ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1138246

ENERGY ANALYSIS OF A SMALL CAPACITY SI ENGINE FUELED WITH LEAN AIR GAS MIXTURE

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Abstract

In this paper, the results of the theoretical study of an internal combustion engine, fuelled with lean air – gas mixtures, are presented. Energetic property calculations were done for several chosen gaseous fuels such as methane, landfill gas, and producer gas. Based on these fuels, the performance of a theoretical Seiliger-Sabathe cycle was investigated using variable air excess values. The accurate analysis of the various processes taking place in an internal combustion engine is a very complex problem. If these processes were to be analyzed experimentally, it would be more expensive than theoretical analysis. The Seiliger-Sabathe cycle turns out to be help in theoretical analysis of internal combustion engine performance. Dimensionless descriptive parameters (E, ψ) are very useful at this analysis by combining the properties of fuel with initial thermodynamic parameters of the cycle. Moreover, the experimental results of SI engine fuelled with a lean mixture of natural gas are presented for comparative purposes. The experiments were carried out on a petroleum engine with a low engine displacement. A typical SI engine was selected in order to evaluate the potential application of a gaseous fuel (i.e. natural gas). These types of engines are widely available and commonly used in the automotive sector because of low purchase prices and operating costs. It is expected that after minor modifications, the engine can easily operate in a low power co-generation mode. The main objective is to evaluate the performance of the engine under lean air/fuel mixture conditions. The slight impact of air excess ratio on COV_{IMEP} was noticed. The value decreases insensibly with air excess ratio decreasing. Obtained results are located at acceptable levels for power generation sources and are less than 5 %. Although, the more distinct impact was observed regarding to COV_{pmax}. The maximum value was noted for leaner mixture and it amounts to approximately 7.5%.

Keywords: *global warming, spark ignition (SI) engine, natural gas, indicated efficiency, lean mixture*

1. Introduction

Changes in the global climate are of great significance to civilization. Emissions of combustion gases are continuously increasing and contribute to the greenhouse effect. Limiting carbon dioxide and methane emissions is very important in alleviating environmental problems. One way to reduce greenhouse gas emissions could be through the utilization of alternative fuels such as landfill gas, producer gas and other renewable energy resources. That is, using resources, which do not emit greenhouse gases and contribute to major reduction of emissions (considering fuel sustainability and the carbon life cycle). Among all the types of renewable fuels, biomass is considered the most important due to the climate and geopolitical conditions of Poland [1-3].

When considering biomass-processing technologies, fermentation is one of the possible options for biogas production. A possible further alternative is the production of landfill gas using urban organic wastes, which in terms of composition is comparable to biogas. Both gases consist mainly of methane and carbon dioxide, with some traces of other gases. It should be highlighted that methane is a greenhouse gas that similarly to carbon dioxide contributes to global warming and climate change. It has been reported that methane contribution to global warming can be approximately 20 times greater compared to carbon dioxide when released directly to the atmosphere [4]. Therefore, utilization of gases, which contain methane for the purposes of power generation, will lead to a reduction of emissions released into the atmosphere (i.e. emissions from municipal solid waste) and provide potential benefits and additional revenue to the local government. Furthermore, the value of municipal solid waste is rising worldwide every year. Depending on the country of origin, the organic part of urban waste represents approximately 20 to 80% [5].

A second way of utilising biomass fuels such as agricultural and forestry residues, is to gasify them in downdraft fixed bed gasifiers [6, 7]. This type of device can generate a producer gas (low calorific value gas - LCV) that can be used in spark ignition engines for heat and power generation purposes. There are several ways to gasify solid fuels, and various gasifier designs can be employed. The main difference between various gasifiers is based on the flow direction between the oxidizer and the solid fuel. These features determine the physical and chemical properties of the LCV gas. The downdraft fixed bed gasifier has considerably higher technological requirements in terms of the fuel application (i.e. fuel characteristic) compared to an updraft one. The downdraft gasification method is mainly associated with the lower tars pollution. That is the reason of frequent utilization of downdraft fixed bed gasifier at combined heat and power (CHP) systems with low and medium output power [7-9]. Typically, a producer gas composed of: nitrogen, carbon dioxide, moisture, hydrogen, carbon dioxide, methane and tars, and the output temperature range are usually between 500°C and 800°C.

2. Characteristic of gaseous fuels for small CHP systems

2.1 The main properties of gaseous fuels

Gaseous fuels are attractive for use in internal combustion engine (ICE) because of their wide ignition limits and ability to form homogeneous mixtures. Moreover, gaseous fuels usually have high hydrogen to carbon ratio, and thus relatively low CO_2 emissions are possible when they are used in SI engines. Natural gas is readily available from deep underground natural rock formations, or as petroleum-based fuels, while biogas and producer gas can be obtained from renewable sources.

There are various significant parameters that can be used to characterize the usefulness of a fuel to powering the internal combustion engine. The first one is the resistance of a fuel to knocking. For gaseous fuels, the knocking tendency is described by the methane number (MN). Occurrence of knocking during combustion leads to increased emissions and decreased engine efficiency. In Fig. 1, the approximate values of methane number for chosen gaseous fuels are presented. All values concern stoichiometric mixtures.

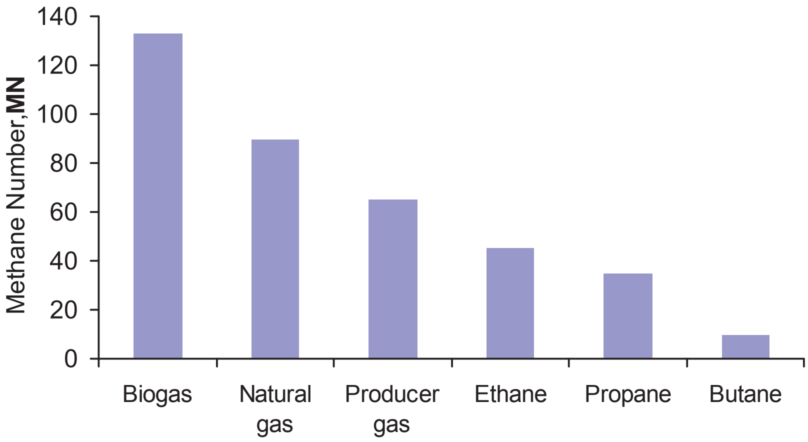

Fig. 1. Methane Number of chosen gaseous fuels

The methane number is defined as: The percentage by volume of methane blended with hydrogen that exactly corresponds to the knock intensity of the unknown gas mixture under specified operating conditions in a knock testing engine. The methane – carbon dioxide mixtures are used as reference mixtures for the range of MN above 100. For example, a blend of 40% hydrogen and 60% methane constituted a methane number (MN) of 60. To achieve the methane number higher than 100, e.g. 140 the reference fuel should consist of 40% carbon dioxide and 60% methane.

Results in Fig. 1 show that there are significant differences between MN values of fuels, which could be use in SI engine. A higher value of MN correlates to higher resistance to knocking. Fuel with high MN values can be combusted at high compression ratio SI engine. Theoretically, a higher compression ratio leads to increased engine thermal efficiency. In this case, the engine fuelled with biogas could be operated with higher efficiency than with e.g. a producer gas. It is however noteworthy that high MN value alone, is not enough to ensure correct engine work. Different fuels need different amount of air to create expected air-gas mixtures, which can be burned in an engine. Thus, the calorific value ($e_{d,v}$) of an air-gas mixture, the flammability limits, and the flame speed, are the next important parameters to, consider. Properties of several gaseous fuels are presented in Tab. 1.

Tab. 1. Fuel properties [10-13]

Gas	Formula (Composition)	**LHV** *MJ/Nm³*	**$e_{d,v}$** *MJ/dm³*	Flammability Limits λ (ϕ) Lower	Higher	Laminar Flame speed (ϕ = 1) *cm/s*	**AFR**	**V_{amin}** *Nm³/Nm³*
Methane	CH_4	35.8	3.18	2 (0.5)	0.6 (1.68)	35	17.2	9.52
Biogas	$CH_4 \approx 60\%$ $CO_2 \approx 39\%$ other: CO, H_2, H_2S	21.5	3	2.16 (0.46)	0.83 (1.21)	25	6.11	5.72
Natural gas GZ 50	$CH_4 \approx 98.5\%$, $CO_2 \approx 0.1\%$, $N2 \approx 1\%$ other: C_2H_6, C_3H_8, C_4H10_6,	35.3	3.15	2.03 (0.49)	0.6 (1.66)	34	16.9	9.4
Producer gas	$CO \approx 24\%$, $H_2 \approx 23\%$, $CO_2 \approx 9\%$, $CH_4 \approx 2\%$, $N_2 \approx 42\%$	6.2	2.58	2.17 (0.46)	0.6 (1.67)	50 ±5	1.63	1.31
Carbon monoxide	CO	12.6	3.54	2.94 (0.34)	74.2 (6.85)	45	2.45	2.38
Hydrogen	H_2	10.8	3.03	10 (0.1)	0.15 (6.85)	270	34.33	2.38
Propane	C_3H_8	91	3.42	2.06 (0.49)	0.4 (2.5)	40	15.6	23.81
Butane	C_4H_{10}	118.6	3.44	1.7 (0.57)	0.33 (3.06)	115	15.4	30.95

The calorific value of air-gas mixture was calculated using following equation:

$$e_{d,v} = \frac{p_1}{(MR)T_1} \frac{LHV}{\left[\dfrac{1}{M_f} + \lambda n'_{a,\min}(1 + X_{za} + \delta_{ex})\right]}, \tag{1}$$

where:

p_1, *Pa*; T_1, *K* – in-cylinder thermodynamic parameters after filling process,
(MR) = 8.3145 kJ/kmol K; LHV, $^{J}/_{kg}$ – Lower heating value,
M_f, $^{kg}/_{kmol}$ – fuel atomic weight,
λ – air excess ratio,
X_{za}, $kmol_{H2O}/kmol_a$ – molar humidity ratio.

The ($e_{d,v}$) results presented in Tab. 1 were computed for accepted data according to:

- air-gas mixture temperature $T_1 = 273.15$ K and pressure $p_1 = 101.325$ kPa,
- relative air humidity, $\varphi = 45\%$ (to determine molar humidity ratio),
- LHV – as a value from Tab. 1 (row 3),
- the residual exhaust gases fraction $\delta_{ex} = 5\%$ (as a mass fraction in fresh air).
- the air excess ratio $\lambda = 1$.

The influence of the air excess ratio on the calorific value of air-gas mixture for several gaseous fuels is presented in Fig. 2.

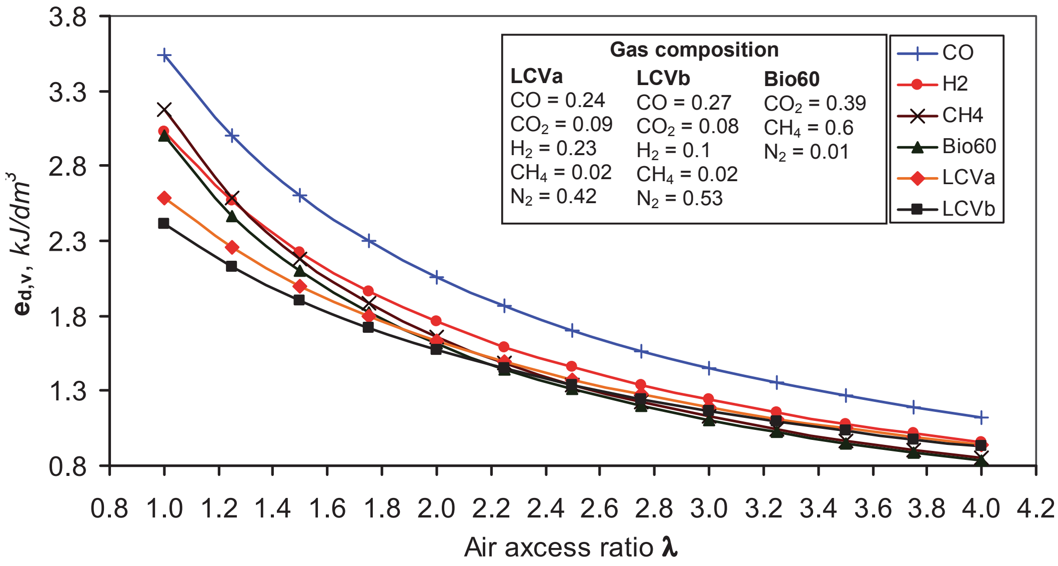

Fig. 2. Calorific value of air-gas mixture for chosen gaseous fuels

Figure 2 shows that for most fuels at the range of stoichiometric mixtures (i.e. $\lambda = 1$) the differences between the values of ($e_{d,v}$) are the highest. The value of ($e_{d,v}$) obtained for fuels such as LCVa, LCVb, Bio60 and CH4 is almost the same for air excess ratio of 2.35. Carbon monoxide had the highest values of ($e_{d,v}$) for all the chosen values of the air excess ratio.

It is noteworthy that the calorific value of an air gas mixture is directly related to engine performance. The higher the value of ($e_{d,v}$) the higher the indicated mean effective pressure (IMEP) of the engine can be obtained. Therefore, the best energetic properties in terms of engine fuel, achieved by butane. The values of lower flammability limits (LFL) determine the proper engine work with lean air-gas mixture. The hydrogen has the higher flammability range comparing to other fuels presented in Tab. 1. Therefore, the presence of hydrogen in gas composition should be beneficial due to lean mixture burning. Moreover, the hydrogen has many other desirable properties that can improve the performance and exhaust emissions of the gas engine [14]. Flame velocity of hydrogen is much higher than other fuels (see Tab. 2); therefore, the flame velocity of the combustible mixture will be substantially increased by the addition of hydrogen to e.g. natural gas. Thereby the combustion process will occur with less heat losses. In addition, the quenching gap of hydrogen is approximately 3 times less than natural gas. Hereby, the hydrogen addition will promote complete combustion [14].

2.2 The lean mixtures combustion

Combustion of lean air-gas mixtures has been proved to be an effective way for reducing toxic emissions and improving thermal efficiency of SI engines [15]. It is worth noting that combustion of lean mixtures can provide a substantial increase in energy efficiency of the internal combustion engine. The following key factors listed below affect the energy efficiency when using lean gas-fuel mixtures:

– lower pumping losses at a given engine speed is termed part load operation,
– reduced wall heat loss of the combustion chamber since the overall gas temperature is considerably lower compared to that of stoichiometric mixtures,
– reduced dissociation of the high temperature combustion gases that more chemical energy of fuel is released when the piston is close to the top dead centre (TDC) position during expansion phase.

The lean mixture combustion is burdened with the difficulties as well. It can be less stable comparing to stoichiometric mixture combustion. Furthermore, one of the problematic issues with respect to lean mixtures combustion is their difficulty to ignite and slower flame propagation. Under certain conditions the lean mixture may have a greater tendency to incomplete combustion. The interrelation of these factors leads to increased variations of the pressure profile in the successive cycles of the engine operation.

3. Energy analysis

3.1 Theoretical calculations

Theoretical analysis of gaseous fuels usability to power the internal combustion engines can base on Seiliger-Sabathe (S-S) cycle. In this thermodynamic cycle the heat is supplied in two stages, during isochoric and isobaric process. The p-v and T-s diagrams of S-S cycle are presented in Fig. 3. All calculations were done assuming 0^{o}C and 101.325 kPa as values at the first point of the cycle. The temperature after compression was computed using adiabatic equation. At this stage of cycle the specific heat ratio was agreed as a value calculated for mean temperature during compression.

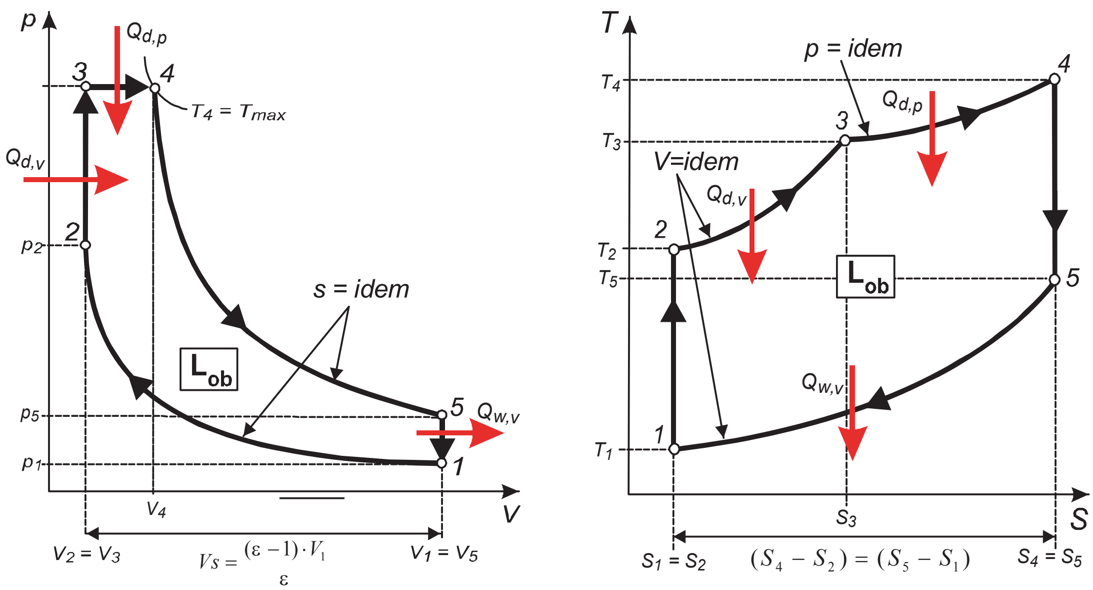

Fig. 3. The Seiliger-Sabathe cycle

The S-S cycle can be characterized by several main parameters. The first one is the heat distribution number (ψ) is defined as a ratio of heat supplied under isochoric process to the whole heat supplied in the thermodynamic cycle, and can be described:

$$\psi = \frac{Q_{d,v}}{Q_{d,v}+Q_{d,p}}, \tag{2}$$

where:
$Q_{d,v}$, J – amount of heat supplied under isochoric process,
$Q_{d,p}$, J – amount of heat supplied under isobaric process.

The second one is the energy -stoichiometric parameter (E) which combines calorific value of fuel and air excess ratio during combustion process with thermodynamic parameters at the first point of the theoretical cycle [16]. The (E) parameter can be calculated by following equation:

$$E = \frac{LHV}{\left(1 + \lambda\, n'_{a,\min}\, M_a\, (1 + \delta_{ex})\right) R\, T_1} \;; \quad E \leq E_{max} \approx 34\,, \tag{3}$$

where:
LHV, *kJ/kg* – lower heating value of analyzed fuel,
λ – air excess ratio,
$n'_{a,min}$ *kmol$_a$/kmol$_f$* – minimum amount of air required to combustion the fuel,
M_a, *kg/kmol* – air atomic weight,
δ_{ex} , *kg$_{ex}$/kg$_{fair}$* – the residual exhaust gases fraction (mass exhaust gases in fresh air);
R, *kJ/kgK* – individual gas constant.

The Seiliger-Sabathe cycle efficiency is obtained using the first law of thermodynamics. Relationships between S-S cycle parameters can bring various equation form of cycle efficiency. In the work the equation based on E, ψ parameters was used and can be described as follow [16]:

$$\eta_o = 1 - \frac{1}{(\kappa - 1)E}\left[\left(1 + \frac{(\kappa-1)E\,\Psi}{\varepsilon^{(\kappa-1)}}\right)\left(1 + \frac{(\kappa-1)E(1-\Psi)}{\kappa\left((\kappa-1)E\,\Psi + \varepsilon^{(\kappa-1)}\right)}\right)^{\kappa} - 1\right], \tag{4}$$

where:
ε – compression ratio,
κ – specific heat ratio.

Presented form of equations (3) and (4) let investigate an impact of fuel properties, mixture stoichiometry (value of air excess ratio) and thermodynamic parameters for value of thermal efficiency. The influence of air excess ratio on Seiliger-Sabathe cycle efficiency for chosen gaseous fuels is presented in Fig. 4. As a working medium of S-S cycle during compression process (1-2, see Fig. 3) an air was analyzed. Calculated gas composition arisen after combustion considered fuels was used as a working medium in subsequent parts of the cycle (processes 2-5, see Fig. 3).

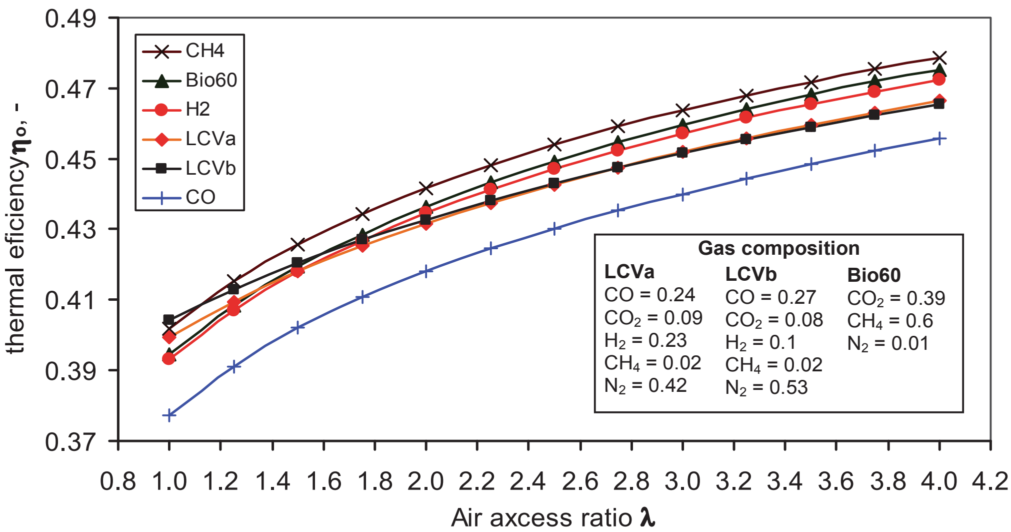

Fig. 4. Seiliger-Sabathe thermal efficiency as a function of air excess ratio

In this work the Seiliger-Sabathe cycle is considered as an equivalent air cycle with the changing composition of working fluid and variable value of specific heat ratio. The value of (E) parameter included in equation (4) was calculated for accepted data as for ($e_{d,v}$) (see point 2.1). Specific heat ratio (κ) was computed using JANAF thermodynamic tables as a function of temperature. The temperature was determined as an average value during the heat transfer to the working medium. Compression ratio (ε) value is 9.3 for all calculation data. All calculation were done for compression ratio $\varepsilon = 9.3$ and heat distribution number $\psi = 0.8$.

The analysis of obtained data (Fig. 4) shows that the change in air excess ratio has a clear influence on the Seiliger-Sabathe cycle efficiency value. The efficiency increase towards leaner air-gas mixtures. This is caused by increasing of specific heat ratio (κ) values for lower temperatures. The difference of efficiency values between analyzed air-fuel mixtures result from various exhaust gas compositions.

As it was mentioned the indicated mean effective pressure (IMEP) is one of the more important parameters regarding to engine performance. The work of theoretical cycle can be described as a relative value e.g. refers to the cycle displacement volume (V_s, see Fig. 3). The relative theoretical cycle work RCW is a similar parameter to the IMEP of real cycle. Considering S-S cycle regarding to dimensionless parameters (E, ψ), the relative cycle work can be described by following equation:

$$\left(\frac{L_0}{V_s}\right) = p_1 \frac{\varepsilon}{\varepsilon-1}\left\{E - \frac{1}{(\kappa-1)}\left[\left(1+\frac{(\kappa-1)\cdot E\cdot\psi}{\varepsilon^{\kappa-1}}\right)\cdot\left(1+\frac{(\kappa-1)\cdot(1-\psi)\cdot E}{\kappa\cdot\left[(\kappa-1)\cdot E\cdot\psi+\varepsilon^{\kappa-1}\right]}\right)^{\kappa}-1\right]\right\}, \quad (5)$$

where:

L_o, J – work of theoretical cycle,

V_s, m^3 – displacement volume.

Values of RCW can be computed to the pressure unit (bar) what is usually used for IMEP calculations. The impact of relative cycle work (L_o/V_s) on thermal efficiency of S-S cycle is presented in Fig. 5.

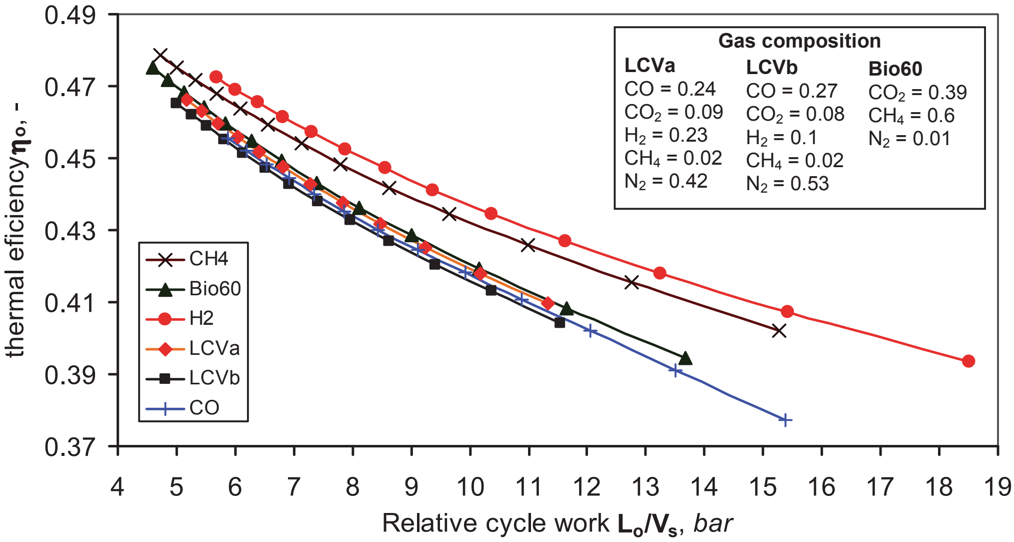

Fig. 5. The impact of relative cycle work L_o/V_s on the thermal efficiency of S-S cycle

From Fig. 5 it is seen that the thermal efficiency of the S-S cycle is clearly decreasing the increasing values of relative cycle work. It is because the leaner mixture is combusted the lower value of RCW is obtained. According to Fig. 4, the efficiency increase towards higher values of air excess ratio what means for the leaner mixtures.

3.2. SI engine performance

The experimental investigation of the natural gas combustion in SI engine was carried out using naturally aspirated engine. Three cylinders SI engine with the capacity of 796 cm^3 and compression ratio equal to 9.3. The engine was originally powered by petrol. For the purpose of experiment and possibility of gaseous fuel application, control and power supply system of the engine have been modified.

During experiment the engine was fuelled with natural gas coming from the gas grid. The experiments were performed at constant rotational speed of $r_o = 1500$ rpm. The study was conducted for the full engine load (i.e. wide open air throttle). The engine operation control factors represent an ignition advance angle and composition of the gas-air mixture (value of air excess ratio). In view of the fact that the engine combustion chamber is not subject to any modifications in the first place the maximum value of excess air ratio at which the operation of the internal combustion engine is stable and there is no "misfire" detected, was checked.

The pressure measurements in the first cylinder were performed using piezoelectric pressure transducer. This type of transducer through so called charge amplifier generates an analogue (voltage) pressure signal which is sampled then at a sufficient frequency by the data acquisition system. In addition, the absolute pressure within the intake manifold was recorded with piezoresistive absolute pressure transducer. At both measurement ducts the pressure signal was sampled at predefined crank angle using encoder. Measurements were carried out with a resolution of 1024 measurement points per revolution of the crankshaft. The encoder was also equipped with a position marker device for indicating the position of a piston in a cylinder. Each sets of measurement consisted of 100 consecutive engine cycles.

Figure 6 shows the impact of air excess ratio λ on the indicated efficiency and IMEP when engine is fuelled with natural gas GZ50. The engine was operated on optimal spark timing to get maximum brake torque (MBT). From the results, it can be concluded that the value of the engine indicated efficiency increased toward the higher values of air excess ratio, while the IMEP shape is opposite. Lower value of IMEP for higher air excess ratio depends primarily on the air-gas mixture calorific value ($e_{d,v}$, see Fig. 2). It is worth noting that both presented parameters fairly well agree quantitative and qualitative with theoretical data.

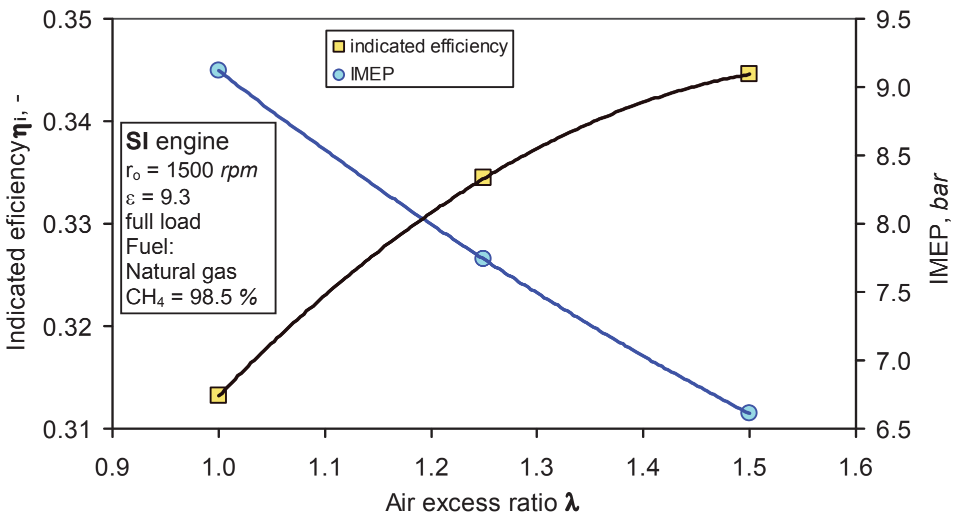

Fig. 6. The impact of air excess ratio on Indicated efficiency and IMEP of SI engine

One significant problem associated with lean mixture is that the combustion process is likely to be less stable cycle by cycle. The charge cylinder filling process, initiation of ignition and air-fuel mixture homogeneity has a significant impact on engine's working cycle shape. One of the most important reasons, which lead to high cycle to cycle variations (CCV), is misfire of the air-fuel mixture. The misfire directly affects to performance of the engine, particularly: mean effective pressure, efficiency and pollutants. The limited repeatability of engine's working cycle shape exist even when the ignition process occur correct.

There are two characteristic coefficients that characterize cycle to cycle variations. The first one is a coefficient of variation the in-cylinder maximum pressure (COV_{pmax}). While, the second coefficient is based on the indicated mean effective pressure (COV_{IMEP}). The influence of air excess ratio on both mentioned coefficients are presented in Fig. 7.

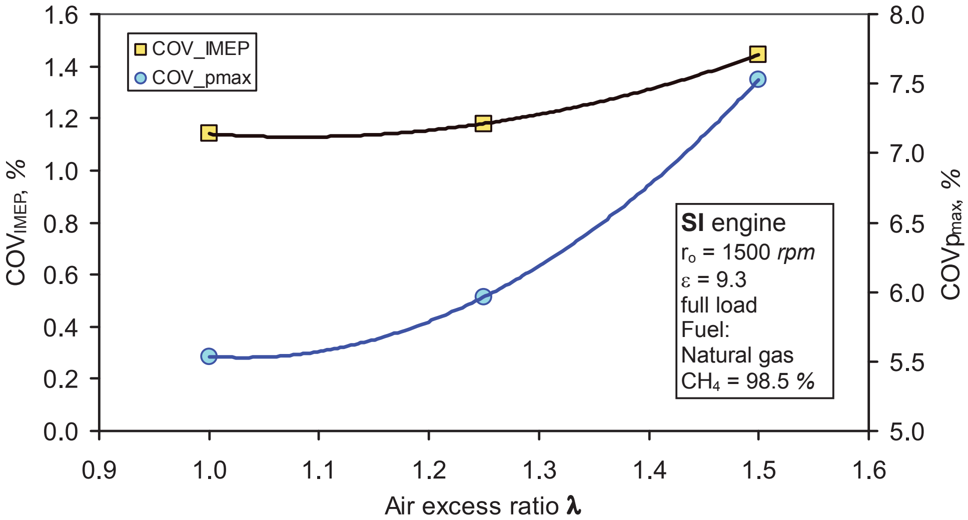

Fig. 7. The impact of air excess ratio on COV_{IMEP} and $COVp_{max}$ of SI engine

The coefficient of variation in the indicated mean effective pressure seems to be relevant to characterize repeatability of engine working cycles shape. Its high values lead to fluctuations of engine speed and power. For proper work of engine with power generator the coefficient should not exceed 5%. It can be seen in the Fig. 7 that for used air excess ratio range the value of COV_{IMEP} meet this condition. The COV_{pmax} values are significantly higher than values obtained for COV_{IMEP}. Decreased values of COV_{pmax} could be achieved after optimization the air-gas mixing process and ignition system. The lower value of COV_{pmax} should be achieved for the more complete combustion of the charge.

4. Conclusions

There are many important parameters characterized usefulness of fuel to powering the internal combustion engines. The engine performance depends on fuels properties, such as: methane number, calorific value of air-fuel mixture, laminar flame speed and flammability limits. All listed fuels properties were studied for several gaseous fuels by literature review and partially by author's calculations. The methane number value is indirect limiting the maximum engine efficiency exploitation due to the knock occurrence possibility. Calorific value of the air-gas mixture directly influences on the IMEP of the engine. Laminar flame speed is a parameter that affecting the combustion and determines the rate of energy released during the process. It also has a significant effect on the performance and pollutant emissions of internal combustion engines. The higher flammability limit value (LFL) limits an engine operating range for lean mixtures.

The accurate analysis of the various processes taking place in an internal combustion engine is a very complex problem. If these processes were to be analyzed experimentally, it would be more expensive than theoretical analysis. The Seiliger-Sabathe cycle turns out to be help in theoretical analysis of internal combustion engine performance. Dimensionless descriptive parameters (E, ψ) are very useful at this analysis by combining the properties of fuel with initial thermodynamic parameters of the cycle.

The slight impact of air excess ratio on COV_{IMEP} was noticed. The value decreases insensibly with air excess ratio decreasing. Although, the more distinct impact was observed regarding to COV_{pmax}. Higher variations of engine working cycles fuelled with leaner air-gas mixture are mainly caused by lower presence of combustible components probability near spark plug electrodes (higher volume fractions of N_2 - inert gas). The correct air-gas mixture preparation plays very important role in this case. The values are obtained at acceptable levels for power generation sources and are less than 5%.

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