ON THE POSSIBILITY TO RECOVER THE DECREASE OF THE SPARK IGNITION ENGINES OUTPUT AT LIQUIFIED PETROLEUM GAS FUELING

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This paper describes a study concerning a possible solution designated to recover the decrease of the gasoline spark ignition engines output which occurs in the case of transition to an alternative fuel like liquefied petroleum gas. This solution involves the engine's dual fuelling with a mixture of LPG and hydrogen rich gas – HRG – the gas resulting from water electrolysis. The method used for this study consisted in investigating the working performance of a spark ignition engine fuelled successively with gasoline and with liquefied petroleum gas combined with different HRG flows rates – of 5, 10 and 20 lpm – corresponding to certain supplement mass fractions of 3.7; 7.9 and 15.3% respectively. The engine operating condition selected was n = 2500 rpm, engine speed, MAP = 370 mbar, intake manifold absolute pressure, engine load, and λ = 1.14 the relative air-fuel ratio. This engine condition represents an increased frequency operating condition that is typical for both urban and interurban traffic. It has been also studied the effect of a possible increasing of the compression ratio associated with the using of the minimum HRG flow rate on the engine's performance and efficiency. This study was made by numerical simulation by using the AVL BOOST code v. 2009.

Keywords: dual fueling, hydrogen, HRG, performance, compression ratio

1. Introduction

The transition from the fossil fuel to the renewable hydrogen, by using certain combinations of hydrogen, in small amounts, with conventional fuels, brings significant reductions of the pollutant emissions produced by internal combustion engine. Using hydrogen as combustion stimulant makes possible the observances of future demands for the reduction of the pollutant emissions of internal combustions engines, imposed in the European countries. The present paper addresses the possibilities of recovering the performance of an internal combustion engine operating on gasoline when is fuelled with LPG-HRG (Liquefied Petroleum Gas – Hydrogen Rich Gas) mixtures. [1], [2], [3].

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An experimental study was performed initially by the authors of this paper on a spark ignition engine having the main specifications shown in table 1. The engine was integrated in a specially designed test bed, schematically shown in figure 1.

Table 1

Test engine specifications

Engine Type	Four cylinders, four strokes
Displacement	1397 cm^3
Bore	76 mm
Stroke	77 mm
Compression ratio	9.5:1
Fuelling system	carburettor

The replacement of gasoline with LPG or LPG-HRG mixture it was found, by analyzing the experimental data, that no HRG flow rate (5, 10 or 20 l/min), regardless of the working conditions of the engine (speed and load) combined with LPG was able to bring the engine to the same performance as the gasoline fueled engine, in the area of the stoichiometric and relatively slightly rich mixtures.

The alternative LPG (propane and butane) and HRG fuels, especially hydrogen, are gases with higher resistance to knock compared to gasoline – table 2. [4]

The process of an experimental research activity on an engine with a modified compression ratio is difficult and definitely expensive. For this reason, as initial stage of the study, investigation method was considered using the theoretical procedure of working with the simulation software, AVL BOOST v.2009. The study was performed under the test conditions of 2500 rpm and MAP = 370 mbar (Manifold Absolute Pressure) and relatively lean mixtures $\lambda \sim 1.14$, which is significant in terms of pollutant emissions.

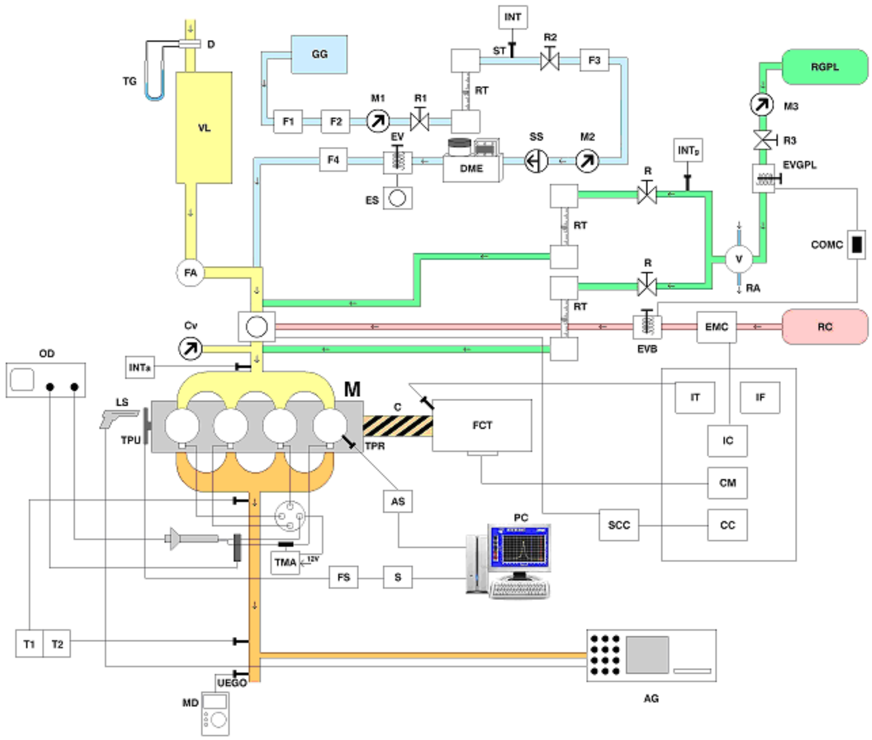


Fig.1 General schematic of the experimental test bed **Symbol – Meaning**

M – DACIA 1410 engine, type102-13	D – Diaphragm
FCT – Eddy current dynamometer	TG – Manometer U tube
C – Coupling shaft	LS - Stroboscopic lamp
GG – HRG generator	OD - Oscilloscope
F1, F2, F3, F4 – Fire extinguisher	TPU – Crank Angle Encoder
M1, M2, M3 -Manometers	T1, T2 – Temperature gauge
VM – Manual regulator	TPR - Pressure transducer
RT – Gas flowmeter	TMA – Spark ignition sensor
INT – Temperature numeric indicator	FS – Signal filter
DME – Electronic Mass Flowmeter	S – Power supply
ST – Temperature Probe	AS – Charge amplifier
SS – Check valve	PC – Data acquisition computer
EV – Solenoid valve	AG – Gas analyzer
ES – Emergency stop valve	UEGO - Universal Exhaust Gas Oxygen Sensor
RGPL –LPG tank	MD – Digital multimeter
R1, R2, R3 – Multi-circuit taps	IT – Dynamometer temperature sensor
EVGPL – LPG solenoid valve	IF – Dynamometer indicator
COMC – Fuel Switcher	IC – Fuel consumption indicator
V - Vaporizer	CM – Torque control unit
FA- Air Filter	CC - Throttle control unit
VL - Surge tank	SCC - Servomechanism throttle position

Under these operating conditions it was analyzed the possibility of recovering the performance of the gasoline fueled engine at LPG fueling by increasing the compression ratio. It was taken into account the fact that considering this load conditions and the LPG supply and, possibly, a HRG supplement, the compression ratio increase does not induce some risk of knock occurrence.

Table 2

The properties of hydrogen compared to gasoline and propane [5], [6]

Properties	**Gasoline (Izo-octan)**	**Propane**	**Hydrogen**
Chemical formula	(C_8H_{18})	C_3H_8	H_2
Molecular weight [kg/kmol]	~107 (114)	44.1	2.02
Minimum ignition energy [mJ]	0.24	0.26	0.02
Flame propagation speed [cm/s]	41.5	46	237
Diffusion coefficient [cm^2/s]	0.05	0.12	0.61
Stoichiometric air-fuel ratio [-]	14.7	15.6	34
Vapours density (at t = 20°C; p = 1atm) [kg/m^3]	0.275	-	0.08376
Liquid density (at the normal boiling point and p = 1 atm) [kg/m^3]	700	-	70.8
Energy density [MJ/Nm^3]	3.83	3.68	3.21
Relative energy density [MJ/Nm^3]	1.00	0.96	0.84
Higher Heating value (at 25°C ; p = 1atm) [kJ/g]	47.5	50.36	141.86
Higher Heating value (at 25°C ; p = 1atm) [kJ/g]	44.5	45.6	119.93
Ignition temperature [°C]	-43	-104	-253
Self-ignition temperature [°C]	240 - 480	490	585
Octane number research	95	105	130+ (weak mixtures)

2. Results

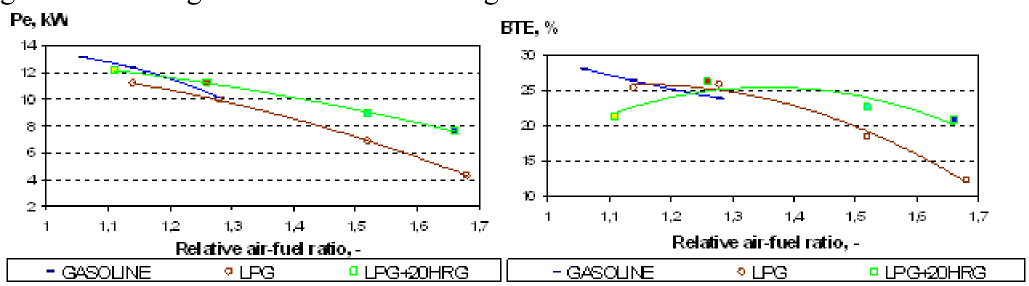
a. Experimental data

The experimental study revealed the main performance characteristics of the specified engine when operated with LPG + HRG mixtures. Under conditions of the optimal spark timings, the following main conclusions are to be shown:

1. By the partial substitution of LPG with HRG it was obtained, for the same load and speed, an increase of the brake output relative to the level corresponding to LPG fueling, but the maximum output similar to gasoline fuelling was not reached – figure 2 a.

2. The HRG supplement under certain conditions of LPG substitution determined the increase of the brake thermal efficiency of the LPG+HRG fuelled

engine relative to LPG fueling, but the brake thermal efficiency corresponding to gasoline fuelling was not recovered – Fig. 2 b.



a. b.

Fig.2 a-The brake output; b-Brake Thermal Efficiency, at n = 2500 rpm and MAP =370 mbar

b. Results obtained by numerical simulation

The study by numerical simulation, using AVL BOOST v.2009 software, had a demonstrative feature. The possible limits of the compression ratio increasing were imposed by the knock avoidance condition at high or full load engine operation.

Model construction

The AVL Workspace is a complete software package dedicated to internal combustion engines development. It includes BOOST, CRUISE, EXCITE and FIRE software products.

For the actual study it was used AVL BOOST v.2009 code, which is the software dedicated for thermodynamic calculations of the operating cycle and IMPRESS, which is the instrument used for the graphic processing of the obtained results.

AVL BOOST v.2009 is a complex software that can be used for a wide range of parametric studies such as the evaluation of performance at different engine operating conditions, for dimensional optimization of the engine's design parameters or for comparison of different engine concepts, that are part of the pre-processing data structure.

The construction of the engine's model by means of AVL BOOST is related to a detailed analysis of the experimental results and to a parametric study as a predictive stage in the engines development activity [7]. It allows the identification of the relevant variables that are meant to reach the goal of this research, but also the possibility of prediction on the engine behavior.

Figure 3 shows the symbolic model of the engine used in the experimental investigation. This is made by means of the pre-established constitutive elements in AVL BOOST, as follows: connection pipes (1→16), plenums (PL1, PL2, PL3 and PL4), air filter (CL1), throttle (TH1), HRG injectors (I1), respectively LPG (I2), engine's cylinders (C1, C2, C3 and C4), measuring points (MP1, MP2 and

MP3) and connections of the system with the environment by the system boundaries, on intake and exhaust (SB1 and SB2). [8], [9]

For every component element of the model the constructive and functional parameters corresponding to the investigated conditions are inserted.

The engine is considered to be an assembly of identical cylinders, which exchange mass and energy through the system boundaries with the environment.

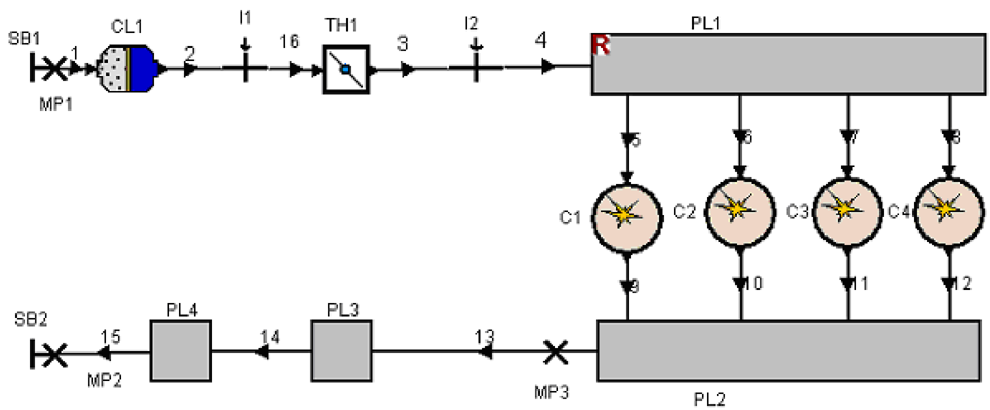


Fig. 3 The engine's symbolic model created for simulation its operation conditions.

It is supposed that the cylinder's charge represents an homogeneous mixture of reaction products and air, in which there are no gradients of temperature, pressure waves and nonequilibrium composition.

On the basis of these assumptions and in accordance with the energy balance figured in figure 4, represented by the energy conservation equation (1) applied to an open thermodynamic system and by mass conservation equation (2):

$$\frac{d(m_c \cdot u)}{d\alpha} = -p_c \cdot \frac{dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - h_{BB} \cdot \frac{dm_{BB}}{d\alpha} + \sum \frac{dm_i}{d\alpha} \cdot h_i - \sum \frac{dm_e}{d\alpha} \cdot h - q_{ev} \cdot f \cdot \frac{dm_{ev}}{d\alpha} \qquad (1)$$

$$\frac{dm_c}{d\alpha} = \sum \frac{dm_i}{d\alpha} - \sum \frac{dm_e}{d\alpha} - \frac{dm_{BB}}{d\alpha} + \frac{dm_{ev}}{d\alpha} \qquad (2)$$

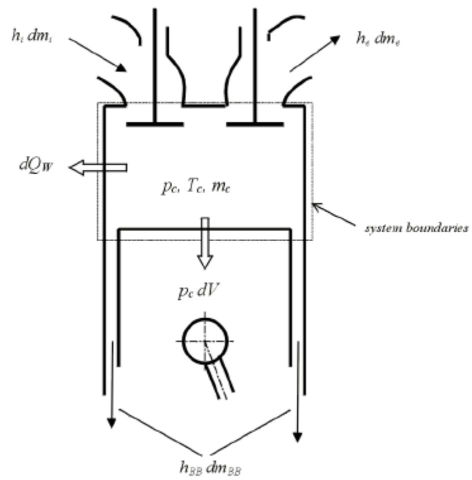


Fig. 4 The energetic balance-sheet in the engine cylinder, [6].

where:

$\frac{d(m_c \cdot u)}{d\alpha}$ - change of the internal energy in the cylinder;

$-p_c \cdot \frac{dV}{d\alpha}$ - piston work;

$\frac{dQ_F}{d\alpha}$ - fuel heat input;

$\sum \frac{dQ_W}{d\alpha}$ - wall heat losses;

$-h_{BB} \cdot \frac{dm_{BB}}{d\alpha}$ - enthalpy flow due to blow-by;

p_c - cylinder pressure;

V - cylinder volume;

α - crank angle;

h_{BB} - enthalpy of blow-by;

$\frac{dm_{BB}}{d\alpha}$ - blow-by mass flow;

dm_i - mass element flowing into the cylinder ;

dm_e - mass element flowing out of the cylinder;

h_i - enthalpy of the in-flowing mass;

h_e - enthalpy of the mass leaving the cylinder;

q_{ev} - evaporation heat of the fuel ;

f - fraction of evaporation heat from the cylinder charge;

m_{ev} - evaporating fuel.

The heat exchange to the combustion chamber walls, such as the cylinder head, the piston and the cylinder liner is calculated by the equation (3):

$$Q_{wi} = A_i \cdot \alpha_w \cdot (T_c - T_{wi}) \tag{3}$$

where Q_{wi} is the heat transferred by convection to the combustion chamber walls, to the cylinder head, to the piston and the liner; A_i is the area of the cylinder head, of the piston and the liner; α_w is the convective coefficient of heat exchange; T_c is the mean temperature of the cylinder charge; T_{wi} is the temperature of the cylinder head walls, of the piston head and the cylinder liner**.** [6]

The heat exchange model adopted for this study was Woschni 1990. According to this model the heat exchange coefficient correlates well with the engine's operating condition by to means of the indicated mean effective pressure through equation (4):

$$\alpha_w = 130 \cdot D^{-0.2} \cdot p_c^{0.8} \cdot T_c^{-0.53} \cdot \left\{ c_1 \cdot c_m \cdot \left[1 + 2\left(\frac{V_{TDC}}{V}\right)^2 \cdot IMEP^{-0.2} \right] \right\}^{0.8} \tag{4}$$

where α_w is the convective coefficient of heat exchange; D is the cylinder bore; p_c is the current pressure in the cylinder; T_c is the mean current temperature of the load in the cylinder; c_1 = 2.28 + 0.308 * c_u/c_m, c_m is the average speed of the piston; c_u is the circumferential speed; V_{TDC} is the volume of the combustion chamber at Top Dead Center (the minimum volume of the combustion chamber); V is the current volume of the combustion chamber; $IMEP$ is the indicated mean effective pressure.

The combustion of the fuel was expressed through a conventional (5) VIBE-type function, where:

$$\frac{dx}{d\alpha} = \frac{a}{\Delta\alpha_c} \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{(m+1)}} \tag{5}$$

where:

$$dx = \frac{dQ}{Q} \tag{6}$$

$$y = \frac{\alpha - \alpha_0}{\Delta\alpha_c} \tag{7}$$

Q - total fuel heat input;
α - crank angle;
α_0 - start of combustion;
$\Delta\alpha_c$ - combustion duration;
m - shape parameter (m = 2)
a - Vibe parameter (a = 6.9 for complete combustion)

The simulation of the fuel combustion into the engine cylinders was performed under the following assumptions:

- the air-fuel mixture preparation is made outside the cylinders, considering that the mixture is homogeneous at the beginning of the combustion, the air-fuel ratio is constant during the combustion and both the burned and unburned gases have the same pressure and temperature, although the chemical composition is different;
- the induction of LPG and HRG is made through the I1 and I2 injectors, considering it a continuous injection with an established constant flow rates, depending on the investigated experimental conditions ;
- the mass flows rates of the component gases of the fuel mixture are determined by species (propane, butane, H_2), as well as other parameters that characterize the flow of these gases (ex flow coefficients).

The calibration of the model

The calibration of the model has been made by comparison of the pressure indicated diagrams, considering the concordance with the experimental data for three fuels: gasoline, LPG, LPG+5HRG, under the engine's operating conditions n=2500 rpm, λ=1.14, MAP=370 mbar and optimum spark timings– Fig. 6.

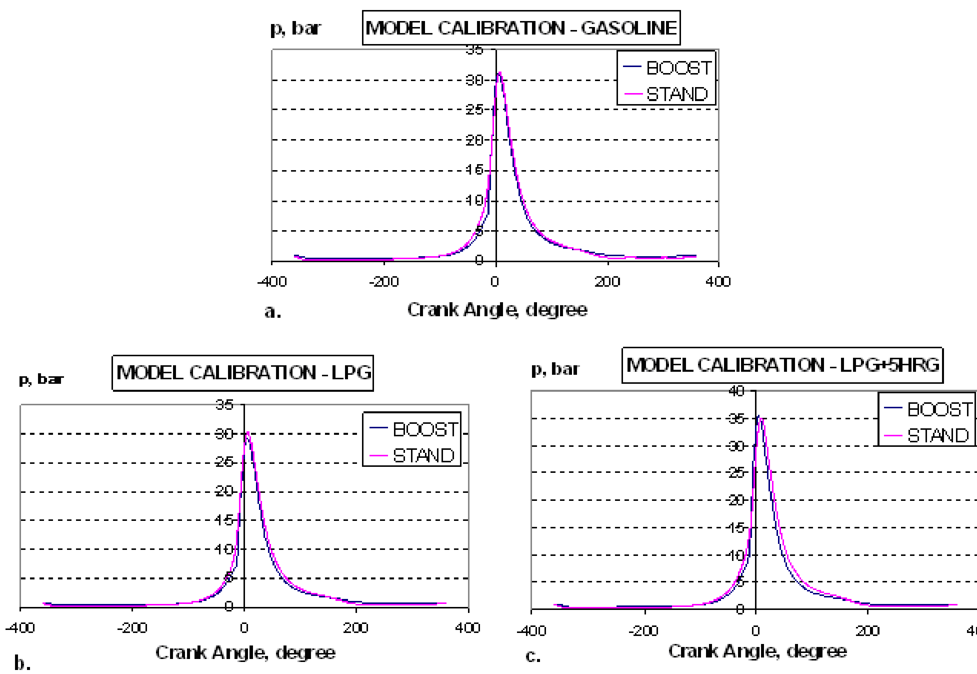


Fig. 6 Comparison between the indicated diagrams: AVL BOOST model – experimental data under the operating conditions n=2500rpm, λ=1.14, MAP=370 mbar, optimum timings for: a. Gasoline, b. LPG, c. LPG+5HRG.

During the calibration operation of the model it was also analyzed the concordance of another series of parameters, such as: the brake output, the fuel and air consumption, the relative air-fuel ratio - λ, the pressures and the temperatures in the measuring points.

Maximum deviations have been detected between the calculated values and the experimental ones for the air consumption Ca (6.6%) and for engine output P (4.21%) – table 3.

Table 3

Data comparison experimental stand – numerical simulation

GASOLINE					
Origin of the value	**p_{max} [bar]**	**P [kW]**	**C_c [g/s]**	**C_a [g/s]**	**λ [-]**
Experimental	41.72	12.35	1.078	17.35	1.14
Simulation	41.38	12.48	1.112	17.7	1.12
Deviation	-0.81%	1.05%	3.89%	1.98%	-1.75%
LPG					
Origin of the value	**p_{max} [bar]**	**P [kW]**	**C_c [g/s]**	**C_a [g/s]**	**λ [-]**
Experimental	43.1	11.19	0.953	16.302	1.14
Simulation	42.89	11.1	0.957	16.308	1.107
Deviation	-0.49%	-0.80%	0.42%	0.04%	-2.89%
LPG+5LPM HRG					
Origin of the value	**p_{max} [bar]**	**P [kW]**	**C_c [g/s]**	**C_a [g/s]**	**λ [-]**
Experimental	40.92	11.64	1.055	16.27	1.12
Simulation	41.29	12.13	1.055	16.73	1.14
Deviation	0.9%	4.21%	0%	2.83%	1.79%

The study of engine's operating with increased compression ratio and LPG +5 HRG fueling

Within the calibration operation, related to the chosen combustion model, based on the VIBE function and on the homogeneous mixture, we inserted the values of parameters α_0 - the start of combustion, CD – combustion duration and *m* – the shape parameter of heat released (considered to be constant during the analysis) were adjusted.

Once the VIBE calibration parameters have been established they have been considered as constant parameters. [10], [11]

It is known that for the increased compression ratios the optimum spark timings decrease. This is, first of all, the effect of shortening of the first stage of combustion process. On the other hand it may be noticed the tendency of reducing the entire combustion duration – CD – by increasing the intensity of the chemical transformations in the combustion zone. [12]

The following working procedure was adopted establishing the effective engine output dependency on α_0 and CD parameters:

- as for the engine working conditions chosen for this study, defined by n = 2500 rpm, λ = 1.14, MAP = 370 mbar, specific optimum spark timings, compression ratio ε = 9.5, for each investigated fuel (Gasoline, LPG, LPG+5HRG), from the experimental database, the pair of values of the two combustion parameters: α_0 – the start of combustion (for gasoline α_0 = -22 °CA, for LPG α_0 = -22 °CA and for LPG+5HRG, α_0 = -22…-16 °CA) and CD – combustion duration (for gasoline CD = 48 °CA, for LPG, CD = 59 °CA and for LPG+5HRG, CD = 55…63 °CA) were considered.

Information regarding the engine performance developed for higher compression ratios, up to ε = 12 in steps of 0.25 units (Fig. 7) have been obtained keeping constant α_0 and CD previously defined based on experimental results.

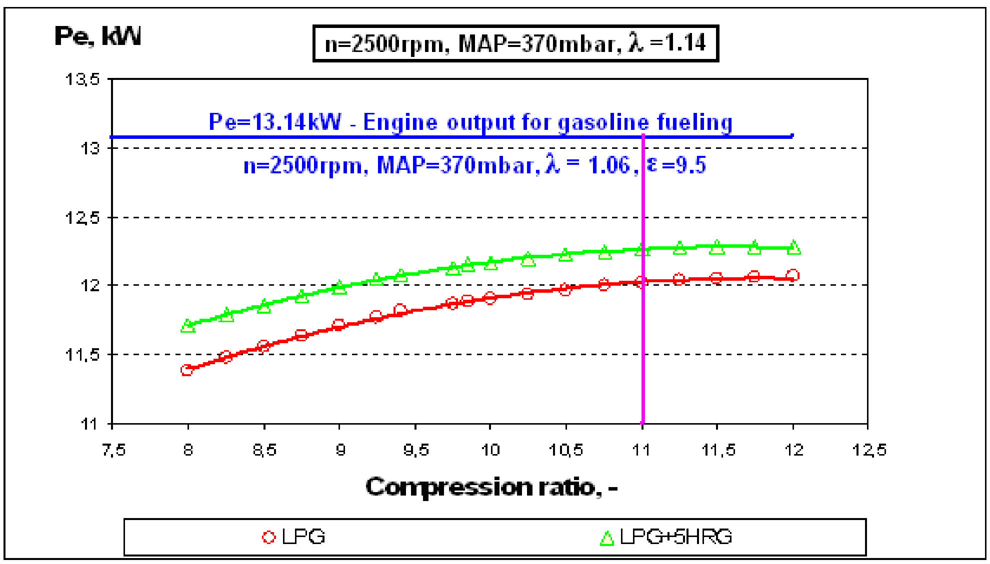


Fig. 7 The variation of the engine output relative to compression ratio, considering (α_0, CD) values obtained for ε = 9.5

It can be notice that under the established conditions the recovery of the effective engine output developed for gasoline P_e = 13.14 kW was not possible for compression ratio ε = 11.

- further on, it was performed the parametric study of the engine output, efficiency and emissions for compression ratio ε = 11, considering as variable the two control parameters of combustion, α_0 and CD knowing the influence of the compression ratio on combustion process. Within the parametric study, it has been imposed to simulate cycles having α_0 smaller than that experimentally established, for compression ratio ε = 9.5, (α_0 = -12…0° CA), and combustion durations – CD therefore (CD = 55, 60 and 65 ° CA), were selected.

Fig. 8 shows the simulation results obtained for ε = 11 and engine operation condition of n=2500 rpm, λ=1.06 , MAP=370 mbar.

It can be emphasized that the increase of the compression ratio from 9.5 to 11 cannot ensure the recovery of the gasoline engine output (considering the mentioned working conditions) using only LPG.

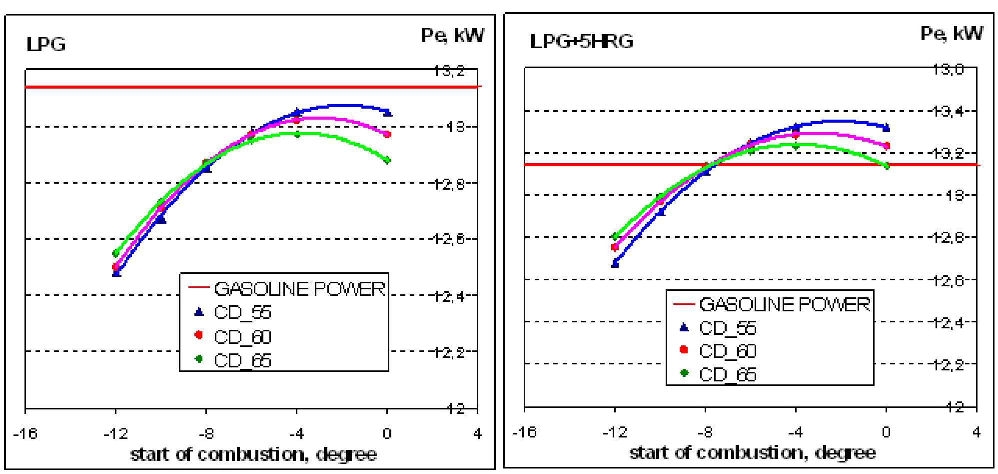
The measure of increasing the compression ratio to ε = 11 associated with the use of a mixed fuel – LPG+5HRG, having a mass fraction X_a = 3.7% HRG under the established conditions, proved feasible in terms of output recovery.

Analyzing the simulations results, the following remarks can be made:

- using only LPG, the increase of the compression ratio to ε = 11 determines a predictable increase of the effective output, but it does not reach the same performance as the gasoline engine 13.14 kW – Fig. 8 - a.

- by increasing the compression ratio to ε = 11, on a spark ignition engine fueled with LPG combined with HRG, the effective output of the gasoline engine can be reached, even exceeded; thus, using LPG and 3.7% (mass) HRG, the objective of recovering the effective output of the engine is achieved for all the three values proposed for the combustion duration.

- considering CD = 65 °CA as impracticable (higher than corresponding to ε = 9.5) the following groups: CD = 55 °CA, α_0 = - 8…- 4 °C and CD = 60 °CA, α_0 = - 8…- 4 °CA, became the actual imposed conditions; the increasing of HRG fraction over 3.7% induced a substantially over passing of the gasoline engine output.



a. b.

Fig. 8 The output characteristics relative to start of combustion α_0 at different combustion durations: a. LPG, b. LPG+5HRG (3.7% HRG)

Consequently, the range CD = 55…60° CA and α_0 = - 8…- 4° CA for LPG using combined with 3.7% (mass) HRG represented the interesting domains for output recovery considering LPG enhanced with small fraction of HRG.

A major advantage of using LPG + 5 HRG at high compression ratio ε = 11 is the increase by 9% of brake thermal efficiency in comparison with pure LPG fuelling at ε = 9.5 – table 4.

Table 4

The brake efficiency for start of combustion α_0=-8°CR

Brake thermal efficiency	**Gasoline** n=2500rpm,λ=1.06 MAP=370mbar	**LPG** n=2500rpm,λ=1.14, MAP=370mbar			**LPG+5HRG** n=2500rpm,λ=1.14, MAP=370mbar		
			ε = 11			ε = 11	
	ε = 9.5	ε = 9.5	CD=55 °CA	CD=60 °CA	ε = 9.5	CD=55 °CA	CD=60 °CA
$\eta_{e[\%]}$	28.02	25.36	27.648	27.647	21.22	27.15	27.09

3. Conclusions

The study regarding the possibility to recover the engine's output when gasoline fueling is used in the case of combining LPG with HRG by increasing the compression ratio, on the basis of the simulation made in AVL BOOST software, lead to the following main conclusions:

1. The gasoline engine operation, under the conditions n = 2500 rpm, MAP = 370 mbar, λ = 1.06 and ε = 9.5, experimentally studied, have been adequately reproduced by calculation.

2. The effective output of the gasoline engine was not recovered when pure LPG fuel was used by increasing the compression ratio to ε = 11 and by optimizing the start of combustion and the combustion duration.

3. The recovery of the effective output of the gasoline engine was possible only when LPG and HRG fuels were used by adding 5 l/min HRG flow (3.7% HRG mass) and by increasing the compression ratio to 11, by optimizing the start of combustion and the combustion duration too. Under these circumstances, the brake fuel conversion efficiency was closer to gasoline operation.

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NOMENCLATURE:

CD	- Combustion Duration [deg];
LPG	- Liquefied Petroleum Gas;
HRG	- Hydrogen Rich Gas;

LPG+5HRG	- LPG-HRG mixture with 5 l/min HRG flow rate;
Pe	- Effective output [kW];
CR	- Crankshaft angle [deg];
SOC	- Start of combustion [° CA];
n	- Engine speed [rpm];
MAP	- Intake manifold absolute pressure [bar];
pi	- Indicated mean effective pressure [bar];
pmax	- Maximum pressure in cylinder [bar];
re	- Brake efficiency [%];
α_0	- Start of combustion angular position [°CA];
α_s	- Ignition timing
ε	- Compression ratio [-];
λ	- Relative air-fuel ratio [-].

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