V* diagram (deviation of the combustion curve from the compression or expansion straight line). During each operating mode (change of ignition timing angle, change of fuel) 197 consecutive cycles were recorded at the mentioned operating speed of the internal combustion engine of 1500 rpm and full load.

Results and discussion

Among the most important indicators of the performance parameters that can be obtained from the indication of the pressure in the cylinder is the value of the indicated mean effective pressure (*IMEP*). The following graph (Fig. 4) shows averaged IMEP curves (based on the 197 consecutive cycles for each evaluated engine conditions) depending on the angle at which 50% of the fuel is burned (the indicator that best describes the combustion process). The ignition timing angle was gradually reduced from a value of 40°CA BTDC to a value where the combustion engine did not experience misfiring. The graph also shows the dispersion of individual cycles, which represents the so-called parameter coefficient of variation (COV), which is determined as the ratio of the standard deviation to the arithmetic mean. The reference fuel against which the results were compared is pure methane. The highest IMEP values (0.867 MPa) were achieved when operating on synthesis gas G3, which had the highest methane content (20% vol.). In its optimal operating mode (φ_{ign} = 34°CA BTDC), the value of COV_{IMEP} was 1.16%, which can also be seen in the dispersion of the individual values. On the contrary, the lowest values of IMEP (0.576 MPa) were measured during the operation of the combustion engine on synthesis gas labelled G6, which has had the highest proportion of a hydrogen (30% vol.). The largest dispersion of IMEP and thus the highest value of COV (3.87%) has had syngas G1 with zero representation of methane.

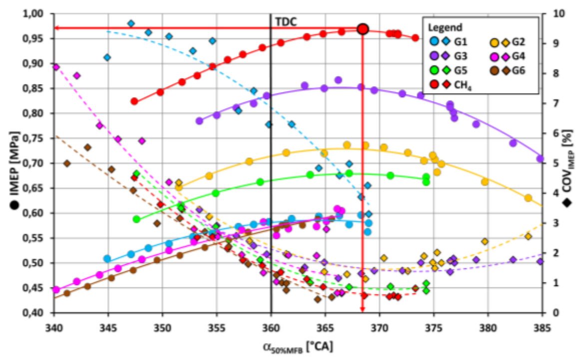

***Fig. 4:** Course of the indicated mean effective pressure (IMEP) and coefficient of variation (COV_{IMEP}) dependent on the angle of the crankshaft where the 50 % of the fuel mass is burned ($\alpha_{50\%MFB}$) for methane and researched gases (G1-G6). Conditions: 1500 rpm, full load, stoichiometric mixture.*

Fuel combustion parameters have a fundamental influence on the operation of the combustion engine. One of such significant parameters describing the course of fuel combustion is the angle $\alpha_{50\%MFB}$, which refers to the moment when 50% of the fuel is burned. This angle directly affects the indicated mean effective pressure of the cycle and thus also the output economic, as well as performance parameters of the combustion engine [22, 23, 24, 25, 26]. Several authors agree that in order to achieve the highest efficiency and indicated mean effective pressure, the ideal angle $\alpha_{50\%MFB}$ for spark-ignition engines is in the range of 8 to 10° of a crankshaft rotation after top dead centre (CA ATDC) [24, 25, 26, 27]. However, some authors extend this interval to 5 to 11° CA ATDC [22]. As can be seen from the graph (Fig. 4), with decreasing ignition timing angle, the angle $\alpha_{50\%MFB}$ moves beyond the top dead centre (TDC), but due to the problems with the uniformity of the combustion engine operation (misfiring), the full control characteristic of IMEP was not achieved for some of the gases, which have a high hydrogen content in the mixture (G6). The smallest optimal ignition timing angle for G6 was 21°CA BTDC, where the IMEP value of 0.576 MPa was reached at the position $\alpha_{50\%MFB}$ 5.5°CA ATDC. For the other synthesis gases, the angle $\alpha_{50\%MFB}$ at which the highest performance parameters were achieved varied in the interval from 6.1°CA ATDC (G4) to 8.3°CA ATDC (G1). The decrease in performance parameters compared to the operation on methane was from 10% to 40%.

Following graph (Fig. 5) shows the averaged pressure curves in the time of compression and expansion during the engine operation with the optimal ignition timing angle for each fuel, which changed due to the different composition of the synthesis gas compared to the reference fuel, which was methane. The highest values of the maximal pressure (5.86 MPa) were achieved by synthesis gas labelled G3, which has the highest proportion of methane in the mixture. Due to the large negative area before TDC and the rapid decrease in the pressure curve in the expansion stroke, it has lower performance parameters compared to methane, although the maximal pressure differs only slightly (a difference of 0.2 MPa). Conversely, the lowest values of the maximal pressure (4.56 MPa) was achieved when burning G1.

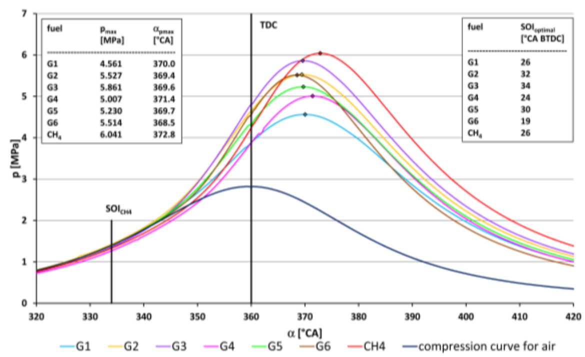

***Fig. 5:** Pressure profile p in the cylinder of an internal combustion engine during operation on methane and synthesis gases (G1-G6). Conditions: 1500 rpm, full load, stoichiometric mixture, optimal start of ignition (SOI) angle for each fuel, compression curve is for air.*

In addition to the coefficient of variation of the indicated mean effective pressure, the parameter that also relates to the stability of the engine operation as well as the life expectancy of the engine is also the coefficient of variation of the maximal pressure. When operating on the reference fuel (methane), its value was 6.8%. Among the synthesis gases, the highest value of COV_{pmax} (approximately 8%) was reached while using gases marked G1, 2 and 5, on the other hand, the lowest value (4.8%) was observed when operating on the G6, which contained the highest proportion of hydrogen in the fuel mixture. An important parameter revealed by increased noise during combustion and also a shorter service life of individual engine components is the high pressure rise rate. When operating on methane, the pressure rise rate value was 0.225 MPa/1°CA. When operating on synthesis gases, the highest pressure rise rate (0.233 MPa/1°CA) was achieved with gas labelled G6, due to the increased amount of hydrogen contributing to a faster course of a heat release. On the contrary, the lowest value (0.160 MPa/1°CA) was recorded when operating on synthesis gas G1, where carbon monoxide was represented in the highest amount (30% vol.). In general, the higher the proportion of internal gases in the mixture, the lower the pressure rise rate. There is a visible correlation between the length of the main combustion (10-90% MFB) and the pressure rise rate value. The longer the duration of the main combustion, the lower the pressure rise rate. In general, it can be stated that increasing the proportion of hydrogen in the mixture increases the value of the pressure rise rate.

The following graph (Fig. 6) shows the dependences of the maximal pressure from the value of the angle when 50% of the fuel mass is burned. The plot curves were created by gradually changing the ignition timing from a value of 40°CA BTDC up to the ignition timing value, where stable engine operation was recorded (without missing combustion). The step change was 2°CA and during each measurement 197 consecutive cycles were recorded. By graphically displaying and subsequent analysis of the maximal pressures for the reference fuel the outcome where value of the maximal pressure settles at approx. 8.5 MPa is achieved and by increasing the ignition timing, this value practically does not change. This is due to the fact that the main combustion takes place before TDC, i.e. at the time of the compression stroke and at the same time the piston has not yet reached TDC and therefore the reduction of the stroke volume is still taking place. The course of mentioned dependence has an inverted S-shape. Inflection point has a value of approximately 6.1 MPa and 8.5°CA ATDC. This inflection point region represents engine operation on pure methane with an optimum ignition timing angle (26°CA BTDC). As can be seen from the following graph (Fig. 7), the value of $\alpha_{50\%MFB}$ (8.6°CA ATDC) for the optimal ignition timing angle (26°CA BTDC) is approximately the same as the position of the inflection point. The graph shows the incompleteness of the curves for G4 and G6 gases, where the values beyond the inflection point were not reached due to the instability of the combustion during the engine operation at a small ignition timing angle. The graph also shows the dispersion of the maximal pressure, which is numerically expressed by the coefficient of variation and corresponds to the previous coefficients of variation.

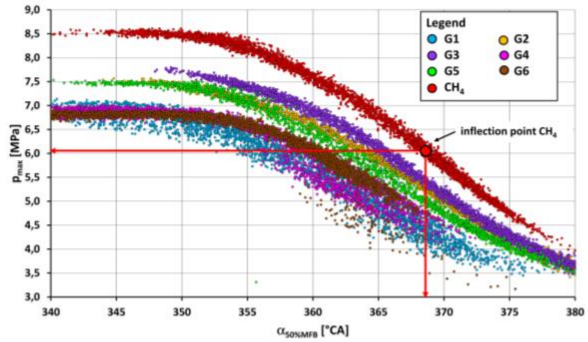

Fig. 6: The course of the maximal pressure depending on the angle at which 50 % of the fuel for methane and synthesis gases (G1-G6) is burned. Conditions: 1500 rpm, full load, stoichiometric mixture.

In general, it can be stated that the location of the inflection point of the curves in the previous graph corresponds to the optimal operation of the combustion engine with the highest performance parameters.

The graph bellow (Fig. 7) shows the fuel burnout curves (Mass Fraction Burned - MFB) depending on the crankshaft rotation angle for different synthesis gas mixtures compared to methane. The ignition delay (time between spark (Start Of Ignition - SOI) and moment of visible combustion (Start Of Combustion - SOC)) is approximately 12.5°CA at the optimal ignition timing angle for methane. Main combustion (10-90% MFB) takes 24.4°CA. The total burning time (time between SOC and EOC) is approximately 56°CA for methane.

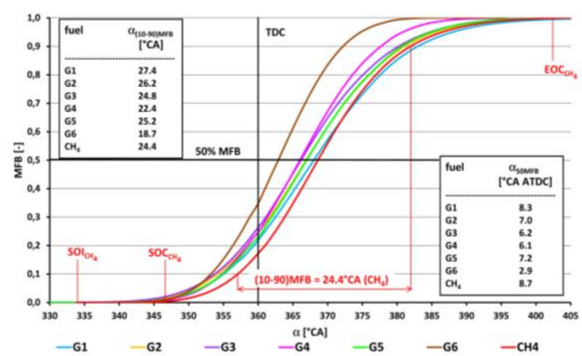

Fig.7: Course of fuel burnout as a dependent on the crankshaft angle for methane and synthesis gases (G1-G6)(MFB - Mass Fraction Burned, α - Crankshaft Rotation Angle, TDC – Top Dead Centre, SOI – Start of Ignition, SOC – Start of Combustion, EOC – End of Combustion) Conditions:1500 rpm, full load, stoichiometric mixture, optimal SOI angle for each fuel

If we compare the combustion processes of synthesis gases at optimal ignition timing angles, the fastest fuel burnout is with gas (G6) with the largest proportion of hydrogen, when the main combustion time is 19.8°CA. Conversely, G1 gas, which has the highest proportion of carbon monoxide (30% vol.) and does not contain methane has the longest fuel burn-up time (27.4°CA). In the initial phase of heat release, i.e. in the moment from the spark to the angle when 5% of the fuel mass is burned. This time is the shortest (12.2°CA) when burning synthesis gas (G6) with the highest proportion of hydrogen. On the contrary, the longest initial combustion time (24.8°CA) was documented while burning G3 gas, which has the highest proportion of methane.

The coefficient of variation of the position angle (COV_{α}), when a given mass fraction of the fuel is burned, generally increases with the increasing mass fraction of the burned fuel. The highest COV values in each phase of combustion were recorded when operating on G1 gas ($COV\alpha_{10\%MFB}$ = 0.50%, $COV\alpha_{50\%MFB}$ = 0.79%, $COV\alpha_{90\%MFB}$ = 1.41%). On the contrary, the lowest variation parameters were acquired while operating the gas with the highest proportion of hydrogen (G6) ($COV\alpha_{10\%MFB}$ = 0.26%, $COV\alpha_{50\%MFB}$ = 0.42%, $COV\alpha_{90\%MFB}$ = 0.61%). For the reference fuel (CH_4) were the values of coefficients of variation as follows: $COV\alpha_{10\%MFB}$ = 0.36%, $COV\alpha_{50\%MFB}$ = 0.53%, $COV\alpha_{90\%MFB}$ = 0.71%.

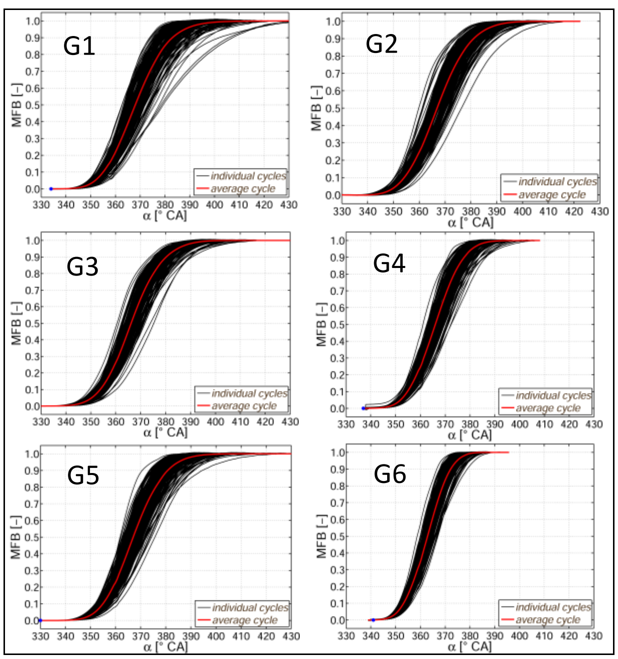

Fig.8: The course of mass fraction burned (MFB) for measured synthesis gases (G1 – G6) at optimal start of ignition angle (conditions: speed 1500 rpm, full load, stoichiometric ratio, 197 consecutive cycles)

Course of the fuel burnout is depicted in the graph (Fig.8) this consists of 197 subsequent cycles for each evaluated gas with highlighted average value during optimal ignition timing. From the course of the individual MFBs spread of this value is clearly visible which is documented by above mentioned coefficients of variation.

Conclusion

Synthesis gases produced from renewable energy sources, to which belongs also municipal waste, are a sustainable source of energy that contributes to the reduction of the carbon footprint and ensures the diversification of energy sources. From the analysis of the pressure course in the cylinder as well as from the composition of the fuel, it can generally be concluded that the volumetric calorific value of the stoichiometric mixture has the greatest influence among all the known parameters determining the output combustion engine performance. To a lesser extent, the quality of cylinder filling with a fresh mixture, which is characterized by the volumetric efficiency and the rate of heat release, when it is ideal to burn the fuel as quickly as possible in the area close to the top dead centre, when the pressure course in the cylinder is closest to the ideal isochoric circulation have also influence on the integral engine parameters.

Summarizing the basic knowledge in the operation of the combustion engine on synthesis gases of different composition with a focus on its internal parameters are given in the following points:

- for each fuel composition, it was necessary to set the optimal ignition timing, which ranged from 21°CA BTDC for G6 gas with the highest proportion of hydrogen (30% vol.) to a value of 34°CA BTDC for G3 gas, which, on the other hand, had a high proportion of methane (20% vol.) and a low proportion of hydrogen (10% vol.). The generally high proportion of inert gases (60% vol.) contributes to the prolonged combustion of the mixture, and for this reason it is necessary to increase the ignition timing angle.
- a decrease in performance parameters (from 10 to 40%) in the form of IMEP was recorded for all investigated gases compared to the methane operation. The lowest value of IMEP (0.576 MPa) was acquired when operating on G6 with the lowest calorific value of the mixture (2.147 MJ/m^3), and on the contrary, the highest value of IMEP (0.867 MPa) was achieved with the G3 gas

with the highest calorific value of the mixture (2.619 MJ/m^3).

- highest value of the maximal pressure (5.86 MPa) was reached when operating on synthesis gas G3 and the lowest value of the maximal pressure (4.56 MPa) was observed during the operation of the syngas labelled G1. The coefficient of variation of the maximal pressure was the lowest (approx. 4.8%) when operating on gas mixtures with a high proportion of hydrogen (G6), and on the contrary, the highest values (around 8%) where reached while burning G1, 2 and 5 gases.
- in general, it can be stated that by increasing the proportion of hydrogen in the mixture, the pressure rise rate also increases. The highest value (0.233 MPa/1°CA) was gathered while operating on G6 gas, which has also the highest hydrogen content (30% vol.). Conversely, the lowest value (0.160 MPa/1°CA) was for G1 gas with a high proportion of carbon monoxide (30% vol.). The pressure rise rate is correlating with the length of the main combustion event and there is a directly proportional dependence between them, i.e. the shorter the burning time, the higher the pressure rise rate.
- the main combustion event is generally shorter with the larger proportion of hydrogen in the mixture and, on the contrary, it is prolonged by the increasing proportion of methane or carbon monoxide. Inert gases are generally known for slowing down the burning.

Presented results point to the high potential of using synthesis gases produced from municipal waste in stationary combustion engines as alternative sources of energy for cogeneration units. Their use will achieve a reduction in the environmental burden of emitted greenhouse gases (closed carbon cycle, or utilization of the waste) as well as a reduction in the amount of the landfilled municipal waste.

Acknowledgements

This publication was realized with support of Operational Program Integrated Infrastructure 2014 - 2020 of the project: Innovative Solutions for Propulsion, Power and Safety Components of Transport Vehicles, code ITMS 313011V334, co-financed by the European Regional Development Fund.

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