Realization of the Atkinson-Miller cycle in spark-ignition engine by means of the fully variable inlet valve control system

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Abstract The theoretical analysis of the charge exchange process in a spark ignition engine has been presented. This process has significant impact on the effectiveness of engine operation because it is related to the necessity of overcoming the flow resistance, followed by the necessity of doing a work, so-called the charge exchange work. The flow resistance caused by the throttling valve is especially high during the part load operation. The open Atkinson-Miller cycle has been assumed as a model of processes taking place in the engine. Using fully variable inlet valve timing the A-M cycle can be realized according to two systems: system with late inlet valve closing and system with early inlet valve closing. The systems have been analysed individually and comparatively with the open Seiliger-Sabathe cycle which is a theoretical cycle for the classical throttle governing of the engine load. Benefits resulting from application of the systems with independent inlet valve control have been assessed on the basis of the selected parameters: fuel dose, cycle work, charge exchange work and a cycle efficiency. The use of the analysed systems to governing of the SI engine load will enable to eliminate a throttling valve from the system inlet and reduce the charge exchange work, especially within the range of part load operation.

Keywords: Spark-ignition engine; Variable inlet valve control; Open theoretical cycle; Charge exchange process

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Nomenclature

E_0	–	energy-stoichiometric parameter
E_w	–	waste energy, J
L_c	–	cycle work, J
$L_{c,max}$	–	maximal work of the Seiliger-Sabathe cycle (without flow resistance in a charge exchange system), J
M	–	molecular weight, kg/kmol
m_f	–	fuel dose, kg
$m_{f,0}$	–	basic fuel dose, kg
$m_{f,SS}$	–	fuel dose for Seiliger-Sabathe cycle, kg
m_0	–	maximum mass of the fresh combustion mixture, kg
$n_{a,min}$	–	theoretical air demand, kmol/kg fuel
p	–	pressure, Pa
p_0	–	ambient pressure, Pa
Δp_e	–	pressure drop in the exhaust system, Pa
Δp_i	–	pressure drop in the inlet system during filling, Pa
$\Delta p_{1,A}$	–	pressure drop in the inlet system during back-flow of the mixture, Pa
Q	–	heat, J
(MR)	–	universal gas constant, J/(kmol K)
T	–	temperature, K
T_0	–	temperature of the fresh charge, K
V	–	volume, m^3
$V_{1,A}$	–	cylinder volume at the moment of the intake valve closing, m^3
X	–	humidity, kmolH$_2$O/kmol d.g

Greek symbols

γ, ϕ	–	load parameters
ε	–	compression ratio
ε_A	–	isentropic compression ratio
η_t	–	thermal efficiency of a cycle
κ	–	isentropic exponent
λ	–	excess air number
λ_0	–	excess air number for maximal mass of the fresh charge
μ	–	relative charge exchange work
Ψ	–	heat distribution number

Subscripts

a	–	air
c	–	cycle
e	–	exhaust
ex	–	exchange
i	–	intake
p	–	p = idem
s	–	supplied
0	–	ambient

t	–	thermal
v	–	$V = idem$
w	–	waste
m	–	combustion mixture

1 Introduction

Realization of a charge exchange process in the combustion engine requires extra work to surmount flow resistances in inlet and exhaust systems hence execution of the corresponding work co-called the charge exchange work. The charge exchange system has an essential impact on the effectiveness of engine work. Each element installed in the charge exchange system generates flow resistance of a fresh charge in the intake system and combustion products in the exhaust system. These resistances bring about an increase of the charge exchange work which contributes to the decrease of internal and effective work of the engine. Increase of the charge exchange work for part load operation in the spark-ignition (SI) engine is connected with a method of load control. Quantity governing with the aid of a throttle, installed in the intake system, is disadvantageous especially from thermodynamic point of view because throttling generates losses of exergy [1,2].

The use of the independent intake and exhaust valve actuation has been proposed in order to increase the efficiency of the open, ideal cycle and effective efficiency of the spark-ignition engine [3]. Theoretical research of the system with late inlet valve closing (LIVC) and the system with early inlet valve closing (EIVC) has been carried out. These systems enable elimination of a choke valve from the intake system of SI engine.

Independent variable valve actuation enables control of engine work also according to the other systems [4,5], e.g.:

- early exhaust valve closing, making an internal exhaust gas recirculation possible;

- fully variable inlet and exhaust valves actuation.

Investigations on variable valve actuation are conducted by many scientific as well as research and development centres [6–8,10–12]. This justifies the importance of the presented problem within which the theoretical research on the systems with independent inlet valve control has been conducted. Problem of energy efficiency increase of engines can be resolved also by the over ways, e.g., cycle improvement, heat recovery, using of alternative fuels [13,14].

The open Seiliger-Sabathe cycle, Fig. 1, is the reference cycle for an assessment of advantages and effectiveness of work gaining in consequence of application of the analysed systems (LIVC and EIVC). The open Seiliger-Sabathe cycle is the ideal cycle taking into consideration flow resistances in the inlet and exhaust systems.

Figure 1: Open Seiliger-Sabathe cycle.
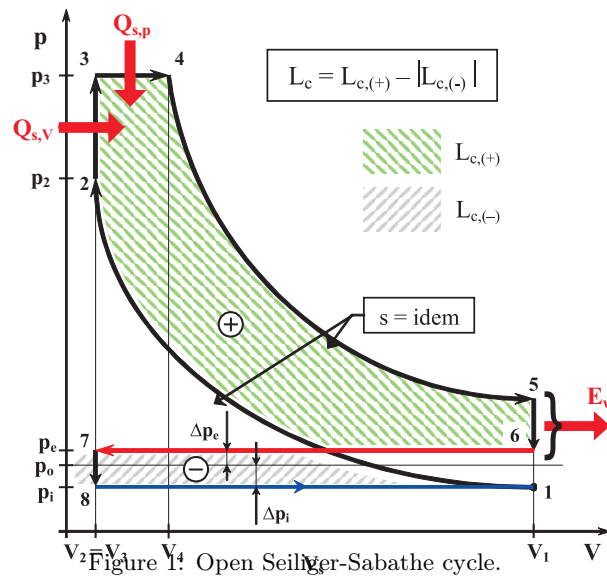
2 Realization of the Atkinson-Miller cycle in spark-ignition engine

The open, theoretical Atkinson-Miller cycle has been assumed as the model of processes executed in a spark ignition engine. The open cycle has been obtained by modification of the ideal cycle by addition of the processes characterizing the charge exchange. Using fully variable inlet valve timing the Atkinson-Miller cycle can be realized according to two systems [3]:

- system with late inlet valve closing – Fig. 2,
- system with early inlet valve closing – Fig. 3.

More detailed analysis of parameters is presented as an example only for the LIVC system in the next Section 3. Majority of the figures contain comparison of both LIVC and EIVC systems as well as the open, ideal

Seiliger-Sabathe cycle with classic throttle governing of engine load as the reference cycle for evaluation of benefits and the work effectiveness of the analysed systems.
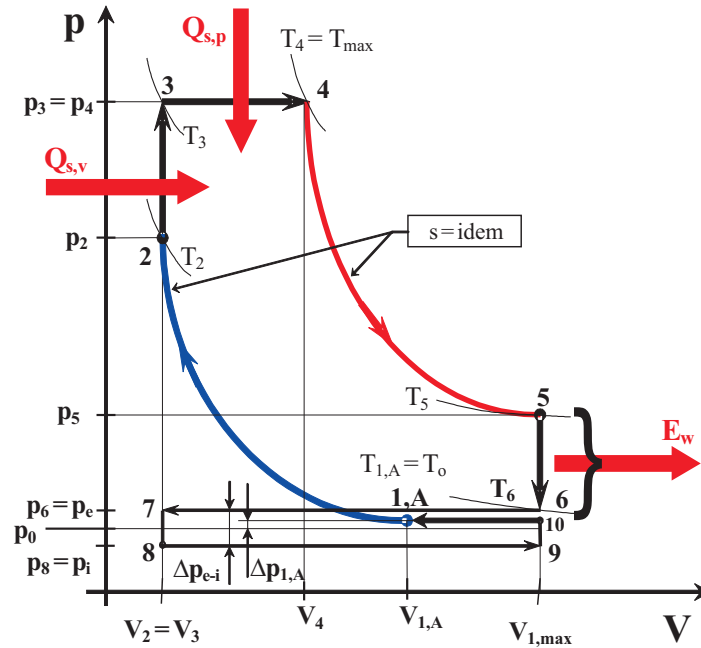

Figure 2: Open, ideal cycle for the system with late inlet valve closing (LIVC).

3 The system with late inlet valve closing (LIVC)

3.1 Basic characteristics of the cycle

The volume $V_{1,A}$ of a cylinder, at which the intake valve closing occurs during compression stroke, is the control parameter of load (the filling). Simultaneously, this is the parameter adjusting the mass of a fresh air-fuel mixture fed into a cylinder. The volume $V_{1,A}$ can be divided by the minimal cylinder volume V_2, defining the isentropic compression ratio:

$$\varepsilon_A = \frac{V_{1,A}}{V_2} \,, \qquad 1 < \varepsilon_A \le \varepsilon \,. \tag{1}$$

The relative values of the control parameter, ε_A, (in relation to the compression ratio ε) depending on the cycle work are presented in Fig. 4.
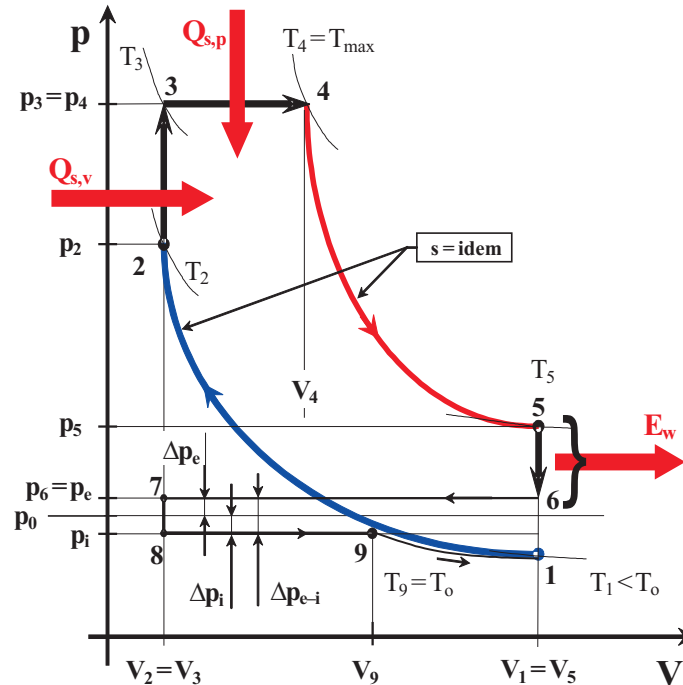
Figure 3: Open, ideal cycle for the system with early inlet valve closing (EIVC).

Near-linear interdependence between the control parameter and the cycle work is favourable in respect of load governing.

The pressure drop Δp_e determines the flow resistance in the exhaust system and the pressure drop $\Delta \mathrm{p}_i$ determines the flow resistances in the intake system. Whereas pressure drop $\Delta p_{1,A}$ determines the resistance of the back-flow of the air-fuel mixture, which excess is pushed again into the inlet manifold during compression stroke. In the cycle analysis, the following assumptions were made:

- the filling process finishes at the point '1,A', Fig. 2; at the volume $V_{1,A} \leq V_{1,max}$, pressure $p_{1,A} = p_0 + \Delta p_{1,A}$ and temperature $T_{1,A} = T_0$, where p_0 is the ambient pressure;

- compression ratio $\varepsilon = 10$,

- heat distribution number $\Psi = 0.9$ (for SI engine) [4],

- relative pressure drop in the inlet and exhaust systems $\Delta p_i/p_0 = \Delta p_e/p_0 = \Delta p_{1,A}/p_0 = 0.1$.
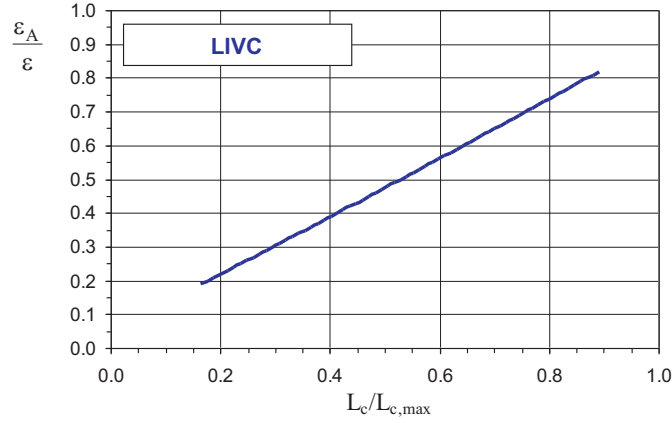
Figure 4: Control parameter $\varepsilon_A/\varepsilon$ of the system with LIVC cycle versus cycle work.

3.2   Fuel dose

The maximum mass, m_0, of the fresh charge is fed into a cylinder at an absence of the exhaust residue and when intake valve closing occurs at bottom dead centre, then

$$V_{1,A} = V_{1,max}, \text{ that is } \varepsilon_A = \varepsilon ,$$

and for simultaneous absence of the flow resistance in the inlet and exhaust systems

$$\Delta p_e = 0, \Delta p_i = 0, \Delta p_{1,A} = 0 , \quad \text{then} \quad p_{1,A} = p_0 .$$

The maximum mass of the fresh mixture can be described by the formula

$$m_0 = \frac{p_0 \, V_{1,\max}}{(MR) \, T_0} M_m , \tag{2}$$

where M_m is the molar mass of the fresh mixture.

For the made assumptions, the basic fuel dose amounts to

$$m_{f,0} = \frac{p_0 \, V_{1,\max}}{(MR) \, T_0} \; \frac{M_m}{[1 + \lambda_0 \, n_{a,\min} M_a \, (1 + X_a)]} \; . \tag{3}$$

The fuel dose m_f depends on the engine load. The basic parameters influencing the fuel dose are the following:

$V_{1,A}$ –   cylinder volume at the moment of the intake valve closing that is isentropic compression ratio ε_A (the control parameter),

T_0 –   temperature of the fresh charge,

Δp_i –   pressure drop in the inlet system, during filling,

$\Delta p_{1,A}$ –   pressure drop in the inlet system, during back-flow of the mixture,

λ –   excess air number.

For partial load, the cylinder volume, $V_{1,A}$, of the intake valve closing ranges

$$V_2 < V_{1,A} \le V_{1,\max} , \text{ hence } 1 < \varepsilon_A \le \varepsilon .$$

The flow resistances in the inlet and exhaust systems are taken into consideration:

$$\Delta p_e \ge 0, \ \Delta p_i \ge 0 , \ \Delta p_{1,A} \ge 0 \text{ so } p_{1,A} \ge p_0$$

and an assumption is made that the temperature of the fresh charge is equal to the ambient temperature T_0. Then, the fuel mass, m_f, amounts to

$$m_f = \frac{p_{1,A} V_{1,A}}{(MR) T_0} \frac{M_m}{[1 + \lambda n_{a,\min} M_a (1 + X_a)]} . \tag{4}$$

Relative fuel dose for the partial loads of an engine results from Eqs. (3) and (4)

$$\frac{m_f}{m_{f,0}} = \frac{p_{1,A} V_{1,A}}{p_0 V_{1,\max}} \frac{1 + \lambda_0 n_{a,\min} M_a (1 + X_a)}{1 + \lambda n_{a,\min} M_a (1 + X_a)} . \tag{5}$$

For the assumption that $\lambda = idem$, the following relation is obtained:

$$m_f = m_{f,0} \frac{p_{1,A} V_{1,A}}{p_0 V_{1,\max}} , \tag{6}$$

that can also be noted as

$$m_f = m_{f,0} \left(1 + \frac{\Delta p_{1,A}}{p_0}\right) \frac{\varepsilon_A}{\varepsilon} . \tag{7}$$

A change of the engine load is achieved by a change of the fuel dose, m_f, and the isentropic compression ratio, ε_A, which is the principal control parameter. The relative fuel dose, $m_f/m_{f,0}$, depending on the work of the LIVC cycle is presented in Fig. 5.

Relative reduction of the fuel dose, $\Delta m_f/m_{f,SS}$, for the LIVC cycle and EIVC cycle, in comparison with the system with the classic, throttle
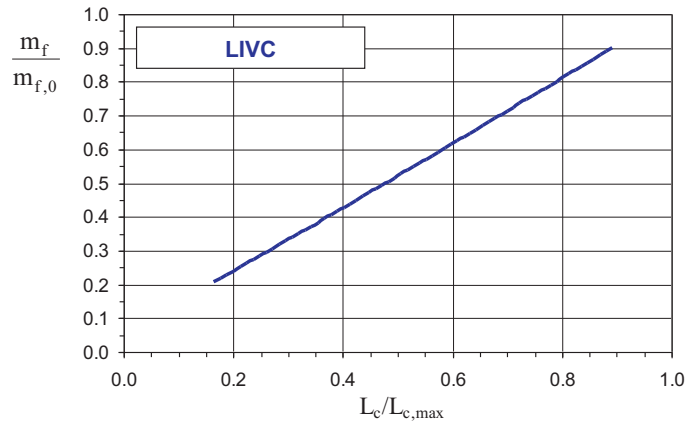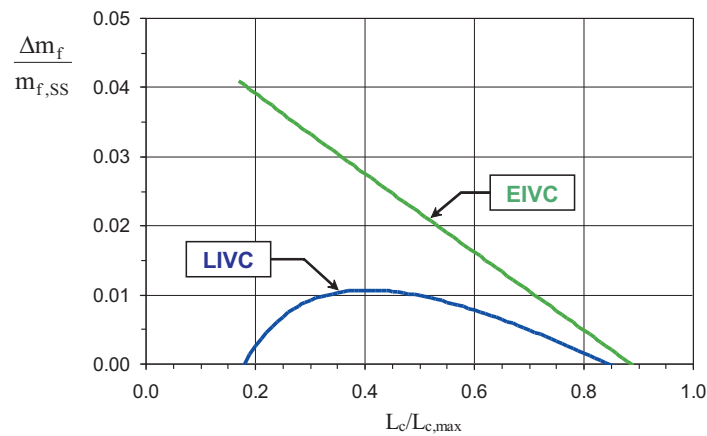
Figure 5: Relative fuel dose $m_f/m_{f,0}$ versus work of the LIVC cycle.

Figure 6: Relative reduction of the fuel dose for the LIVC cycle and the EIVC cycle compared with the open Seiliger-Sabathe cycle.

governing (Seiliger-Sabathe cycle) is illustrated in Fig. 6. For the LIVC the peak decrease of the fuel dose is achieved for the load $L_c/L_{c,max} = 0.4$. Unfortunately, the fuel economy for this system is not large and amounts to slightly above 1%. Much greater fuel economy is effected using the EIVC system. This system gives 4% decrease of the fuel consumption at low load of the engine.

3.3   Work of the cycle

The work of the LIVC cycle, Fig. 2, can be expressed as the sum of the component absolute works:

$$L_c = L_{1,A-2} + L_{2-3} + L_{3-4} + L_{4-5} + L_{5-6} +$$
$$+ L_{6-7} + L_{7-8} + L_{8-9} + L_{9-10} + L_{10-1,A} \;. \tag{8}$$

The cycle work, L_c, formulated below relatively, is received by inserting relations expressing the absolute works of the individual processes to (8):

$$\frac{L_c}{p_{1,A}\,V_{1,A}} = -\frac{\varepsilon_A^{(\kappa-1)} - 1}{\kappa - 1} + \gamma\,(\varphi - 1)\,\varepsilon_A^{(\kappa-1)} + \frac{\gamma\,\varphi}{\kappa - 1} +$$
$$+ \left(\frac{\varepsilon_A}{\varepsilon}\right)^{(\kappa-1)} \left[\varepsilon^{(\kappa-1)} - \varphi^{(\kappa-1)}\right]$$
$$- \left(\frac{\frac{\Delta p_i}{p_0} + \frac{\Delta p_e}{p_0}}{1 + \frac{\Delta p_{1,A}}{p_0}}\right) \left(\frac{\varepsilon - 1}{\varepsilon_A}\right) - \left(\frac{\varepsilon - \varepsilon_A}{\varepsilon_A}\right) \;, \tag{9}$$

where γ and ϕ are the load parameters [4]: $\gamma = \frac{p_3}{p_2}$, $\varphi = \frac{V_4}{V_3}$.

The specific work $L_c/(p_0 V_{1,max})$ of the LIVC cycle versus control parameter $\varepsilon_A/\varepsilon$ is presented in Fig. 7 and the cycle work in relation to the maximum work of the ideal Seiliger-Sabathe cycle is illustrated in Fig. 8. Characteristic curves in both figures are near-linear which is beneficial for governing reasons.
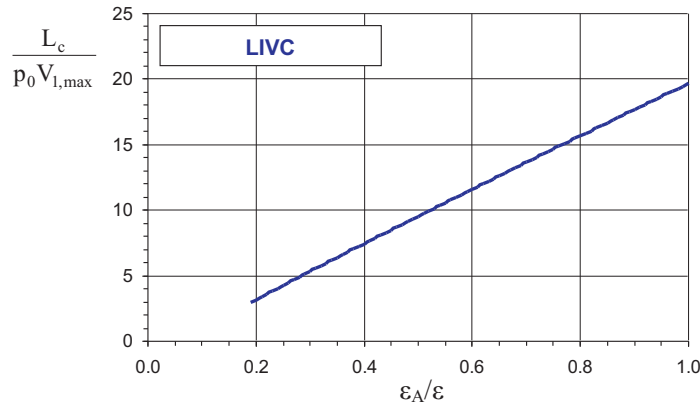

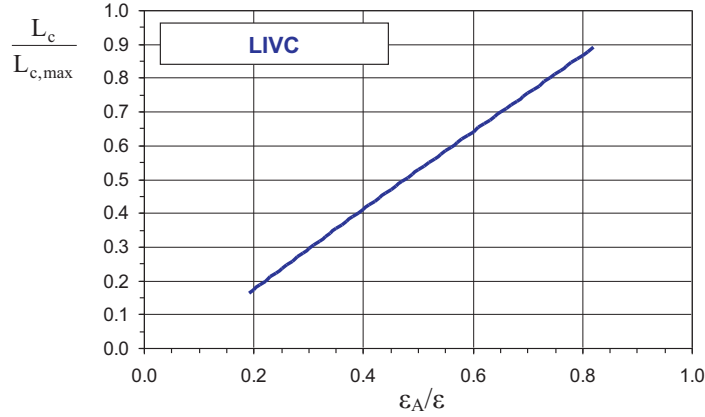Figure 7: Specific work $L_c/(p_0 V_{1,max})$ of the LIVC cycle versus control parameter $\varepsilon_A/\varepsilon$.

Figure 8: Ratio of work of the LIVC cycle to the maximal work of the Seiliger-Sabathe cycle versus control parameter $\varepsilon_A/\varepsilon$.

3.4 Charge exchange work

The charge exchange work L_{ex} of the LIVC cycle, Fig. 2, can be expressed as the sum of the component useful works:

$$L_{ex} = L_{6-7} + L_{7-8} + L_{8-9} + L_{9-10} + L_{10-1,A} \,. \tag{10}$$

The specific charge exchange work, L_{ex}, in relation to $(p_{1,A}V_{1,A})$ is obtained inserting relations expressing the useful works of the individual processes to the formula (10)

$$\frac{L_{ex}}{p_{1,A}\,V_{1,A}} = -\frac{(\varepsilon - 1)\left(\frac{\Delta p_i}{p_0} + \frac{\Delta p_e}{p_0}\right) + (\varepsilon - \varepsilon_A)\,\frac{\Delta p_{1,A}}{p_0}}{\varepsilon_A\left(1 + \frac{\Delta p_{1,A}}{p_0}\right)} \,. \tag{11}$$

The index μ of the relative charge exchange work is calculated by definition:

$$\mu = \frac{|L_{ex}|}{L_c} = \frac{\left|\frac{L_{ex}}{p_{1,A}\,V_{1,A}}\right|}{\frac{L_c}{p_{1,A}\,V_{1,A}}} \,, \tag{12}$$

as a ratio of the charge exchange work (11) to the cycle work (9).

The specific charge exchange work $L_{ex}/(p_0V_{1,max})$ for the LIVC and the EIVC (for comparison) depending on the cycle work is presented in Fig. 9. Absolute value of the charge exchange work for the EIVC system decreases when the cycle work decreases that is especially advantageous for

an engine work at low load resulting in increase of the energy efficiency. By this reason, within the range of small loads, considerable reduction of the relative charge exchange work, μ, of the EIVC system in relation to the LIVC system and the Seiliger-Sabathe cycle is observed which value is very small and amounts to only circa 4%, Fig. 10.
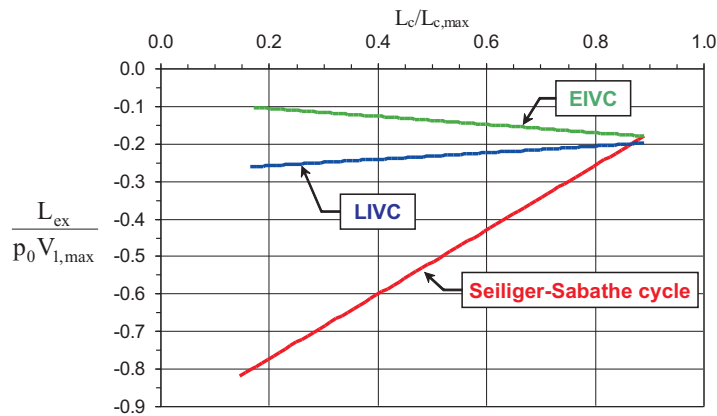

Figure 9: Comparison of the specific charge exchange works $L_{ex}/(p_0 V_{1,max})$ for the LIVC, EIVC and Seiliger-Sabathe cycles.
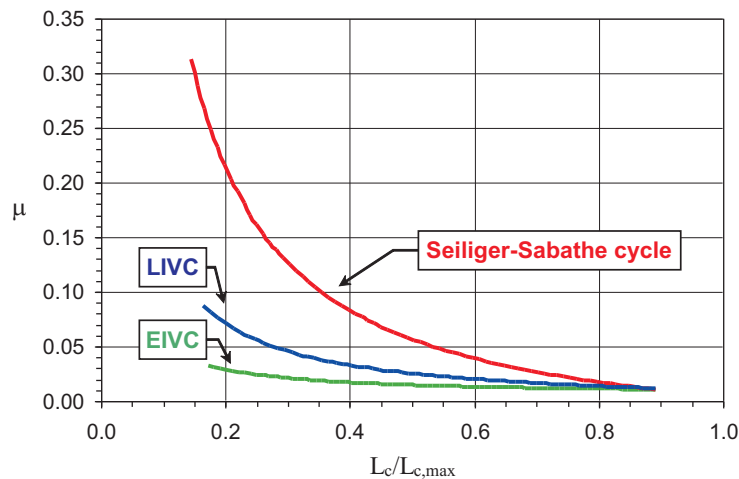

Figure 10: Comparison of the relative charge exchange works, μ, for the LIVC, EIVC and Seiliger-Sabathe cycles.

The open, ideal Seiliger-Sabathe cycle with generally applied, classic throttle governing of an engine load, being a model of the internal processes

proceeding in the typical SI engine, is the reference cycle for evaluation of benefits and the work efficiency in consequence of use of the systems with LIVC or EIVC. Therefore for comparison, characteristics of the specific charge exchange work and the relative charge exchange work for the open Seiliger-Sabathe cycle are also presented in Figs. 9 and 10. These works for the EIVC cycle are considerably smaller particularly within the range of part load operation.

3.5   Efficiency of the cycle

Efficiency of a cycle is defined as a ratio of the cycle work, L_c, to the supplied heat, Q_s:

$$\eta_t = \frac{L_c}{Q_s} \,, \tag{13}$$

which can also be formulated using the relative quantities

$$\eta_t = \frac{\frac{L_c}{p_{1,A} V_{1,A}}}{\frac{Q_s}{p_{1,A} V_{1,A}}} = \frac{\frac{L_c}{p_{1,A} V_{1,A}}}{E_0} \,. \tag{14}$$

Next, inserting the energy-stoichiometric parameter E_0 [3,4] and (9) to (14), the following formula is obtained:

$$\eta_t = \frac{\kappa - 1}{\varepsilon_A^{(\kappa-1)} \left[\gamma - 1 + \kappa\gamma \left(\varphi - 1 \right) \right]}$$

$$\times \left\{ \begin{array}{l} -\frac{\varepsilon_A^{(\kappa-1)}-1}{\kappa-1} + \gamma \left(\varphi - 1 \right) \varepsilon_A^{(\kappa-1)} + \frac{\gamma\varphi}{\kappa-1} \left(\frac{\varepsilon_A}{\varepsilon} \right)^{(\kappa-1)} \left[\varepsilon^{(\kappa-1)} - \varphi^{(\kappa-1)} \right] - \\ -\left(\frac{\frac{\Delta p_i}{p_0} + \frac{\Delta p_e}{p_0}}{1 + \frac{\Delta p_{1,A}}{p_0}} \right) \left(\frac{\varepsilon - 1}{\varepsilon_A} \right) - \left(\frac{\varepsilon - \varepsilon_A}{\varepsilon_A} \right) \end{array} \right\} \,. \tag{15}$$

Comparison of the cycle efficiency for the systems LIVC, EIVC and the open Seiliger-Sabathe cycle depending on the cycle works is presented in Fig. 11. The efficiency of the LIVC cycle is higher than the efficiency of the open Seiliger-Sabathe cycle only within the range of medium load. Unfortunately, this increase of efficiency is inconsiderable. Whereas the efficiency of the EIVC cycle is about 2% higher within the range of low load of an engine.
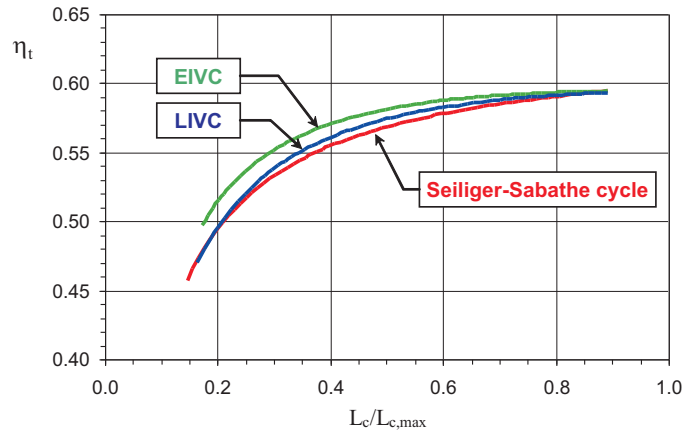
Figure 11: Comparison of efficiencies η_t of the LIVC, EIVC and Seiliger-Sabathe cycles.

4 Summary

There are two methods of realisation of the Atkinson-Miller cycle using fully variable inlet valve control. These are the late inlet valve closing (LIVC) and the early inlet valve closing (EIVC). The second system is more effective.

The open, ideal Seiliger-Sabathe cycle with classic throttle governing of the engine load is the reference cycle for evaluation of benefits and the work effectiveness of analysed systems. Effects of use of investigated systems can be expressed best of all by the energy efficiency of the cycle. Unfortunately, the efficiency of the LIVC cycle is not considerably higher than the efficiency of the Seiliger-Sabathe cycle. Thus, reduction of the fuel consumption is not too big as well. This means that load governing of the SI engine according to the system with late intake valve closing does not produce desired results. Better results are attained for the EIVC system – higher energy efficiency of the cycle and lower fuel consumption.

The use of the analysed systems to governing of the SI engine load will enable to eliminate a throttling valve from the system inlet and reduce the charge exchange work, especially within the range of part load operation. Decrease of the charge exchange work leads to increase of the internal and effective works, which results in increase of the effective efficiency of the spark ignition engine.

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