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Ideal cycle of combustion engine with natural gas as a fuel

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Abstract

Vitázek I., Klúčik J., Jablonický J., Vereš P. (2016): **Ideal cycle of combustion engine with natural gas as a fuel**. Res. Agr. Eng., 62 (Special Issue): S14–20.

The aim of the paper is to present a detailed methodology of calculations of parameters of an ideal working cycle of spark-ignition combustion engine. Natural gas in the form of compressed natural gas (CNG) was used as a fuel. A theoretical ideal cycle is currently described in simplified way. The paper introduces calculations considering excess air, residual space in the cylinder of the engine and the course of properties of gases in dependence on temperature. The thermodynamics of ideal gas mixture was used. A computer program for clear, quick and accurate calculations of this relatively complicated system of relations was designed. The presented methodology of calculations broadens the scope of the theory of combustion engines and enables a precise determination of parameters of combustion engine with natural gas as a fuel.

Keywords: thermodynamic parameters; CNG; stoichiometry; precise calculation; thermal efficiency

Methods of reducing fuel consumption and production of gaseous emissions nowadays pose a challenge for automotive industry. Using natural gas as a fuel is one of the options. Cars are labelled with the abbreviation of compressed natural gas (CNG) and are adapted for this fuel by the manufacturer. In thermodynamics, the theoretical ideal cycle of combustion engine is a fundamental part for further study of the theory of engines (Nožička 2008; Stone 2012; Struchtrup 2014). To ensure uncomplicated calculations, theoretically exact course of the cycle with constant amounts of gases, excluding excess air and residual volume, is assumed (Shavit, Gutfinger 2008; Jablonický et al. 2010; Vitázek 2012). The paper puts forward theoretically ideal working cycle of combustion engine. Excess air, residual space and the course of properties of gases in dependence on temperature are considered in calculations. The computer program was also designed in order to facilitate considerably the usage of presented complex calculations. Regarding the scope of the paper, the program is not dealt with in details. The paper is focused on the working cycle of combustion engine with gaseous fuel (Vitázek et al. 2007). Apart from CNG, liquefied petroleum gas (LPG) and compressed hydrogen are also used. This work is concerned with gaseous fuel – natural gas, because of its widespread use as an ecological fuel (Müllerová et al. 2012, Jablonický et al. 2013, Winterbone, Turan 2015).

MATERIAL AND METHODS

The ideal working cycle of combustion engine with isochoric heat supply is analysed. The analysis draws from the theory of combustion engine (Neuberger et al. 2007, Stone 2012). Basic information about natural gas (CNG) is obtained from the data published by SPP, a.s. (www.spp.sk). Values of en-

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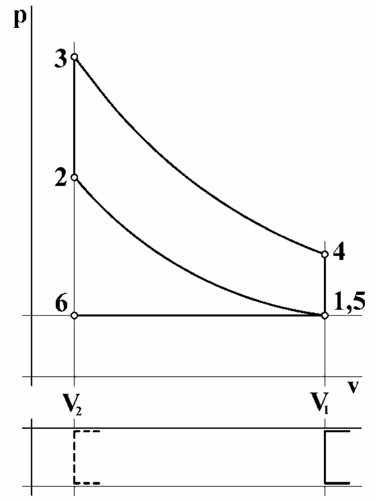

Fig. 1 Theoretical p-v diagram

V_1 – total cylinder volume, V_2 – compression cylinder volume

thalpies of the particular gases are taken from the tables (Ražnievič 1969). All analyses and calculations are presented for mass unit (kg), considering these relations are the simplest. The thermodynamics of ideal gas mixture is used (Struchtrup 2014, Caton 2015). The calculation of a specific case of combustion engine with CNG fuel is carried out by means of the original computer program.

Theoretical diagram of the working cycle. Fig. 1 shows the theoretical diagram of the working cycle of combustion engine with isochoric heat supply in p-v coordinates. Fig. 2 demonstrates the scheme of substance movement in internal combustion engine:

$$m_n = m_A + m_{NG} \tag{1}$$

$$m_1 = m_n + m_{re} \tag{2}$$

$$m_2 = m_C + m_{re} \tag{3}$$

$$m_3 = m_4 = m_2 \tag{4}$$

$$m_5 = m_C \tag{5}$$

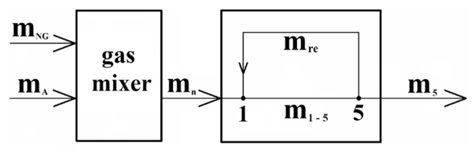

Fig. 2 Scheme of substance movement in internal combustion engine

m_{NG} – mass of natural gas; m_A – mass of atmospheric air, m_n – mass of the mixture of natural gas and atmospheric air; m_{re} – mass of non-removed flue gases; m_{1-5} – mass of working mixture in particular stages of the cycle; m_5 – mass of flue gases

Total cylinder volume is V_1, and compression (residual) cylinder volume is V_2.
Swept volume is:

$$V_S = V_1 - V_2 \tag{6}$$

Compression ratio:

$$\varepsilon = \frac{V_1}{V_2} = \frac{V_1}{V_1 - V_S} \tag{7}$$

For calculation of V_2 volume, the following relation is derived:

$$V_1 = \frac{V_S}{\varepsilon - 1} \tag{8}$$

Fuel – natural gas. Natural gas (NG) for combustion engines is internationally recognized as CNG. Composition and properties for NG (2016) according to SPP, a.s. are presented on their website (http://www.spp.sk/sk/velki-zakaznici/zemny-plyn/o-zemnom-plyne/emisie/).

The parameters of invoiced NG are as follows:
$\rho = 0.7169$ kg/m^3; $Q_i^V = 34.920$ MJ/m^3; $Q_i^m = 48.7097$ MJ/kg.

With composition:

$c = 0.7319$ $h = 0.2392$ $co_2 = 0.0096$ $n = 0.0193$
$M_{NG} = 16.524$

where: ρ – density of NG; Q_i^V calorific value per m^3; Q_i^m – calorific value per kg; c, h, co_2, n – mass fractions of the components (carbon, hydrogen, CO_2, nitrogen); M_{NG} – molar mass

Stoichiometric equations for combustion of 1 kg of NG:

$$1 \text{ kg C} + 2.6642 \text{ kg } O_2 = 3.662 \text{ kg } CO_2 \tag{9}$$

$$1 \text{ kg } H_2 + 7.9381 \text{ kg } O_2 = 8.9381 \text{ kg } H_2O \tag{10}$$

Atmospheric air. Atmospheric air serves as a source of oxygen in the process of combustion of NG. The following mass composition of atmospheric air is introduced in (Chyský 1977):

Dry air:

nitrogen N_2

$\sigma_{N_2} = 75.5\%$ $M_{N_2} = 28.016$ $\kappa_{N_2} = 1.4$

oxygen O_2

$\sigma_{O_2} = 23.20\%$ $M_{O_2} = 32.000$ $\kappa_{O_2} = 1.4$

argon + inert gas

$\sigma_{Ar+} = 1.3\%$ $M_{Ar+} = 40.000$ $\kappa_{Ar+} = 1.67$

carbon dioxide CO_2 $M_{CO_2} = 44.01$ $\kappa_{CO_2} = 1.31$

Moisture:

water vapour $M_{H_2O} = 18.016$ $n_{H_2O} = 1.3$

where: σ – mass fraction of gas component; M – molar mass; κ – izoentropic exponent; n – adiabatic exponent

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In literature, there are presented tables of enthalpies for particular gases up to 3,000°C (Ražnievič 1969). Values for selected ranges of temperature were adopted from these tables. Equations were derived by means of linear regression. These equations are to be found in a computer program with suitable precision. Correlation coefficient always exceeds $r = 0.9999$.

Regarding atmospheric air, the following basic parameters are measured: temperature t_0, relative humidity φ_0, total pressure p_0.

Working mixture. The course of ideal working cycle of combustion engine with spark ignition for 1 kg of NG is described. Engine with compression ratio ε and mixture with excess air α is considered.

The mass of oxygen O_2 for the stoichiometric combustion of 1 kg of NG is:

$$m_{O_2 i} = 2.6642 \times c + 7.9381 \times h = 3.848698 (\text{kg } O_2/\text{kg NG}) \quad (11)$$

Mass of atmospheric air:

$$m_{Ai} = \frac{m_{O_2 i}}{\sigma_{O_2}} = \frac{3.848698}{0.232} = 16.5892 \quad (\text{kg}) \quad (12)$$

where: σ_{O2} – mass fraction of oxygen in the air

Thereby arises:

$$m_{CO_2} = 3.6642 \times 0.7319 = 2.6818 \quad (\text{kg } CO_2) \quad (13)$$

$$\Delta m_W = 8.9381 \times 0.2392 = 2.13799 \quad (\text{kg } H_2O) \quad (14)$$

Sucked-in mixture for 1 kg of NG. The process with excess air m_{Ap} is considered. Hence, the mass of air is:

$$m_A = m_{Ai} + m_{Ap} \quad (15)$$

where:

$$\alpha = m_A/m_{Ai} \quad \text{hence} \quad m_A = \alpha \times m_{Ai} \quad (16, 17)$$

where: m_{Ai} – air mass for ideal combustion of gas; α – surplus of air

The composition of sucked-in mixture is:

$$m_{nNG} = 1 \text{ kg and } m_{nA} = m_{Ai} \times \alpha = 16.5892 \times \alpha \quad (18)$$

$$m_{nW} = m_{nA} \times x_0 \quad (19)$$

where: m_n – mass of the component (gas, dry air, humidity); x_0 – specific humidity

Total mass:

$$m_{nC} = m_{nNG} + m_{nA} + m_{nW} \quad (20)$$

where:

$$m_{nNG} = 1 \text{ kg and } m_{nN} = m_{nA} \times \sigma_{N_2} \quad (21)$$

$$m_{nO} = m_{nA} \times \sigma_{O_2} \quad (22)$$

$$m_{nAr+} = m_{nA} \times \sigma_{Ar+} \quad (23)$$

$$m_{nW} = m_{nA} \times x_0 \quad (24)$$

Molar mass:

$$M_n = \frac{1}{\sum \frac{\sigma_{ni}}{M_i}} = \frac{m_{nC}}{\frac{m_{nNG}}{M_{NG}} + \frac{m_{nN}}{M_N} + \frac{m_{nO}}{M_O} + \frac{m_{nAr+}}{M_{Ar+}} + \frac{m_{nW}}{M_{H_2O}}} \quad (25)$$

where: σ_{ni} – mass fractions of components; M_i – molar masses of the components

Individual gas constant:

$$r_n = 8314.4/M_n \quad (\text{J/(kg·K)}) \quad (26)$$

Ideal combustion gases for 1 kg of NG. The following stoichiometric relations were used if the composition of total mass is $m_{CC} \equiv m_{nC}$

Then: moisture $m_{CW} = 8.936 + m_{nW}$ (27)

dry component $m_{Csp} = m_{nC} - m_{CW}$ (28)

If $m_{CN} = m_{nN}$, then:

$$m_{CO} = m_{Ai} (\alpha - 1) \times \sigma_{O_2} \quad (29)$$

$$m_{CAr+} = m_{nAr+} \quad (30)$$

where: m_{CN} – total mass of nitrogen; m_{CO} – oxygen; m_{CAr+} – argon and inert gases

Molar mass:

$$M_C = \frac{1}{\sum \frac{\sigma_{Ci}}{M_i}} = \frac{m_{nC}}{\sum \frac{m_{Ci}}{M_i}} \quad (31)$$

where: σ_{Ci} – mass fractions of the components in the total flue gases; M_i – molar masses of the components

Individual gas constant:

$$r_C = 8314.4/M_C \quad (\text{J/(kg.K)}) \quad (32)$$

Working mixture (1) for 1 kg of NG. Working mixture consists of ignition compound with volume V_s and mass m_{nC} and of the share of non-removed flue gases with volume V_2 and mass m_{re}.

Coefficient of residual flue gases:

$$s_{re} = m_{re}/m_{nC} \quad (33)$$

Total mass of working mixture is:

$$m_{1C} = m_{nC} + m_{re} = m_{nC}(1 + s_{re}) \quad (34)$$

The following relation was derived for coefficient of residual flue gases:

$$s_{re} = \frac{r_n}{r_C(\varepsilon - 1)} \quad (35)$$

where: r_n – individual gas constant (relation 26); r_C – individual gas constant (relation 32); ε – compression ratio (relation 7)

Particular components for total mass of working mixture according to Eq. 34 are:

$$m_{1N} = m_{nN} + s_{re} \times m_{SN} \quad (36)$$

$$m_{1O} = m_{nO} + s_{re} \times m_{SO} \quad (37)$$

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$$m_{1Ar+} = m_{nAr+}(1 + s_{re}) \quad (38)$$

$$m_{1W} = m_{nO} + s_{re} \times m_{SW} \quad (39)$$

$$m_{1CO_2} = s_{re} \times m_{SCO_2} \quad (40)$$

where: m_1 – mass of the components (nitrogen, oxygen, argon, humidity, carbon dioxide CO_2)

Molar mass:

$$M_{C1} = \frac{1}{\sum \frac{\sigma_{1i}}{M_i}} = \frac{m_{1C}}{\sum \frac{m_{1i}}{M_i}} \quad (41)$$

Individual gas constant:

$$r_1 = 83144/M_1 \qquad (\mathrm{J/(kg \cdot K)}) \quad (42)$$

Adiabatic exponent:

$$\kappa_1 = \Sigma\, \sigma_{1i} \times \kappa_i = \frac{\Sigma m_{1i} \times \kappa_i}{m_{1C}} \quad (43)$$

where: κ_i – adiabatic exponent of component; σ_i – mass fraction of component; m_{1i} – mass of component; m_{1C} – total mass of the mixture

RESULTS AND DISCUSSION

Adiabatic compression 1–2

Working mixture m_1. Pressure at the end of compression:

$$p_2 = p_1 \times \varepsilon^{\kappa_1} \qquad (\mathrm{Pa}) \quad (44)$$

Temperature at the end of compression:

$$T_2 = T_1 \times \varepsilon^{(\kappa_1 - 1)} \qquad (\mathrm{K}) \quad (45)$$

where: ε – compression ratio; κ_1 – adiabatic exponent

Ražnievič (1969) tables provide only data of enthalpies of particular gases up to 3,000°C, while for 0°C it is $i_i = 0$. Relations obtained by linear regression for temperature range 100–500°C are to be found in a computer program. The internal energy of working mixture at the end of compression 2 is:

$$u_2 = i_2 - r_2 \times t_2 = \Sigma\sigma_{1i} \times i_{1i} - r_2 \times t_2 = (\Sigma m_{1i} \times i_{1i})/m_{1C} - r_2 \times t_2 \qquad (\mathrm{kJ/kg}) \quad (46)$$

where: i_2 – enthalpy of mixture; r_2 – individual gas constant; t_2 – temperature at the end of compression

Isochoric combustion 2–3. Working mixture m_1 is ignited in 2 and by explosion it burns to flue gas mixture m_3. Volume remains constant $v_3 = v_4$; total mass of the mixture remains constant $m_{3C} = m_{2C}$. The mixture is made up entirely of flue gases; therefore, $M_3 = M_C$, $r_3 = r_C$, $\kappa_3 = \kappa_C$.

Composition of total mass

$$m_{3C} = m_{1C} = m_{nC}(1+s_{re}) \quad (47)$$

where:

$$m_{3N} = m_{nN}(1+s_{re}) \quad (48)$$

$$m_{3O} = m_{CO}(1+s_{re}) \quad (49)$$

$$m_{3Ar+} = m_{nAr+}(1+s_{re}) \quad (50)$$

$$m_{3CO_2} = m_{CCO_2}(1+s_{re}) \quad (51)$$

$$m_{3W} = m_{SW}(1+s_{re}) \quad (52)$$

Thermal balance of combustion:

$$m_{1C} \times u_2 + Q_i = m_{1C} \times u_3 \quad (53)$$

where: Q_i – calorific value of the fuel (natural gas)

The internal energy of hot flue gases is:

$$u_3 = u_2 + \frac{Q_i}{m_{1C}} \qquad (\mathrm{kJ/kg}) \quad (54)$$

Relations obtained by linear regression from tabular values (Ražnievič 1969) for the range 2,000–3,000°C were used in the computer program. The relation for u_3 is as follows:

$$u_3 = \Sigma\sigma_{3i} \times i_{3i} - r_3 \times t_3 = (\Sigma m_{3i} \times i_{3i})/m_{1C} - r_S \times t_3 \qquad (\mathrm{kJ/kg}) \quad (55)$$

Hence, the relation for temperature t_3:

$$t_3 = \frac{u_3 + B}{A} \qquad (^\circ\mathrm{C}) \quad (56)$$

where: A, B – constants of linear equation

Pressure at the end of combustion p_3 is:

$$p_2 \times v_2 = r_1 \times T_2 \quad (57)$$

$$p_3 \times v_3 = r_C \times T_3 \quad (58)$$

If $v_2 = v_3$, then

$$p_3 = p_2 \frac{r_C \times T_3}{r_1 \times T_2} \qquad (\mathrm{Pa}) \quad (59)$$

where: T_2 – thermodynamic temperature at the end of the compression; T_3 – thermodynamic temperature at the end of the combustion process

Adiabatic expansion 3–4. Composition remains constant; therefore, $M_4 = M_3 = M_C$ and $r_4 = r_3 = r_C$. Pressure at the end of expansion p_4:

$$p_4 = \frac{p_3}{\varepsilon^{\kappa_3}} \qquad (\mathrm{Pa}) \quad (60)$$

where: ε – compression ratio; κ_3 – adiabatic exponent

Temperature at the end of expansion T_4:

$$T_4 = \frac{T_3}{\varepsilon^{(\kappa_3 - 1)}} \qquad (\mathrm{K}) \quad (61)$$

Relations for the enthalpy of specific gases are gained by means of linear regression from tables (Ražnievič 1969) for the range of 500–1,500°C. They are to be found in the computer program. In-

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ternal energy of the mixture at the end of expansion u_4 is:

$$u_4 = i_4 - r_4 \times t_4 = \Sigma\sigma_{4i} \times i_{4i} - r_4 \times t_4 = (\Sigma m_{3i} \times i_{4i})/m_{1C} - r_C \times t_4 \quad \text{(kJ/kg)} \qquad (62)$$

where: i_4 – enthalpy of mixture; r_4 – individual gas constant; t_4 – temperature at the end of expansion; σ_i – mass fraction of components; i_i – enthalpy of components

Isochoric heat removal 4–5. Isochoric heat removal is theoretically considered the weakest process in the ideal cycle, because reality departs from the theoretical ideal cycle to the greatest extent.

Theoretical assumptions are $v_5 = v_1$, $T_5 = T_1$ and $p_5 = p_1$. Relations for internal energy of particular gases are obtained by linear regression from tables (Ražnievič 1969) for the range 0–100°C and then used in the computer program. The internal energy of gas mixture at the end of isochoric heat removal is:

$$u_5 = i_5 - r_5 \times t_5 = \Sigma\sigma_{5i} \times i_{5i} - r_5 \times t_5 = (\Sigma m_{3i} \times i_{5i})/m_{1C} - r_C \times t_5 \quad \text{(kJ/kg)} \qquad (63)$$

where: i_5 – enthalpy of mixture; r_5 – individual gas constant; t_5 – temperature at the end of the heat removal; σ_i – mass fractions of components; i_i – enthalpy of components

The paper presents only the final form of the relations without their derivation. The calculations are carried out for given four-stroke piston engine with cylinder volume V_{Va} (l) and revolutions per minute n.

Characteristic indicators

Thermal efficiency:

$$\eta_t = \frac{A_{it}}{Q_p} = \frac{Q_p - Q_o}{Q_p} = 1 - \frac{u_5 - u_5}{u_3 - u_2} \qquad (64)$$

where: A_{it} – work obtained from thermal cycle, Q_p – supplied heat, Q_o – removed heat, u – internal energy

This relation derived for ideal spark-ignition cycles can be found in textbooks:

$$\eta_t = 1 - \left(\frac{1}{\varepsilon}\right)^{\kappa-1} \qquad (65)$$

Fuel consumption – natural gas. Natural gas is described and invoiced in m^3 of NG.

Consumption of NG per hour:

$$V_{1hNG} = \frac{V_{Va}(\varepsilon - 1) \times n \times 30 \times p_1}{\rho_{NG} \times \varepsilon \times m_{1C} \times r_1 \times T_1} \quad \text{(m}^3\text{/h)} \qquad (66)$$

where: V_{Va} – engine displacement; ε – compression ratio; n – revolutions per minute; p_1 – pressure of mixture; ρ_{NG} – density of gas; m_{1C} – total mass of mixture

Per 1 second:

$$V_{1sNG} = V_{1hNG}/3{,}600 \quad \text{(m}^3\text{/s)} \qquad (67)$$

Theoretical engine performance:

$$P = \frac{V_{1hNG} \times Q_i^V \times \eta_t}{3{,}600} = V_{1sNG} \times Q_i^V \times \eta_t \quad \text{(W)} \qquad (68)$$

where: Q_i^V – calorific value of NG; η_t – thermal efficiency

Theoretical heat removal by flue gases:

$$\dot{Q}_V = \frac{V_{1hNG} \times Q_i^V (1 - \eta_t)}{3{,}600} = V_{1sNG} \times Q_i^V (1 - \eta_t) \quad \text{(W)} \qquad (69)$$

Torque:

$$M_t = 9.5492\,(P/n) \quad \text{(N·m)} \qquad (70)$$

Mean theoretical overpressure. Mean theoretical overpressure is hypothetical overpressure that would press on the piston during one stroke and would perform the same work as an actual variable pressure that presses on the piston:

$$p_{it} = A_{it}/V_s \quad \text{(Pa)} \qquad (71)$$

where: A_{it} – work obtained from thermal cycle; V_S – swept volume

For 1 kg of NG:

$$A_{it} = q_n \times \eta_t \qquad (72)$$

$$V_s = \frac{m_{1C} \times r_1 \times T_1}{p_1} \qquad (73)$$

where: q_n – the amount of heat supplied by 1 kg of NG; η_t – thermal efficiency

Specific fuel consumption – is amount of m^3 of NG that is needed for the production of unit of work (J, kJ, MJ):

$$V_{ms} = \frac{1}{Q_i^V \times \eta_t} \quad \text{(m}^3\text{/J)} \qquad (74)$$

Litre performance – is performance that attributes to the unit of engine displacement:

$$P_i = \frac{P \times \varepsilon}{V_{Va}(\varepsilon - 1)} \quad \text{(W/l)} \qquad (75)$$

where: P – theoretical engine performance; V_{Va} –engine displacement; ε – compression ratio

Flow of sucked-in air – is flow of the air that enters the gas mixer:

$$\dot{V}_n = \frac{m_{nC} \times r_n \times T_0}{p_a} V_{1sNG} \times \rho_{NG} \quad \text{(m}^3\text{/s)} \qquad (76)$$

where: m_{nC} – total mass of mixture; r_n – individual gas constant; T_o – ambient air temperature, p_a – atmospheric pressure, V_{1sNG} – gas consumption per 1 s, ρ_{NG} – density of NG

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Flow of flue gases:

$$\dot{V}_{cg} = \frac{m_{CC} \times r_5 \times T_5}{p_5} \; V_{1sNG} \times \rho_{NG} \qquad (m^3/s) \qquad (77)$$

Flue gases – mass composition

nitrogen N_2

$$m_{1sN_2} = m_{CN_2} \times V_{1sNG} \times \rho_{NG}/m_{CC} \qquad (kg/s) \qquad (78)$$

oxygen O_2

$$m_{1sO_2} = m_{CO_2} \times V_{1sNG} \times \rho_{NG}/m_{CC} \qquad (kg/s) \qquad (79)$$

argon + inert. gas A_{r+}

$$m_{1sAr} = m_{CAr} \times V_{1sNG} \times \rho_{NG}/m_{CC} \qquad (kg/s) \qquad (80)$$

carbon dioxide CO_2

$$m_{1sCO_2} = m_{CCO_2} \times V_{1sNG} \times \rho_{NG}/m_{CC} \qquad (kg/s) \qquad (81)$$

water vapour H_2O

$$m_{1sW} = m_{CW} \times V_{1sNG} \times \rho_{NG}/m_{CC} \qquad (kg/s) \qquad (82)$$

where: m_C – total mass of the individual components (nitrogen, oxygen, argon, carbon dioxide, water vapour); V_{1sNG} – gas consumption per 1 s; ρ_{NG} – density of NG; m_{CC} – total mass of mixture

Exhaust gases – volumetric share of O_2 and CO_2

oxygen O_2 $$x_{O_2} = \left(\frac{m_{CO_2}}{M_{O_2}} \Big/ \frac{m_{CC}}{M_C}\right) 100 \qquad (\%) \qquad (83)$$

carbon dioxide CO_2 $$x_{CO_2} = \left(\frac{m_{CCO_2}}{M_{CO_2}} \Big/ \frac{m_{CC}}{M_C}\right) 100 \qquad (\%) \qquad (84)$$

where: M – molar masses (oxygen, carbon dioxide)

Program for calculations. The scope of this paper indicates that the prospective user needs considerable amount of tabular data and the calculation itself is rather extensive and time-consuming. Therefore, the computing program was designed. It consists of three parts:

(1) Questionnaire for basic and determining quantities that is filled in by the user.

(2) Program for calculations.

(3) Report including results of the calculations.

The example of calculation by means of the program is not included in the paper.

This part of the paper introduces a detailed methodology of calculations of parameters of an ideal working cycle of spark-ignition combustion engine. Ideal cycles are being dealt with in a number of scientific works (Shavit, Gutfinger 2008; Stone 2012; Caton 2015). Properties of the fuel are not taken into account. The work (Stone 2012) presents comparison of thermal efficiency of thermodynamic and mechanical cycles. Thermodynamics of combustion, air-standard engine cycles and reciprocating internal combustion engines are described by Winterbone and Turan (2015). However, aspects such as excess air, residual space in the cylinder of the engine and the course of properties of gases in dependence on temperature were not considered in relation to CNG in any of the works mentioned above.

CONCLUSION

Current thermodynamics explains the theoretical ideal cycle of spark-ignition combustion engine in the most simplified way as possible. This paper describes the course of ideal working cycle of an actual combustion engine for gaseous fuel – natural gas. To sum up, this theoretically precise calculation of combustion engine working cycle is considerably demanding for the user who has to deal with the thermodynamics of gas mixture, linear and nonlinear regression of numerous thermal tabular values and calculation of a substantial amount of auxiliary and final data. In conclusion, the presented methodology may be successfully applied, provided that the prospective user has access to the computer program designed for these specific purposes.

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