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MAIN CONDITIONS AND EFFECTIVENESS OF GAS FUEL USE FOR POWERING OF DUAL FUEL IC SELF-IGNITION ENGINE

Summary. Internal combustion engines are fuelled mostly with liquid fuels (gasoline, diesel). Nowadays the gaseous fuels are applied as driving fuel of combustion engines. In case of spark ignition engines the liquid fuel (petrol) can be totally replaced by the gas fuels. This possibility in case of compression engines is essentially restricted through the higher self-ignition temperatures of the combustible gases in comparison to classical diesel oil. Solution if this problem can be achieved by using of the dual fuel system, where for ignition of the prepared fuel gas - air mixture a specified amount of the liquid fuel (diesel oil) should be additionally injected into the combustion chamber. For assurance that the combustion process proceeds without mistakes and completely, some basic conditions should be satisfied. In the frame of this work, three main aspects of this problem are taken into account: a. filling efficiency of the engine, b. stoichiometry of the combustion, c. performance of mechanical parameters (torque, power). A complex analysis of these conditions has been done and some achieved important results are presented in the paper.

PODSTAWOWE UWARUNKOWANIA ORAZ EFEKTYWNOŚĆ WYKORZYSTANIA PALIWA GAZOWEGO DO NAPĘDU SILNIKA SPALINOWEGO Z ZAPŁONEM SAMOCZYNNYM

Streszczenie. Silniki spalinowe są zwykle zasilane paliwami ciekłymi (benzyna, ON). Aktualnie do napędu tych silników często wykorzystywane są paliwa gazowe, przy czym zazwyczaj stosowany jest zapłon iskrowy. W przypadku silników z zapłonem samoczynnym wymagane jest użycie podwójnego systemu paliwowego (tzw. systemu dualnego), w ramach którego łącznie z paliwem gazowym podawana jest nieduża dawka paliwa ciekłego, inicjująca zapłon przygotowanej gazowo-powietrznej mieszanki palnej. W celu zapewnienia, że proces spalania będzie przebiegać prawidłowo, a pracujący silnik osiągnie wymagany moment obrotowy (moc efektywną), wiele uwarunkowań powinno być ściśle dotrzymanych. W artykule przeprowadzono teoretyczno-obliczeniową analizę uwarunkowań związanych z prawidłowym funkcjonowaniem silnika spalinowego, wyposażonego w dwupaliwowy układ zasilania w aspekcie uwarunkowań stechiometrycznych oraz pozyskiwanego momentu obrotowego.

1. INTRODUCTION

Internal combustion engines (both spark and compression ignition) are fuelled mostly [1] with liquid fuels (petrol, diesel oil). Nowadays more and more the gaseous fuels (for instance: natural gas (NG), coal bed gas (CBG), compressed natural gas (CNG), synthetic natural gas (SNG), liquefied petroleum gas (LPG) and biogas (BG)) are applied as driving fuel of IC engines [2, 3, 5].

One of the numerous gases which can be used for fueling of the IC engines is the methane CH_4 and next the natural gas (NG);its main component is methane, whereby mostly $CH_4 \approx (88 - 95)$ %.

In case of spark ignition engines, in which the ignition of the earlier prepared vaporised fuel-air mixture is realised by the spark energy source, the liquid fuel (petrol) can be replaced totally without additional troubles by the gas fuels. This possibility is essentially restricted in case of compression engines, because ignitability of the gaseous fuels is mostly not so good as ignitability of the diesel oil. Self-ignition of the fuel gas appears only at considerable higher temperatures in comparison to self-ignition of classical diesel oil. Solution if this problem can be achieved by using of the dual fuel system. The diesel engine will be fulfilled with the fuel gas basically, but for ignition of the prepared fuel gas-air mixture a specified amount of the liquid fuel (diesel oil) should be additionally injected into the combustion chamber [4, 6, 7, 9].

The main objective of this study is a general analysis of conditions concerning proper functioning of the internal combustion engine, equipped with a dual fuel system. From the operation conditions of the system is apparent, that the internal combustion engine equipped with a dual fuel system should deliver the required effective torque and adequate effective power (for given engine speed $\dot{r}_0$).

For assurance that the whole combustion process in the engine cylinder proceeds smoothly, evenly and completely some basic conditions should be satisfied [3, 4, 8].

In the frame of this work, three main aspects of this problem are taken into account [5, 6]:

a) filling efficiency of the engine cylinder (amount of the intake fresh air and fuel dose),
b) stoichiometry of the combustion process (combustion mixture, air excess ratio),
c) performance of mechanical parameters (torque, power output, engine rotational speed).

A complex (quality and quantity) analysis of given basic conditions connected with the dual fuel combustion process has been elaborated and results for the selected gases are presented in the paper.

The obtained results allow to assess the appropriateness of use of the different available gaseous fuels, and as well appropriate programming of experimental research in the field of rational use of different gases as the driving force of the internal combustion engines.

2. BASIC PERFORMANCE PARAMETERS OF DIESEL ENGINE

The compression ignition IC engine at the normal working state (Fig. 1) can be characterised by the basic mechanical quantities and output parameters: $N_{e,0}$ - effective power, $M_{e,0}$ - effective torque, $\dot{r}_0$ - range of revolution number, $\eta_{e,0}$, - energy efficiency, and $\dot{m}_{b,0}$ - mass flux of the consumed fuel. On the input side of the engine important is $\dot{m}_{a,0}$ - mass flux of the intake air (can be expressed as $\dot{n}_{a,0}$, *kmol/s*), which influences directly the filling ratio of the engine and each cylinder.

The effective efficiency of the working internal combustion engine is defined as:

$$\eta_{e,0} = \frac{N_{e,0}}{\dot{m}_{b,0} \cdot H_{u,b}}, \tag{1}$$

where: $H_{u,b}$, *kJ/kg,* - is the lover heating value of the liquid fuel.

The combustion process proceeds at the required value of the air excess ratio, calculated as:

$$\lambda_0 = \frac{\dot{m}_{a,0} \cdot z_{a,O_2}}{\dot{m}_{b,0} \cdot n'_{O_2,min,b} \cdot M_a} = \frac{\dot{n}_{a,0} \cdot z_{a,O_2}}{\dot{m}_{b,0} \cdot n'_{O_2,min,b}}, \tag{2}$$

where: $n'_{O_2,min,b}$, *kmolO₂/kg b.*, - minimum specific oxygen demand of the liquid fuel, z_{a,O_2} - content of the oxygen in the intake air (~ 0.21), M_a, *kg/kmol*, - molar mass of filling air ($M_a \approx 29.1$ *kg/kmol*).

The following analysis will be made for one cycle (one period) of work and in relation to one cylinder of the combustion engine; and so adequate to eq. (2) it can be written:

$$\lambda_0 = \frac{n_{a,0} \cdot z_{a,O_2}}{m_{b,0} \cdot n'_{O_2,min,b}} \quad . \tag{3}$$

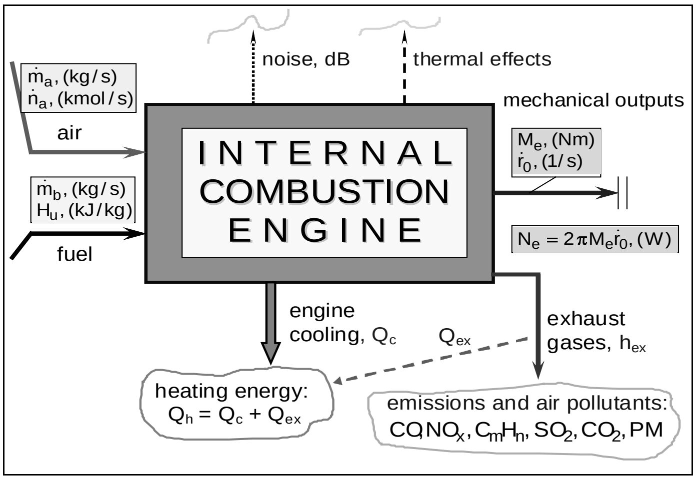

Fig. 1. Classic IC engine system and its parameters
Rys. 1. Klasyczny układ silnika spalinowego i jego parametry

The mass of the air filling each cylinder in one period results from the state equation of the gas:

$$n_{a,0} = \frac{p_1 \cdot V_0}{z \cdot (MR) \cdot T_1} , \tag{4}$$

where: (MR) = 8314.3 *J/(kmol K)* - is the universal constant of the gas, (p_1, T_1) - thermal parameters of the air at the end of the filling process, $n_{a,0}$, *kmol/cycle,* - amount of the air fed into the each engine cylinder during the filling process, V_0 - volume of engine cylinders, z - number of engine cylinders.

Based on the equations (3), (4) it can be found, that using the essential combustion condition ($\lambda_0 \geq 1$) the acceptable amount $m_{b,0}$ of the injected liquid fuel equals;

$$m_{b,0} = n_{a,0} \cdot \frac{z_{a,O_2}}{\lambda_0 \cdot n'_{O_2,min,b}} = \frac{z_{a,O_2}}{\lambda_0 \cdot n'_{O_2,min,b}} \cdot \frac{p_1 \cdot V_0}{z \cdot (MR) \cdot T_1} . \tag{5}$$

The effective torque $M_{e,0}$ should be interpreted as the work performed on 1 radian of rotating shaft. It can be determined, using relation (1) from equation:

$$M_{e,0} = \frac{z}{2\pi \cdot k} \cdot \frac{H_{u,b} \cdot z_{a,O_2}}{\lambda_0 \cdot n'_{O_2,min,b}} \cdot n_{a,0} \cdot \eta_{e,0} , \tag{6}$$

whereby the amount $n_{a,0}$ of the fresh air results from eq. (4), and k - calculation parameter (k ≡ 2 for the 4-stroke engine, and k ≡ 1 for the 2-stroke engine). The performance characteristic of the IC engine includes all values of basic mechanical quantities and output parameters.

3. DIESEL ENGINE WORKING WITH THE DUAL FUEL SYSTEM

It will be assumed that combustion engine is fulfilled basically with the fuel gas (e.g. biogas), but for ignition of the prepared gas-air mixture a specified amount of the liquid fuel (diesel oil) should be injected additionally into the combustion chamber.

The scheme of analysed diesel engine equipped with the dual fuel system is shown in the Fig. 2.

For one cycle of the engine work following amounts of substance are fed into the cylinder: n_g, *kmol/Periode,* - amount of the fuel gas (e.g. biogas), n_a, *kmol/Periode,* - amount of the fresh air, m_b, *kmol/Periode,* - mass of the liquid fuel (diesel oil).

In the frame of these three main aspects of the problem are taken into account: a. filling of the engine cylinder, b. stoichiometry of the combustion process, c. performance of mechanical parameters (torque, power output, engine rotational speed).

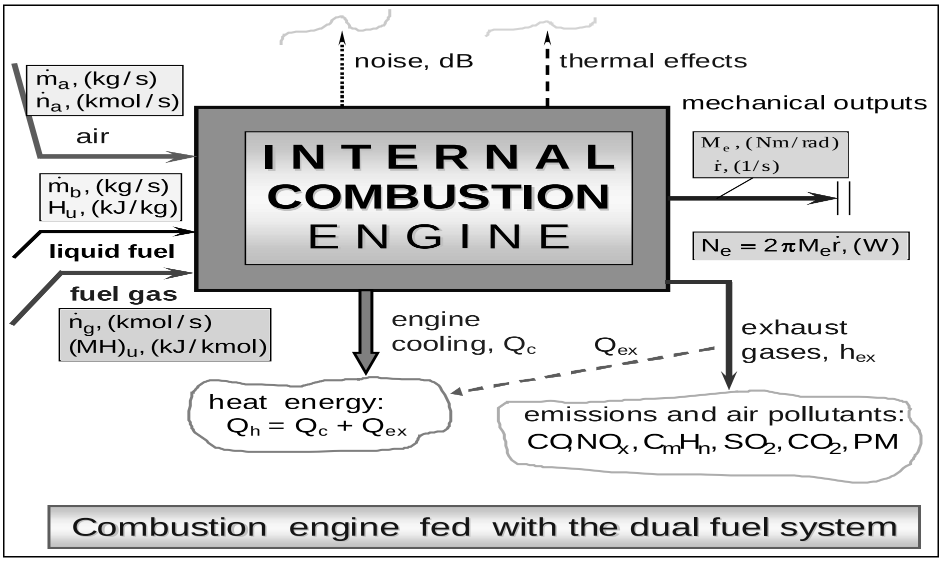

Fig. 2. Internal combustion engine with a dual fuel supply system
Rys. 2. Silnik spalinowy z dwupaliwowym systemem zasilania

After the cylinder filling there is the air-gas mixture in this volume, for which the state equation can be written in form:

$$p_1 \cdot V_0 = z \cdot (n_a + n_g) \cdot (MR) \cdot T_1 \,. \tag{7}$$

From equations (4) and (7) it can be achieved, that: $n_{a,0} = (n_a + n_g)$, and also:

$$\frac{n_g}{n_{a,0}} + \frac{n_a}{n_{a,0}} = 1 \,. \tag{8}$$

It will be assumed, that the filling ratio (volumetric filling efficiency) of each cylinder stays constant in these two cases. The combustion process does not begin till then the liquid fuel is injected.

Main stoichiometric conditions for the dual fuel system

For assurance that the whole combustion process in the engine cylinder proceeds completely, some basic stoichiometric conditions should be satisfied.

The essential amount of the oxygen staying in the cylinder after filling process will be consumed by the fuel gas as well as by the injected liquid fuel.

The basic stoichiometric relation, adequate to the eq. (5) can be written as follows:

$$\lambda \cdot (m_b \cdot n'_{O_2,min,b} + n_g \cdot n'_{O_2,min,g}) = n_a \cdot z_{a,O_2} \,, \tag{9}$$

where: λ - is the oxygen (air) excess number of the whole dual combustion, $n'_{O_2,min,g}$, *kmol O_2/kg g.,* - the minimum specific oxygen demand of the fuel gas.

Using the equations (3) and (5) the equation (9) can be written in following form:

$$\left(\frac{n_g}{n_{a,0}}\right)\cdot\left[1+\lambda_0\cdot\frac{n'_{O_2,min,g}}{z_{a,O_2}}\cdot\left(\frac{\lambda}{\lambda_0}\right)\right]-\left[1-\left(\frac{\lambda}{\lambda_0}\right)\cdot\left(\frac{m_b}{m_{b,0}}\right)\right]=0 \tag{10}$$

On the ground of the relation (10) it can be concluded, that each systematic growing of the amount of the fuel gas ($n_g\uparrow$) leads to decreasing of the mass of the injected liquid fuel ($m_b\downarrow$). This dependence given by equation (10) for the whole region of main parameters is shown in the Fig. 3.

The investigated function takes the explicit form:

$$\left(\frac{n_g}{n_{a,0}}\right)=\frac{1}{(1+R\cdot S)}\cdot\left[1-R\cdot\left(\frac{m_b}{m_{b,0}}\right)\right], \tag{11}$$

whereby the relative parameters:

$$R\overset{df}{=}\left(\frac{\lambda}{\lambda_0}\right), \qquad S\overset{df}{=}\lambda_0\frac{n'_{O_2,min,g}}{z_{a,O_2}}. \tag{12}$$

determine the effectiveness of the substituting of liquid fuel by the different gaseous fuels.

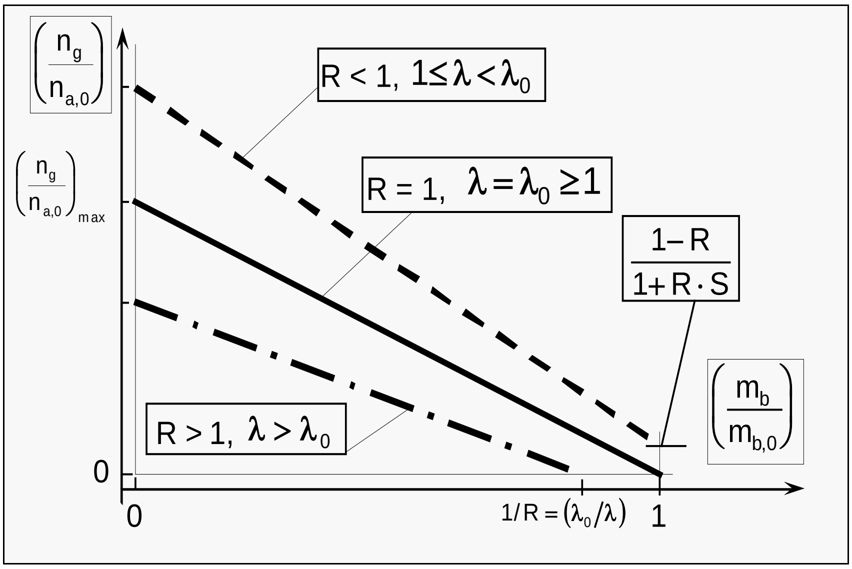

Fig. 3. Relative amount of gaseous fuel and liquid fuel in the engine cylinder
Rys. 3. Względna ilość paliwa gazowego oraz ciekłego w cylindrze silnika

The maximal amount $n_{g,max}$ of the used fuel gas can be determined from equation (11) when the value of injected liquid fuel equals zero, it means that ($m_b \to 0$), and than:

$$\left(\frac{n_g}{n_{a,0}}\right)_{max}=\frac{1}{1+R\cdot S}. \tag{13}$$

If the combustion process by the dual fuel system proceeds stoichiometric (at $\lambda\to1$), than from the equation (13) results the maximum of all maximal values. On the other side the minimum value of the relation ($n_g/n_{a,0}$) equals zero, i.e. $(n_g/n_{a,0})_{min} = 0$, if the air excess ratio is kept ($\lambda = \lambda_0$) without change.

If the values of the air excess ratio λ belong to the range $1 \le \lambda < \lambda_0$ than parameter $R < 1$ and the minimum values of the relative amount of the supplied intake gas ($n_g/n_{a,0}$) are as follows:

$$\left(\frac{n_g}{n_{a,0}}\right)_{min} = \frac{1-R}{1+R\cdot S}\,, \qquad (1/\lambda_0) \le R < 1 \tag{14}$$

and can be achieved when $(m_b/m_{b,0}) = 1$.

For another values of the air excess ratio λ, i.e. if $\lambda > \lambda_0$ than parameter R > 1, and the minimum values of the relative gas amount $(n_g/n_{a,0})$ equal $(n_g/n_{a,0})_{min,min} = 0$ and can be achieved at $(m_b/m_{b,0}) = (\lambda_0/\lambda)$. These characteristic values are shown in the Fig. 3 too.

Additionally the minimum relative amount $(m_b/m_{b,0})_{min}$ of injected liquid fuel should be established too. This can be determined only experimentally and individually for each applied fuel gas as well as for the concrete combustion engine (e.g. depending on its compression ratio).

Basic input-output energy conditions for the dual fuel system

Combustion engine working with the dual fuel system has to perform simultaneously the needed effective mechanical torque M_e and power output N_e at given engine speed $\dot{r}_e$.

For one cycle of engine work, determined amount of chemical energy E_{ch} will be delivered into the combustion chamber of the combustion engine, which is calculated as:

$$E_{ch} = m_b \cdot H_{u,b} + n_g \cdot (MH_u)_g \;, \tag{15}$$

whereby: $(MH_u)_g$, *kJ/kmol*, - is lover heating value of the supplied gas fuel.

Chemical energy (15) supplied into the IC engine influences directly the mechanical engine performance parameters: torque M_e and power output N_e at given engine speed $\dot{r}_e$.

Dependencies characterising the combustion engine working with the dual fuel system, take now adequate to the equation (6) the following forms:

- for the effective torque M_e

$$M_e = \eta_e \cdot \frac{z}{2\pi \cdot k} \cdot \left[m_b \cdot H_{u,b} + n_g \cdot (MH_u)_g \right], \tag{16}$$

- for the effective power output N_e

$$N_e = \eta_e \cdot \frac{z \cdot \dot{r}}{k} \cdot \left[m_b \cdot H_{u,b} + n_g \cdot (MH_u)_g \right]. \tag{17}$$

The equations (17) and (6), after using equation (5) can be put in common formula:

$$\left(\frac{\eta_{e,0}}{\eta_e}\right) \cdot \left(\frac{M_e}{M_{e,0}}\right) = \left(\frac{m_b}{m_{b,0}}\right) + \lambda_0 \cdot \frac{(MH_u)_g}{H_{u,b}} \cdot \frac{n'_{O_2,min,b}}{z_{a,O_2}} \cdot \left(\frac{n_g}{n_{a,0}}\right) \;. \tag{18}$$

The mechanical power output N_e can be determined by using equations (17), (5) and written as:

$$\left(\frac{\eta_{e,0}}{\eta_e}\right) \cdot \left(\frac{N_e}{N_{e,0}}\right) = \left(\frac{\dot{r}}{\dot{r}_0}\right) \cdot \left[\left(\frac{m_b}{m_{b,0}}\right) + \lambda_0 \cdot \frac{(MH_u)_g}{H_{u,b}} \cdot \frac{n'_{O_2,min,b}}{z_{a,O_2}} \cdot \left(\frac{n_g}{n_{a,0}}\right) \right]. \tag{19}$$

The main unknown quantities (n_g, M_e, N_e) of the engine working with the dual fuel system can be evaluated using the above given equations system (10), (18), (19).

The relative portion of injected liquid fuel $(m_b/m_{b,0})$ and amount of gas fuel $(n_g/n_{a,0})$ are additionally connected - eq. (11), through the range (Fig. 3) of the effective air excess ratio $\lambda \ge 1$.

4. FUEL GAS APPLICATION AND EFFECTIVENESS OF THE ENGINE POWERING

The mechanical engine performance parameters (torque M_e and power output N_e at given engine speed $\dot{r}_e$) are influenced by relative amount of fuel gas $(n_g/n_{a,0})$, used as driving force of the engine.

The evaluated equation (18), which expresses the direct influence of the relative amount $(n_g/n_{a,0})$ of fuel gas used in the fuel dual system on the engine torque $(M_e/M_{e,0})$, can be given in a short form as:

$$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right)=\left(\frac{m_b}{m_{b,0}}\right)+E\cdot\left(\frac{n_g}{n_{a,0}}\right), \tag{20}$$

whereby:

$$E\overset{df}{=}\lambda_0\cdot\frac{(MH_u)_g}{H_{u,b}}\cdot\frac{n'_{O_2,min,b}}{z_{a,O_2}}. \tag{21}$$

The energy eq. (20) should be analysed together with the stoichiometric relation (11), therefore after setting equation (11) into equation (20), the searched relation is:

$$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right)=\frac{1}{R}+\left[E-\left(S+\frac{1}{R}\right)\right]\cdot\left(\frac{n_g}{n_{a,0}}\right). \tag{22}$$

From the equation (22) it can be deducted that depending of the parameters: E, R, S) different influences of the fuel gas amount ($n_g/n_{a,0}$) on the performed engine torque ($M_e/M_{e,0}$) will be indicated.

The shape of the achieved relation (22) has been shown in the Fig. 4.

Three typical cases can be observed (Fig. 4) by gradual replacement of the liquid fuel ($m_b/m_{b,0}$) through the gaseous fuel ($n_g/n_{a,0}$) and by keeping the effective air excess ratio λ, parameter R:

- relative engine torque $[(\eta_{e,0}\cdot M_e)/(\eta_e\cdot M_{e,0})]$ grows if relation: E > (S+1/R) occurs,
- relative engine torque $[(\eta_{e,0}\cdot M_e)/(\eta_e\cdot M_{e,0})]$ drops for values: E < (S+1/R),
- relative torque $[(\eta_{e,0}\cdot M_e)/(\eta_e\cdot M_{e,0})]$ remains unchanged if parameters: E = (S+1/R).

For the case when E = (S + 1/R) the specific relative air excess ratio $(\lambda/\lambda_0)_{gr}$ equals:

$$\left(\frac{\lambda}{\lambda_0}\right)_{gr}=\frac{z_{a,O_2}}{\lambda_0\cdot n'_{O_2,min,b}\cdot\left[\left(\frac{(MH_u)_g}{H_{u,b}}\right)-\left(\frac{n'_{O_2,min,g}}{n'_{O_2,min,b}}\right)\right]} \tag{23}$$

whereby the condition $\lambda \geq 1$ should be taken into account.

For given air excess ratio λ the relative amounts of fuels: - liquid ($m_b/m_{b,0}$) and - gaseous ($n_g/n_{a,0}$) can be principally independent selected, but the stoichiometric condition (10) should be fulfilled.

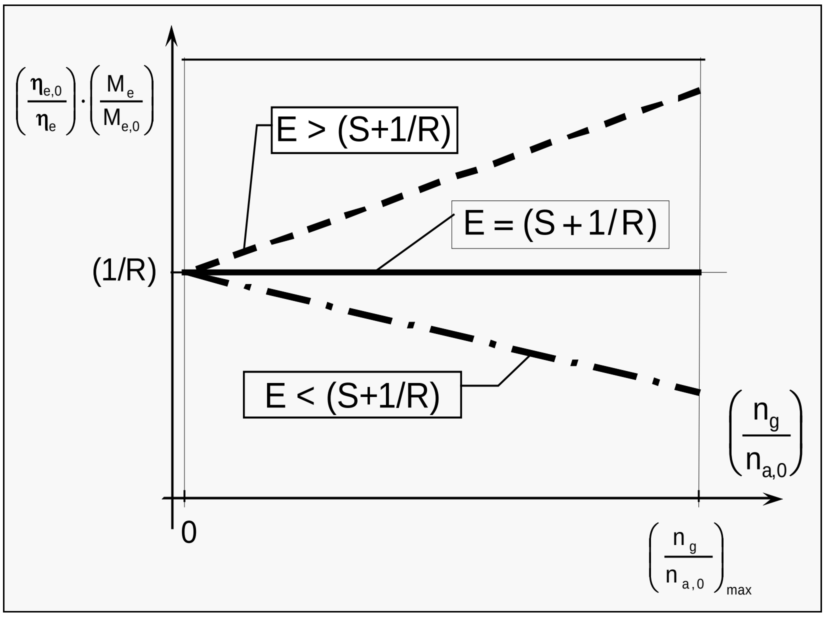

Fig. 4. Performance parameters of the engine working with the dual fuel system
Rys. 4. Parametry eksploatacyjne silnika spalinowego zasilanego dwupaliwowo

Example of the performed analysis for chosen gas fuels

The sample calculations, verifying the correctness of elaborated algorithm – relations (20), (22) were performed, to confirm the influence of the different criterial parameters (E, R, S) on the effectiveness of the liquid fuel ($m_b/m_{b,0}$) substituting by the gaseous fuel ($n_g/n_{a,0}$).

The methane CH_4 and natural gas ($CH_4 \approx 88$ - 95 %) can be used for fuelling of the IC engines. The calorific value equals $(MH_u)_g = 805.2$ *MJ/kmol*, and oxygen demand $n'_{O_2,min,g} = 2$ *kmol O_2/kmol g*.

Coal-Bed-Gas ($CH_4 \approx 35$ - 45 % and (N_2, O_2) are the rest) and Bio-Gas are also very adequate gas fuels, which can be used as driving force of IC engine. Coal-Bed-Gas is obtained during the methane removal from coal seams, whereas Bio-Gas is spontaneous generated in municipal landfills systems.

The Bio-Gas is a gaseous mixture, which contains combustible (mainly: methane CH_4, and H_2S, CO, C_mH_n) components, as well as inert (non-combustible: e.g. carbon dioxide CO_2, nitrogen N_2) compounds. As a model composition of the dry Bio-Gas will be accepted the following values: methane $[CH_4] = 40$ %, and the rest as carbon dioxide $[CO_2] \cong 60$ %.

It was assumed for calculations the same content of the methane $CH_4 = 40$ % - both in the Bio-Gas as well in the Coal-Bed-Gas, so then the lower heating value of each gas is the same, about $(MH_u)_g = 325.1$ *MJ/kmol g*. The minimum oxygen demand $n'_{O_2,min}$ for Coal-Bed-Gas equals about $n'_{O_2,min,C-B-G} \approx 0.672$ *kmol O_2/kmol g.*, and for Bio-Gas $n'_{O_2,min,B-G} \approx 0{,}804$ *kmol O_2/kmol g.*

The chosen properties of the liquid fuel: - calorific value $H_{u,b} = 42.5$ *MJ/kg*, - minimum oxygen demand: $n_{O2,min,b} = 0.112$ *kmol O_2/kg b.*, - predetermined referential air excess ratio equals: $\lambda_0 = 1.8$.

Achieved values of decisive parameters (S, R – eq. (12), E – eq. (21)) for the reference point are:

- for Bio-Gas: $S_{B-G} = 6.89$ E = 7.32 R = 1.0 [E – (S+1/R)] = - 0.568
- for Coal-Bed-Gas: $S_{C-b-G} = 5.76$ E = 7.32 R = 1.0 [E – (S+1/R)] = 0.563

In this case (reference medium load of the IC engine) the analyzed function (22) takes the form:

- for Bio-Gas: $$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right) = 1 - 0.568\cdot\left(\frac{n_g}{n_{a,0}}\right), \quad \text{and} \tag{24}$$

- for Coal-bed-Gas: $$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right) = 1 + 0.563\cdot\left(\frac{n_g}{n_{a,0}}\right). \tag{25}$$

The increase in the share of the relative amount (chemical energy) of supplied gas fuel can in the reference state (**$\lambda = \lambda_0$** , whereby **$\lambda_0 = 1.8$**) of the engine work influence different (depend on the gas properties) on the achieved relative engine torque (Fig. 5).

The achieved results inform, that by substituting the liquid fuel: - if the supplied gas is the Coal-Bed-Gas then can be indicated a significant growth of the engine torque (Fig. 5), and conversely; - if the supplied gas is the Bio-Gas then will be observed a clear decrease of the relative engine torque. The decisive factor in this regard is the value of the criteria expression [E – (S+1/R)].

At stoichiometric conditions (high load of the IC engine), when oxygen (air) excess ratio equals: $\lambda = 1.0$ then the typical influence parameters (at $\lambda_0 = 1.8$) are:

- for Bio-Gas: $S_{B-G} = 6.89$ E = 7.32 R = 0.5556 [E – (S+1/R)] = - 1.371
- for Coal-Bed-Gas: $S_{C-b-G} = 5.76$ E = 7.32 R = 0.5556 [E – (S+1/R)] = - 0.240 .

For analyzed stoichiometric case (full load of the IC engine, rich combustible mixture, $\lambda \approx 1{,}0$) the performing function (22) takes the form:

- for Bio-Gas: $$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right) = 1{,}8 - 1.371\cdot\left(\frac{n_g}{n_{a,0}}\right), \tag{26}$$

- for Coal-Bed-Gas: $$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right) = 1{,}8 - 0.240\cdot\left(\frac{n_g}{n_{a,0}}\right). \tag{27}$$

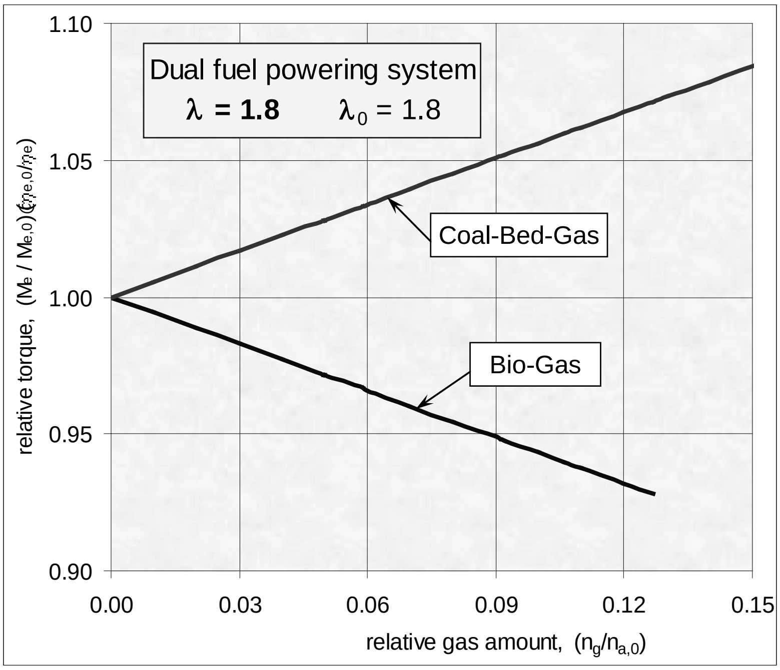

Fig. 5. Influence of gaseous fuels on the relative torque of the engine working at the reference state
Rys. 5. Wpływ ilości paliwa gazowego na moment obrotowy silnika przy jego nominalnym obciążeniu

The increase in the share of the relative amount (and proportional of the chemical energy) of supplied gas fuel (Fig. 6) can at the stoichiometric state (for $\lambda = 1.0$ at $\lambda_0 = 1.8$) of the engine work directly influence (depend on the gas properties) on the achieved relative engine torque.

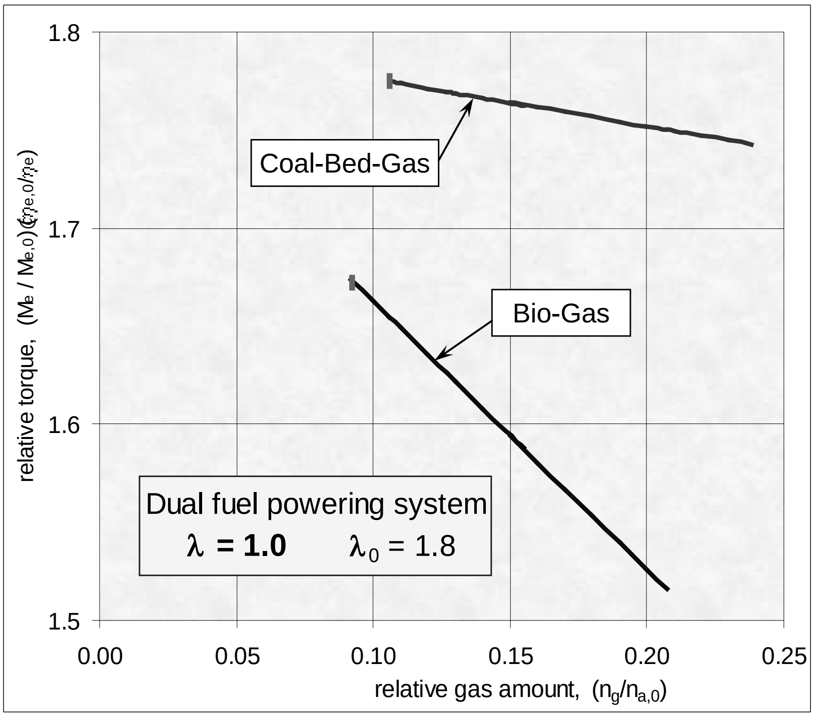

Fig. 6. Influence of gaseous fuels on the relative torque of the IC engine working at the full load state
Rys. 6. Wpływ ilości paliwa gazowego na osiągany moment obrotowy silnika przy jego pełnym obciążeniu

By substituting the liquid fuel by the gas fuels at full load (λ = 1.0) of the IC engine - in analyzed two cases (Coal-Bed-Gas, Bio-Gas) increasing of the supplied gas fuel amount (the relative amount of diesel drops respectively) leads to effective decreasing (Fig. 6) of the achieved relative engine torque.

At the low load of the IC engine the lean combustible mixture (λ >> 1.0) for IC engine powering should be used, e.g. keeping the oxygen excess ratio on the level $\lambda \approx 5.0$.

In this characteristic case the basic parameters are:

- for Bio-Gas: S_{B-G} = 6.89 E = 7.32 R = 2.778 [E – (S+1/R)] = 0.071
- for Coal-Bed-Gas: S_{C-b-G} = 5.76 E = 7.32 R = 2.778 [E – (S+1/R)] = 1.201

In this case (lean combustible mixture, low load of the IC engine) the function (22) takes the form:

- for Bio-Gas:

$$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right)=0{,}36+0.071\cdot\left(\frac{n_g}{n_{a,0}}\right), \quad \text{and} \tag{28}$$

- for Coal-Bed-Gas:

$$\left(\frac{\eta_{e,0}}{\eta_e}\right)\cdot\left(\frac{M_e}{M_{e,0}}\right)=0{,}36+1.201\cdot\left(\frac{n_g}{n_{a,0}}\right). \tag{29}$$

The increase in the share of the relative amount (chemical energy) of supplied gas fuel can in the reference state ($\lambda = \lambda_0 = 1.8$) of the engine work influence different (depend on the gas properties) on the achieved relative engine torque (Fig. 7).

The achieved results inform, that by substituting of the liquid fuel by the gas fuel, at two analyzed cases (Coal-Bed-Gas, Bio-Gas) of the supplied gas, the systematic growth (Fig. 7) of the engine torque is occurred. This is specially important for the Bio-Gas effective utilization.

The decisive factors in this regard are the positive values of the criteria expression [E – (S+1/R)].

It is very important to emphasize that by growing of the relative gas fuel amount parallel the relative liquid fuel (diesel) dose should be respectively diminished – eq. (10), and Fig. 3, because the oxygen excess ratio λ should be kept on the set up level λ = idem.

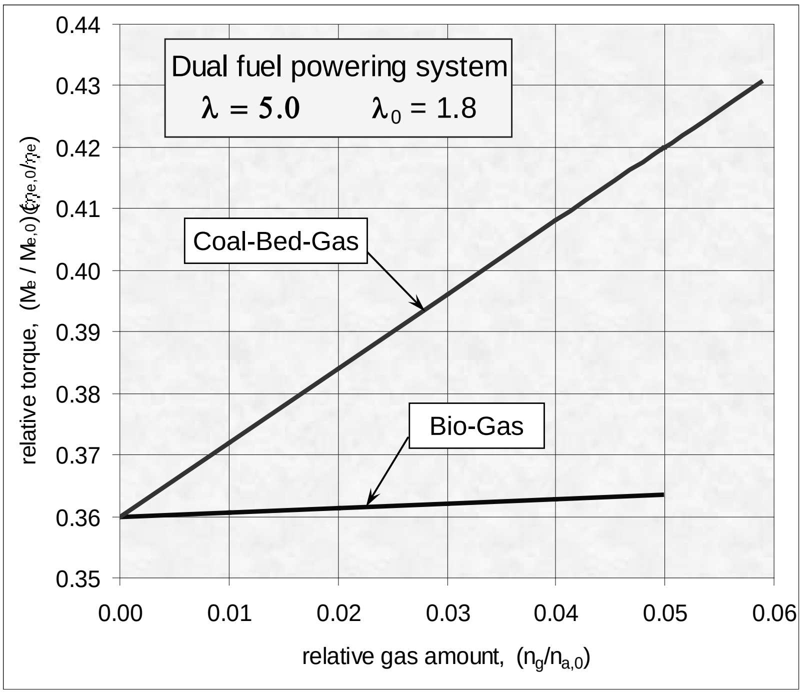

Fig. 7. Influence of gaseous fuels on the relative torque of the IC engine working at the low load state
Rys. 7. Wpływ ilości paliwa gazowego na osiągany moment obrotowy silnika przy jego niskim obciążeniu

The indicated different possibilities can be also confirmed in the Fig. 8, where for assumed data, is presented the performance characteristic of the IC engine, equipped with a dual fuel system (separately for the Bio-Gas – on the top and for Coal-Bed-Gas – at the bottom).

The above given examples show that (e.g. keeping λ = idem) the gradual replacement of liquid (diesel) fuel by the gas fuel causes different effects (depending on the gas type and kind of combustible mixture: lean, medium, rich) with respect to obtained relative effective engine torque.

Concrete values of the parameters E, resulting from eq. (21) and similarly S – eq. (12) - depend mainly from the fuel properties, while the quantity R – eq. (12) is especially important from the point of view of the qualitative IC engine control (changes of the air excess ratio λ) and work adjustment.

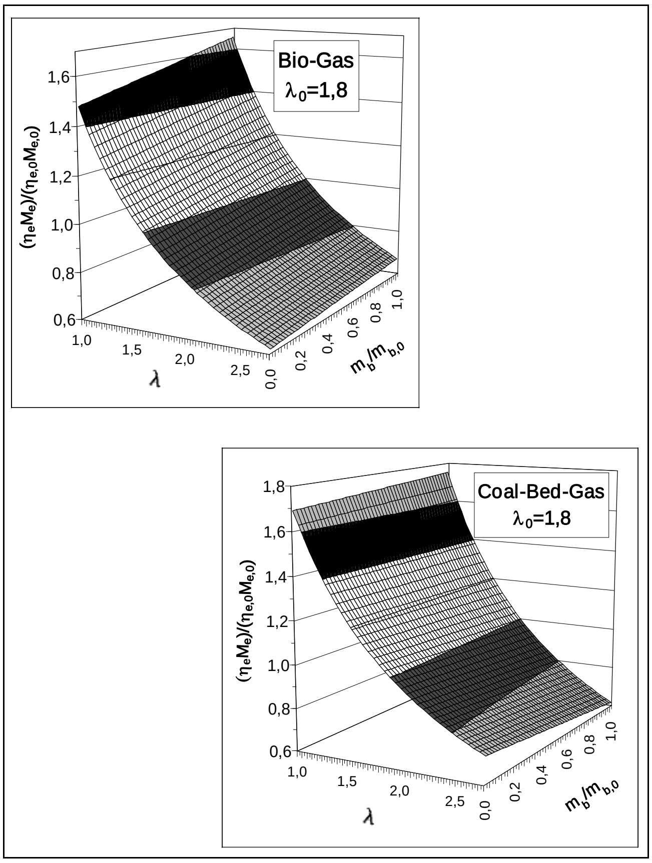

Fig. 8. Effectiveness of the use of different gaseous fuels for IC engines powering
Rys. 8. Efektywność silnika spalinowego zasilanego różnymi paliwami gazowymi

Additional, important practical problem deals to the minimum relative amount $(m_b/m_{b,0})_{min}$ of the injected liquid fuel, which in each case of engine load (at different oxygen excess ratio (λ = var), qualitative IC engine control) should obtain reliable self-ignition of the prepared diesel-air-gas combustible mixture. The task may be solved for given IC engine power station on the experimental way.

Numerous experiments carried out with different IC gas-diesel engines lead to the conclusion that the minimum dose $m_{b,0}$ of the liquid fuel (diesel), which in any case ensure auto-ignition of the combustible mixture, should provide the liquid fuel oxygen excess ratio λ_0 on the level $\lambda_0 = 5.0$.

In the whole ranges of the IC engine performance parameters (engine torque M_e, revolution $\dot{r}_0$) the dose of injected liquid fuel (diesel) should be kept on the level m_b = idem, and $(m_b/m_{b,0}) \equiv 1$, resulting directly from the assumed value of the liquid fuel oxygen excess ratio λ_0.

Using the elaborated algorithm - relation (11) after adding a portion of gas fuel, the actual oxygen excess ratio (λ = var), can be calculated as:

$$\lambda = \lambda_0 \cdot \frac{1 - \left(\dfrac{n_g}{n_{a,0}} \right)}{1 + S \cdot \left(\dfrac{n_g}{n_{a,0}} \right)} \tag{30}$$

where parameter S - eq. (12), depends on the gas fuel composition.

Increase of the delivered gas fuel amount leads systematically to decrease the actual oxygen excess ratio λ of the whole combustible mixture. The minimum value of the oxygen excess ratio λ = 1 will be achieved after delivery of the maximal gas fuel amount, which equals:

$$\left(\frac{n_{g,max}}{n_{a,0}} \right) = \frac{(1 - R_{min})}{(1 + R_{min} \cdot S)} \tag{31}$$

where: $$R_{min} = \frac{1}{\lambda_0}, \qquad R_{min} \approx 0.2, \quad for \ \lambda_0 = 5.0 \ . \tag{32}$$

The influence of the relative amount ($n_g/n_{a,0}$) of gas fuel used in the fuel dual system powering on the IC engine relative torque [($\eta_{e,0} \cdot M_e$)/($\eta_e \cdot M_{e,0}$)], can be calculated using eq. (22), whereby the characteristic parameter R - eq. (12) changes its value.

The above given example shows that the regular adding of the new gas fuel portions to the liquid (diesel) fuel, causes characteristic effect (depending on the gas type) of the oxygen excess ratio diminishing - in the range $1 \le \lambda \le \lambda_0$, which is important for the qualitative control of the IC engine.

5. CONCLUSIONS

Diesel engine equipped with the dual fuel (gaseous, liquid) system has to perform simultaneously for the needed effective mechanical torque and power output. For assurance that the combustion process in the engine proceeds without mistakes and completely, some basic conditions should be satisfied.

This problem can be solved by adapting of the effective air excess ratio, depending on the kind and propriety of the used gas and actual mechanical parameters of the engine.

The carried out analysis has shown, that substituting of the good quality liquid fuel (diesel oil) through the gaseous fuel mainly leads to the decreasing of the performed torque M_e (respectively also the effective power output N_e); however specified gases can be identified (e.g.: Coal-Bed-Gas, C_2H_2, CO and other) which causes a different effect. In this respect it is possible, through adjustment of the air excess ratio λ respectively and according to the operational requirements, to correct the achieved values of IC engine performance parameter.

With respect to obtained relative effective engine torque, the analyzed examples show that (by keeping λ = idem) the gradual replacement of liquid (diesel) fuel by the gas fuel may cause different (Fig. 5, 6, 7) effects (depending on the gas type and kind of combustible mixture: lean, medium, rich).

In frame of the qualitative control of the IC engine (minimum liquid fuel amount m_b = idem) increase of the delivered gas fuel amount leads systematically to decrease the oxygen excess ratio λ of the whole combustible mixture.

Nomenclature

$N_{e,0}$ - effective power output, *kW* $M_{e,0}$ - effective torque, *Nm/rad,*

$\dot{r}_0$ - revolution number (engine speed), *rev/s* $\eta_{e,0}$ - effective efficiency,

$\dot{m}_{b,0}$ - mass flux of the fuel, *kg/s* $\dot{m}_{a,0}$ - mass flux of the intake air, *kg/s*

$n'_{O_2,min}$ - minimum oxygen demand, *kmol O_2/(fuel specific amount)*

H_u - lover heating value, *kJ/kg* , *(MH_u)* - lover heating value, *kJ/kmol*
λ - air excess ratio, *(fuel specific amount ≡ 'kg' - for liquid fuel,'kmol' - for gaseous fuel).*

References

1. Benson, R.S. & Horlok, J.H. & Winterbone, D.E. *The thermodynamics and gas dynamics of internal combustion engines*. New York: Oxford University Press. 2002.
2. Henham, A.W.E. & Makkar, M.K. Combustion of simulated biogas in a dual-fuel diesel engine. In: *Florence World Energy Research Symposium 'Clean Energy for the new century'*. Florence, Italy. 1997. P. 1187-1194.
3. Papagiannakis, R.G. & Kotsiopoulos, Zannis, T.C. & Yfantis, E.A. & Hountalas, D.T. & Rakopoulos, C.D. Theoretical study of the effects of engine parameters on performance and emissions of a pilot ignited natural gas diesel engine. *Energy*. 2010. Vol. 35 (2). P. 1129-1138.
4. Postrzednik, S. & Żmudka, Z. Investigation on the improving of ICE energy conversion efficiency at different engine load. J*ournal of KONES – Powertrain and Transport*. 2009. Vol. 16. No. 3. P. 375-384.
5. Postrzednik, S. Engine gas-fuel from coal for Combustion Engines. *Scientific Magazine - Combustion Engines*. 2009. No. SC1. P. 62-67.
6. Postrzednik, S. & Żmudka, Z. & Przybyła, G. Basic conditions and effectiveness of replacement of conventional liquid fuel with gaseous fuel to combustion engine drive. *Scientific Magazine - Combustion Engines*. 2013. No. 3 (154). SC-162. P. 140-141.
7. Rakopoulos, C. & Kyritsis, D. Comparative second-law analysis of internal combustion engine operation for methane, methanol, and dodecanese fuels. *Energy*. 2001. Vol. 26 (7). P. 705-722.
8. Raman, P. & Ram, N.K. Performance analysis of an internal combustion engine operated on producer gas, in comparison with the performance of the natural gas and diesel engines. *Energy*. 2013. Vol. 63(15). P. 317-333.
9. Stelmasiak, Z. Possibility of Improvement of Some Parameters of Dual Fuel CI Engine by Pilot Dose Division. *Journal of Polish CIMAC*. 2011. Vol. 7. No 1. P. 181-190.

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