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LATE INTAKE VALVE CLOSING AS A WAY OF THE THROTTLELESS CONTROL OF SI ENGINE LOAD

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Abstract

The paper tackles the problems connected with the charge exchange in internal combustion engines. The theoretical analysis of the charge exchange process in the SI engine has been presented. The realization of the charge exchange process is connected with the necessity of overcoming the flow resistances, then with the necessity of doing a work, so-called the charge exchange work. The flow resistance caused by throttling valve is especially high at the partial load running of an engine. A system with independent, late intake valve closing has been analysed. The use of the analysed system to governing of an engine load will enable to eliminate a throttling valve from inlet system and reduce the charge exchange work, especially within the range of partial load. The decrease of the charge exchange work leads to an increase of the internal and effective works, which results in an increase of the effective efficiency of the spark ignition engine. The open, theoretical Atkinson-Miller cycle has been assumed as a model of processes proceeding in the engine with variable intake valve actuation. The system has been analysed individually and comparatively with open Seiliger-Sabathe cycle, which is theoretical cycle for the classic throttle governing of engine load. Benefits resulting from application of the system with late intake valve closing have been assessed on the basis of the selected parameters: a fuel dose, a cycle work, relative charge exchange work and cycle efficiency.

Keywords: *spark-ignition engine, variable valve actuation, late intake valve closing, open Atkinson-Miller cycle, charge exchange process, cycle efficiency*

1. Introduction

Realization of a charge exchange process in a combustion engine requires surmounting flow resistances in inlet and exhaust systems so execution of an adequate work co-called the charge exchange work. A charge exchange system has an essential impact on effectiveness of an engine work. Each element installed in the charge exchange system generates flow resistance of a fresh charge in an intake system and combustion products in an exhaust system. These resistances bring about an increase of the charge exchange work, which contributes to a decrease of the internal, and the effective work of an engine. Increase of the charge exchange work for partial load in the spark-ignition engine is connected with a method of load control. Quantity governing with the aid of a throttle, installed in an intake system, is disadvantageous especially from thermodynamic point of view because throttling generates losses of exergy [7, 8].

The use of the independent intake and exhaust valve actuation has been proposed in order to increase of efficiency of the open, ideal cycle and effective efficiency of the spark-ignition engine [9]. Theoretical research of the system with late intake valve closing has been carried out. This system enables elimination of a choke valve from an intake system of SI engine. The open Seiliger-Sabathe cycle (presented in the Fig. 1) is the reference cycle for an assessment of advantages and effectiveness of work gaining in consequence of application of the late intake valve closing.

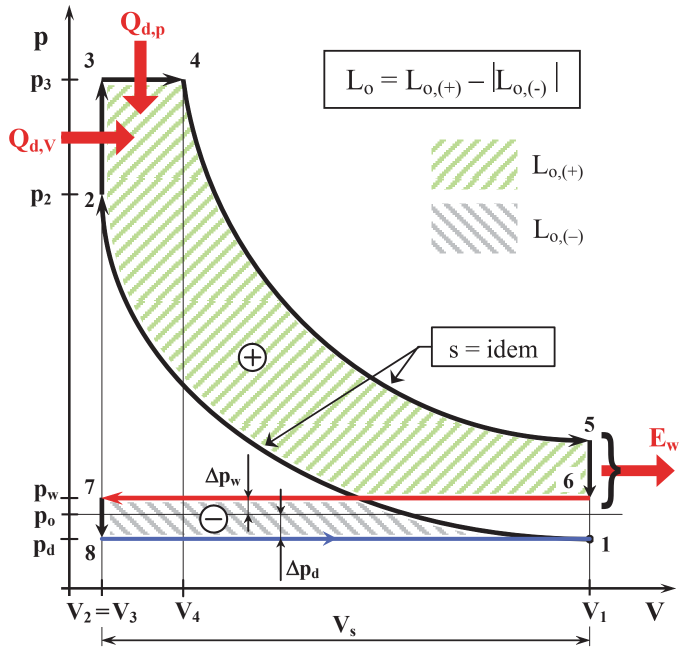

Fig. 1. Open Seiliger-Sabathe cycle, i.e. ideal cycle taking resistances in the inlet and exhaust systems into consideration

2. The system with late intake valve closing

2.1. Basic characteristics of the cycle

The open, theoretical Atkinson-Miller cycle has been assumed as the model of the processes proceeding in an engine with late intake valve closing. The open cycle has been obtained by modification of the ideal cycle by addition of the processes characterizing the charge exchange (the Fig. 2) [5].

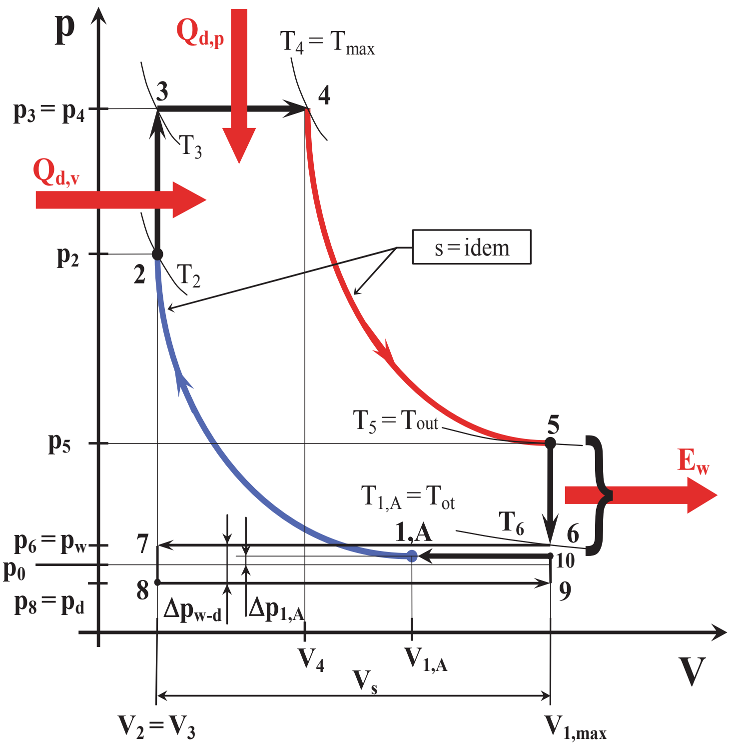

Fig. 2. System with late intake valve closing – the open, theoretical Atkinson-Miller cycle

The volume $V_{1,A}$ of a cylinder, at which an intake valve closing occurs during compression stroke, is the control parameter of load (the filling). Simultaneously, this is 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 the Fig. 3. Near-linear interdependence between the control parameter and the cycle work is favourable in respect of load governing.

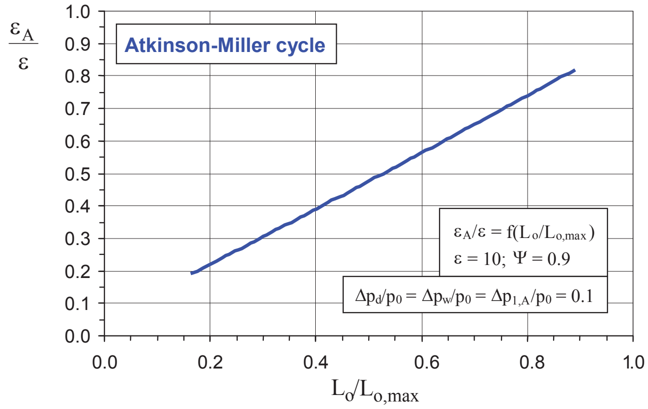

Fig. 3. Control parameter $\varepsilon_A/\varepsilon$ of the open Atkinson-Miller cycle versus cycle work

The pressure drop Δp_w determines the flow resistance in an exhaust system and the pressure drop Δp_d determines the flow resistances in an 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 an inlet manifold during compression stroke. In the cycle analysis, the assumption was made that the filling process finishes in the point „1,A"at (the Fig. 2):

the volume $V_{1,A} \le V_{1,max}$, the pressure $p_{1,A} = p_0 + \Delta p_{1,A}$ and the temperature $T_{1,A} = T_0$.

2.2. Fuel dose

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

$$\Delta p_w = 0 \ \Delta p_d = 0, \ \Delta p_{1,A} = 0 \text{ then } p_{1,A} = p_0.$$

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

$$m_0 = \frac{p_0 V_{1,max}}{(MR)T_0} M_m, \quad p_0 \approx p_{ot}, \quad T_0 \approx T_{ot}, \tag{2}$$

where: M_m – molar mass of the fresh mixture.

For the made assumptions, the basic fuel dose amounts to:

$$m_{p,0} = \frac{p_0 V_{1,max}}{(MR)T_0} \frac{M_m}{\left[1 + \lambda_0 n'_{a,min} M_a (1 + X_a)\right]}. \tag{3}$$

The fuel dose m_p depends on an 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_d – 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:

$$\Delta p_w \ge 0 \quad \Delta p_d \ge 0 \quad \Delta p_{1,A} \ge 0 \text{ so } p_{1,A} \ge p_0$$

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_p amounts to:

$$m_p = \frac{p_{1,A} V_{1,A}}{(MR) T_0} \frac{M_m}{\left[1 + \lambda n'_{a,\min} M_a \left(1 + X_a\right)\right]}. \tag{4}$$

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

$$\frac{m_p}{m_{p,0}} = \frac{p_{1,A} V_{1,A}}{p_0 V_{1,\max}} \frac{1 + \lambda_0 n'_{a,\min} M_a \left(1 + X_a\right)}{1 + \lambda n'_{a,\min} M_a \left(1 + X_a\right)}. \tag{5}$$

For the assumption that λ = idem, the following relation is obtained:

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

that can also be noted as:

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

Therefore, a change of the engine load is achieved by the change of the fuel dose m_p and the isentropic compression ratio ε_A is the principal control parameter. The relative fuel dose $m_p/m_{p,0}$ depending on the work of the open Atkinson-Miller cycle is presented in the Fig. 4.

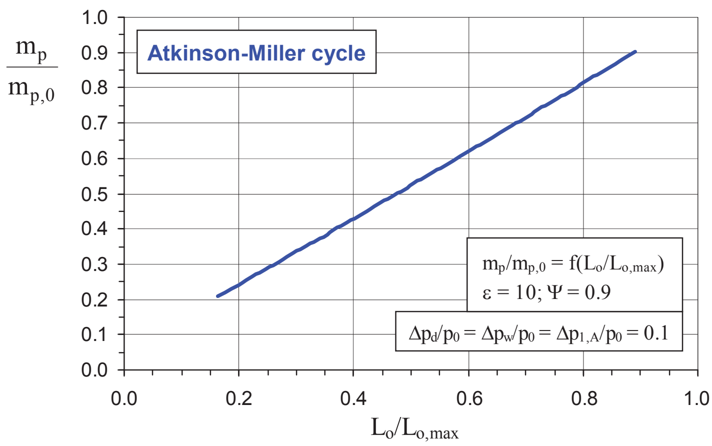

Fig. 4. Relative fuel dose $m_p/m_{p,0}$ versus work of the open Atkinson-Miller cycle

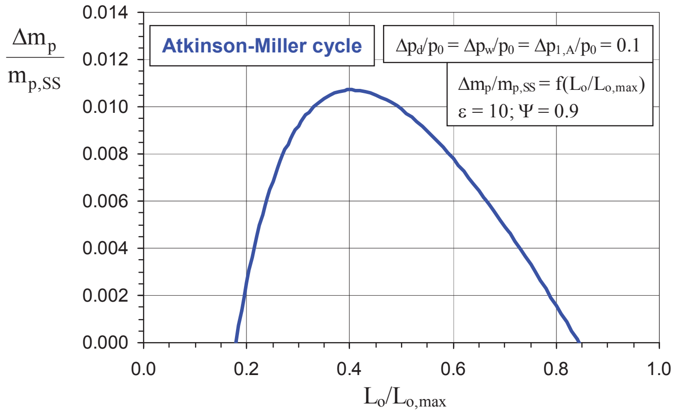

Fig. 5. Relative reduction of the fuel dose for the open Atkinson-Miller cycle compared with the open Seiliger-Sabathe cycle

Relative reduction of the fuel dose $\Delta m_p/m_{p,SS}$ for the open Atkinson-Miller cycle, in comparison with the system with the classic, throttle governing (the open Seiliger-Sabathe cycle) is illustrated in the Fig. 5. The peak decrease of the fuel dose is achieved for the load $L_o/L_{o,max} = 0.4$. Unfortunately, the fuel economy is not large and amounts to slightly above 1%.

2.3. Work of the cycle

The work of the open, ideal Atkinson-Miller cycle (the Fig. 2) can be expressed as the sum of the component absolute works [7]:

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

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

$$\frac{L_o}{p_{1,A}V_{1,A}} = -\frac{\varepsilon_A^{(\kappa-1)}-1}{\kappa-1} + \gamma(\quad -1)\varepsilon_A^{(\kappa-1)} + \frac{\gamma}{\kappa-1}\left(\frac{\varepsilon_A}{\varepsilon}\right)^{(\kappa-1)}\left[\varepsilon^{(\kappa-1)} - \quad {}^{(\kappa-1)}\right] - $$
$$-\left(\frac{\frac{\Delta p_d}{p_0}+\frac{\Delta p_w}{p_0}}{1+\frac{\Delta p_{1,A}}{p_0}}\right)\left(\left(\frac{\varepsilon-1}{\varepsilon_A}\right)-\left(\frac{\varepsilon-\varepsilon_A}{\varepsilon_A}\right)\right), \tag{9}$$

where γ and φ are the load parameters [5]: $\gamma = \frac{p_3}{p_2}$, $\varphi = \frac{V_4}{V_3}$.

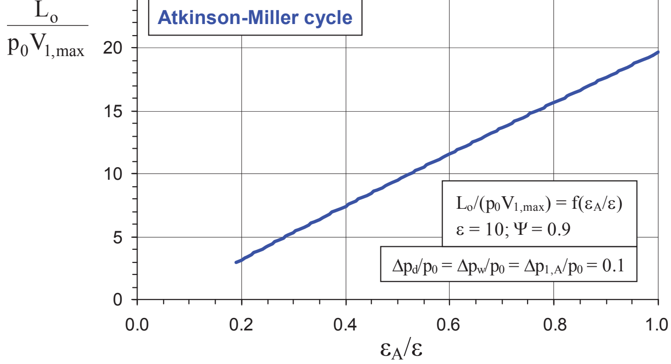

Fig. 6. Specific work $L_o/(p_0V_{1,max})$ of the open Atkinson-Miller cycle versus control parameter $\varepsilon_A/\varepsilon$

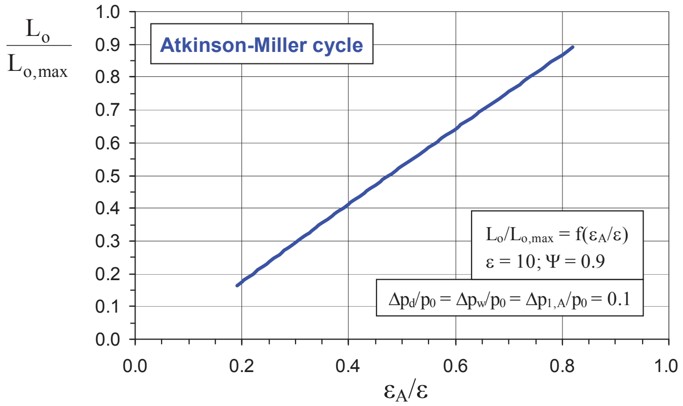

Fig. 7. Ratio of work of the open Atkinsom-Miller cycle to the maximal work of the Seiliger-Sabathe cycle versus control parameter $\varepsilon_A/\varepsilon$

The specific work $L_o/(p_0V_{1,max})$ of the open Atkinson-Miller cycle versus control parameter $\varepsilon_A/\varepsilon$ is presented in the Fig. 6 and the cycle work in relation to the maximum work of the ideal Seiliger-Sabathe cycle is illustrated in the Fig. 7. Characteristic curves in the both figures are near linear which is beneficial for governing reasons.

2.4. Charge exchange work

The charge exchange work L_w of the open, ideal Atkinson-Miller cycle (the Fig. 2) can be expressed as the sum of the component useful works [4]:

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

The specific charge exchange work L_w 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):

$$\frac{L_w}{p_{1,A}\,V_{1,A}} = -\frac{(\varepsilon-1)\left(\frac{\Delta p_d}{p_0}+\frac{\Delta p_w}{p_0}\right)+(\varepsilon-\varepsilon_A)\frac{\Delta p_{1,A}}{p_0}}{\varepsilon_A\left(1+\frac{\Delta p_{1,A}}{p_0}\right)}\,. \tag{11}$$

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

$$\mu = \frac{|L_w|}{L_o} = \frac{\left|\frac{L_w}{p_{1,A}\,V_{1,A}}\right|}{\frac{L_o}{p_{1,A}\,V_{1,A}}}\,, \tag{12}$$

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

The specific charge exchange work $L_w/(p_0V_{1,max})$ for the open Atkinson-Miller cycle depending on the cycle work is presented in the Fig. 8. Absolute value of the charge exchange work increases when the cycle work decreases. By this reason, within the range of small loads, an increase of the relative charge exchange work μ is observed which value amount to circa 9% (the Fig. 9).

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 system with late intake valve closing. Therefore, for comparison, characteristics of the specific charge exchange work and the relative charge exchange work for the open Seiliger-Sabathe cycle are presented in the Fig. 8 and 9 respectively. These works for Atkinson-Miller cycle are considerably smaller particularly within the range of low loads.

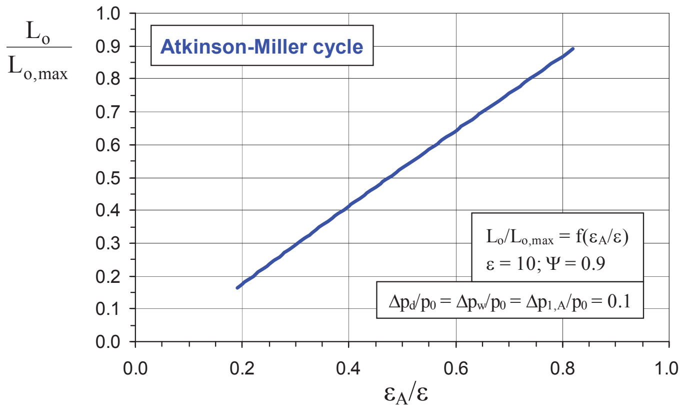

Fig. 8. Comparison of the specific charge exchange works $L_w/(p_0V_1)$ for the Atkinson-Miller and Seiliger-Sabathe cycles versus work of the cycles

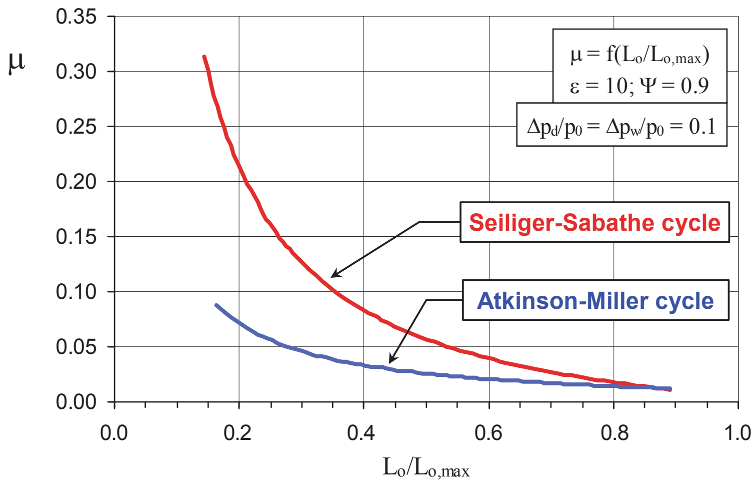

Fig. 9. Comparison of the relative charge exchange works μ for the Atkinson-Miller and Seiliger-Sabathe cycles versus work of the cycles

2.5. Efficiency of the cycle

Efficiency of an ideal cycle is defined as a ratio of the cycle work L_o to the supplied heat Q_d:

$$\eta_o = \frac{L_o}{Q_d}, \tag{13}$$

which can also be formulated using the relative quantities?

$$\eta_o = \frac{\dfrac{L_o}{p_{1,A}\,V_{1,A}}}{\dfrac{Q_d}{p_{1,A}\,V_{1,A}}} = \frac{\dfrac{L_o}{p_{1,A}\,V_{1,A}}}{E_0}. \tag{14}$$

Next, inserting the energy-stoichiometric parameter E_0 [5, 9] and (9) to (14), the following formula is obtained:

$$\eta_o = \frac{\kappa - 1}{\varepsilon_A^{(\kappa-1)}\left[\gamma - 1 + \kappa\gamma(\varphi - 1)\right]}\left\{\begin{array}{l} -\dfrac{\varepsilon_A^{(\kappa-1)} - 1}{\kappa - 1} + \gamma(\varphi - 1)\varepsilon_A^{(\kappa-1)} + \dfrac{\gamma\varphi}{\kappa - 1}\left(\dfrac{\varepsilon_A}{\varepsilon}\right)^{(\kappa-1)}\left[\varepsilon^{(\kappa-1)} - \varphi^{(\kappa-1)}\right] - \\ -\left(\dfrac{\dfrac{\Delta p_d}{p_0} + \dfrac{\Delta p_w}{p_0}}{1 + \dfrac{\Delta p_{1,A}}{p_0}}\right)\left(\dfrac{\varepsilon - 1}{\varepsilon_A}\right) - \left(\dfrac{\varepsilon - \varepsilon_A}{\varepsilon_A}\right) \end{array}\right\} . \quad (15)$$

The efficiency η_o is significant parameter, which enables assessment of the cycle in the energy aspect. Comparison of the cycle efficiencies for the system with late intake valve closing and the open Seiliger-Sabathe cycle depending on the cycle works is presented un the Fig. 10.

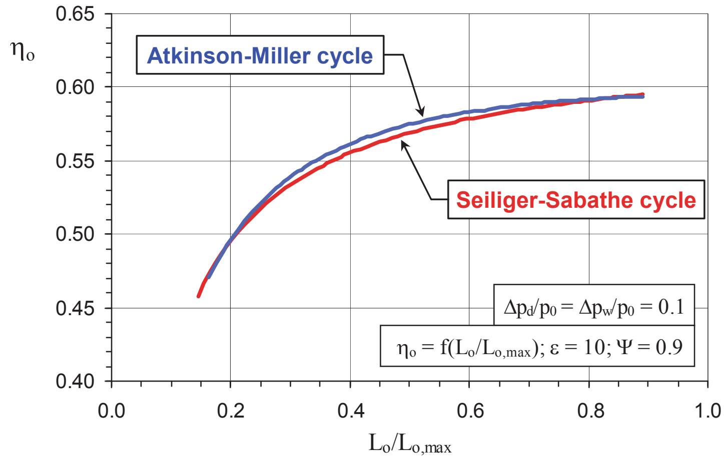

Fig. 10. Comparison of efficiencies η_o of the Atkinson-Miller and Seiliger-Sabathe cycles versus work of the cycles

The efficiency of the open Atkinson-Miller cycle is higher than the efficiency of the open Seiliger-Sabathe cycle only within the range of medium load. Unfortunately, this increase of the efficiency is inconsiderable.

3. Conclusion

Investigations on variable valve actuation are conducted by many scientific as well as research and development centres [1-3]. This testifies topicality of the presented problems within which the theoretical research on the system with independent late intake valve closing has been conducted. The open, ideal Atkinson-Miller cycle has been assumed as a model of the processes proceeding in a combustion engine working according to the analysed system.

The open, ideal Seiliger-Sabathe cycle with classic throttle governing of an engine load is the reference cycle for evaluation of benefits and the work effectiveness of the system with late intake valve closing. Effects of use of the investigated system can be expressed best of all by the energy efficiency of the cycle. Unfortunately, the efficiency of the open Atkinson-Miller 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.

Independent variable valve actuation enables control of an engine work also according to the other systems [5, 9] e.g.:

- early intake valve closing,
- early exhaust valve closing, making an internal exhaust gas recirculation possible,
- fully variable both valve actuation.

Analysis of the above systems has shown that they are more effective [5, 9] and will be presented in future publications.

References

[1] Cope, D., Wright, A., Corcoran, C., Pasch, K., *Fully flexible electromagnetic valve actuator: design, modeling, and measurements*, SAE Technical Paper, No. 2008-01-1350, 2008.

[2] Franca, O. M., *Impact of the Miller cycle in the efficiency of an FVVT engine during part load operation*, SAE Tech. Paper, No 2009-36-0081.

[3] Haas, M., Rauch M., *Electro-hydraulic fully variable valve train system*, ATZ autotechnology, 02, Vol. 10, 2010.

[4] Postrzednik, S., *Termodynamika zjawisk przepływowych*, Wydawnictwo Politechniki Śląskiej, Gliwice 2006.

[5] Postrzednik, S., Żmudka, Z., *Termodynamiczne oraz ekologiczne uwarunkowania eksploatacji tłokowych silników spalinowych*, Wydawnictwo Politechniki Śląskiej, Gliwice 2007.

[6] Postrzednik, S., Żmudka, Z., *Badania aplikacyjne nad procedurami niezależnego sterowania zaworami silnika spalinowego*, Projekt badawczy nr N502 026 32/2190. PBU-30/RIE-6/07, 05.2007 – 06.2009, Sprawozdanie, Gliwice 2009.

[7] Szargut, J., *Termodynamika techniczna*, Wydawnictwo Politechniki Śląskiej, Gliwice 2000.

[8] Żmudka, Z., Postrzednik, S., *Flow resistance in the engine inlet-exhaust system as affected by the engine parameters*, Combustion Engines, No. 1 (136), 2009.

[9] Żmudka, Z., *Energetyczne i ekologiczne aspekty doskonalenia procesu wymiany ładunku w silniku spalinowym*, Wydawnictwo Politechniki Śląskiej, Gliwice 2010.