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HYDROGEN RICH GASES COMBUSTION IN THE IC ENGINE

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Abstract

Experimental results of combusting three different syngases in an internal combustion (IC) spark ignition engine are presented in this paper. The syngases used for tests varied each from the other with hydrogen content, which was of 10, 15 and 60%. Other combustible gases as CO and CH_4 were also changed. Thus, the lower heating value of the syngases was of 2.7, 4.6 and 17.2 MJ/nm^3, respectively. Combustion tests were performed at stoichiometric ratio of syngas-air mixture, with variable spark timing and constant compression ratio of 10. On the basis of in-cylinder combustion pressure histories the indicated mean effective pressure (IMEP) was computed and presented versus spark timing and vs location of the middle combustion phase expressed by the 50% of mass fraction burned (MFB). Additionally, the 0-10% MFB and 10-90% MFB were also determined. Furthermore, the paper contains theoretical determination of the three fuel quantities, which can affect combustion duration and heat release rate during burning the syngases in the IC engine. They are as follows: laminar flame speed, ignition delay and adiabatic flame temperature. Final results does not show satisfactory correlation between LFS computed at NTP and real combustion phasing. Furthermore, both long combustion duration and long 0-10% MFB leading to unstable combustion were observed for the syngas with the lowest LHV of 2.7 MJ/nm^3.

Keywords: *syngas, coke gas, hydrogen, combustion engine*

1. Introduction

The biomass based gases can be managed as renewable, what makes them attractive fuels for energetic utilization. Several works have been conducted in the field of combusting these gaseous fuels obtained from gasification of biomass, waste wood, energetic crops etc. Research has been focused on two issues: gasification and combustion. As regards combustion technologies, an IC engine can be considered as the most efficient machine for generating mechanical work easily converted into electric power in a power generator. Producer gases obtained from gasification of biomass with low organic content are characterized with low calorific value. It leads to several problems with their proper combustion in the engine. Thus, as engine fuel for the spark ignited engine, low Btu gases should be enriched with natural gas to avoid misfires and provide good stability of engine work. This problem can be also overcome through burning low Btu gases in a compression ignition engine with a diesel pilot. Among the others, E. Tomita et al. carried out investigation on syngas combustion in a compression ignition engine [1], works on natural gas combustion in the engine have been conducted by Kowalewicz et al. [2].

As known, the Lower Heating Value (LHV) of the syngas depends on combustibles content. A gas generated during gasification mainly contains hydrogen and carbon monoxide as combustible gases. There is also methane presented in a producer gas, but its content usually does not exceed 5% by volume in a gas generated without steam reforming. Other major components of a producer gas are non-flammable gases as nitrogen and carbon dioxide. Thus, from this point of view, it is desirable to keep high hydrogen content in a producer gas. Such a producer gas, called coke gas, with relatively high hydrogen content can be obtained from coal gasification. Research in this field is conducted by Postrzednik [3] and others. Each research work on syngas combustion in the IC engine usually concerns syngas of specific chemical composition. Investigation, involved in this paper, presents results of combustion of three different syngases in the IC SI engine.

2. Experimental set-up

Three syngases of different combustible and non-combustible content were used for tests. Properties and chemical composition of these syngases are presented in the Tab. 1.

Syngas 1 - was managed as a wood-gas obtained from properly conducted gasification process.

Syngas 2 - is a typical producer gas obtained from gasification conducted under non-optimal conditions.

Syngas 3 - is similar to a coke gas, which is by-product of coal gasification during coke prodution.

Syngas 1 was tentatively prepared and stored as pressurized in a cylinder. Then it was injected to the CFR engine intake port with a gaseous injector.

Syngases 2 and 3 were prepared on the basis of 5 standard gases N_2, H_2, CO_2, CO and CH_4. The gases were stirred at required ratio at atmospheric pressure in a mixer and then the modeled syngas was delivered to the intake port of the engine.

Tab. 1. Syngases properties

Fuel		H_2	CO	CH_4	CO_2	N_2	LHV_{Vol}	A/F_{stoic}	LHV_{mix}
		%	%	%	%	%	MJ/nm^3	nm^3/nm^3	MJ/nm^3
Syngas 1	Vol %	15	25	5	5	50	4.58	1.43	1.89
	Energy %	35.0	26.8	38.2	-	-			
Syngas 2	Vol %	10	12	3	15	60	2.71	0.81	1.50
	Energy %	39.5	21.7	38.8	-	-			
Syngas 3	Vol %	60	5	30	5	-	17.16	4.40	3.18
	Energy %	37.4	1.4	61.2	-	-			

LHV_{Vol} - Lower Heating Value of the fuel by volume,
LHV_{mix} - Lower Heating Value of the combustible mixture of fuel with air.

As seen in the Tab. 1 the LHV_{mix} of air and the poorest syngas 2 at stoichiometric ratio is lower only by 20% when compared with the LHV_{mix} of the syngas 1, however, these fuels differ much more from each other in their LHV and incombustible ballast (N_2, CO_2).

The syngases were burned in the IC engines at parameters as depicted in the Tab. 2.

Tab. 2. Tests conditions

Fuel	Engine	Spark Timing deg ATDC	Compression ratio -	Lambda -	IMEP kPa	Temperature ºC
Syngas 1	CFR, 611ccm Waukesha	var (–5... –25) @ 900rpm	10	1	approx.550	85
Syngas 2	Deutz FL511 825ccm	var (–20... –35) @ 1500rpm	10	1	approx.400	85
Syngas 3	Deutz FL511 825ccm	var (–2... –25) @ 1500rpm	10	1	approx.500	110

Both the engines were adopted to burn several types of gaseous fuels. The test benches for the each engine were described in details in previously published papers of KONES [5, 6].

3. Results and discussion

The results are split into sections. In the 1st section results from computing particular fuel properties are presented. The 2nd section describes experimental results of syngas combustion.

LFS, Ignition Delay and Adiabatic Temperature modelling

At the first phase of combustion, expressed by 0-10%MFB (Mass Fraction Burned) both the flame propagation speed and the ignition delay of the air-fuel mixture play crucial role and affect combustion start and flame kernel development. It is difficult to determine real flame speed at turbulent environment in the engine cylinder, but might it be possible to apply these quantities as indicators to show the combustion kernel growth, even though the LFS is determined at normal pressure and temperature conditions (NTP). Both the LFS and ignition delay for the three syngases were determined with Chemkin [7] and depicted in the Fig. 1a and 2.

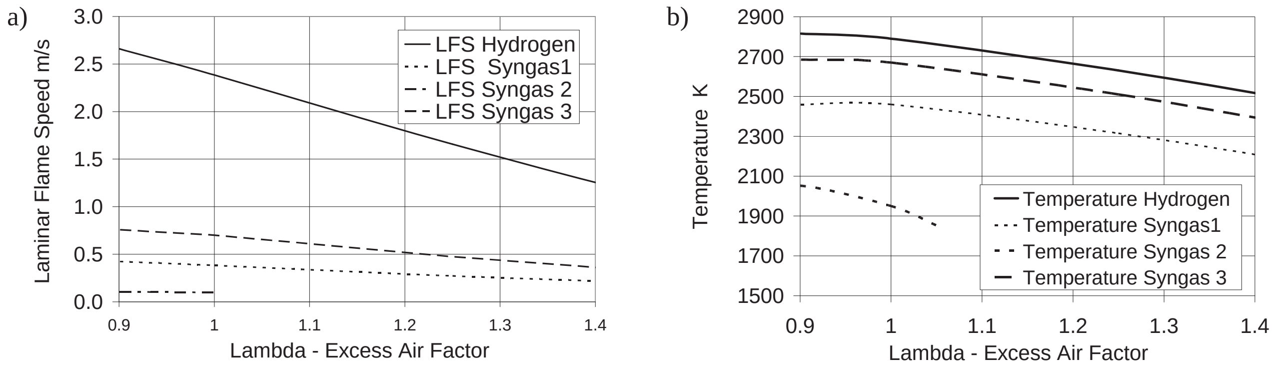

Fig. 1. Laminar flame speed (LFS) (a) and adiabatic temperature (b) vs excess air lambda for various mixtures of gaseous fuels and air at NTP conditions 25°C, 101,325 Pa

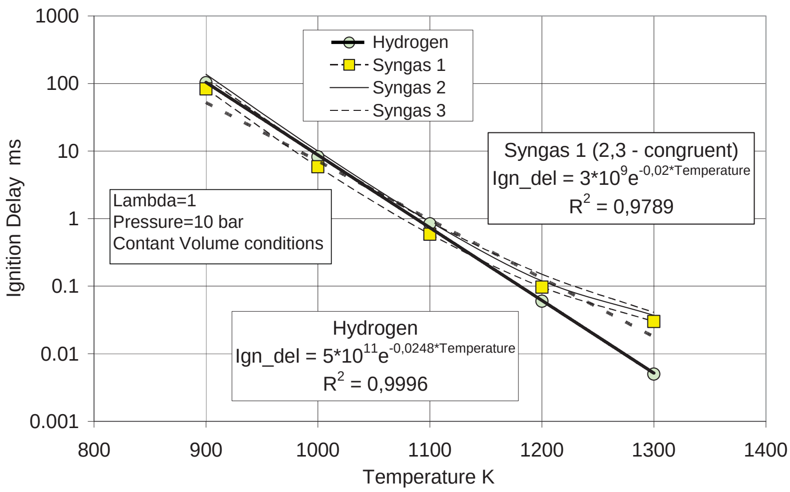

Fig. 2. Ignition delay vs temperature for hydrogen and various syngases at stoichiometric ratio with air

Additionally, adiabatic temperature of the combustible mixtures at constant-volume conditions were computed and presented as function of excess air lambda in the Fig. 1b. On the basis of this temperature one can conclude about real combustion peak temperature of the engine work cycle.

As plotted, the LFS decreases with lambda increase. However, the LFS for syngas 2 does not vary significantly with lambda change. Furthermore, the LFS decreases with H_2 content decrease for these syngases. The fig.1 also shows the LFS and the adiabatic flame temperature for hydrogen as the reference curves and makes comparison between these fuels. One can notice that hydrogen-air at lambda of 1.8-2.0 has the same LFS as the mentioned syngases at stoichiometric ratio. So, it can be concluded that hydrogen-air of lambda 2.0 might be burnt at the same rate as e.g. the syngas 1 at lambda = 1.

As far as the ignition delay is concerned, there is not significant difference between the syngases-air mixtures. The crucial difference of almost one order is noticed when compare the syngas ignition delay plots and hydrogen ignition delay at temperature over 1200K. Below that temperature all the plots are almost superimposed on each other in the log scale. In the spark ignited engine the ignition delay might not significantly affect the 0-10%MFB unless the ignition is initiated as self-ignition above the auto-ignition temperature of the combustible mixture.

Experimental results of syngas combustion

As mentioned earlier, tests were conducted on two test-beds with two different engines. As the in-cylinder combustion pressure is the origin data from all the engine tests, they are presented in the Fig. 3 with variable spark timing for these 3 syngases. Each the pressure plot depicted in the fig.3 is a mean trace of 100 consecutive pressure traces. The thick solid line is in the each Fig. 3.a), 3.b) and 3.c) emphasizes the optimal pressure trace for the maximal indicated mean effective pressure (IMEP).

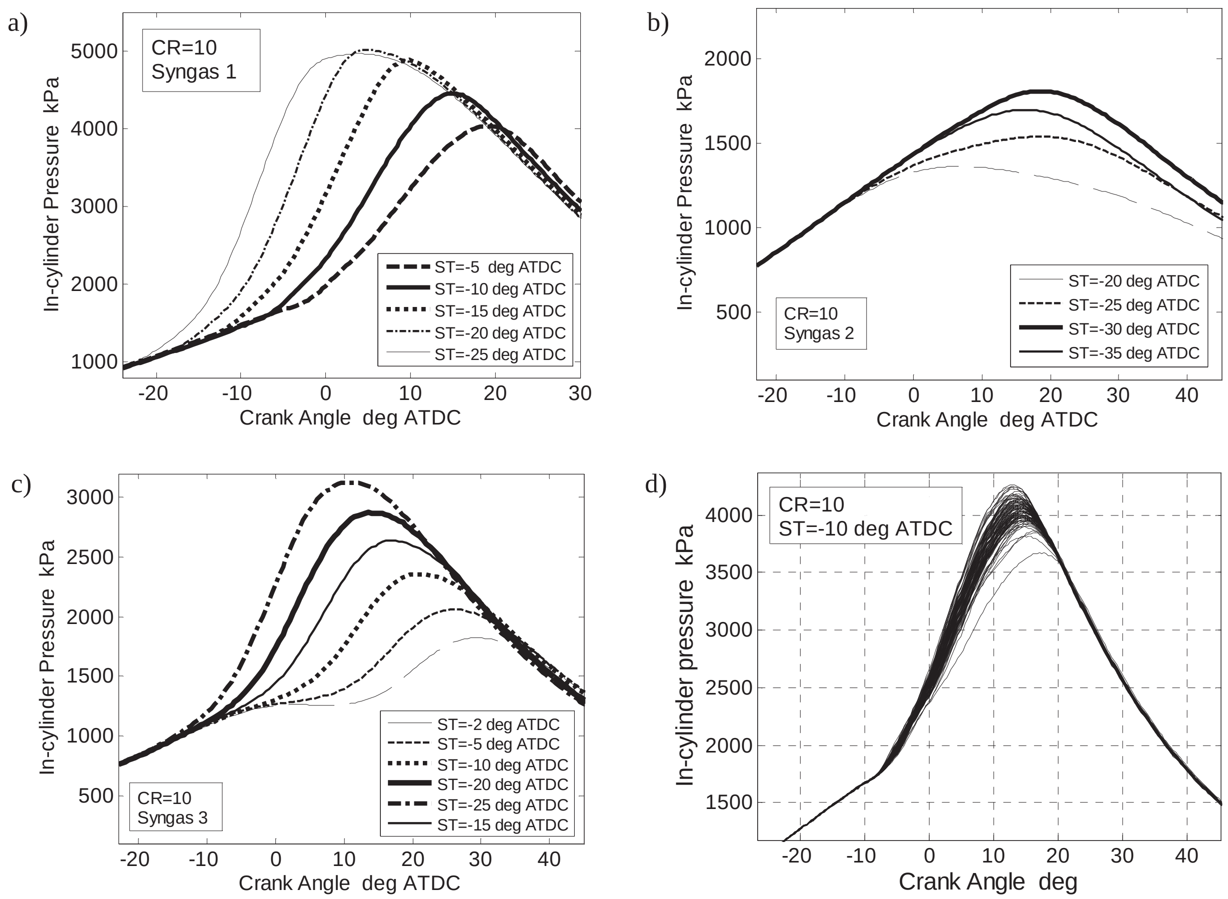

Fig. 3. In-cylinder combustion pressure histories for: a) syngas 1, b) syngas 2), c) syngas 3; at various spark timings, d) Set of 100 in-cylinder pressure traces for syngas 1

Figure 3d shows 100 consecutive combustion events of the syngas 1. They are useful to compute combustion stability, which can be expressed by the COV_{IMEP}. The determined COV_{IMEP} was of 0.22% for the exemplary plot of syngas 1 combustion presented in the Fig. 3d. COV_{IMEP} for other tests does not exceed 3% except the test of syngas 2 combusted at very advanced spark timing over 40 deg. The COV_{IMEP} for that test series was over 10%. This area was marked as unstable combustion in the Fig. 10.

The Fig. 4 shows IMEP as function of spark timing for the tests presented in the Fig. 3. As depicted, the maximum of IMEP goes with spark timing of approximately –10 crank angle (CA) degrees after top dead centre (ATDC) for syngas combustion. With applying the syngas 2 and 3 the engine produces maximal IMEP at spark timing of –30 deg ATDC and –20 deg ATDC, respectively.

The Fig. 5 shows normalized mass fraction burned (MFB) profiles for combustion of the syngas 1 at various spark timings as presented in the Fig. 3a. Combustion phasing can be easily determined from the MFB profiles [8]. As mentioned earlier, the initial phase 0-10% MFB is responsible for flame development and it strictly affect the further combustion phases. The second phase, defined as 10-90% MFB, is usually considered as the main combustion.

Next, the rate of MFB as MFB derivative d(MFB)/d(CA) over crank angle was computed. It was presented in the Fig. 6. The MFB rate looks similarly to the rate of heat release as the MFB profile matches the heat released during combustion. The only difference is that both the MFB and the rate of MFB are dimensionless quantities, thus makes it possible to compare each with the other profiles. The maximum of MFB rate for the each series of combustion increases with advancing spark timing. Higher rate of MFB, coming from faster combustion, makes the 10-90% MFB shortened. Additionally, acceleration of combustion can be caused by knock onset, which rapidly speeds up combustion process. Following Karim [9] hydrogen based gases are willing to generate combustion knock. In these tests the combustion knock was not detected, however, combustion at spark timing of –25deg ATDC was on the knock border for syngases 1 and 3 in the engine working at compression ratio (CR) of 10. Additional combustion test for the same syngas 1 was carried out at CR = 12 and the spark timing of –10 deg ATDC. The MFB rates of these two combustion events are plotted in the Fig. 7. As seen, the maximal MFB rate for the combustion test at CR = 12 is approximately twice higher than the MFB rate for the test at CR = 10. It provides credible symptoms for knock onset in the combustion at CR = 12. It can be also noticed that the location of the maximal MFB rate for knocking combustion events is shifted to the end combustion phase as is typical for gasoline combustion.

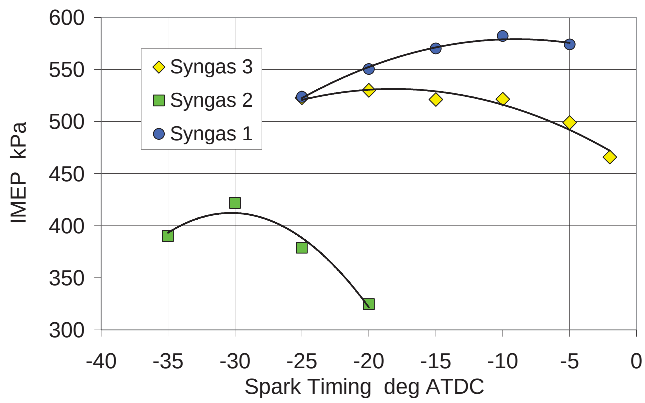

Fig. 4. IMEP vs spark timing for combustion tests of syngases 1, 2 and 3

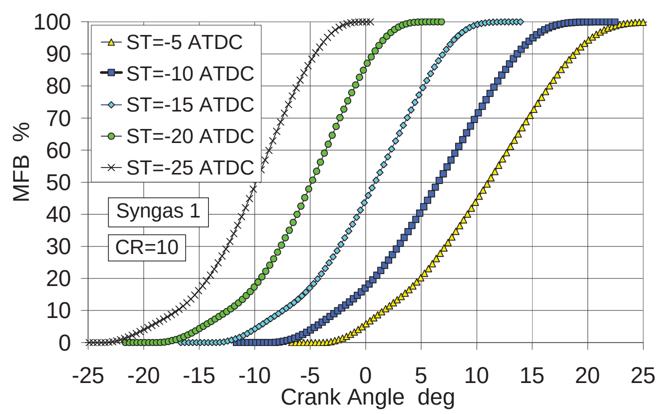

Fig. 5. MFB profile of combusting syngas 1 with various spark timings

As observed, the most advanced spark timing is for the mixture of air and the syngas 2. This fuel contains the minimal content of combustibles (CO, H_2, CH_4). Altogether, CO, H_2 and CH_4 occupy only 25% by volume of the entire fuel, unlike syngas 1 and syngas 3 which contain combustible substance of 45% and 95% respectively. Thus, this dilution effect by the non-flammable fuel content combined together with hydrogen energy fraction of the fuel can influence on the initial combustion phase represented by the 0-10%MFB. The longest 0-10%MFB is for the syngas 2, which is the worst

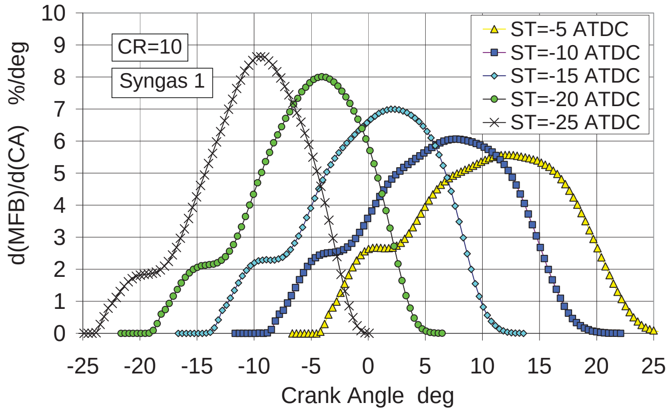

Fig. 6. MFB rate of combusting syngas 1 with various spark timings

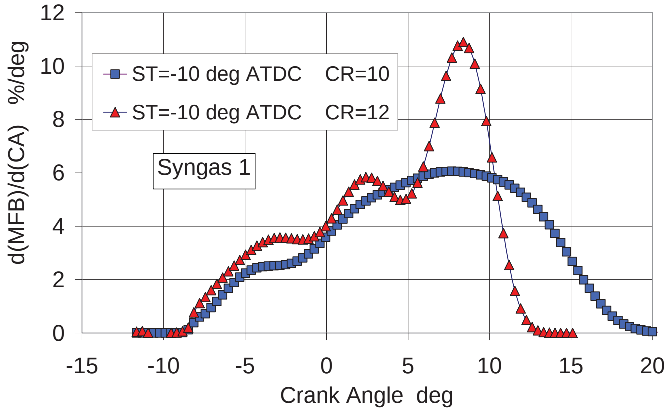

Fig. 7. MFB rate of syngas 1 combustion at compression ratio of 10 and 12

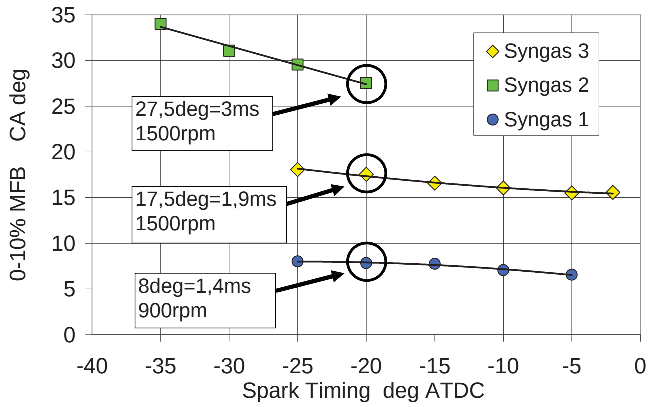

Fig. 8. 0-10% MFB vs spark timing for syngases 1, 2 and 3

syngas as regards its LHV. The shortest 0-10%MFB is for the syngas 1, even though, the syngas 3 has the highest LHV and its combustible gases are less diluted, but makes it difficult to compare these results due to syngas 1 was combusted in another engine than syngases 2 and 3. Similar correlation as for 0-10%MFB is presented for 10-90%MFB drawn in the Fig. 9.

Finally the Fig. 10 shows IMEP against location of 50%MFB commonly called as CA50. As anyone can notice, the maximum of IMEP is generated when middle combustion expressed by the CA50 is located in the range between 6 and 9 deg ATDC. Only the syngas 2 is combusted outside these optimal limits. To force the CA50 be located in the range of 6-9 deg ATDC, the syngas 2 should be combusted at spark timing advanced to 40 or more deg before TDC. As presented in the Fig. 8 advancing spark timing leads to increase in 0-10%MFB due to lower temperature of preignition phase and combustion becomes less stable. In these tests made it difficult to keep steady conditions for combustion of syngas 2 to obtain its CA50 at this optimal range with respect to maximum of IMEP.

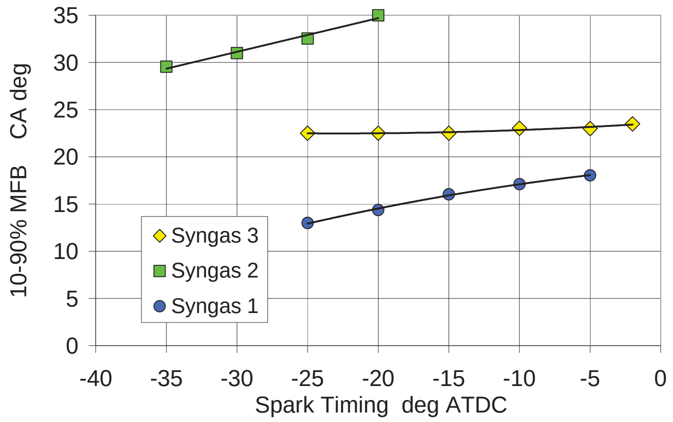

Fig. 9. 10-90% MFB vs spark timing for syngases 1, 2 and 3

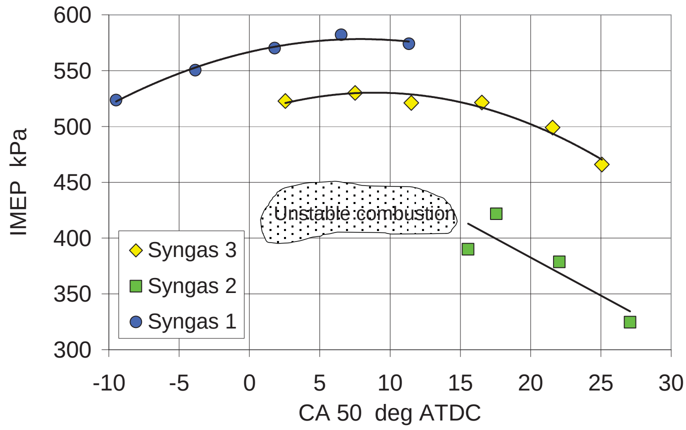

Fig. 10. IMEP vs CA50 for syngases 1, 2 and 3

4. Conlusions

1. Combustion stability of syngases 1 and 3 is satisfactory good, thus, from this standpoint it provides good conditions for the engine working as a drive for a power generator for electricity production. Unstable combustion was observed for the syngas 2 when the spark timing was advanced to 40 deg BTDC to obtain maximal IMEP of the engine.
2. High percentage of incombustible gases in the syngas (eg. syngas 2) leads to lengthen the 0-10% MFB what results in advancing the spark timing to 40-50 deg before TDC for syngas 2.
3. Due to hydrogen content in the syngas, it has tendency to generate combustion knock. Other combustible gases as methane and carbon monoxide are knock resistant fuels. Moreover, carbon dioxide has also anti-knock feature because of its high thermal capacity. Thus, the combustion knock was not observed in the conducted tests at the compression ratio of 10. However, at spark timing advanced to 25 deg, the syngases 1 and 3 can generate marginal combustion knock, just at the knock threshold. Higher knock was observed when the syngas 1 was burned at CR = 12.
4. When one compares the LFS with both the 0-10% MFB and the 10-90% MFB, then should be stated, in general such a simple formula between these quantities for the tests conducted in this investigation does not exist. It was probably caused by different conditions of computing LFS (at NTP) and the real combustion taking place in the IC engine. Additionally, the final results come from two different engines.

Tab. 3. Comparison between LFS (at NTP), 0-10%MFB and 10-90%MFB at ST=-20deg ATDC, CR=10

Fuel+air at lambda = 1	LFS at NTP	0-10% MFB	0-10% MFB	10-90% MFB	Engine
	cm/s	CA deg	ms	CA deg	
Syngas 1	39	8	1.4	14.5	CFR
Syngas 2	10	27.5	3.0	34.9	Deutz
Syngas 3	70	17.5	1.9	22.5	Deutz

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References

[1] Tomita, E., Fukatani, N., Kawahara, N., Marayama, K., Komoda, T., *Combustion in a supercharged biomass gas engine with micro-pilot ignition – effects of injection pressure and amount of diesel fuel*, 34th European Congress KONES 2008, European Science Society of Powertrain and Transport, Journal of KONES Powertrain and Transport, Vol. 14, No. 2, pp. 513-520, 2007.

[2] Kowalewicz, A., Wojtyniak, M., *Natural gas engines – problems and challenges*, 33th European Congress KONES 2007, European Science Society of Powertrain and Transport, Journal of Kones Powertrain and Transport, Vol. 14, No. 2, pp. 273-282, 2007.

[3] Postrzednik, S., *Gazowe paliwo silnikowe pozyskiwane na bazie węglowej*, Combustion Engines, PTNSS–2009–SC1, pp. 62-67, 2009.

[4] Smith, J. A., Bartley, G. J., *Stoichiometric Operation of a Gas Engine Utilizing Synthesis Gas and EGR for NO_x Control*, Transactions of the ASME, Journal of Engineering for Gas Turbines and Power 622, Vol. 122, 2000.

[5] Borecki, R., Szwaja, S., Pyrc, M., *Dual-Fuel Hydrogen-Diesel Compression Ignition Engine*, 34th European Congress KONES 2008, European Science Society of Powertrain and Transport, Journal of KONES Powertrain and Transport, Vol. 15, No. 4, pp. 49-52, 2008.

[6] Szwaja, S., Naber, J. D., *Impact Of Leaning Hydrogen-Air Mixtures On Engine Combustion Knock*, 34th European Congress KONES 2008, European Science Society of Powertrain and Transport, Journal of Kones Powertrain and Transport, Vol. 15, No. 2, pp. 483-492, 2008.

[7] Smith, P., Golden, D., Frenklach, M., Moriarty, N., Eiteneer, B., Goldenberg, M., Bowman, C., Hanson, R., Song, S., Gardiner, W., Lissianski, V., Qin, Z., Grimech30 Homepage, http:// www.me.berkeley.edu/gri mech, 1999.

[8] Heywood, J. B., *Internal Combustion Engines Fundamentals*, McGraw Hill Inc, 1988.

[9] Karim, A. G., *The onset of knock in gas fueled spark ignition engines prediction and experiment*, 33th European Congress KONES 2007, European Science Society of Powertrain and Transport, Journal of KONES Powertrain and Transport, Vol. 14, No. 4, pp. 165-175, 2007.