ANALYSIS AND COMPARISON OF MODERN IC ENGINES COMPARATIVE CYCLES BY USING A GENERAL THERMODYNAMIC CYCLE

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INTRODUCTION

Describe the process of conversion heat into mechanical work in heat engines by a closed, thermodynamic cycle started Sadi Carnot (1796. ÷ 1832.), introducing the famous Carnot's cycle [8]. This cycle has the highest efficiency, but because of the wide changes interval of the characteristic state variables (pressure and volume) cannot be implemented in real heat engine.

With the advent of heat IC engines (Otto and Diesel), are introduced and the corresponding comparative cycles with isochoric (Otto) and isobaric heat addition (Diesel) cycle. The largest contribution to the development of Diesel engines represents, of course, the discovery of the fuel injection system under high pressure (James McKechnie, 1910).. This innovation led to the modification of the comparative Diesel engines cycle in which now used the combined method of the isochoric-isobaric heat addition in the literature known as: Sabathe's, Seiliger's or the combined cycle.

Regardless what is in the comparative cycle idealize the phases of the actual IC engines cycle (isochoric, isobaric and isochoric-isobaric heat addition, adiabatic compression and expansion and isochoric heat rejection), their study can define the basic relationship between the characteristic cycle parameters and efficiency, ie, the specific work that apply to the actual cycles. It is usual that this dependences is determined by the combined cycle, as well as the general, and from them, under pre-defined conditions, obtained expressions for the thermal efficiency and specific work of Otto and Diesel cycles.

Intensive development of Otto and Diesel engines for passenger cars, caused by strict regulations to improve the environmental characteristics and fuel consumption while preserving performance, influenced and the character of the cycles of modern engines. For this reason it was necessary to modify the existing comparative cycles and / or introduce new ones. Modern technologies that require or enable the implementation of new cycles are certainly GDI, HCCI engines, overboost, variable systems (VCR, VVT, VVL), EGR, fuel injection during the course of expansion, hybrid drive, etc.

Of the new comparative cycles which fullest characterize the actual working cycles of modern engines are certainly the most important: Atkinson's [2], Miller's [1], [4], [5] and HEHC [7] cycle, and cycles with isothermal expansion (Gruden [6]). Comparison of these cycles according to the criteria of economy (thermal efficiency) and effectiveness (specific work), involves the determination of the analytical expression of these values for each cycle individually, then assessment of the impact of cycle characteristic parameters on their values. In this paper just presents a method of using the general thermodynamic cycle (GTC) [3], for analysis and comparison of all existing comparative heat engine cycles.

In general thermodynamic cycle heat addition and rejection is isochoric-isobaric-isothermal (VPT) and the processes of compression and expansion are adiabatic, which allows to the appropriate combinations of these state changes derive all of the existing

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cycles (130). The formulas for the thermal efficiency, specific work and the values of the state variables (p, V, T, S) in the characteristic points of the general thermodynamic cycle are modeled by using Simulink program, so that the output can get graphical representations of state changes in the course of the cycle in the form of p-V and T-S diagrams, as well as diagrams depending on the thermal efficiency and specific work of any selected parameter. Setting a concrete cycle is carried out for selecting appropriate values of the input parameters in the model of the general thermodynamic cycle. Using this method can also be optimized values of the characteristic parameters of any cycle on the criterion of the thermal efficiency and specific work.

THE GENERAL THERMODYNAMIC CYCLE

The general thermodynamic cycle [1] is shown in p-V and T-s coordinate systems, Figure 1 and 2. As may bee seen from Figure 1 and 2, the general thermodynamic cycle (GTC) is obtained as a section of the cycle with maximal specific work (index “w”) and cycle with maximal thermal efficiency (index “c” – Carnot cycle).

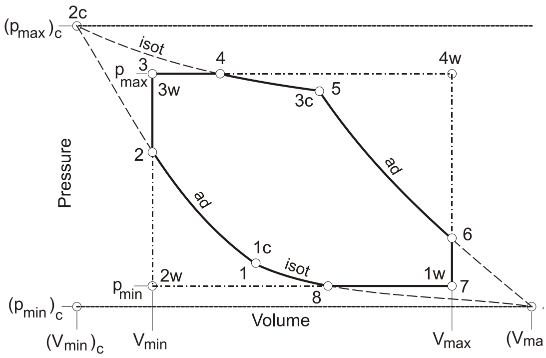

Figure 1 GTC cycle in p-V coordinates

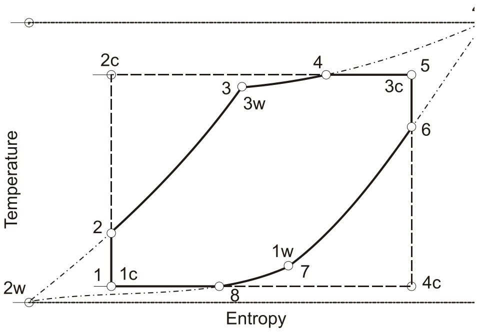

Figure 2 GTC cycle in T-s coordinates

This cycle is composed of the following processes:

- heat addition at constant volume (2 – 3), constant pressure (3 – 4) and constant temperature (4 – 5),
- compression (1 – 2) and expansion (5 – 6) without heat exchange with the surroundings (adiabatic processes),
- heat rejection at constant volume (6 – 7), constant pressure (7 – 8) and constant temperature (8 – 1).

The method of forming new cycle using the general thermodynamic cycle can be presented by schema in Figure 3

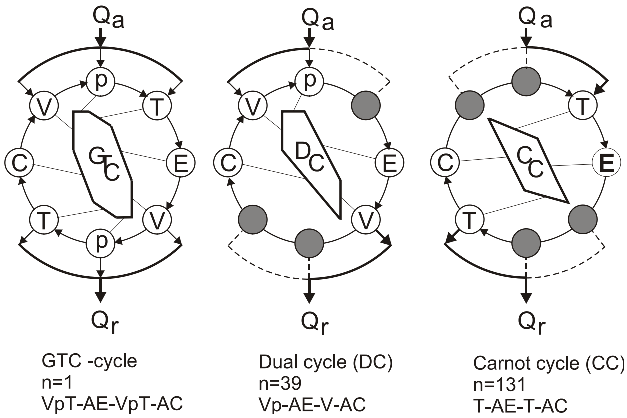

Figure 3 Schema of the new cycles formation from GTC cycle

The indices in Figure 3 have following meanings:

- Qa –total amount of heat added,
- QaV , Qap , QaT -amounts of heat added by constant volume, constant pressure and constant temperature, respectively,
- Qr –total amount of heat rejected,
- QrV , Qrp , QrT -amounts of heat rejected by constant volume, constant pressure and constant temperature, respectively,
- AC -adiabatic compression,
- AE -adiabatic expansion.

Combining the state changes in accordance with schema in Figure 3 it can obtain 130 new cycles. These cycles comprise all know cycles of the heat machines as well as cycles that up to now are not used. In Table 1, top 20 cycles in form of the excel table are presented.

For determination of the thermal efficiency and mean pressure (specific work) of the GTC, following dimensionless parameters are defined:

$\varepsilon = V_{max} / V_{min}$ (compression ratio), $V_{max} = V_6 = V_7$; $V_{min} = V_2 = V_3$
$x_p = p_{max} / p_{min}$, $p_{max} = p_3 = p_4$; $p_{min} = p_7 = p_8$
$x_T = T_{max} / T_{min}$, $T_{max} = T_4 = T_5$; $T_{min} = T_1 = T_8$
$\alpha_1 = p_3 / p_2$, $\alpha_2 = p_6 / p_7$
$\rho_1 = V_4 / V_3$, $\rho_2 = V_7 / V_8$
$\delta_1 = V_5 / V_4$, $\delta_2 = V_8 / V_1$

By using the relationships between these parameters it can write expression:

$$\frac{\alpha_1}{\alpha_2} = \left(\frac{\rho_2}{\rho_1}\right)^{\kappa} \left(\frac{\delta_2}{\delta_1}\right)^{\kappa-1} \tag{1}$$

Table 1 Excel table example of the heat machines cycles

Cycles designation	Cycle Nr.	Dimensionless input parameters						
		ε	α1	ρ1	ρ2	δ1	δ2	κ
V-p-T-AE-V-p-T-AC	1	10	3	1.3	1.3	1.5	1.5	1.4
V-p-T-V-p-T-AC	2	10	3	1.3	1.3	7.692308	1.5	1.4
V-p-T-AE-V-p-T	3	10	3	1.3	1.3	1.5	7.692308	1.4
V-p-T-V-p-T	4	10	3	1.3	1.3	7.692308	7.692308	1.4
V-p-T-AE-V-p-AC	5	10	3	1.3	1.3	1.5	1	1.4
V-p-T-V-p-AC	6	10	3	1.3	1.3	7.692308	1	1.4
V-p-T-AE-V-p	7	10	50	1.3	10	1.5	1	1.4
V-p-T-V-p	8	10	50	1.3	10	7.692308	1	1.4
V-p-T-AE-V-T-AC	9	10	3	1.3	1	1.5	1.5	1.4
V-p-T-V-T-AC	10	10	3	1.3	1	7.692308	1.5	1.4
V-p-T-AE-V-T	11	10	3	1.3	1	1.5	10	1.4
V-p-T-V-T	12	10	3	1.3	1	7.692308	10	1.4
V-p-T-AE-p-T-AC	13	10	3	1.3	1.3	1.5	1.5	1.4
V-p-T-p-T-AC	14	10	3	1.3	1.3	7.692308	1.5	1.4
V-p-T-AE-p-T	15	10	3	1.3	1.3	1.5	7.692308	1.4
V-p-T-p-T	16	10	3	1.3	1.3	7.692308	7.692308	1.4
V-p-T-AE-V-AC	17	10	3	1.3	1	1.5	1	1.4
V-p-T-V-AC	18	10	3	1.3	1	7.692308	1	1.4
V-p-T-AE-p-AC	19	10	3	1.3	1.3	1.5	1	1.4
V-p-T-p-AC	20	10	3	1.3	1.3	7.692308	1	1.4

THERMAL EFFICIENCY AND MEAN PRESSURE OF GTC

In accordance with the second law of thermodynamic, the thermal efficiency of the GTC is given by formula:

$$\eta_t = \frac{Q_a - Q_r}{Q_a} \tag{2}$$

where are:

$$Q_a = c_v T_{max}\left[\frac{1}{\alpha_1 \rho_1}(\alpha_1 - 1) + \frac{\kappa}{\rho_1}(\rho_1 - 1) + (\kappa - 1)\ln\delta_1\right] \tag{3}$$

$$Q_r = c_v T_{min}\left[\rho_2(\alpha_2 - 1) + \kappa(\rho_2 - 1) + (\kappa - 1)\ln\delta_2\right] \tag{4}$$

In upper expressions c_v is specific heat at constant volume, $\kappa = c_p / c_v$ is adiabatic exponent and mater amount is 1 (ideal gas). The maximal and minimal temperature values are determined by expression:

$$T_{max} = T_{min}\alpha_1\rho_1\left(\frac{\varepsilon}{\rho_2\delta_2}\right)^{(\kappa-1)} \tag{5}$$

Substituting Q_a and Q_r into formula (2) we obtain the expression for η_t :

$$\eta_t = 1 - \frac{1}{x_T}\alpha_1\rho_1 f(param) \tag{6}$$

where is:

$$f(param) = \frac{\rho_2(\alpha_2 - 1) + \kappa(\rho_2 - 1) + (\kappa - 1)\ln\delta_2}{(\alpha_1 - 1) + \kappa\alpha_1(\rho_1 - 1) + (\kappa - 1)\alpha_1\rho_1\ln\delta_1}$$

By using the relation (1), the variables number in equation (6) can be reduced for one. Equation (6) may be rewritten as a follow:

$$\eta_t = 1 - \frac{1}{\varepsilon^{\kappa-1}}(\rho_2\delta_2)^{\kappa-1} f(param) \tag{7}$$

and

$$\eta_t = 1 - \frac{1}{x_p^{\frac{\kappa-1}{\kappa}}}(\alpha_1\delta_2)^{\frac{\kappa-1}{\kappa}} f(param) \tag{8}$$

The expressions for thermal efficiency η_t, (6), (7) and (8) as a function of x_T, ε and x_p are given respectively.

The cycle mean pressure (specific work), is determined by expression:

$$p_t = \frac{W_{cyc}}{V_{max} - V_{min}} \tag{9}$$

where is: W_{cyc} –cycle work which is determined in p-V coordinates by the area inside the contour describing the GTC.

Taking into account the know dependences for ideal (air) cycles:

$$W_{cyc} = Q_a - Q_r \tag{10}$$

and

$$V_{max} - V_{min} = \frac{(\varepsilon - 1)}{\varepsilon}V_{max} \tag{11}$$

the equation (7) may be rewritten in following form:

$$p_t = p_{max}\frac{\rho_1 A}{\varepsilon - 1}\left(1 - \frac{1}{x_p}\frac{\varepsilon}{\rho_1\rho_2}\frac{B}{A}\right) \tag{12}$$

where are: $$A=\left(1-\frac{1}{\rho_1}\right)+\ln\delta_1+\frac{1}{\kappa-1}\left[1-\left(\frac{\rho_1\delta_1}{\varepsilon}\right)^{\kappa-1}\right]$$

$$B=(\rho_2-1)+\ln\delta_2+\frac{1}{\kappa-1}\left[1-\left(\frac{\varepsilon}{\rho_2\delta_2}\right)^{\kappa-1}\right]$$

$$p_{max}=p_{min}\,\alpha_1\left(\frac{\varepsilon}{\rho_2}\right)^{\kappa}\delta_2^{1-\kappa}$$

The equations (6),(7),(8) and (12), it can apply to all of the 131 cycles. The expressions for η_t and p_t , for a concrete cycle are obtained by inserting the corresponding values of the dimensionless parameters in these equations. In this manner it possible to estimate the influence of various parameters on the thermal efficiency and mean pressure for all cycles.

CYCLES COMPARISON

The comparison of heat-machine cycles is performed with purpose to choice an optimal cycle in consideration of the η_t and p_t values. By reason of larger parameters influence, the cycles comparison is performed for definite comparison cases. In literature following three comparison cases the most frequently are used:

$p_{max}=const.,\ Q_r=const.,\ \varepsilon\neq const.,\ Q_a\neq const.$

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$\varepsilon=const.,\ Q_a=const.,\ p_{max}\neq const.,\ Q_r\neq const.$

FIRST COMPARISON CASE

In this case the temperature T_1 and pressure p_{min} are known. For adopted parameters values: x_p and Q_r it calculated:

$p_{max}=x_p\,p_{min}=const.$

$Q_{rc}=Q_r/c_v\,T_1=const.$

and from equation for Q_r the compression ratio ε:

$$\varepsilon=\left(\frac{x_p\rho_2\rho_1^{\kappa}\delta_1^{\kappa-1}}{Q_{rc}-\kappa(\rho_2-1)-(\kappa-1)\ln\delta_2+\rho_2}\right)^{\frac{1}{\kappa}}\qquad(13)$$

Substituting the parameters values ($\rho_1,\ \rho_2,\ \delta_1,\ \delta_2$) for every cycles (Table 1), in expression (13), we obtain corresponding compression ratio value, and then the thermal efficiency and mean pressure of this cycle. By variation of the $\rho_1,\ \rho_2,\ \delta_1,\ \delta_2$ values it can estimate their influence on the η_t and p_t for this comparison case.

SECOND COMPARISON CASE

This comparison case is the same as preliminary besides: instead Q_r=*const.* it adopted Q_a=*const.*, and then is calculated:

$Q_{ac}=Q_a/c_vT_1=const.$

From the equation for Q_a the compression ratio ε now is defined in form:

$$\varepsilon^{\kappa} + C_1\,\varepsilon + C_2 = 0 \qquad (14)$$

$$C_1=Q_{ac}\,(\rho_2\,\delta_2)^{\kappa-1}$$

$$C_2 = x_p \rho_2^{\kappa}\delta_2^{\kappa-1}\left[\kappa(1-\rho_1)+(1-\kappa)\rho_1 \ln\delta_1 - 1\right]$$

By solution of the equation (14), are obtained the values for ε, and then the thermal efficiency η_t and cycle mean pressure p_t.

THIRD COMPARISON CASE

In this case for adopted parameters values:

$\varepsilon=const.$

$Q_a=const.$

it calculated:

$Q_{ac}=Q_a / c_vT_1=const.$

The equation for Q_a gives now the relationship between the others five parameters: α_1, ρ_1, ρ_2, δ_1, δ_2. This equation enables that for four adopted parameters values determines the fifth. Finally, as well as in preliminary cases, it determined the values η_t and p_t.

NEW METHOD FOR ANALYSIS AND COMPARISON OF THE COMPARATIVE IC ENGINES CYCLES

Based on the definition of the general thermodynamic cycle, the formula for calculating the state variables of the characteristic points of the cycle and the expression of the thermal efficiency and specifice work, developed a simulation model in Simulink which block diagram is shown in Figure 4.

Dimensionless values of input parameters (ε ... κ) may be constant (const.), or a variable in the range of its minimum (1.0) to the maximum value (max), where the selection of these variants is performed by using the switch S2. Constant values can be assigned discretely by input blocks or excel spreadsheets table (Table 1), wherein the choice of one or other variants perform with switch S1. The advantage of using excel tables is that at the input can be taken into account functional dependencies between different parameters. The output values are : added (Qa) and rejected (Qr) the amount of heat, thermal efficiency (ηt), cycle mean pressure (pt), p-V and T-S diagrams and depending on any two selected parameters, (eg. ηt = f (ε)). By setting the appropriate input values of dimensionless parameter (ε ... δ2), according to the described procedures of the individual cycles formation from the GTC , using this model it is possible for each of 130 cycles obtain the listed output parameters.

When analyzing the heat engines working cycles the special significance has their comparison by criteria of: economy (ηt) and effectiveness (pt), why is developed a simulation model for cycles comparison by foregoing criteria (three cases of comparison) shows by block diagram in Figure 4.

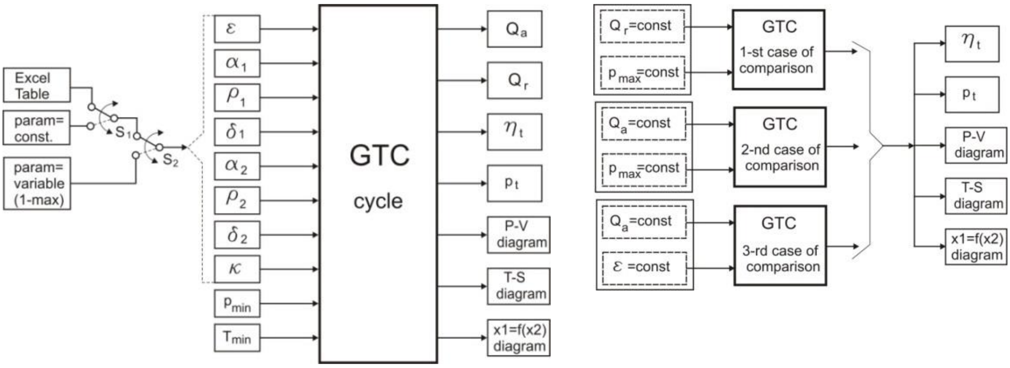

Figure 4 Simulation models for cycles analysis and comparison

In order to verify the developed model was compared to the three main IC engines comparative cycles (Otto, Diesel and Seiliger), and the results for the two comparison case ,(Qa = const., pmax = const), are shown by diagrams in the Figure 5 to 8 . The amount of heat Qa is determined by using a general thermodynamic cycle (the block diagram in Figure 4), for a combined cycle with the following values of dimensionless parameters: $\varepsilon = 15$, $\alpha 1=2$, $\rho 1 = 1.5$ ($\delta 1=1$, $\rho 2 = 1$, $\delta 2 = 1$) , $\kappa = 1.4$ and in dimensionless form is: $Qa / Cv\ Tmin = 7.09$ [-].

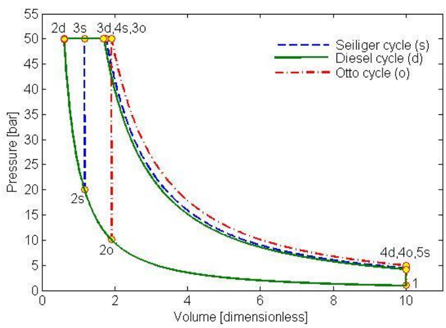

Figure 5 p-V diagram for 2- comparison case

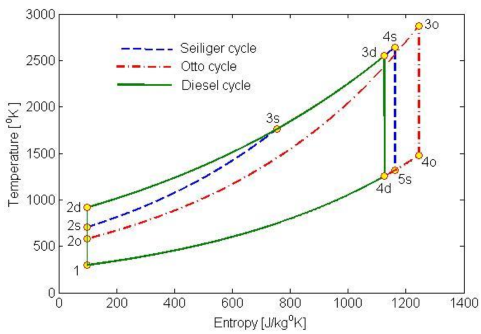

Figure 6 T-s diagram for 2- comparison case

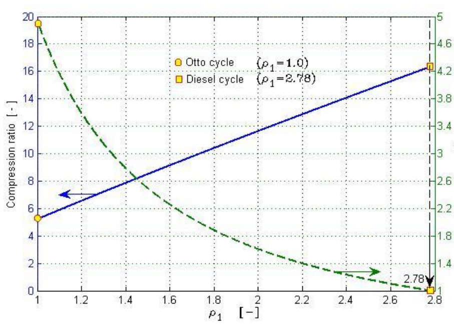

Figure 7 Diagrams depending on parameters ε, α1 and ρ1 for 2- comparison

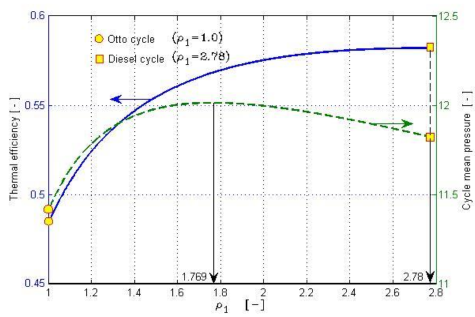

Figure 8 Diagrams depending on parameters ηt , pt and ρ1 for 2- comparison

case *case*

cP-V and T-s diagrams of basic comparative cycles obtained as graphical output of models in Figure 4, for second comparisons case are shown in Figure 5 and 6.

The diagrams in Figure 8 showing the influence of the previous expansion coefficient (ρ1) on the values of the thermal efficiency and mean cycle pressure all three cycles, wherein these values are obtained for the individual cycles by: ρ1 = 1.0 for Otto, ρ1 = 2.78 for Diesel and 1.0 <ρ1 <2.78 for Seiliger's cycle. These diagrams also confirm the known fact that the highest thermal efficiency in this case the comparison has diesel cycle (because of the highest value of compression ratio), while the maximum value of the mean cycle pressure occurs in Seiliger's cycle by the value of the previous expansion coefficient: ρ1 = 1.769.

COMPARATIVE CYCLES OF THE MODERN IC ENGINES

One of the well-known measure of improvement of efficiency, in particular Otto engines, is the use of the cycle with the extended expansion (the expansion stroke is larger than the compression stroke). Such a cycle was first introduced by James Atkinson in 1887, a patent registration of the engine with special piston mechanism that allows extended expansion (Figure 9). Modification of this cycle was made by Miller in 1957, who, instead of a special mechanism, the extended expansion realized by later or earlier closing of the intake valve which involves the use of the valve train with variable valve timing. In addition, this method of the compression stroke regulation leads to a decrease of the charging coefficient, which requires use of a supercharging system.

On Figure 9 are shown p-V diagrams: Atkinson's (1-2-3-4-1), Otto's (1-2-3-4"- 1) and Miller's cycle (1-2-4'-5 -1). Line 1'-2' and 1" - 2 ", corresponding to different lengths of compression stroke of Miller cycle and using them can determine the value of the required overboost pressure: pk1 and pk2.

HEHC cycle is a combination of the Atkinson, Otto and Diesel cycle (diagram in Figure10) and belongs to the group of cycles with a high fuel economy (High Efficiency Hybrid Cycle).

Application of catalyst systems for aftertreatment of exhaust gases in modern engines actualised the cycle with isothermal expansion because of the higher exhaust gases temperatures. P-V diagram of a such (Gruden's) cycle is shown in Figure10.

Using the general thermodynamic cycle and the above described procedure is carried out the comparison of specified cycles with the comparative cycles of the Otto and diesel engines. The comparison results are shown with diagrams in the following figures.

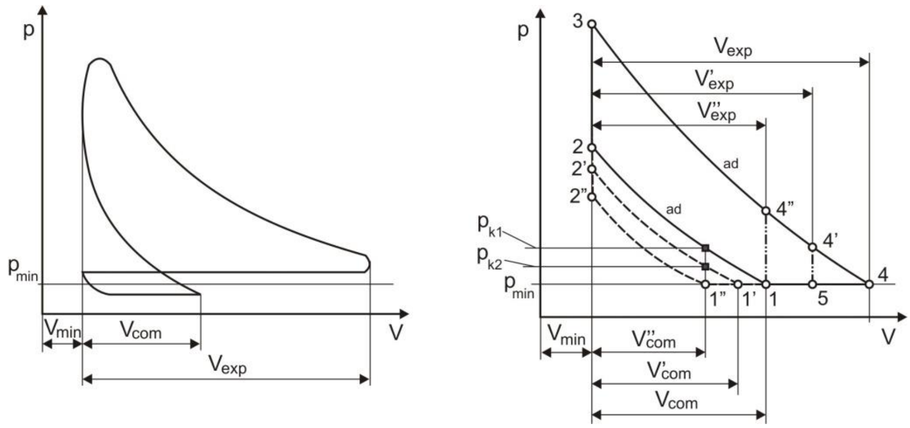

Figure 9 p-V diagrams of Atkinson (left) and Miller (right) cycles

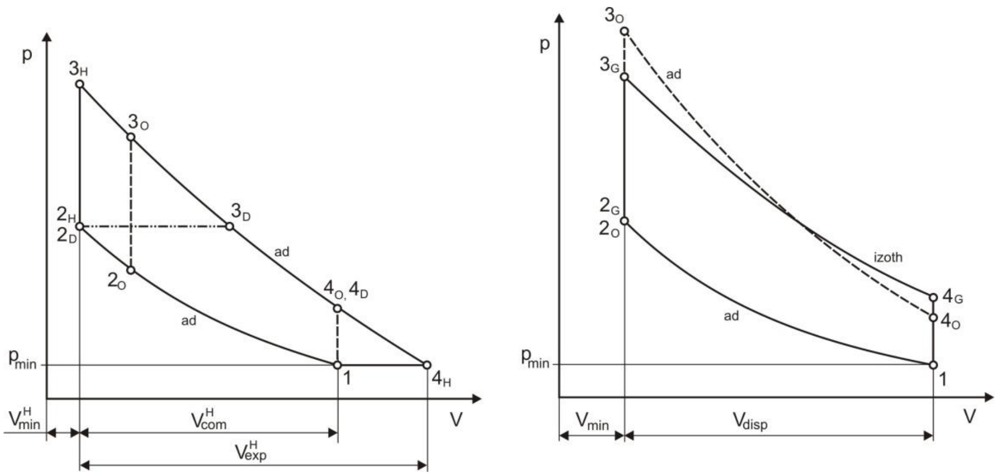

Figure 10 p-V diagrams HEHC (left) and Gruden (right) cycles

On Figure 9 are shown p-V diagrams: Atkinson's (1-2-3-4-1), Otto's (1-2-3-4"- 1) and Miller's cycle (1-2-4'-5 -1). Line 1'-2' and 1" - 2 ", corresponding to different lengths of compression stroke of Miller cycle and using them can determine the value of the required overboost pressure: pk1 and pk2.

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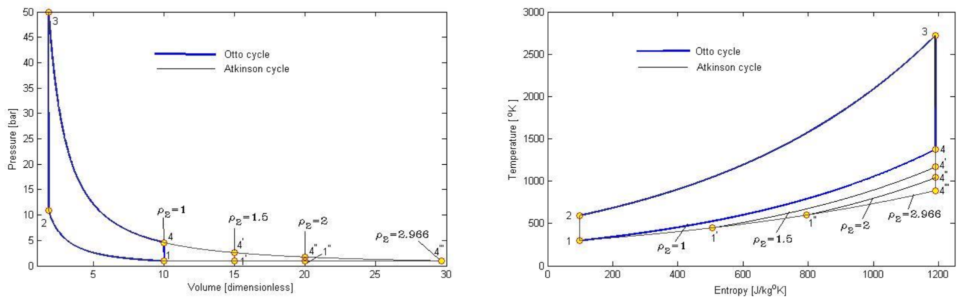

Figure 11 p-V and T-s diagrams of the otto and Atkinson cycles for 2-comparison case

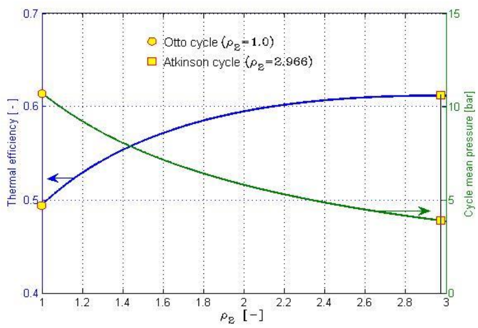

Figure 12 Diagrams depending on parameters: ηt , pt and ρ2 for 2- comparison case

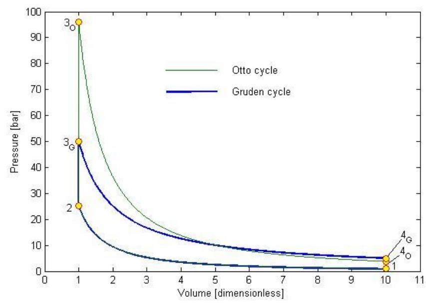

Figure 13 p-V diagrams of the otto and Gruden cycles for 3-comparison case

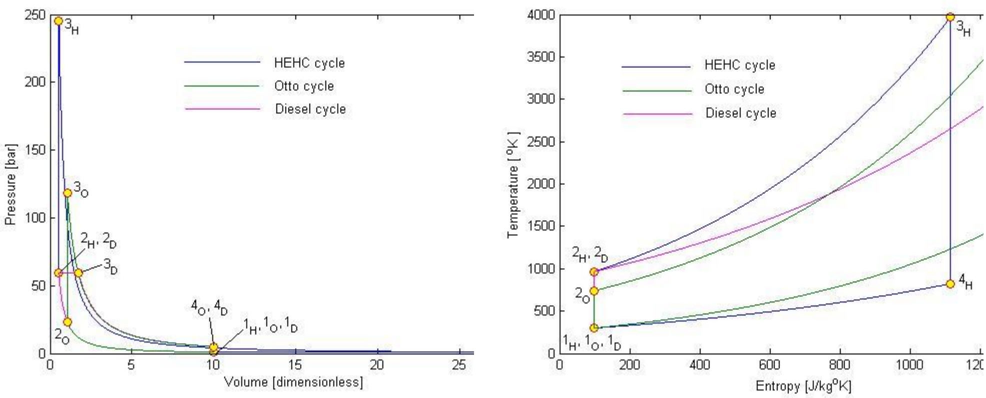

Figure 16 p-V and T-s diagrams of the otto diesel and HEHC cycles for 3-comparison case

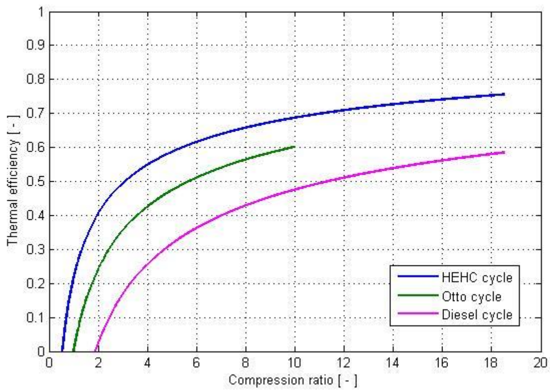

Figure 17 Thermal efficiency of the HEHC, otto and diesel cycle vs compression ratio for 3-comparison case

CONCLUSION

The method of analysis and comparison of the comparative IC engines cycles, based on a general thermodynamic cycle (GTC), provides the following benefits:

- Generate all the possible cycles of heat into work conversion (total: 130).
- Graphic presentation of each of the selected cycle in p-V and T-S coordinate systems.
- Graphic presentation of depending on two or more selected characteristic parameters of the cycle.
- Comparison of cycles per criteria set in advance with the graphical interpretation of the results of comparisons.

The above benefits are presented in the paper the examples of the analysis and comparison of the classic Otto, Diesel and combined cycles as well as cycles of modern combustion engines: Atkinson, Miller, Gruden, HEHC. The proposed method can be applied to all existing and new heat engines cycles..

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