IMPROVING OF IC ENGINE EFFICIENCY THROUGH DROPPING OF THE CHARGE EXCHANGE WORK

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Abstract

Normally the charge exchange occurs once during each one engine cycle realized. The main idea presented in the paper leads to diminishing of the ICE charge exchange work. Elaborated proposition bases on the elimination of chosen charge exchange processes and through this the dropping of the charge exchange work can be achieved.

Work of internal combustion engines, which are used as the driving source of cars, occurs not only at the full load, but mostly at the part load. In this range the energy efficiency η_e – eq. (1) is significant lower as in the optimal (nominal field) stage of the performance parameters. One of the numerous reasons of this state is regular growing of the relative load exchange work of the IC engine. On the base of the experimental results and using the elaborated formulas it has been calculated that the relative charge exchange work can achieve value up to 40 % at the part load (e.g. idle run) of the IC engine.

Both of this mentioned problems can be solved using presented in the paper a new concept of the reference cycle (called as eco-cycle) of IC engine. The work of the engine basing on the eco-cycle occurs in two 3-stroke stages: the fresh air is delivered only once for both stages, but in range of each stage a new portion of fuel is burned. The fuel combustion for the whole cycle can be performed at relatively low values of air excess $\lambda_{el} \geq 1$. The mostly taken into account criteria for valuation of internal combustion engine refer to its energy efficiency and to emission. Diminishing of emission (gaseous phase and solid particles) from combustion engines can be achieved by realisation of two groups of measures: primary (inside-engine) doings, secondary doings, comprising catalysts and solid particle filters. The proposed solution bases on the possibilities of diminishing of the emission, which is contained in the primary measures. The elaborated eco-cycle leads to the diminishing of the toxic substance emission and to the improving of engine efficiency.

Keywords: energy efficiency, IC engine, emission, engine cycle, charge exchange work.

1. Introduction

Piston combustion engine belongs to the internal combustion heat machines which periodically performs the work in frames of the realised thermodynamic cycle [1], [3]. The basic criteria taken into account during working out and quality assessment of new designs of internal combustion engines are among other things [2], [4]:

a) emission of pollutants and other toxic substances into the environment,
b) efficiency of energy conversion processes taking place in the system,
c) reliability and correctness of the used working system.

Diminishing of emission of toxic substances (components in the gaseous phase: CO, NO_x, C_mH_n, SO_y, and likewise solid particles: soot, condensed hydrocarbons) from combustion engines can be achieved by realisation of two groups of measures [5], [6]:

1. primary (otherwise inside-engine), in this:
   - combustion of the lean air-fuel mixtures (at the high oxygen excess),

- gradation of the fuel feed (multistage injection of the fuel),
- recirculation of combustion products (turning back of flu gases),
- after-burning (re-burning) of combustible components of exhaust gases,
- loading (injection) of additional water into a cylinder volume (combustion of fuel-water mixture, humidification of recirculating flu gases and supercharging air);

2. secondary (in other words outside-engine) comprising multifunction catalysts and solid particle filters.

Effective energy efficiency η_e of IC engine depends on the energy efficiency η_0 of the reference ideal thermodynamic cycle, expressed as [2], [4]:

$$\eta_e = \frac{N_e}{\dot{m}_p H_{u,p}}, \quad \text{and} \quad \eta_0 = \frac{N_0}{\dot{m}_p H_{u,p}} \tag{1}$$

where: N_e, kW – effective power output of the real IC engine, $\dot{m}_p$, kg/s – mass flux of the fuel consumed, $H_{u,p}$, kJ/kg – specific lover calorific value of the supplied fuel, N_0, kW – power output of the IC engine working due to the reference ideal thermodynamic cycle,

whereby: $$\eta_e = \eta_0 \, \xi_i \, \xi_m, \quad \text{and} \quad \xi_i = \frac{N_i}{N_0}, \quad \xi_m = \frac{N_e}{N_i} \tag{2}$$

where: ξ_i – internal goodness rate of the engine, ξ_m – mechanical goodness rate of the IC engine.

Therefore improving the structure of the reference cycle leads to reaching of better effective energy efficiency of the real internal combustion engine. As standard reference of each real thermodynamic cycle of any IC engine is the ideal thermodynamic cycle, traditional called as theoretical reference cycle (e.g. the Seiliger – Sabathe cycle). An supplemented theoretical cycle, which consists additionally the isothermal after combustion phase, is the Eichelberg cycle (shown in the fig. 1), where supplying of the heat (Q_d) occurs in three phases: first ($Q_{d,v}$) isochoricly (2–3), second ($Q_{d,p}$) isobaricly (3–4) and next ($Q_{d,T}$) isothermally (4–5). The heat output (Q_w) from the considered cycle is realised once and isochoricly (6-1).

The total heat supplied into the system in frame of one period of the work is the sum of this three components: $$Q_d = Q_{d,v} + Q_{d,p} + Q_{d,T} \tag{3}$$

Two parameters (ψ, ϑ) defined as relations: $$\psi \overset{df}{=} \frac{Q_{d,v}}{Q_d}, \quad \vartheta \overset{df}{=} \frac{Q_{d,p}}{Q_d}, \quad 0 \le (\psi + \vartheta) \le 1 \tag{4}$$

can be used as follows: $$Q_{d,v} = \psi Q_d, \quad Q_{d,p} = \vartheta Q_d, \quad Q_{d,T} = (1 - (\psi + \vartheta))Q_d \tag{5}$$

The work performance L_{ob} of the system results from the energy balance set for one period:

$L_{ob} = Q_d - Q_w$, and equals:

$$L_{ob} = \frac{1}{\psi} Q_{d,v} - Q_w \qquad L_{ob} = \frac{1}{\vartheta} Q_{d,p} - Q_w, \qquad L_{ob} = \frac{1}{1 - (\psi + \vartheta)} Q_{d,T} - Q_w \tag{6}$$

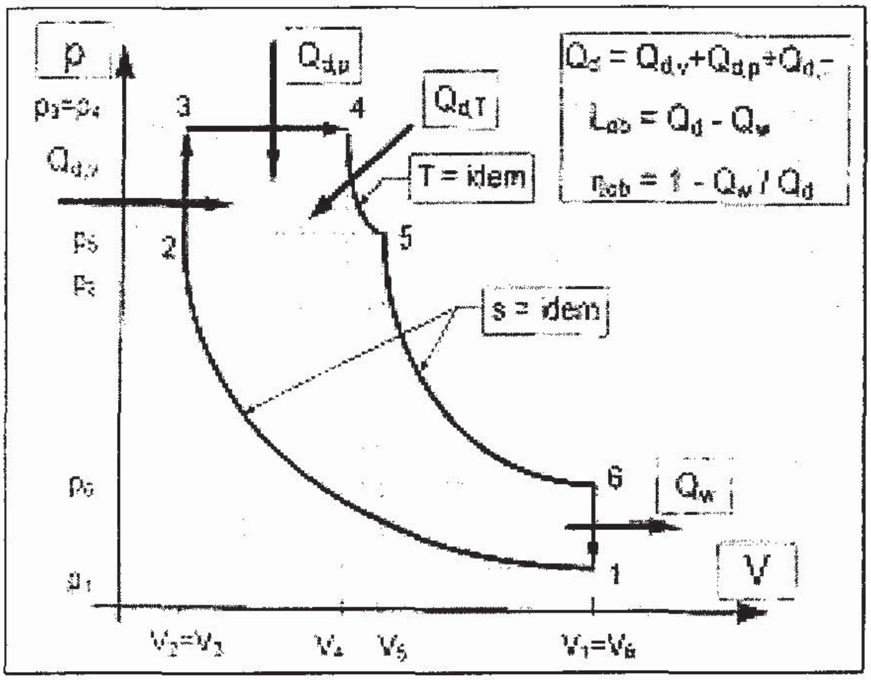

Fig. 1. Typical reference cycle (Eichelberg) of IC engine

Depending on the actual values of the parameters ψ, ϑ three particular cases of the reference engine cycle can be indicated:

- if the sum: $(\psi + \vartheta) = 1$, then the typical Seiliger - Sabathe cycle is achieved,
- for the values: $(\psi = 1,\ \vartheta = 0)$ the considered cycle becomes the Otto cycle (as reference cycle for spark ignition engines),
- if contemporaneously $(\psi = 0,\ \vartheta = 1)$ then the investigated cycle is identical with the Diesel cycle (as reference cycle for self ignition engines).

The thermal efficiency η_0 of the considered thermodynamic cycle can be expressed as:

$$\eta_0 \overset{df}{=} \frac{L_{ob}}{Q_d}, \qquad \eta_0 = 1 - \psi\frac{Q_w}{Q_{d,v}}, \qquad \eta_0 = 1 - \vartheta\frac{Q_w}{Q_{d,p}}, \qquad \eta_0 = 1 - \left(1 - (\psi + \vartheta)\right)\frac{Q_w}{Q_{d,T}} \tag{7}$$

Introduction of the isothermal phase (T = idem) within the thermodynamic cycle is important, because it refers to the maximal temperature (T_{max}) of the whole cycle, on which value depends the possibility and rate of the formation of the nitrogen oxides NO_x, carbon monoxide CO, and the out-burning ratio of the injected fuel [6].

Effective energy efficiency η_e of the real IC engine should be treated as a function of its actual performance parameters (shown in fig. 2): $\eta_e = F(M_e,\ \dot{n}_o),\ \ \eta_e = F(N_e,\ \dot{n}_o)$ (8)

where: M_e, Nm/rad – effective torque, $\dot{n}_o$, rev./min – engine speed (revolutions).

Instead of effective energy efficiency η_e the specific (relative) fuel consumption b_e can be used:

$$b_e = \frac{\dot{m}_p}{N_e}\ , \text{kg/kWs} \ \ \text{(or: g/kWh – fig. 2)}, \quad \text{whereby:} \ \ \eta_e\, b_e\, H_{u,p} = 1 \tag{9}$$

The energy efficiency of each heat engine can not be greater than thermal efficiency of the ideal engine working according to Carnot cycle. ·

Work of internal combustion engines, which are used as the driving source of cars, occurs not only at the full load, but mostly at the part load. In this range the energy efficiency η_e – eq. (1), (8) is significant lower as in the optimal (nominal field) stage of the performance parameters – fig. 2. One of the numerous reasons of this state is regular growing of the relative load exchange work at the part load (e.g. idle run) of the IC engine.

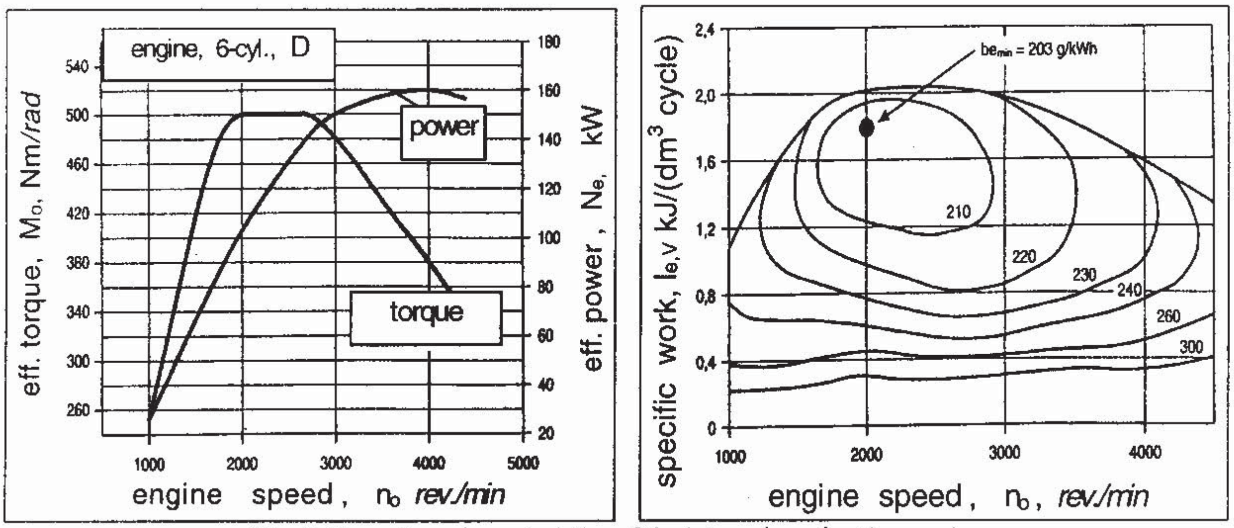

Fig. 2. Operating characteristics of the internal combustion engine

The mentioned problem was at first theoretically analysed, thanks this the systematic dropping of the energy efficiency has been confirmed, and for illustration the achieved approximate results are shown in the fig. 3.

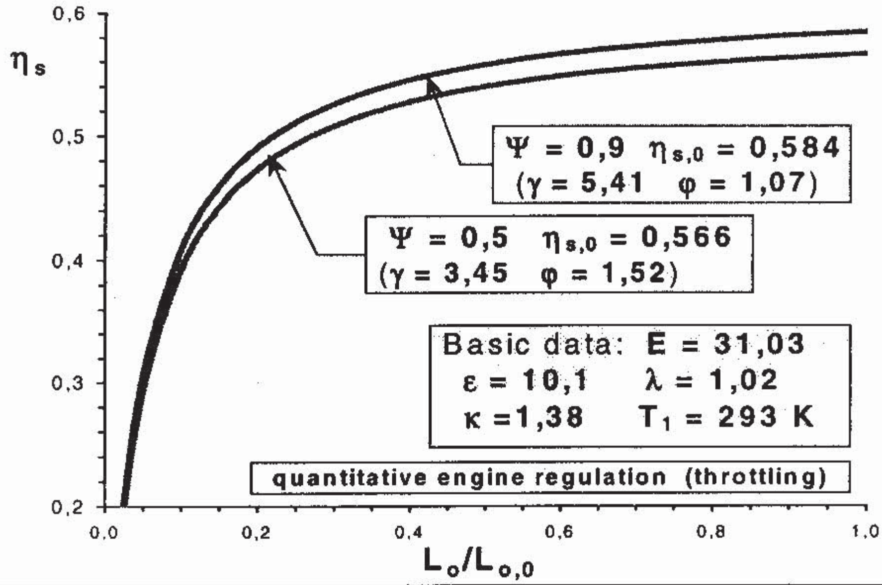

Fig. 3. Influence of the relative load on the engine energy efficiency

Using the elaborated formulas and on the base of the achieved theoretical results it has been calculated that the relative load exchange work can significant influence the values of energy efficiency (up to 55 % at the part load, e.g. idle run) of the IC engine.

Next, on the base of the experimental results and using the elaborated formulas it has been calculated that the relative load exchange work can achieve value up to 40% at the part load (e.g. idle run) of the IC engine. Results are shown in the fig. 4.

As consequence of the growing of the relative load exchange work is the regular and significant drop of the engine energy efficiency; from ca. 55% down to ca. 25%. The engine speed influences the real investigation results too (fig. 4). The main reason of this effect is the throttling process (causing exergy losses) occurring in the inlet and outlet channels. The newest proposals for solving of this problem is based on applying of the fully electronic control of the motion of inlet and outlet valves.

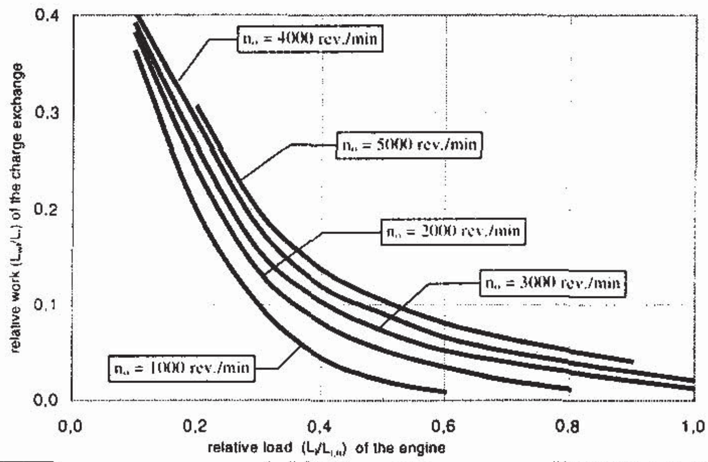

Fig. 4. Influence of load ratio on the relative exchange load work of the IC engine

The main idea presented in the paper leads to diminishing of the ICE charge exchange work. Normally the charge exchange occurs once during each engine cycle realized. Elaborated proposition bases on the elimination of chosen charge exchange processes and through this the dropping of the charge exchange work can be achieved. The first step is introduction of only one charge exchange, but for two fuel injection, and ipso facto two work output stages.

The mentioned problem can be solved using presented in the paper a new concept of the reference cycle (called as eco-cycle) of IC engine.

2. New developed concept of the eco-cycle

New concept of the thermodynamic cycle (conventionally: eco-cycle) of IC engine contains most of the elements leading to diminishing of toxic substances emission [4]. The work of the combustion engine bases on the performance of two 3-stroke stages.

During the first stage the engine cylinder is full filled with the fresh charge (mostly with the air), and after this process the cylinder charge is compressed. At the end of the compression the first portion of the fuel is injected and first stage of combustion process occurs, and afterwards the whole charge expands and decompresses.

The second stage of the eco-cycle begins with the isochoric cooling of the charge. This effect can be achieved by injection of liquid water into the volume of hot part-combustion products in the cylinder; the injected water is heated and vaporises immediately, what results with the dropping of the charge temperature and pressure.

The achieved new mixture is compressed again, and then after injection of the second portion of fuel, the second stage of combustion process occurs. The whole charge expands and decompresses, and next the open expansion and outflow of flu gases process. In the range of each stage a new portion of fuel is injected into the combustion chamber, so the combustion of the prepared combustion mixture, energy release and heat output take place in two stages too.

Below the functioning of the considered eco-cycle is discussed and main stages of this thermodynamic eco-cycle are in detail described.

In the **I. stage** (illustrated in fig. 5) the following steps are realised:

- filling of the engine cylinder (0-1) with the fresh air charge,
- compression (1-2) of the fresh charge (change of the cylinder volume: from V_0 to V_k),
- initial phase of the fuel injection and mixture combustion (energy release and heat output): approximately – isochoricly (2-3), and next – isobaricly (3-4),

- first expansion (4-5) of the working medium (I stage of the work performance),
- isochoric (at the cylinder volume V_0) cooling (5-6) of the charge (e.g. by injection and next vaporising of the liquid water), which results in the temperature and pressure dropping.

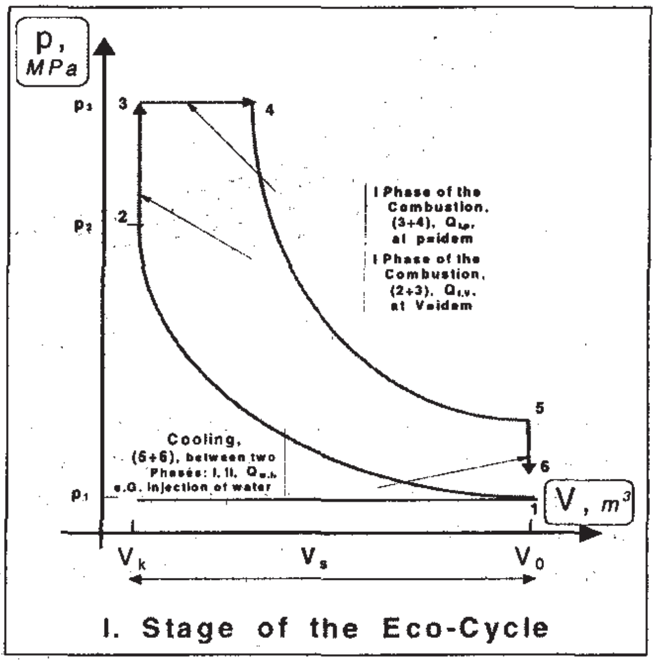

Fig. 5. Main components of the first stage of the elaborated eco-cycle

During the **II. stage** (shown in fig. 6) the following steps are realised:

- renewed compression (6-7) of the working medium (provided by change of the cylinder and charge volume: from V_0 to V_k),
- second phase of the fuel injection and mixture combustion (energy release and heat output): approximately – isochoricly (7-8), and next – isobaricly (8-9),
- final expansion (9-10) of the combustion products (II stage of the work performance),
- open expansion and outflow of flu gases.

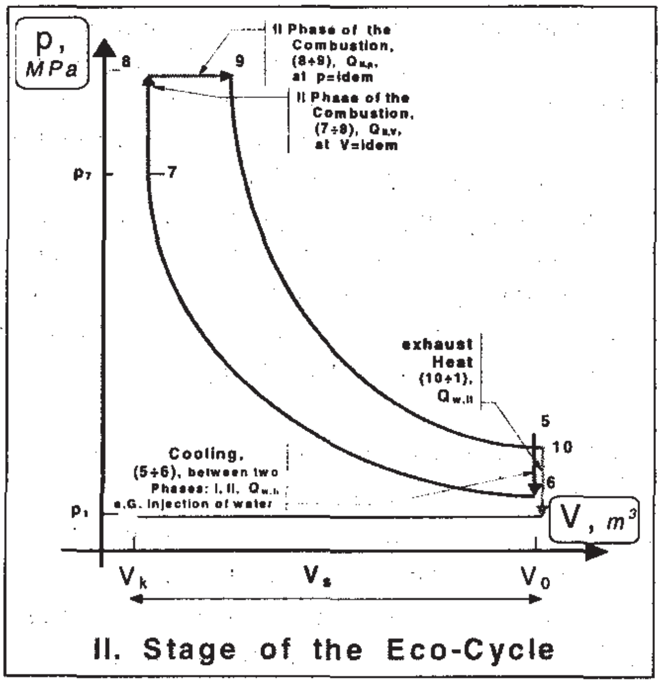

Fig. 6. Main elements of the second stage of the proposed eco-cycle

The second combustion stage (containing anew the isochoric and next isobaric phases) processes by nearly stoichiometric combustion conditions (the actual oxygen excess ratio equals one $\lambda_2 \geq \cong 1$), but also in the presence of significant amount of the inert substances (recirculating gases), what efficiently limits the excessive temperature rise in the combustion chamber, and through this diminishes the formation of the nitrogen oxides NO_x. In the second stage the reburning of earlier (in the first stage) unburned gaseous (hydrocarbons C_mH_n, carbon monoxide CO) and solid (soot) substances takes place.

The elaborated concept of the reference thermodynamic cycle of IC engine in the composed form is presented in the fig. 7.

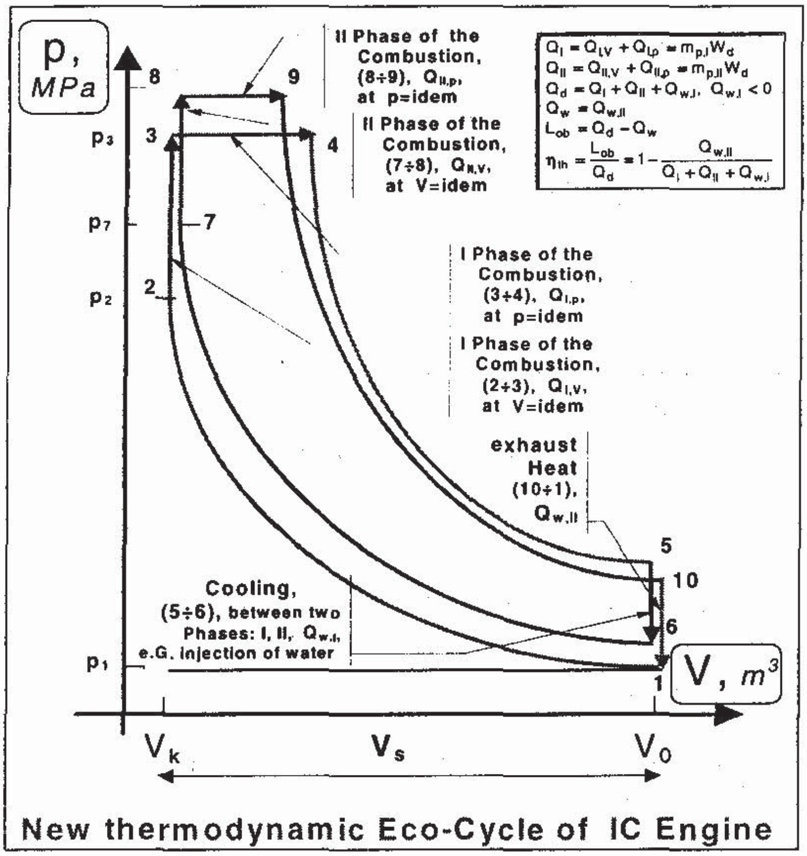

Fig. 7. Full composition of the thermodynamic reference eco-cycle

The characteristic feature of the proposed solution is among other things that it contains almost all ways of the diminishing of the toxic substance emission:

- combustion of the lean air-fuel mixtures,
- multistage injection of the fuel,
- recirculation of combustion flu gases,
- after-burning of combustible components,
- loading of additional water into the cylinder,

appearing in the primary measures.

The first stage of combustion process (containing the isochoric and next isobaric phases) is signified through this, that is realised in the range of the lean combustion mixtures, it means at the high air (oxygen) excess $\lambda_1 > 1$. The recirculation of the flu gases is realised by keeping of the whole charge in the cylinder volume between both stages, and renewed compression at the beginning of the second stage of the eco-engine.

The engine expansion effective work is performed twice within the cycle, but the exchange charge work only once; therefore the proposed thermodynamic eco-cycle of IC engine possesses features of the 3-stroke cycle.

3. Basic stoichiometric relations

Mass of the fuel m_p, kg/(cyl. cycle) delivered (injected) into the each cylinder of the engine during one realised cycle results from the equation:

$$m_p = \dot{m}_p \frac{k}{z\,\dot{n}_0} \tag{10}$$

where: $\dot{m}_p$, kg/s – mass flux of the fuel, z – cylinder number of the engine,

k = 2 rev./cycle, in case of 2-revolution engine (4-stroke engine), k = 1 rev./cycle, in case of 1-revolution engine (2-stroke engine), $\dot{n}_0$, rev./s – revolution number.

If a determined amount (m_p, eq. (10)) of fuel should be fully burned, the adequate (with the excess λ_0) amount n_a of the air (oxygen O_2) should be delivered into the cylinder each time, in accordance to the stoichiometric dependence:

$$\lambda_0 = \frac{n_a}{m_p\, n'_{a,min}} = \frac{n_a\, z_{a,O_2}}{m_p\, n'_{O_2,min}} = \frac{n_{O_2}}{m_p\, n'_{O_2,min}} \tag{11}$$

where: $n'_{a,min}$, $\frac{\text{kmol pow.}}{\text{kg pal.}}$, $n'_{O_2,min}$, $\frac{\text{kmol } O_2}{\text{kg pal.}}$, $z_{a,O_2} \approx 0.21$.

In case of the proposed eco-cycle the fuel combustion is organised in two stages, according to the distribution:

$$m_{p,I} = \Psi\, m_p\,, \qquad m_{p,II} = (1-\Psi)\, m_p \tag{12}$$

where: $\Psi = \frac{m_{p,I}}{m_p}$, $0 \le \Psi \le 1$ is the ratio of the inter-stage fuel distribution.

The excess of the air (oxygen) in each stage of the fuel combustion equals suitably:

$$\lambda_1 = \frac{n_a}{m_{p,I}\, n'_{a,min}} = \frac{n_a}{\Psi\, m_p\, n'_{a,min}} = \frac{\lambda_0}{\Psi}\,, \qquad \lambda_1 \ge \lambda_0 \tag{13}$$

and

$$\lambda_2 = \frac{n_a - n_{a,I,min}}{m_{p,2}\, n'_{a,min}} = \frac{n_a - \psi\, m_p\, n'_{a,min}}{(1-\Psi)\, m_p\, n'_{a,min}} = \frac{\lambda_0 - \Psi}{1-\Psi}\,, \qquad \lambda_2 \ge \lambda_0 \tag{14}$$

The achieved mutually conditioned dependencies are shown in the fig. 8.

For the two separate values (λ_1, λ_2) of air (oxygen) excess, the effective air excess λ_0 at the outflow of flu gases from engine reaches value:

$$\lambda_0 = \frac{\lambda_1\, \lambda_2}{\lambda_1 + \lambda_2 - 1}\,, \qquad \lambda_0 \le \lambda_i\,, \qquad i = 1,\ 2 \tag{15}$$

The achieved results show that the fuel combustion for the whole eco-cycle can be performed at relatively low values of air excess $\lambda_{ef} \ge 1$, nevertheless locally in each stage the oxygen excess can be freely high (especially in the I stage of the process).

Depending on the value of the number of the inter-stage fuel distribution Ψ two characteristic regions can be indicated, which are connected in the common point Ψ^*, whereby:

$$\Psi^* = \lambda_0 - \sqrt{\lambda_0(\lambda_0 - 1)} \tag{16}$$

The air excess ratio λ^* in the border point Ψ^* is equal to:

$$\lambda^* = \frac{\lambda_0}{\lambda_0 - \sqrt{\lambda_0(\lambda_0 - 1)}}\,, \qquad \lambda^* \ge \lambda \tag{17}$$

For the range: $0 < \Psi < \Psi^*$, $\lambda_1 > \lambda_2 > \lambda_0$, while for: $\Psi^* < \Psi < 1$, $\lambda_2 > \lambda_1 > \lambda_0$ (18)

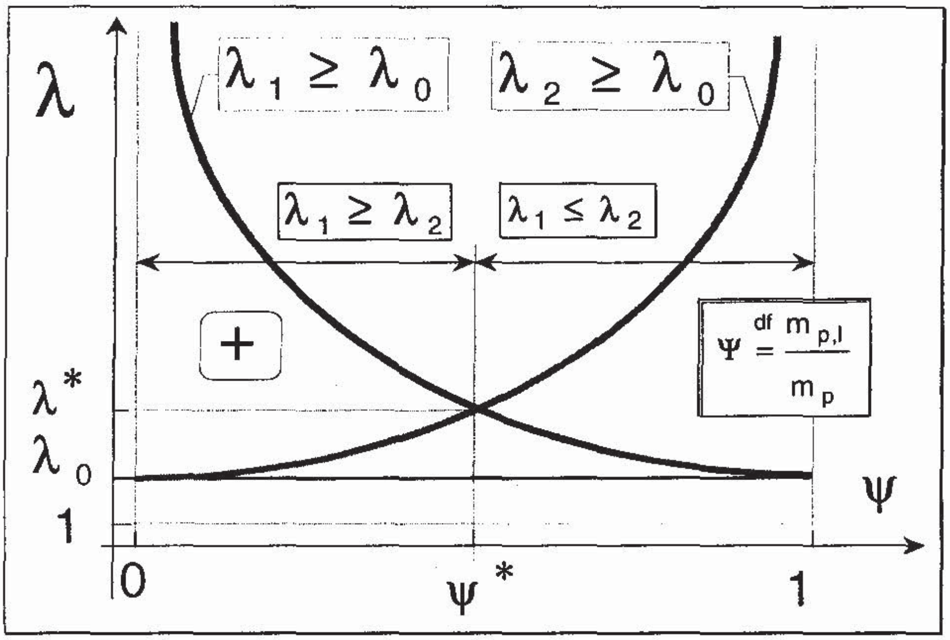

Fig. 8. Basic dependencies between air excess ratios for two stages of the eco-cycle

The elaborated system is very important because in this case for engines with the combustion of lean fuel-air mixtures (air (oxygen) excess $\lambda_1 > 1$ in the stage) the 3-way catalysts can be applied, through this that the effective air excess (observed in the flu gases outflow from engine) can reach values of $\lambda_{ef} \approx 1$. For this case the adequate values of the air excess (λ_1, λ_2) can be calculated using relations (13), (14).

4. Energy analysis – work output, efficiency

The heat Q_d supplied into the system during one period of the eco-cycle is the sum of heats connected with the indicated stages and phases:

$$Q_d = Q_{d,I} + Q_{d,II} + Q_{d,w} \tag{19}$$

where: $Q_{d,I} = Q_{I,v} + Q_{I,p} = m_{p,I} H_u$, $\quad Q_{d,II} = Q_{II,v} + Q_{II,p} = m_{p,II} H_u$, $\quad Q_{d,w} = -m_w r$

and: H_u, kJ/kg – calorific value of the fuel; $m_{p,I}$, $m_{p,II}$ – amount of the fuel injected in each of stages of the eco-cycle – eq. (12); r, kJ/kg – enthalpy of vaporisation of the water; m_w, kg – amount of the cooling water injected into cylinder.

Using definition of the inter-stage fuel distribution Ψ ratio – defined by the eq. (12), the adequate supplied heats can be written as follows:

$$Q_{d,I} = m_{p,I} H_u = \Psi m_p H_u, \qquad Q_{d,II} = m_{p,II} H_u = (1-\Psi) m_p H_u \tag{20}$$

an the total supplied heat is: $$Q_d = (m_{p,I} + m_{p,II}) H_u + Q_{d,w} = m_p H_u + Q_{d,w} \tag{21}$$

The relative amount of cooling water equals: $$\varphi = \frac{m_w}{m_p}, \quad \text{and} \quad \varphi_{max} = \frac{m_{w,max}}{m_p} \tag{22}$$

where: $m_{w,max}$ refers to the maximal mass of water, giving the saturated state of the moist gas.

Heat outflow from system in amount of Q_w proceeds (by V = idem) at the end of the II. stage. The main part of this energy is the enthalpy of the flu gases leaving the engine.

The work L_{ob} of the discussed theoretical eco-cycle results:

$$L_{ob} = Q_d - Q_w, \quad L_{ob} = (Q_{d,I} + Q_{d,II} + Q_{d,w}) - Q_w \tag{23}$$

and then according to the definition (1) energy efficiency η_{ob} of the eco-cycle (heat engine) can be written as: $\eta_0 \stackrel{df}{=} \frac{L_{ob}}{Q_d}$, $\eta_0 = 1 - \frac{Q_w}{Q_{d,I} + Q_{d,II} + Q_{d,w}}$ (24)

Influence of the amount of the used cooling water m_w, for different values of the effective air excess λ_0, on the achieved values of the energy efficiency η_0 of the proposed eco-cycle is shown in the fig. 9.

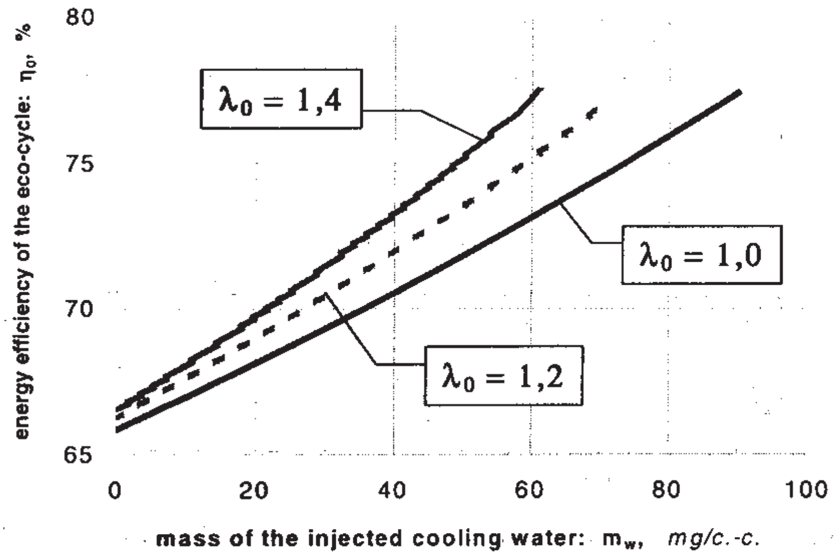

Fig. 9. Influence of the main parameters on the energy efficiency of the eco-engine

It can be observed that the following parameters have the main influence on the work performance and energy efficiency:

- compression ratio of the engine,
- effective air excess λ_0,
- the ratio Ψ of the inter-stage fuel distribution,
- the relative amount φ of the used cooling water (shown in the fig. 10).

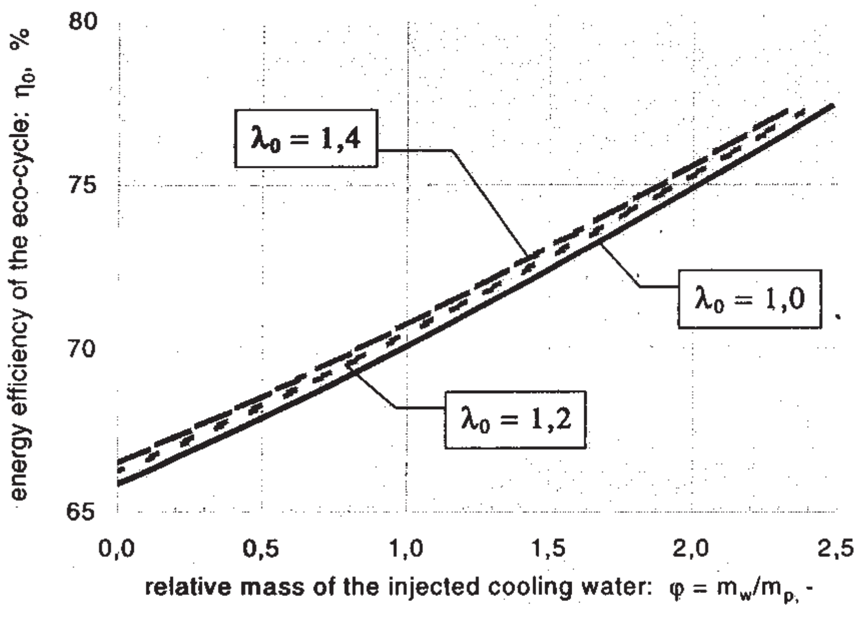

Fig. 10. Influence of the dimensionless parameters on the efficiency of the eco-engine

Basing on the above given solution of the new reference thermodynamic eco-cycle of the internal combustion engine, different influence analysis of the system parameters were carried out. It has been confirmed, that the proposed system leads to improving of the engine work efficiency. The expansion work within one whole eco-cycle is performed twice (equals L_I in the first stage and L_{II} in the second, S_L = 3 strokes/stage of work). Therefore the proposed eco-cycle of the combustion engine can be classified as 3-stroke cycle category.

5. Conclusion

Regular growing of the relative load exchange work of the IC engine causes dropping of the engine efficiency. On the base of the theoretical analyses and experimental results it has been calculated that the relative charge exchange work can achieve value up to 40 % at the part load (e.g. idle run) of the IC engine. As consequence of the growing of the relative load exchange work is the regular and significant drop of the engine energy efficiency: from ca. 55% down to ca. 25%, whereby the engine speed influences the real investigation results too.

The main reason of this effect is the throttling process (causing exergy losses) occurring in the inlet and outlet channels. It is directly connected with the quantitative regulation method common used in the IC engines.

Improving the structure of the reference thermodynamic cycle leads to reaching of better effective energy efficiency of the real internal combustion engine, in the whole range of its operating parameters and especially at the part load.

The paper presents a new concept of theoretical thermodynamic cycle (conventionally called as eco - cycle) of internal combustion engine, which contains most of the primary undertakings leading to diminishing of toxic substances emission:

- combustion of the lean air-fuel mixtures, multistage injection of the fuel,
- recirculation of combustion flu gases,
- after-burning of combustible components,
- loading of additional water into a cylinder.

Using the proposed solution (thermodynamic eco-cycle) in case of engines with the combustion of lean fuel-air mixtures (air (oxygen) excess $\lambda_I > 1$ in the stage) the 3-way catalysts can be applied, thanks this that the effective air excess (observed in the flu gases outflow from engine) can reach values of $\lambda_{ef} \approx 1$.

The proposed system (eco-cycle) leads to the diminishing of the toxic substance emission and simultaneously to improving of engine work efficiency - among other things - through abatement of the IC engine charge exchange work especially at its the part load.

The another newest proposals for solving of this problem are based on applying of the fully electronic control of the motion of inlet and outlet valves; the idea shown and in details described in the paper gives a alternative solution of this important problem.

Acknowledgments

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