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NUMERICAL INVESTIGATION ON THE EFFECTS OF GASOLINE AND HYDROGEN BLENDS ON SI ENGINE COMBUSTION

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Even small amount additive (10…15% by volume from whole air amount) of hydrogen (H_2) into spark ignition (SI) engines obviously effects ecological parameters and engine efficiency because of H_2 exclusive properties.

SI engine work process simulation was made using *AVL Boost* simulation software. Analysis of results showed that engine power depends a lot on H_2 supply technique into engine; NO_x amount in exhaust gases directly proportional to the amount of H_2, however, making mixture leaner up to $\lambda = 1.6$, it is possible to reach significant NO_x decrease. Increased amount of H_2 as an additive in fuel, changes H/C ratio in fuel mixture, also hydrogen improves properties of the mixture (particularly lean) and combustion of hydrocarbons what can be a reason of decreased HC emissions in exhaust gases.

Hydrogen and gasoline mixture, engine efficiency, exhaust gases, nitrous oxides, hydrocarbons, simulation.

Introduction

The most common method of energy production in the world is the burning of fossil fuels (oil products, natural gas, and coal). Large amounts of energy, which is currently very intensively used by humankind for their own needs, accumulated in these raw materials in the form of carbon and hydrogen many thousand years ago. However, the fuel combustion products CO, CO_2, SO_x, NO_x, HC, solid particles, etc., pose serious environmental problems. Therefore, the search for alternative energy "accumulators" is the main goal of the whole mankind.

Hydrogen is carbon-free, renewable, easy (relatively) derived alternative energy "accumulator", which can be easily adapted for usage in internal combustion engines. The usage of pure hydrogen is characterized by some drawbacks because of its extremely low density, e.g., decrease in engine power, torque and coefficient of performance at low engine speed (Kahraman *et al.*, 2006). However, enrichment of conventional fuel with hydrogen helps to avoid the aforementioned drawbacks.

Hydrogen flame propagation speed is approximately 5 times higher than the one of other known hydrocarbon fuels (Table 1). Mixture combustion range is extremely wide, therefore, even a very small additive of hydrogen into liquid or gaseous hydrocarbon fuel can significantly expand the ignition limits of these mixtures (Rimkus, 2013). Hydrogen additive reduces the induction period and improves the period of flame propagation (Ji, Wang, 2011). 3% of hydrogen (by volume) increases flame propagation speed from 11.08 m/s (standard engine) to 15.2 m/s (37%), while 6% of hydrogen – up to 17.78 m/s (60%) (Ji *et al.*, 2013). Under high air excess, even a small amount of H_2 (up to 2% by volume) accelerates and improves combustion, which results in improvement of process efficiency, however, due to intensified heat release (peak temperature) NO_x concentration increases in combustion products (D'Andrea *et al.*, 2004). However, besides already mentioned characteristics of hydrogen usage some of the previous research also mentions the reduction of NO_x (Jingding *et al.*, 1998). 8% (by volume from total quantity of mixture) of hydrogen additive already allows to achieve that engine COP would not decrease as the mixture leans down (Ji, Wang, 2011). Meanwhile, Rimkus' (Rimkus, 2011) papers reveal that as spark ignition engine runs on leaned mixture and as ~0,15% (of intake air volume) HHO (Brown gas) is supplied, indicator coefficient of performance of engine increases up to 5%.

As hydrogen and air mixture combusts, due to higher flame speed and *combustion rate*, compared to combustion of gasoline and air (Table 1), combustion duration significantly decreases in comparison with the one of gasoline and air (Mardi *et al.*, 2014). It is also determined that as indicator pressure increases in combustion chamber, combustion temperature also grows, thus, influencing the increase of NO_x emission.

Table 1. Fuel properties (Bain *et al.*, 1998)

Properties	Hydrogen	Gasoline (izo-octan)
Chemical formula	H_2	C_8H_{18}
Ignition energy, mJ	0,02	0,24
Flame propagation speed, cm/s	237	41,5
Diffusion coefficient, cm^2/s	0,61	0,05
Higher heating value, MJ/kg	142	47,3
Lower heating value, MJ/kg	120	44
Molecular mass	2,02	≈107...114
Octane number	106	100
Stoichiometric ratio	34,3	14,7
Density, kg/m^3	0,09	750

The analysis of scientific literature resources revealed a number of experimental studies on spark engine running on different mixtures of hydrogen and various fuels, as well as a lot of modelling tests, where the combustion process of various pure fuels is modelled, however, no scientific article, modelling the combustion of gasoline and hydrogen, has been found. Therefore, the aim of this research is to develop a mathematical model of spark ignition engine running on gasoline and hydrogen mixture by using the simulation tool *AVL Boost*, and to find the main energy and environmental indicators of the engine on the basis of aforementioned model.

Research methodology

Assessment of hydrogen influence on internal combustion engine performance indicators was performed on the basis of bench tests and numerical modelling of *Nissan Qashqai* spark ignition engine *HR 16DE* (Table 2). The study was performed at constant engine speed (n = 2000 min^{-1}), throttle opening position (15%) and ignition advance angle $\Theta = 18^{\circ}$ before TDC. Bench tests were performed by using gasoline (G), at air excess coefficient of $\lambda = 1$. Numerical modelling was performed by using gasoline and mixtures of gasoline and hydrogen. It was accepted that there will be 10% of hydrogen (G + 10% H_2); and 15% of hydrogen (G + 15% H_2) in the intake air volume. Modelling was performed for flammable mixture of various compositions (λ = 0.9...1.6).

Table 2. Engine *Nissan HR 16 DE* technical data

Parameter	**Value**
Number of cylinders	4
Cylinder bore, mm	78
Piston stroke, mm	83.6
Displacement, cm^3	1598
Nominal power, kW (min^{-1})	84 (6000)
Maximum engine torque, Nm (min^{-1})	156 (4400)
Compression ratio, ε	10.7
Number of valves per cylinder	4

Bench tests were performed during initial stage of the study by using test equipment of Internal Combustion Engines Laboratory of Department of Automobile Transport of Vilnius Gediminas Technical University (Fig. 1). The tested engine is controlled by programmable ECU *MoTeC M800*. While braking with eddy-current load stand *AMX 200/100*, engine's effective torque M_e is measured. Pressure in cylinder is determined by pressure sensor *AVL ZI31*, built-in in spark plug, and recorded by using *AVL DiTEST DPM 800*. Gasoline (G) hourly consumption (B_{d_G}) is measured by electronic fuel consumption gauge *AMX 212F*. Exhaust emission was tested with *AVL DiSmoke 4000* emission tester.

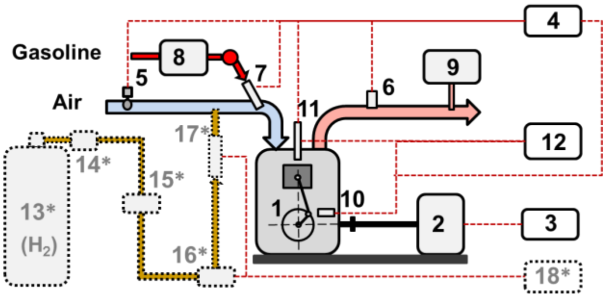

Fig. 1. Scheme of engine stand research equipment: 1 – *Nissan HR 16DE* engine; 2 – engine load stand *AMX 200/100*; 3 – electronic control unit of load stand; 4 – control unit of engine *MoTeC M800*; 5 – throttle control servomotor; 6 – oxygen sensor; 7 – gasoline injector; 8 – gasoline consumption gauge *AMX 212F*; 9 – exhaust gas analyser *AVL DiSmoke 4000*; 10 – crankshaft position sensor ; 11 – spark plug with built-in pressure sensor *AVL ZI31*; 12 – equipment for measuring the pressure in cylinder *AVL DiTEST DPM 800*; 13 – compressed H_2 gas cylinder; 14 – high pressure reducer; 15 – gas meter; 16 – low pressure reducer; 17 – gas injector; 18 – gas equipment supply control unit; ***** – additional H_2 gas supply equipment

Numerical modelling of engine performance was implemented with *AVL Boost* by creating the model of internal combustion engine, using visual icons (Fig. 2).

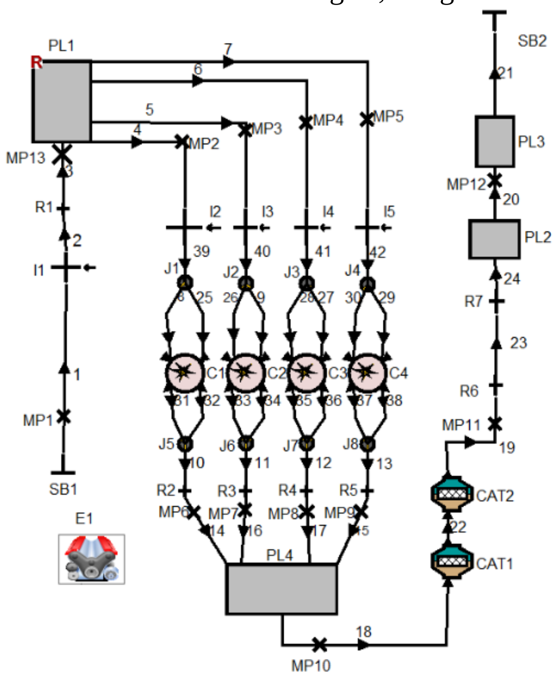

Fig. 2. Model of internal combustion engine developed by numerical modelling tool *AVL BOOST*

During modelling, the following marginal conditions were selected: inlet air pressure, quantity, temperature; temperature, pressure of exhaust combustion products; quantity of injected fuel, air excess coefficient, etc. It was selected that fuel mixture (hydrogen and gasoline fuel mixture) is injected into intake manifold by four injectors.

While performing the analysis of combustion process by applying Vibe function (Vibe 1970), the characteristics of heat release of the engine are approximated:

$$\frac{dx}{d\varphi} = \frac{a}{\Delta\varphi_c} \cdot (m_v + 1) \cdot y^{m_v} \cdot e^{-a \cdot y^{(m_v+1)}} \quad ; \tag{1}$$

$$dx = \frac{dQ}{Q}\,, \tag{2}$$

where Q – heat released by fuel during duty cycle; φ – crankshaft rotation angle; m_v – combustion intensity parameter; a – Vibe constant at 99,9% fuel combustion; a = 6,905; y – relative combustion time:

$$y = \frac{\varphi - \varphi_0}{\Delta\varphi_c}, \tag{3}$$

where φ_0 – start of combustion; $\Delta\varphi_c$ – combustion duration.

By integrating Vibe function, we calculate the part of fuel mass, which has burned down from the start of combustion process:

$$x = \int \frac{dx}{d\varphi} \cdot d\varphi = 1 - e^{-a \cdot y^{(m_v+1)}}. \tag{4}$$

A two-zone Vibe function is used for synthesis and analysis of engine duty cycle during numerical modelling. In this case, it is taken into account that the zones of burned and not burned mixture occur during combustion. By taking into account the duty regime of engine, heat release of fuel combustion, heat and gas exchange in cylinder, the analysis showed the change in gas temperature and pressure during cycle. What is more, energy and environmental indicators are calculated. Combustion duration ($\Delta\varphi_c$) and combustion intensity parameter (m_v), as the engine is running on gasoline and hydrogen mixture, is selected depending on the composition of mixture by taking into consideration the properties of fuel (Safari *et al.*, 2009).

Results and their analysis

The intake air volume, determined during bench tests, as the engine was running on gasoline (n = 2000 min^{-1}, throttle 15%, Θ = 18º and λ = 1), is 40.70 m^3/h. It is assumed that this intake gas volume remains constant as the engine intakes air and hydrogen gas mixture: $V = V_{air} + V_{H2}$ = const. = 40.70 m^3/h.

As 10% H_2 is supplied to air, it makes $V_{H2} = 4.07$ m^3/h, and 15% H_2 hydrogen will take $V_{H2} = 6.10$ m^3/h, respectively, therefore, intake air quantity will decrease down to $V_{air} = 36.64$ m^3/h and $V_{air} = 34.60$ m^3/h, respectively. While leaning the mixture from $\lambda = 0.9$ to $\lambda = 1.6$, an increasingly higher air quantity (according to the determined air excess coefficient) will be required for hydrogen combustion. The air quantity unused for H_2 combustion will decrease, and, taking into account the remaining air and λ, the mass of injected gasoline is calculated. While leaning the mixture within the set limits at constant H_2 supply and as gasoline quantity decreases, hydrogen mass concentration grows from 12% to 24% (G + 10% H_2) and from 21% to 47% (G + 15% H_2) (Fig. 3).

Despite the fact that lower calorific value of hydrogen (H_{l_H2} =120 MJ/kg) is 2.76 times higher than the one of gasoline (H_{l_G} =43.5 MJ/kg), as H_2 is additionally supplied in case of stoichiometric mixture, the energy brought by fuel decreases from 155.73 MJ/h (G) 5.8% (G + 10% H_2) and 8.7% (G + 15% H_2) (Fig. 4).

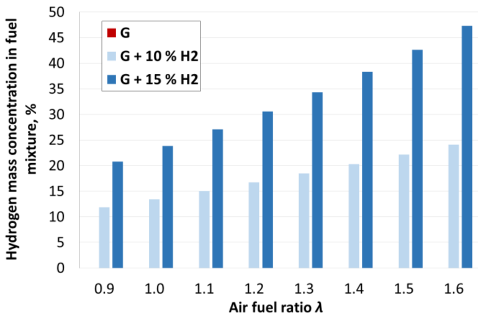

Fig. 3. Dependence of hydrogen mass concentration in fuel mixture on hydrogen concentration and excess coefficient

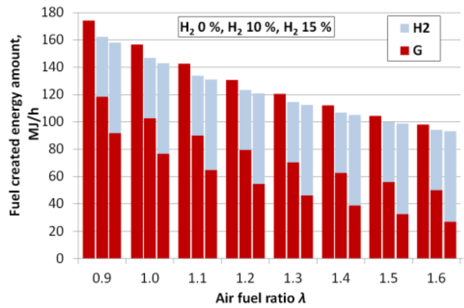

Fig. 4. Dependence of energy quantity brought by hydrogen (H_2) and gasoline (G) on hydrogen concentration and air excess coefficient

When leaning mixture, the energy brought by fuel decreases significantly, since as a result of decrease of quantity of air for gasoline, there is a decrease of injected gasoline mass, as well as in its energy. At $\lambda = 1.6$, the energy brought by gasoline decreases down to 97.33 MJ/h (37.5%). Additional supply of H_2 decreases the energy of fuel mixture by 3.3% (G + 10% H_2) and 4.9% (G + 15% H_2).

The effective power of the engine, determined during bench tests, as the engine was running on gasoline and under stoichiometric mixture, was $P_e = 11.91$ kW. After performing the analysis of engine duty cycle and following the scientific literature resources (Safari *et al.*, 2009; Rakopoulos *et al.*, 2010), combustion duration $\Delta\varphi_c = 47$ and combustion intensity parameter $m_v = 2.5$ was found. By using the aforementioned indicators and upon performing the analysis of combustion process, the determined effective power of the engine $P_e = 12.07$ kW differs from the measured one only by 1.3%, and it means that the created numerical model is quite accurate. During numerical modelling, as hydrogen concentration in mixture varied and composition of combustible mixture was changed, the parameters were selected by taking into account the fact that hydrogen increases combustion intensity and accelerate flame front propagation speed, and combustion intensity decreases in case of leaning the mixture (Heywood, 1988).

Numerical modelling revealed that in case of stoichiometric mixture and additional supply of 10% H_2, engine power decreases by 9.5% (from 12.07 kW to 10.92 kW), and in case of supply of 15% H_2 – 15.5% (from 12.07 kW to 10.2 kW) (Fig. 5). The estimated decrease in power is higher than the determined decrease in energy, brought by fuel, while supplying H_2 gas, since hydrogen changes combustion parameters, and the identified ignition advance angle $\Theta = 18^o$ is not optimal. It is also confirmed by decrease of indicator coefficient of performance η_i (Fig. 6) from 0.351 (G) to 0.334 (-4.8%) (G + 10% H_2) and to 0.327 (-6.8%) (G + 15% H_2).

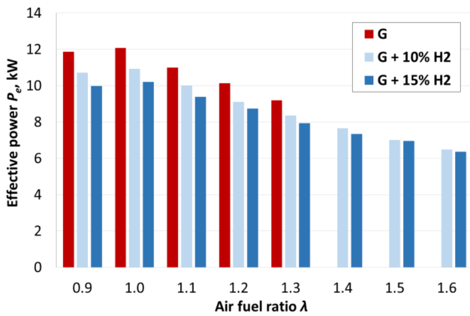

Fig. 5. Dependence of effective power on hydrogen concentration and air excess coefficient

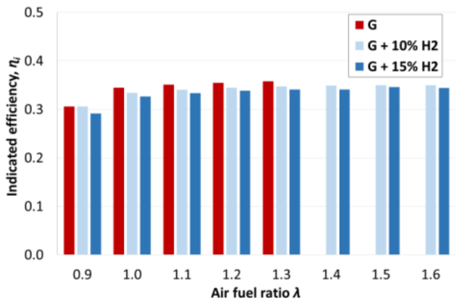

Fig. 6. Dependence of indicator coefficient of performance on hydrogen concentration and air excess coefficient

While leaning the mixture, engine power decreases. In case of engine, running on pure gasoline, $\lambda = 1.3$ is the ignition threshold, i.e., spark plug does not ignite leaner mixtures of gasoline and air. However, significantly leaner gasoline and hydrogen mixtures can be ignited. Following numerical modelling, it was found that in case of hydrogen supply, engine power is 6.46 kW (G + 10% H_2) and 6.36 kW (G + 15% H_2).Indicator coefficient of performance is equal to 0.350 and 0.344, respectively. Energy efficiency of the engine in case of lean mixture ($\lambda = 1.6$) and additional supply of 10% H_2 is close to the one as in case of using stoichiometric mixture of gasoline – air, however, environmental indicators of the engine significantly improve (Fig. 9 and 10). In case of richer mixture ($\lambda = 0.9$), energy and environmental indicators of the engine worsen because of the lack of oxygen, required for combustion, and too early start of combustion.

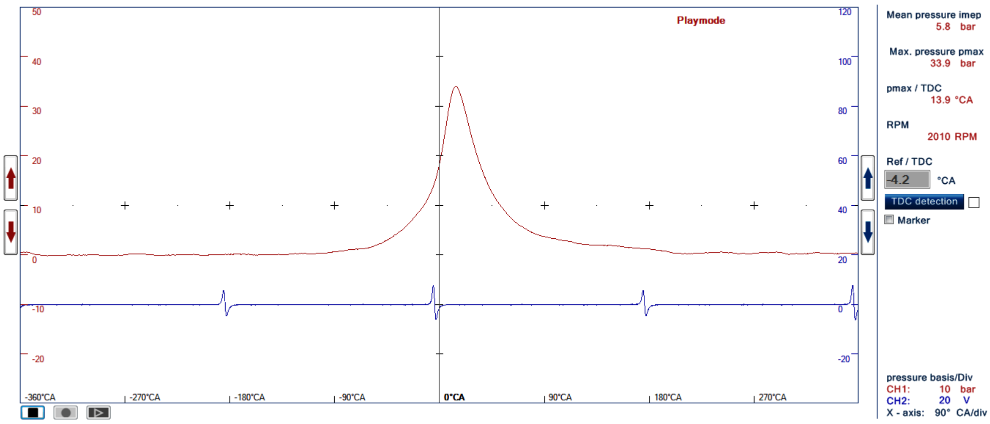

Fig. 7 Pressure in cylinder determined by stand experiments

Pressure in engine cylinder, measured during bench tests, as the engine was running on stoichiometric mixture, reaches p_{max} = 3.39 MPa, as crankshaft turns at 13.9° after TDC (Fig. 7.). Numerical modelling resulted in close indicators p_{max} = 3.50 MPa at 13.6° after TDC. This confirms the properly set modelling parameters.

The maximum values of pressure p_{max} and temperature T_{max}, obtained after making calculations for the provided combustible mixtures, are given in Figure 8. As hydrogen concentration is increased in mixture, combustion takes place more intensively, and the maximum pressure and temperature reached is closer to TDC (φ_{pmax} and φ_{Tmax}). However, in case of leaning the mixture, combustion intensity decreases and approaches to optimum set ignition advance angle. Thus, by additional supply of hydrogen and by leaning the combustible mixture, it is possible to obtain the optimum indicator coefficient of performance for a cycle without additional adjustment of combustion start. However, it is lower than in case of engine running on gasoline, since larger part of thermal energy is transferred to environment due to higher combustion temperature of hydrogen.

It was determined that in case of additional supply of hydrogen and increase of its quantity, there is an increase of NO_x concentration in exhaust gas (Fig. 9). It is mostly affected by free quantity of oxygen (excess air) during combustion, combustion intensity, and maximum temperature. In comparison with pure gasoline, hydrogen additive mostly increases NO_x concentration (up to 60...70%) at λ = 1.1...1.3. However, by comparing the emission of nitrogen oxides as engine runs on gasoline (λ = 1) with emission as engine runs on G + 10% H_2 mixture (λ = 1.6), the concentration of pollutants decreases 96%, when the power goes down only 46%.

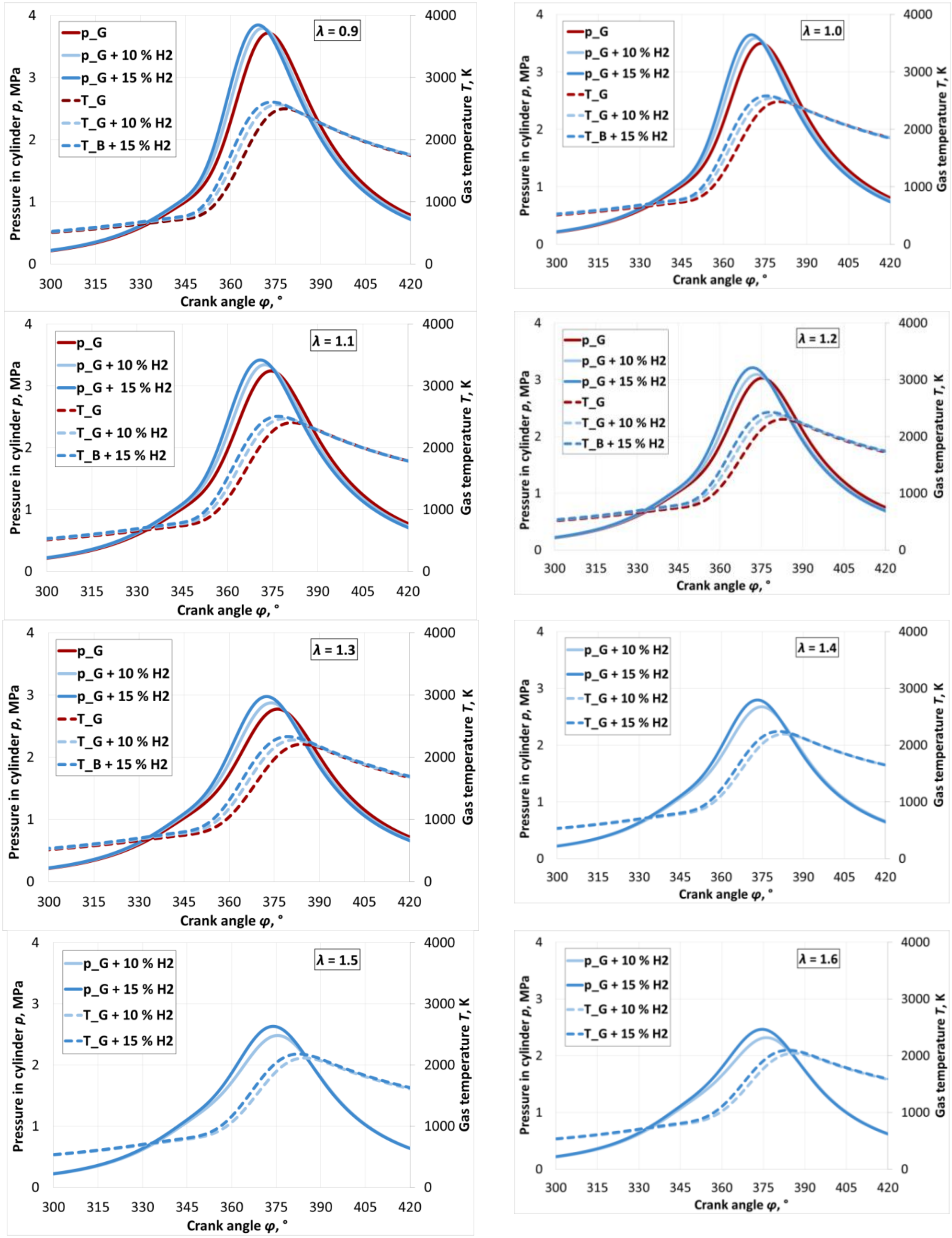

Fig. 8. Dependence of gas pressure (*p*) and temperature (*T*) in cylinder on hydrogen concentration and air excess coefficient

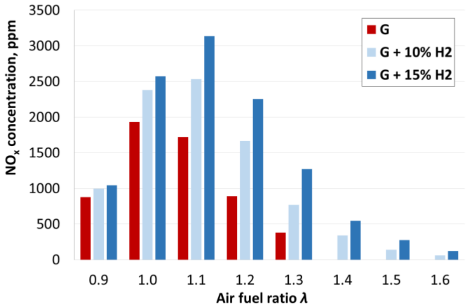

Fig. 9. Dependence of nitrogen oxide concentration in combustion products on hydrogen concentration and air excess coefficient

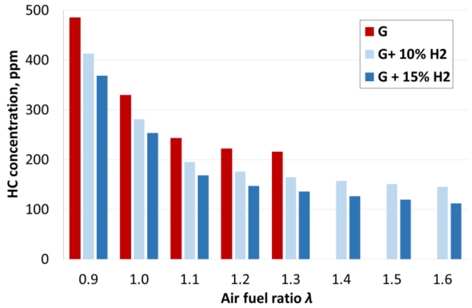

Fig. 10. Dependence of hydrocarbon concentration in combustion products on hydrogen concentration and air excess coefficient

Hydrocarbon concentration in exhaust gases decreases, when leaning the mixture and increasing the concentration of additionally supplied hydrogen (Fig. 10). The mixture with hydrogen can combust even at high air excess ($\lambda = 9.85$), it is characterized by high diffusion (Gupta, 2009). Furthermore, it increases combustion and temperature and, thus, improves oxidation of hydrocarbons. 10% of hydrogen, according to volume in intake air, at different richness of combustible mixture, reduces HC concentration by 15...28%, 15% of hydrogen reduces HC by 23...44%. By comparing the hydrocarbon emission as engine runs on gasoline ($\lambda = 1$) with emission as engine runs on G + 10% H_2 mixture ($\lambda = 1.6$), the concentration of pollutants decreases 56%.

Conclusions

The completed complex experimental and numerical modelling studies of spark ignition engines, as well as analysis of changes of their performance indicators, while using gasoline and the mixture of gasoline and hydrogen fuels, enabled to present the following conclusions on influence of hydrogen gas on energy and environmental indicators of internal combustion engines:

1. Hydrogen, additionally supplied to intake manifold, occupies a certain volume, thus, reducing cylinder filling with air and, in turn, results in decrease of engine power.
2. As hydrogen concentration in combustible mixture increases, intensity and duration of combustion changes, combustion process deviates from the optimum one, and this reduces indicator coefficient of performance. Thermal efficiency of the engine is also reduced by combustion temperature, increased by hydrogen, since larger part of thermal energy of fuel is transferred to environment.
3. By additional supply of hydrogen and by leaning the combustible mixture, it is possible to obtain the optimum indicator coefficient of performance for a cycle without additional adjustment of combustion start.

4. The additive of 15% H_2 in air increases NO_x concentration in exhaust gas up to 60...70% at $\lambda = 1.1...1.3$. However, by comparing the emission of nitrogen oxides as engine runs on gasoline ($\lambda = 1$) with emission as engine runs on G + 10% H_2 mixture ($\lambda = 1.6$), the concentration of pollutants decreases 96%, when the power goes down only 46%.

5. Hydrogen improves hydrocarbon combustion. 10% of hydrogen, according to volume in intake air, in comparison with pure gasoline at $\lambda = 1.1...1.6$, reduces HC concentration by 15...28%, 15% of hydrogen reduces HC by 23...44%.

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