DETAILED ANALYSIS OF THE EFFECT OF THE TURBINE AND COMPRESSOR ISENTROPIC EFFICIENCY ON THE THERMAL AND EXERGY EFFICIENCY OF A BRAYTON CYCLE

by

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Energy and exergy analysis of a Brayton cycle with an ideal gas is given. The irreversibility of the adiabatic processes in turbine and compressor is taken into account through their isentropic efficiencies. The net work per cycle, the thermal efficiency and the two exergy efficiencies are expressed as functions of the four dimensionless variables: the isentropic efficiencies of turbine and compressor, the pressure ratio, and the temperature ratio. It is shown that the maximal values of the net work per cycle, the thermal and the exergy efficiency are achieved when the isentropic efficiencies and temperature ratio are as high as possible, while the different values of pressure ratio that maximize the net work per cycle, the thermal and the exergy efficiencies exist. These pressure ratios increase with the increase of the temperature ratio and the isentropic efficiency of compressor and turbine. The increase of the turbine isentropic efficiency has a greater impact on the increase of the net work per cycle and the thermal efficiency of a Brayton cycle than the same increase of compressor isentropic efficiency. Finally, two goal functions are proposed for thermodynamic optimization of a Brayton cycle for given values of the temperature ratio and the compressor and turbine isentropic efficiencies. The first maximizes the sum of the net work per cycle and thermal efficiency while the second the net work per cycle and exergy efficiency. In both cases the optimal pressure ratio is closer to the pressure ratio that maximizes the net work per cycle.

Key words: *Brayton cycle, net work per cycle, thermal efficiency, exergy efficiency*

Introduction

Climate change and air pollution, especially greenhouse gas emissions, are the environmental issues that are considered the biggest problems of our time. Energy production with the utilization of fossil fuels plays a significant role in the global greenhouse gases emissions. Many possibilities for the reduction of the emissions are evaluated, from the increase of the energy efficiency, followed by a significant increase of the use of renewable energy sources for electricity generation, transportation and other sectors to the low cost solutions such as the implementation of advanced control concepts, particularly those based on the artificial intelligence [1]. Since a part of the energy sector in most countries is focused on the burning

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of fossil fuels, the measures to reduce the emissions of CO_2 include the possibilities of accepting, appropriations, transportation and storage of CO_2 in the reservoirs in general [2]. The ways to improve the energy efficiency are the utilization of combined heat-and-power plants in the generation of electric energy and the replacement of coal by other fuels, such as natural gas [3]. A Brayton cycle is often used in the gas power plants and cogeneration plants. Many papers consider the optimization and exergy analysis of a Brayton cycle [4-10]. In these papers, a wide range of variables that affect the optimum operation of a Brayton cycle is analyzed. Among variables the isentropic efficiency of compressor and turbine often appear. In those papers the influence of the irreversibility of processes through the turbine and compressor on the net work per cycle, or on the thermal efficiency, is not elaborated in detail. Consequently, this paper defines an analytical model of a Brayton cycle with internal irreversibility (friction) through the compressor and turbine. The given analytical expressions quantify the effects of the isentropic efficiencies of compressor and turbine on the net work per cycle, on the thermal efficiency and on the exergy efficiency of a Brayton cycle. Also, two possible optimization strategies for a Brayton cycle are discussed.

Energy analysis of the process

Expression for the dimensionless net work per cycle

A qualitative image in *T-s* diagram is shown in fig. 1, where the points 1, 2, 3, and 4 represent the ideal Brayton cycle with reversible adiabatic processes, whereas the points 1, 2′, 3 and 4′ show the actual Brayton cycle with irreversible adiabatic processes.

The state points 1 and 3 are the same points in both cycles and the temperature ratio $r_T = T_3/T_1$ is assigned while pressure ratio $r_p = p_2/p_1$ is varied. The net work per unit mass per cycle is given by the following equation:

$$w = q - q_0 = c_p[(T_3 - T_2') - (T_4' - T_1)] \tag{1}$$

The temperatures T_2' and T_4' can be expressed, using the isentropic efficiency of compressor (η_{comp}) and turbine (η_{turb}), respectively:

$$T_2' = T_1 - \frac{T_1 - T_2}{\eta_{\text{comp}}} \tag{2}$$

$$T_4' = T_3 - \eta_{\text{turb}}(T_3 - T_4) \tag{3}$$

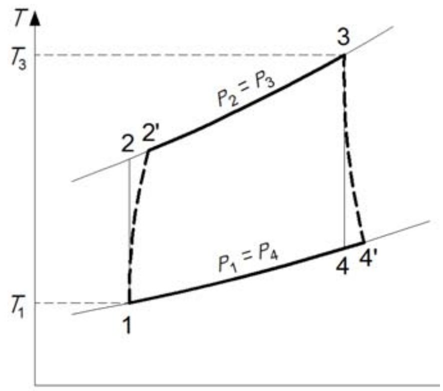

Figure 1. *T-s* diagram of an ideal and actual Brayton cycle

Equation (1) takes the nondimensional form:

$$w_{\text{B}} = \frac{w}{c_p T_1} = \frac{1 - r_p^{\frac{k-1}{k}}}{\eta_{\text{comp}}} + \eta_{\text{turb}}\, r_T \left(1 - r_p^{\frac{1-k}{k}}\right) \tag{4}$$

It is clear from eq. (4) that the net work per cycle will be maximal for the maximal values of η_{comp}, η_{turb} and r_{T}, while the pressure ratio r_{p} for which the net work per cycle has a maximum is defined by:

$$(r_p)_{w_{\mathrm{B\,max}}} = (\eta_{\mathrm{turb}}\eta_{\mathrm{comp}}r_T)^{\frac{k}{2(k-1)}} \tag{5}$$

and the nondimensional maximum net work per cycle:

$$w_{\mathrm{Bmax}} = \frac{(1-\sqrt{\eta_{\mathrm{turb}}\eta_{\mathrm{comp}}r_T})^2}{\eta_{\mathrm{comp}}} \tag{6}$$

Expression for the thermal efficiency

The following expression defines thermal efficiency:

$$\eta = \frac{w}{q} = \frac{w}{c_p(T_3 - T_2')} \tag{7}$$

or in the non-dimensional form:

$$\eta = \frac{1 - r_p^{\frac{k-1}{k}} + \eta_{\mathrm{comp}}\eta_{\mathrm{turb}}r_T\left(1 - r_p^{\frac{1-\kappa}{k}}\right)}{\eta_{\mathrm{comp}}(r_T - 1) + 1 - r_p^{\frac{k-1}{k}}} \tag{8}$$

The thermal efficiency takes its maximal value for the following value of the pressure ratio (with η_{comp}, η_{turb}, and r_T as high as possible):

$$(r_p)_{\eta_{\max}} = \left(\frac{\eta_{\mathrm{turb}}r_T - \sqrt{r_T\eta_{\mathrm{turb}}(r_T - 1)[(1-\eta_{\mathrm{turb}})\eta_{\mathrm{comp}}r_T + 1 - \eta_{\mathrm{comp}}]}}{r_T(\eta_{\mathrm{turb}} - 1) + 1}\right)^{\frac{k}{k-1}} \tag{9}$$

It is obvious from eqs. (5) and (9), that the value of pressure ratio which maximizes the net work per cycle is not the same to the one which maximizes thermal efficiency.

Exergy analysis of the process

Exergy analysis is a practical and useful tool that can be used to improve system efficiency. Many engineers and scientists suggest that exergy analysis is a highly effective method for evaluating and enhancing thermodynamic performance, [11].

The source of irreversibility in considered process is the friction in fluid flow through the compressor and turbine (internal irreversibility) as well as the heat transfer at the finite temperature difference in the heat exchangers (external irreversibility). The upshot of this irreversibility is the exergy destruction, and it is appropriate to carry out the exergy analysis by introducing the exergy efficiency. For the considered Brayton cycle, it is assumed that the temperature of the heat source is equal to the highest temperature of gas in the process $T_{\mathrm{source}} = T_3$, while the temperature of the heat sink is equal to the lowest temperature of gas in the process $T_{\mathrm{sink}} = T_1$. Here, two forms of the exergy efficiency are defined, [12]:

$$\eta_{\mathrm{ex1}} = \frac{w_{\mathrm{B}}}{q\left(1 - \dfrac{T_1}{T_3}\right)} \tag{10}$$

The above formulation is preferred in the analysis of power cycles, and its final nondimensional form is:

$$\eta_{ex1} = \frac{1 - r_p^{\frac{k-1}{k}} + \eta_{comp}\eta_{turb} r_T \left(1 - r_p^{\frac{1-k}{k}}\right)}{\left(1 - \frac{1}{r_T}\right)\left(\eta_{comp}\left(r_T - 1\right) + 1 - r_p^{\frac{k-1}{k}}\right)} \tag{11}$$

According to the second definition of the exergy efficiency, it is the ratio of the net work per cycle to the exergy increase in the heat exchanger, and has the following form:

$$\eta_{ex2} = \frac{w_B}{c_p(T_3 - T_{2'}) - T_1(s_3 - s_{2'})} \tag{12}$$

or in non-dimensional form:

$$\eta_{ex2} = \frac{1 - r_p^{\frac{\kappa-1}{\kappa}} + \eta_{comp}\eta_{turb} r_T \left(1 - r_p^{\frac{1-\kappa}{\kappa}}\right)}{\eta_{comp}\left(r_T - 1 - \ln\frac{\eta_{comp} r_T}{\eta_{comp} - 1 + r_p^{\frac{\kappa-1}{\kappa}}}\right) + 1 - r_p^{\frac{\kappa-1}{\kappa}}} \tag{13}$$

Equations (11) and (13) give different values of the exergy efficiency.

Graphical representation and interpretation of the results

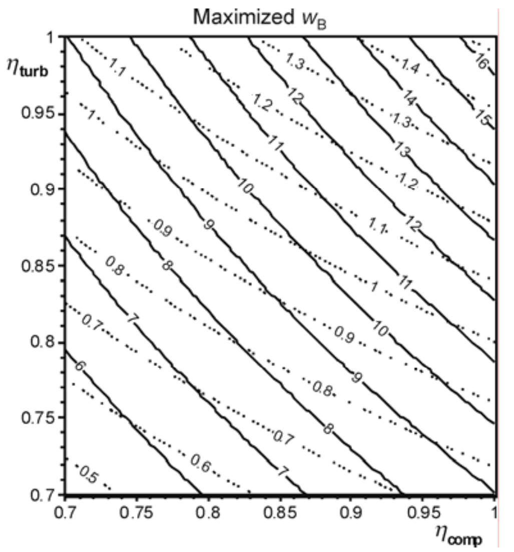

Figure 2. Values of r_p (solid lines) which maximize w_B and corresponding values of w_{Bmax} (dashed lines) as a function of η_{comp} and η_{turb}, for r_T = 5

The four criteria specified by eqs. (4), (8), (11), and (13) are functions of the following four nondimensional quantities: r_p, r_T, η_{turb}, and η_{comp}. The values of the four criteria increase with the increse of r_T, η_{turb}, and η_{comp}, and for these three parameters fixed, there are certain values of r_p which maximize considered criteria. For example, figs. 2 to 5 show the values of r_p which maximize w_B, η, η_{ex1}, and η_{ex2} for r_T = 5, and for the ranges of $0.7 \le \eta_{turb} \le 1$ and $0.7 \le \eta_{comp} \le 1$. It is clear from eqs. (8) and (11) that for given r_T the same value of r_p maximizes η and η_{ex1}.

It is visible in figs. 2 to 5 that r_p which maximizes the four criteria increases with the increase of η_{comp} and η_{turb}. Some numerical values are listed in tab. 1. For example, the increase of the turbine isentopic efficiency from 0.9 to 0.95 will increase the net work from 1.07 to 1.20 (for 0.13), while the same increase of compressor isentropic efficiency (from 0.85 to 0.90) will increase the net work per cycle for 0.07. The increase of pressure ratio is nearly the same (from 10.5 to 11.5). That means that if one can choose to invest in improving compressor or turbine efficiency, it is better to invest in improving of turbine efficiency. The same can be concluded from other three criteria.

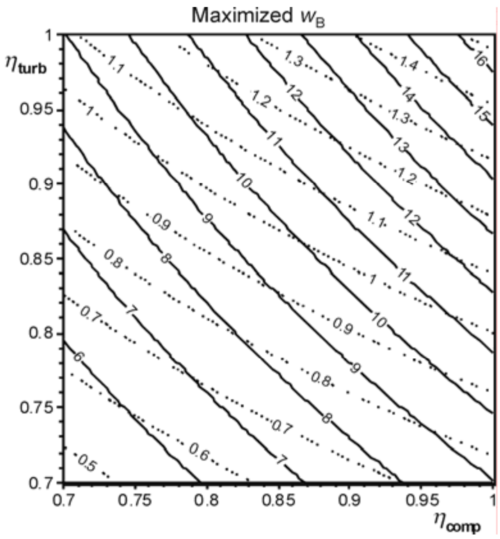

Figure 3. Values of r_p (solid lines) which maximize η and corresponding values of η_{max} (dashed lines) as a function of η_{comp} and η_{turb}, for r_T = 5

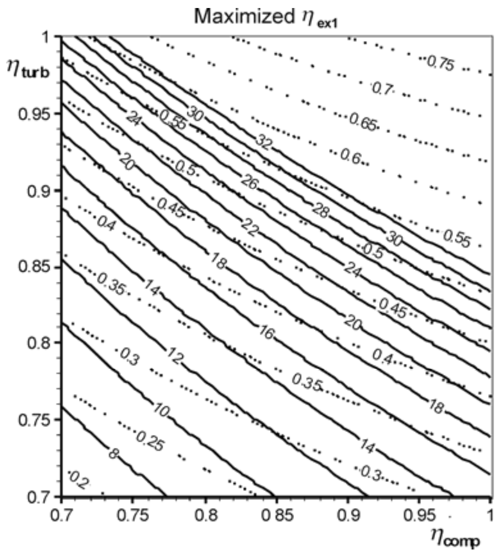

Figure 4. Values of r_p (solid lines) which maximize η_{ex1} and corresponding values of η_{ex1max} (dashed lines) as a function of η_{comp} and η_{turb}, for r_T = 5

In reality compressor and turbine operate with the isentropic efficiencies less than one, and for given values of η_{comp} and η_{turb} it is possible to find pressure ratio which maximizes either net work per cycle or some of efficiencies. Our interest would be to have maximal w_B, η, η_{ex1}, and η_{ex2} at the same time. Since this is impossible, a good compromise might be to have maximal sum of w_B/w_{Bmax} and η/η_{max} or w_B/w_{Bmax} and η_{ex1}/η_{ex2}, where w_{Bmax}, η_{max}, and η/η_{ex2max} refer to maximal values of w_B, η, and η_{ex2} that can be achieved for the given values of η_{comp}, η_{turb}, and r_T. The goal function in the form $w_B/w_{Bmax} + \eta/\eta_{max}$ defines r_p for which the thermal efficiency would be a bit less than its maximum, but greater than the thermal efficiency at maximaum net work per cycle, and the net work per cycle would be a bit less than its maximum but greater than the net work at maximum thermal efficiency, [13].

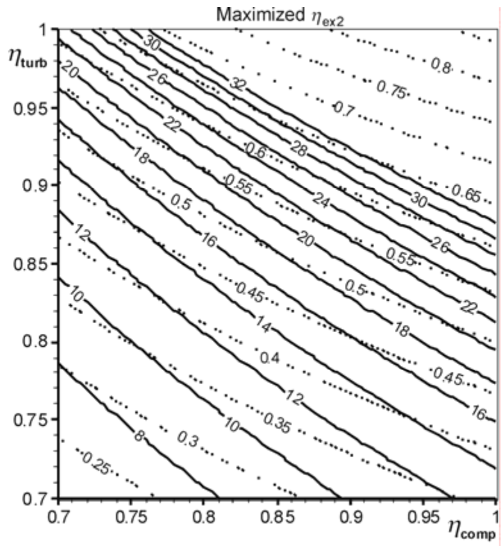

Figure 5. Values of r_p (solid lines) which maximize η_{ex2} and corresponding values of η_{ex2max} (dashed lines) as a function of η_{comp} and η_{turb}, for r_T=5

Table 1. Values of the pressure ratio and corresponding maximal values of the four criteria for two different values of isentropic efficiency of compressor and turbine

r_p w_{Bmax}	$\eta_{turb} = 0.90$	$\eta_{turb} = 0.95$	r_p η_{max}	$\eta_{turb} = 0.90$	$\eta_{turb} = 0.95$
$\eta_{comp} = 0.85$	$r_p = 10.5$ $w_{Bmax} = 1.07$	$r_p = 11.5$ $w_{Bmax} = 1.20$	$\eta_{comp} = 0.85$	$r_p = 27.1$ $\eta_{max} = 0.419$	$r_p = 38.3$ $\eta_{max} = 0.496$
$\eta_{comp} = 0.90$	$r_p = 11.5$ $w_{Bmax} = 1.14$	$r_p = 12.7$ $w_{Bmax} = 1.27$	$\eta_{comp} = 0.90$	$r_p = 32.7$ $\eta_{max} = 0.447$	$r_p = 47.9$ $\eta_{max} = 0.531$
r_p η_{ex1max}	$\eta_{turb} = 0.90$	$\eta_{turb} = 0.95$	r_p η_{ex2max}	$\eta_{turb} = 0.90$	$\eta_{turb} = 0.95$
$\eta_{comp} = 0.85$	$r_p = 27.1$ $\eta_{ex1max} = 0.524$	$r_p = 38.3$ $\eta_{ex1max} = 0.622$	$\eta_{comp} = 0.85$	$r_p = 22.8$ $\eta_{ex2max} = 0.570$	$r_p = 32.0$ $\eta_{ex2max} = 0.665$
$\eta_{comp} = 0.90$	$r_p = 32.7$ $\eta_{ex1max} = 0.559$	$r_p = 47.9$ $\eta_{ex1max} = 0.664$	$\eta_{comp} = 0.90$	$r_p = 27.1$ $\eta_{ex2max} = 0.607$	$r_p = 39.5$ $\eta_{ex2max} = 0.708$

In the next we analyze two goal functions: $w_B/w_{Bmax} + \eta/\eta_{max}$ and $w_B/w_{Bmax} + \eta_{ex1}/\eta_{ex2}$, in the case of $r_T = 5$ and for the two pairs of isentropic efficiencies: $\eta_{turb} = 0.90$, $\eta_{comp} = 0.85$ and $\eta_{turb} = 0.95$, $\eta_{comp} = 0.90$.

Figure 6 shows the ratios w_B/w_{Bmax} and η/η_{max} and their sum as functions of the pressure ratio r_p, for the case $\eta_{turb} = 0.90$ and $\eta_{comp} = 0.85$ and $r_T = 5.0$. Figure 7 shows characteristic points of a Brayton cycle in *T-s* diagram, for the cases presented in fig. 6.

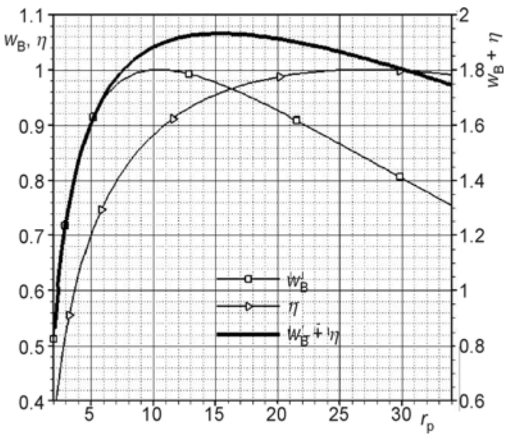

Figure 6. Curves w_B/w_{Bmax}, η/η_{max} and $w_B/w_{Bmax} + \eta/\eta_{max}$ as functions of r_p for $r_T = 5$, $\eta_{turb} = 0.90$ and $\eta_{comp} = 0.85$

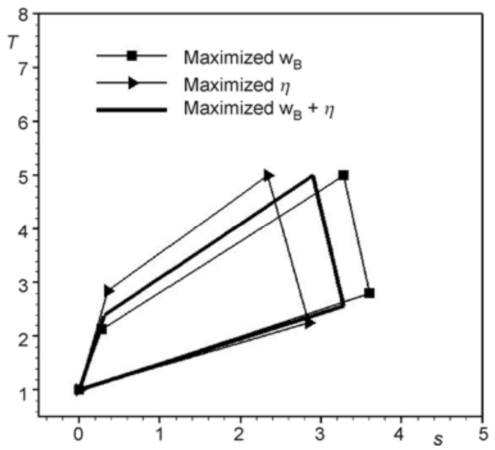

Figure 7. Ilustration of the characteristic points of a *T-s* diagram of a Brayton cycle for the three cases: w_{Bmax}, η_{max} and $(w_B/w_{Bmax}+\eta/\eta_{max})_{max}$

It can be seen that the value of presure ratio for which w_B/w_{Bmax} reaches maximum does not coincide with the pressure ratio in which η/η_{max} reaches maximum. The pressure ratios and the maximal values of the three functions are presented in tab. 2.

Table 2. Values of the pressure ratios and corensponding maximal values of the three functions w_B/w_{Bmax}, η/η_{max} and $w_B/w_{Bmax}+\eta/\eta_{max}$, for r_T = 5, η_{turb} = 0.90 and η_{comp} = 0.85

Curve (symbols)	r_p	w_B	w_B/w_{Bmax}	η	η/η_{max}	$w_B/w_{Bmax}+\eta/\eta_{max}$
Square	10.46	1.075	1.0	0.374	0.892	1.892
Triangle	27.06	0.904	0.841	0.419	1.0	1.841
Solid line	15.44	1.046	0.973	0.402	0.959	1.932

The value of pressure ratio, for which the goal function reaches maximum lies betwen the values of pressure ratio in which the net work per cycle and the thermal efficiency reach maxima, as expected, but closer to the pressure ratio which maximizes the net work per cycle. Figure 8 shows the ratios w_B/w_{Bmax}, η_{ex2}/η_{ex2max} and the sum of w_B/w_{Bmax} and η_{ex2}/η_{ex2max} as functions of r_p. Figure 9 shows a Brayton cycle in the *T-s* diagram for the cases presented in fig. 8.

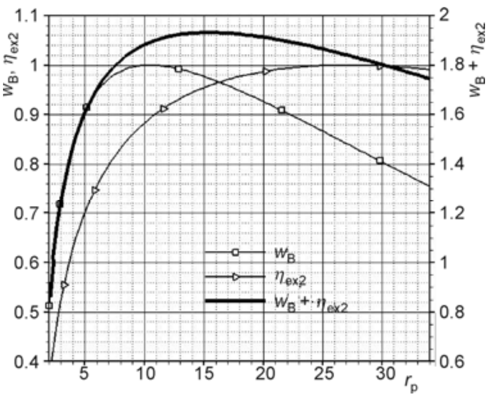

Figure 8. Curves w_B/w_{Bmax}, η_{ex2}/η_{ex2max} and w_B/w_{Bmax} + η_{ex2}/η_{ex2max} as functions of r_p for r_T = 5, η_{turb} = 0.90 and η_{comp} = 0.85

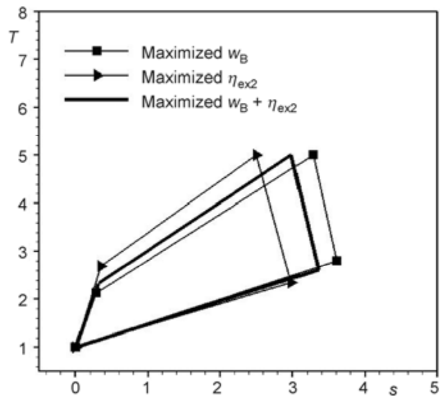

Figure 9. Ilustration of the characteristic points of *T-s* diagram of a Brayton cycle for the three cases: w_{Bmax}, η_{ex2}/η_{ex2max} and $(w_B/w_{Bmax}+\eta_{ex2}/\eta_{ex2max})_{max}$

The pressure ratios and the maximal values of the three functions are presented in tab. 3.

The second goal function gives similar results. The optimal pressure ratio, according the criterium of maximum of the sum of the net work per cycle and the exergy efficiency, lies between the values of the pressure ratio in which maxima of net work per cycle and exergy efficiency are reached, but closer to the pressure ratio which maximizes the net work per cycle.

Table 3. Values of the pressure ratio and corensponding maximal values of the three functions: w_B/w_{Bmax}, η_{ex2}/η_{ex2max}, and $w_B/w_{Bmax}+\eta_{ex2}/\eta_{ex2max}$ at r_T = 5, η_{turb} = 0.90 and η_{comp} = 0.85

Curve (symbols)	r_p	w_B	w_B/w_{Bmax}	η_{ex2}	η_{ex2}/η_{ex2max}	$w_B/w_{Bmax}+\eta_{ex2}/\eta_{ex2max}$
Square	10.46	1.075	1	0.532	0.933	1.933
Triangle	22.75	0.961	0.894	0.570	1	1.894
Solid line	14.12	1.058	0.984	0.555	0.974	1.958

To estimate the effect of the friction in compressor and turbine, we repeated the same procedure for the higher values of η_{turb} = 0.95 and η_{comp} = 0.90. Figure 10 shows curves w_B/w_{Bmax}, η/η_{max} and $w_B/w_{Bmax} + \eta/\eta_{max}$ for r_T = 5 and fig. 11 shows curves w_B/w_{Bmax}, η_{ex2}/η_{ex2max} and $w_B/w_{Bmax} + \eta_{ex2}/\eta_{ex2max}$ for r_T = 5 as functions of r_p.

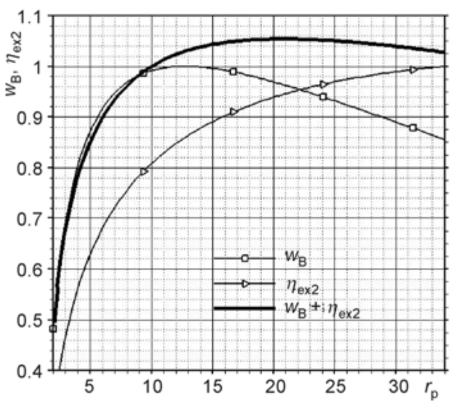

Figure 10. Curves w_B/w_{Bmax}, η/η_{max} and w_B/w_{Bmax} + η/η_{max} as functions of r_p for r_T = 5, η_{turb} = 0.95 and η_{comp} = 0.90

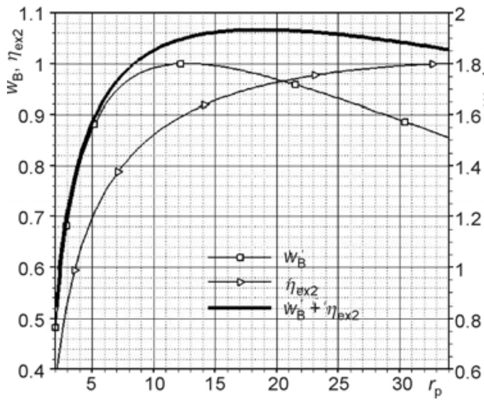

Figure 11. Curves w_B/w_{Bmax}, η_{ex2}/η_{ex2max} and w_B/w_{Bmax} + η_{ex2}/η_{ex2max} as functions of r_p for r_T = 5, η_{turb} = 0.95 and η_{comp} = 0.90

From figures 6, 8, 10, and 11 for the two pairs of values of isentropic efficiencies, it can be seen that the increase of the compressor and turbine isentropic efficiency causes the increase of the net work per cycle, the thermal efficiency and the exergy efficiency. The compressor and turbine isentropic efficiency also has an effect on the values of the pressure ratio in which the maximum net work per cycle and thermal efficiency are obtained. It is also noted that the increase of the isentropic efficiencies moves the values of the optimal pressure ratio, for both quantities w_{Bmax} and $\eta_{max,}$ and also η_{ex2max} to higher values.

The optimal values of pressure ratio according to the given goal functions, in the above presented second case are significantly higher than in the case of the lower values of isenropic efficiencies. Both goal functions give nearly similar values of the pressure ratio. The criterium which includes exergy efficiency results in the slightly lower pressure ratio and the slightly greater net work per cycle.

Table 4. Values of the pressure ratio and corensponding maximal values of the functions w_B/w_{Bmax}, η/η_{max}, η_{ex2}/η_{ex2max}, $w_B/w_{Bmax} + \eta/\eta_{max}$ and $w_B/w_{Bmax} + \eta_{ex2}/\eta_{ex2max}$ for r_T = 5, η_{turb} = 0.95 and η_{comp} = 0.90

r_p	w_B	η	ε_{ex2}	$w_B/w_{Bmax} + \eta/\eta_{max}$	$w_B/w_{Bmax} + \eta_{ex2}/\eta_{ex2max}$
12.71	1.266	0.450	0.637	1.847	1.900
47.88	0.932	0.531	0.705	1.736	1.927
39.50	1.023	0.529	0.708	1.804	1.808
20.67	1.222	0.494	0.683	1.896	1.930
18.69	1.239	0.486	0.675	1.894	1.932

Conclusions

We expressed and analyzed the four nondimensional quantities w_B, η, η_{ex1}, and η_{ex2} of a Brayton cycle as a functions of the four nondimensional variables r_p, r_T, η_{turb} and η_{comp}. To maximize w_B, η, η_{ex1}, and η_{ex2}, it is shown that r_T, η_{turb} and η_{comp} should be as great as possible. For defined values of r_T, η_{turb} and η_{comp} there are three different values of r_p that maximize w_B, η (η_{ex1}), and η_{ex2}. These values of r_p increase (and consequently w_B/w_{Bmax}, η/η_{max}, η_{ex1}/η_{ex1max} and η_{ex2}/η_{ex2max} also rise) with the increase of r_T, η_{turb} and η_{comp}. Effect of improving η_{turb} is greater than the effect of improving η_{comp}.

The value of pressure ratio, which maximizes the net work per cycle is always lower than the values of the pressure ratio which maximize thermal and exergy efficiency. Therefore, different criteria may be used for different purposes, depending on the requests of the maximal net work per cycle or maximal thermal and/or exergy efficiency. We performed optimization according two criteria: maximization of the sum of w_B/w_{Bmax} + η/η_{max} and maximization of the sum of w_B/w_{Bmax} +η_{ex2}/η_{ex2max}. The second criterium results in the slightly lower pressure ratio (closer to the one that maximizes net work) and a bit greater the net work per cycle.

Nomenclature

c_p	– specific heat at constant pressure, [Jkg^{-1}K^{-1}]
h	– specific enthalpy, [J/kg]
k	– specific heat ratio
r_p	– pressure ratio (=p_2/p_1)
r_T	– temperature ratio (=T_3/T_1)
s	– specific entropy, [Jkg^{-1} K^{-1}]
T	– thermodynamic temperature, [K]
w	– net work per cycle, [J/kg]
w_B	– nondimensional net work per cycle [=$w/(c_p/T_1)$]
1	– compressor inlet
2, 2′	– compressor outlet
3	– turbine inlet
4, 4	– turbine outlet

Greek symbols

η	– thermal efficiency
η_{ex1}	– exergy efficiency 1
η_{ex2}	– exergy efficiency 2
η_{comp}	– isentropic efficiency of compressor
η_{turb}	– isentropic efficiency of turbine

References

[1] Mikulandric, A., *et al*., Improvement of Environmental Aspects of Thermal Power Operation by Advanced Control Concepts, *Thermal science*, *16* (2012), 3, pp. 759-772
[2] Garafulic, E., Klarin B., Acceptable Concept of Carbon Dioxide Storage, *Tehnicki vjesnik*, *20* (2013), 1, pp. 161-165
[3] Zaporowski, B., Szczerbowski, R., Energy Analysis of Technological Systems of Natural Gas Fired Combined Heat-and-Power Plants, *Applied Energy*, *75* (2003), 1-2, pp. 43-50
[4] Wu, C., *et al*., Performance of a Regenerative Brayton Heat Engine, *Energy, 21* (1996), 2, pp. 71-76
[5] Sanchez - Orguz, S., *et al*., Thermodynamic Model and Optimization of Multi-Step Irreversible Brayton Cycle, *Energy Conversion and Management*, *51* (2010), 11, pp. 2134-2143
[6] Ferdelji, N., *et al.,* Exergy Analysis of a Co-Generation Plant, *Thermal science*, *12* (2008), 4, pp. 75-88
[7] Haseli, Y., Optimization of Regenerative Brayton Cycle by Maximization of Newly Defined Second Law Efficiency, *Energy Conversion and Management*, *68* (2013), Apr., pp. 133-140
[8] Thong, W., *et al.,* Power and Efficiency Optimization for Combined Brayton and Inverse Brayton Cycle, *Applied Thermal Engineering*, *29* (2009), 14-15, pp. 2885-2894
[9] Cheng, C. Y.,. Chen, C. K., Power Optimization of an Endoreversible Regenerative Brayton Cycle, *Energy, 21* (1996), 4, pp. 241-247
[10] Haseli, Y., *et al.,* Unified Approach to Exergy Efficiency, Environmental Impact and Sustainable Development for Standard Thermodynamic Cycles, *International Journal of Green Energy*, *5* (2008), 1-2, pp.105-119
[11] Ehyaei, M., *et al.,* Exergetic Analysis of an Aircraft Turbojet Engine with an Afterburner, *Thermal science*, *17* (2013), 4, pp. 1181-1194

[12] M. Kanoglu, Y. A., *et al., Efficiency Evaluation of Energy Systems,* Springer Briefs in Energy, Springer Verlag, New York, 2012
[13] Hernandez, A.C., *et al.,* Power and Efficiency in a Regenerative Gas-Turbine Cycle with Multiple Re-heating and Intercooling Stages, *J Phys D Appl Phys, 29* (1996), pp. 1462-1468

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