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Stage performance and optimal compression ratio of a simple–cycle gas turbine

Rafał Laskowski *, Janusz Lewandowski*

Institute of Heat Engineering, Warsaw University of Technology
21/25 Nowowiejska Street, 00-665 Warsaw, Poland

Abstract

In the present paper, the performance of a simple-cycle gas turbine was anal~~yse~~yzed. The first part of the paper focuses on the performance of a gas turbine stage. Assuming the working medium to be an ideal gas, a relation between the current and maximum enthalpy and entropy changes was obtained. Based on this relation, three other relations were obtained: between stage internal efficiency, current and maximum enthalpy and entropy changes, and stage inlet temperature. a relation for stage outlet temperature under changed conditions, taking into account reference conditions, was also proposed. The proposed relations are true when the difference between the inlet and outlet pressures is small, as is the case with the gas turbine stage. In the second part of the paper, determining optimal compression ratio for the whole simple-cycle gas turbine system is described. Most commonly, the optimal compression ratio is assumed for maximum power or thermal efficiency of the whole system. The optimal compression ratio for maximum power is determined analytically, while the one for maximum efficiency iteratively. In the paper, an analytical relation for the optimal compression for maximum thermal efficiency was presented. As practice shows, the real compression ratio for a gas turbine is between the optimum pressure ratio for maximum power and thermal efficiency. In order to determine the real optimal compression ratio for a simple-cycle gas turbine, the applicability of ~~an~~other functions was~~ere~~ examined; besides the enthalpy changes for the turbine and compressor, ~~the~~ other functions also directly take into account entropy changes in the turbine and compressor. In the discussion, identical properties of air and flue gas were assumed, pressure losses in the system were neglected, and the air and flue gas mass flow rates were assumed to be equal.

Keywords: efficiency, gas turbine, optimal compression ratio

1. Introduction

~~A g~~Gas turbines are ~~is~~ widely used in various industrial applications. a simple-cycle gas turbine consists of a compressor, combustion chamber and the~~a~~ gas turbine itself [1, 2]. Figure 1 shows a simple-cycle gas turbine with marked characteristic points.

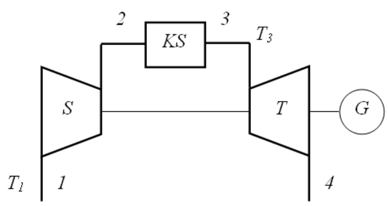

Figure 1: A simple-cycle gas turbine

Although the system shown in Fig. 1 is simple, it is not easy to determine unequivocally the optimal performance parameters. Most commonly, the parameters for maximum unit power or maximum thermal efficiency of the whole system are given [1–12]. Ambient temperature [1–7, 10, 13–17] has a large impact on the achievable efficiency and power of the gas turbine. To determine optimal working conditions of a gas turbine, additional quantities are also used, including a power

*Corresponding author

Email addresses: rlask@itc.pw.edu.pl (Rafał Laskowski), jlew@itc.pw.edu.pl (Janusz Lewandowski)

factor and unit mass flow. As a matter of fact, other objective functions may be provided for which the optimal working conditions would be different. Most often, the objective function (of unit power or thermal efficiency of the system) is provided for the optimal compression ratio. The optimal compression ratio for maximum unit power is analytical in form [2–4], while the one for maximum efficiency is proposed to be determined iteratively [1, 3, 4]. An attempt was made to determine the optimal compression ratio for maximum efficiency analytically. As practice shows, the real compression ratio for a gas turbine is between the optimum pressure ratio for maximum unit power and thermal efficiency. The relations for unit power and thermal efficiency include enthalpy increments in the compressor and turbine and fuel energy; however, entropy changes are not directly taken into account. In the paper, the applicability of functions to determine the real optimal compression ratio of a simple-cycle gas turbine was examined, involving directly, besides enthalpy changes in the compressor and turbine, the entropy changes for the compressor and turbine. By analyz~~s~~ing the enthalpy and entropy changes for the gas turbine stage, a relation was obtained between the current and maximum enthalpy and entropy changes. Based on this proposed relation, three other relations are provided: between stage internal efficiency, current and maximum enthalpy and entropy changes, and stage inlet temperature. a relation for stage outlet temperature under changed conditions, taking into account reference conditions, was also proposed.

2. Mathematical models

In a simple–cycle gas turbine two gaseous media occur: air and flue gas. All the proposed relations were obtained on the assumption that the air and flue gas are considered an ideal gas. Additionally, identical properties of air and flue gas were assumed, pressure losses in the system were neglected, and the air and flue gas mass flow rates were assumed to be equal. In the first part of this section, the focus is on the relations applicable to the gas turbine stage (turbine stage efficiency, temperature downstream of the turbine stage). In the second part of the section, an analytical form of the optimal compression ratio for the maximum thermal efficiency of the simple-cycle gas turbine is provided, and some functions are proposed to determine the real optimal compression, involving directly, besides the enthalpy changes for the turbine and compressor, the entropy changes for the turbine and compressor.

Turbine stage internal efficiency

By definition, the internal efficiency of a stage or a group of stages of a gas turbine is equal to

$$\eta_i = \frac{\Delta i_r}{\Delta i_t} = \frac{i_3 - i_{4r}}{i_3 - i_{4t}} \tag{1}$$

The enthalpy change in the denominator is equal to a maximum enthalpy change for the pressures between the inlet and the outlet (Fig. 1). For an ideal gas with an assumption that is constant, the change ~~equals~~is [5, 18]

$$\Delta i_t = \Delta i_{max} = i_3 - i_{4t} = c_p T_3 \left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{k-1}{k}}\right] \tag{2}$$

The actual enthalpy change for an ideal gas ~~equals~~is [5, 18]

$$\Delta i_r = i_3 - i_{4r} = c_p T_3 \left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}}\right] \tag{3}$$

Considering the relations (2, 3), the internal efficiency of a stage or a group of stages can be written as

$$\eta_i = \frac{1 - \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}}}{\left(\frac{p_4}{p_3}\right)^{\frac{k-1}{k}}} \tag{4}$$

Assuming that the pressure loss between the inlet and outlet ~~equals~~is

$$\Delta p = p_3 - p_4 \tag{5}$$

the internal efficiency can be expressed as [6]

$$\eta_i = \frac{1 - \left(1 - \frac{\Delta p}{p_3}\right)^{\frac{n-1}{n}}}{1 - \left(1 - \frac{\Delta p}{p_3}\right)^{\frac{k-1}{k}}} \tag{6}$$

Using the relation [6], being true for $(\varepsilon \ll 1)$, as with the gas turbine stage,

$$(1 - \varepsilon)^{\beta} \approx 1 - \varepsilon\beta \tag{7}$$

the internal efficiency can be written as [2, 5, 6, 18]

$$\eta_i = \frac{n-1}{n}\frac{k}{k-1} \tag{8}$$

Assuming $(\varepsilon \ll 1)$, the relation 8 can be transformed to a form involving the entropy change:

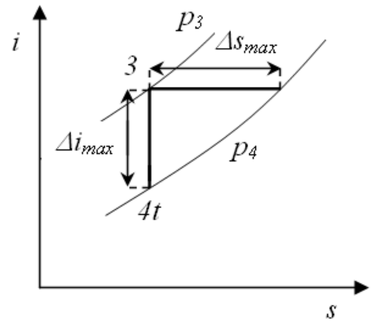

Figure 2: Isentropic expansion and Joule-Thomson expansion with indicated maximum enthalpy and entropy changes

$$\eta_i = \frac{n-1}{n}\frac{k}{k-1} = \frac{k}{k-1}\frac{ln\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}}}{ln\left(\frac{p_4}{p_3}\right)} = \frac{C_p}{R}\frac{ln\left(\frac{T_4}{T_3}\right)}{ln\left(\frac{p4}{p_3}\right)} = 1 - \frac{C_p ln\left(\frac{T_4}{T_3}\right) - R \cdot ln\left(\frac{p4}{p_3}\right)}{-Rln\left(\frac{p4}{p_3}\right)} = 1 - \frac{\Delta s}{-Rln\left(\frac{p4}{p_3}\right)} \tag{9}$$

The maximum entropy change occurs when the enthalpy is constant, i.e., in the Joule-Thomson expansion process (Fig. 2).

Considering flue gas as an ideal gas $\left(C_p = const\right)$, constant temperature $(T_4 = T_3 = const)$ is obtained from the enthalpy relation [19]. Hence, the maximum entropy change for an ideal gas is

$$\Delta s_{max} = -Rln(\left(\frac{p_4}{p_3}\right) \tag{10}$$

Finally, the internal efficiency of the gas turbine stage can be written as

$$\eta_i = 1 - \frac{\Delta s}{\Delta s_{max}} \tag{11}$$

By comparing the relations 1 and 11 the following can be obtained:

$$\frac{\Delta i_r}{\Delta i_{max}} + \frac{\Delta s}{\Delta s_{max}} = 1 \tag{12}$$

Then, the ratio of the maximum enthalpy change and the maximum entropy change for small pressure changes between the inlet and outlet (as is the case with the gas turbine stage) was examined.

$$\frac{\Delta i_r}{\Delta s_{max}} = \frac{C_p T_3\left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{k-1}{k}}\right]}{-Rln\left(\frac{p_4}{p_3}\right)} \tag{13}$$

In the case when the outlet pressure varies little from the inlet pressure, the relation 13 takes an indeterminate form $(0/0)$. In order to obtain a value from the relation 13 for this case, de l'Hospital's principle has to be used. To this end, derivatives of the numerator and denominator in the relation 13 should be determined.

The maximum entropy change can be written as

$$\Delta i_{max} = C_p T_3 \frac{k-1}{k}\int_{p_4}^{p_3}\left(\frac{p}{p_3}\right)^{-\frac{1}{k}} d\left(\frac{p}{p_3}\right) \tag{14}$$

Its differential form is

$$di_{max} = C_p T_3 \frac{k-1}{k}\left(\frac{p}{p_3}\right)^{-\frac{1}{k}} d\left(\frac{p}{p_3}\right) \tag{15}$$

The maximum entropy change can be written as

$$\Delta s_{max} = R\int_{p_4}^{p_3}\frac{dp}{p} \tag{16}$$

Its differential form is

$$ds_{max} = R\frac{dp}{p} \tag{17}$$

Thus, the ratio of the maximum enthalpy change and the maximum entropy change in the differential form becomes

$$\frac{di_{max}}{ds_{max}} = \frac{C_p T_3 \frac{k-1}{k}\left(\frac{p}{p_3}\right)^{-\frac{1}{k}} d\left(\frac{p}{p_3}\right)}{R\frac{dp}{p}} \tag{18}$$

On rearranging, we obtain:

$$\frac{di_{max}}{ds_{max}} = T_3\left(\frac{p}{p_3}\right)^{\frac{k-1}{k}} \tag{19}$$

or

$$\lim p \to p_3 \frac{\Delta i_{max}}{\Delta s_{max}} = T_3 \tag{20}$$

Assuming a small change between the stage inlet and outlet pressure, other relations can be obtained between medium temperature at the inlet, efficiency, and enthalpy and entropy changes. The ratio of the current enthalpy change to the maximum entropy change ~~equals~~is

$$\frac{\Delta i_r}{\Delta s_{max}} = \frac{\Delta i_r}{\Delta i_{max}} \frac{\Delta i_{max}}{\Delta s_{max}} = \eta_i T_3 \tag{21}$$

The ratio of the maximum enthalpy change to the current entropy change is

$$\frac{\Delta i_{max}}{\Delta s} = \frac{T_3}{1-\eta_i} \tag{22}$$

The ratio of the current enthalpy change to the current entropy change is

$$\frac{\Delta i_r}{\Delta s} = \frac{\eta_i}{1-\eta_i} T_3 \tag{23}$$

Outlet temperature downstream the stage

The relation for the internal efficiency of the gas turbine stage is

$$\eta_i = \frac{i_3 - i_{4r}}{i_3 - i_{4t}} = \frac{C_p(T_3 - T_{4r})}{C_p T_3 \left(1 - \pi^{\frac{k-1}{k}}\right)} \tag{24}$$

The relation 24 can be transformed into

$$T_3 \left(1 - \pi^{\frac{k-1}{k}}\right) \eta_i = T_3 - T_{4r} \tag{25}$$

On considering 7, the relation 25 takes the form

$$T_3 \frac{(k-1)}{k} \frac{\Delta p}{p_3} \eta_i = T_3 - T_{4r} \tag{26}$$

Taking into account the reference conditions, one can write

$$\frac{T_3}{T_{3o}} \frac{p_{3o}}{p_3} \frac{\Delta p}{\Delta p_o} \frac{\eta_i}{\eta_{io}} = \frac{T_3 - T_{4r}}{T_{3o} - T_{4ro}} \tag{27}$$

Assuming that the stage inlet parameters and the internal efficiency change within a small range $\left(\frac{T_3}{T_{3o}} \approx 1, \frac{p_{3o}}{p_3} \approx 1, \frac{\eta_i}{\eta_{io}} \approx 1\right)$, the following can be written:

$$\frac{p_3 - p_4}{p_{3o} - p_{4o}} = \frac{T_3 - T_{4r}}{T_{3o} - T_{4o}} \tag{28}$$

Finally, the medium temperature downstream the stage can be expressed as

$$T_{4r} = T_3 - \frac{T_{3o} - T_{4o}}{p_{3o} - p_{4o}} (p_3 - p_4) \tag{29}$$

The optimal compression ratio for maximum efficiency of a simple–cycle gas turbine—the analytical form

For a simple-cycle gas turbine, the optimal compression ratio for maximum efficiency and actual unit power can be distinguished. Relevant relations for the optimal compression ratio for maximum efficiency and actual unit power can be found in the literature [1, 3–5]. In order to simplify further considerations, identical properties of flue gas and air were assumed, pressure losses in the system were neglected, and the flue gas mass flow rate was assumed to be equal to that of the air (the ratio of fuel flow rate to flue gas mass flow rate is close to zero). On these assumptions, the optimal compression ratio for maximum actual unit power has the form [1, 3, 4]

$$\pi_{optN} = \left(\sqrt{\theta \eta_{iT} \eta_{is}}\right)^{\frac{k}{k-1}} \tag{30}$$

Knowing the ratio between the maximum (T_3) and minimum temperatures (T_1) in the system (θ), the internal efficiency of the turbine (η_{iT}) and compressor (η_{is}), and the isentropic exponent (k), the optimal compression ratio for actual maximum unit power can be analytically determined from the relation 30. It is recommended that the optimal compression ratio for maximum efficiency ~~should~~ be determined by successive approximations (iteratively) [1, 4] from the relations below:

$$\pi = \frac{\theta \eta_{iT} \eta_{is}}{1-\eta} \tag{31}$$

$$\eta = \frac{\theta\left(1 - \frac{1}{\alpha}\right)\eta_{iT} - \frac{(\alpha-1)}{\eta_{is}}}{\theta - 1 - \frac{(\alpha-1)}{\eta_{is}}} \tag{32}$$

$$\alpha = \pi^{\frac{k-1}{k}} \tag{33}$$

On these assumptions, an attempt was made to determine the analytical form of the optimal compression ratio for maximum efficiency. To this end, the relations 31, 32 were transformed into a form of a quadratic function

$$(\theta - 1 - \theta\eta_{iT})\,\alpha^2 + 2\theta\eta_{iT}\alpha - \theta\eta_{iT}\,(\eta_{is} - 1 - \theta\eta_{is}) = 0 \tag{34}$$

The equation 34 has two solutions

$$\alpha_1 = \frac{-\theta\eta_{iT} - \sqrt{[\theta\eta_{is}\,(1-\eta_{iT}) + (1-\eta_{is})]\,\theta\eta_{iT}\,(\theta-1)}}{(\theta - 1 - \theta\eta_{iT})} \tag{35}$$

$$\alpha_2 = \frac{-\theta\eta_{iT} + \sqrt{[\theta\eta_{is}(1-\eta_{iT}) + (1-\eta_{is})]\,\theta\eta_{iT}(\theta-1)}}{(\theta - 1 - \theta\eta_{iT})} \tag{36}$$

Valid results were obtained from the second solution. Finally, the optimal compression ratio for maximum efficiency can be written as

$$\pi_{opt\eta} = \left[\frac{-\theta\eta_{iT} + \sqrt{[\theta\eta_{is}(1-\eta_{iT}) + (1-\eta_{is})]\,\theta\eta_{iT}(\theta-1)}}{(\theta - 1 - \theta\eta_{iT})}\right]^{\frac{k}{k-1}} \tag{37}$$

Optimal working conditions of a simple-cycle gas turbine for another objective function

To determine the optimal working conditions (optimal compression ratio) of the simple-cycle gas turbine, a function is proposed involving, besides the enthalpy changes for the turbine and compressor, the entropy changes for the turbine and compressor. The isentropic (theoretical) enthalpy change for the compressor can be written as

$$\Delta i_{st} = C_p T_1 \left(\pi^{\frac{k-1}{k}} - 1\right) \tag{38}$$

The isentropic enthalpy change for the turbine can be expressed as

$$\Delta i_{Tt} = C_p T_3 \left(1 - \frac{1}{\pi^{\frac{k-1}{k}}}\right) \tag{39}$$

The entropy increment for the compressor can be written as

$$\Delta s_s = C_p ln\left(\frac{\eta_{is} + \pi^{\frac{k-1}{k}} - 1}{\pi^{\frac{k-1}{k}}\eta_{is}}\right) \tag{40}$$

The entropy increment for the turbine can be written as

$$\Delta s_T = C_p ln\left(\pi^{\frac{k-1}{k}} - \eta_{iT}\pi^{\frac{k-1}{k}} + \eta_{iT}\right) \tag{41}$$

To determine the optimal compression ratio of the simple-cycle gas turbine, a function is proposed involving a difference between theoretical unit power (the difference between theoretical unit powers of the turbine and compressor) and the sum of entropy increments for the compressor and turbine (the sum of losses in the compressor and turbine). To enable the comparison of these two quantities, they were transformed into a dimensionless form. The theoretical unit power was divided by $(C_p T_1)$, and the sum of entropy increments by (C_p). Transformed into the dimensionless form, the proposed function has the form

$$F = \frac{(\Delta i_{Tt} - \Delta i_{st})}{C_p T_1} - \frac{(\Delta s_T + \Delta s_s)}{C_p} \tag{42}$$

The optimal compression for the maximum value of the function (F) was obtained from the condition $(\frac{dF}{d\pi} = 0)$. On differentiating the function (F) with respect to the compression ratio, the obtained relation was

$$\frac{\theta}{\alpha} + \frac{(\eta_{iT} - 1)\,\alpha}{\alpha - \eta_{iT}\alpha + \eta_{iT}} = \alpha + \frac{(1 + \eta_{is})}{\eta_{is} + \alpha - 1} \tag{43}$$

Due to the~~a~~ complex form of the relation 43, the optimal compression ratio was determined iteratively. The parameter (α) equals 33.

The proposed function F is dimensionless. ~~The proposed concept I~~in order to determine the optimum pressure ratio the proposed concept can be explained as the difference between the theoretical unit power produced in the turbine and consumed by the compressor minus the sum of the losses generated in these two devices. This concept can also be written as a dimensional function in J/kg units

$$f = (\Delta i_{Tt} - \Delta i_{st}) - T_1(\Delta s_T + \Delta s_s) \tag{44}$$

In order to compare these two quantities (the difference ~~of~~between the theoretical unit power produced by the turbine and used by the compressor and the sum of the losses generated in these two devices) the sum of the entropy increments in the turbine and compressor must be multiplied by the temperature (input temperature to the compressor T1 was selected).

From the condition $(\frac{df}{d\pi} = 0)$ for the optimal compression ratio for the maximum value of the function f the same relation was obtained 43.

3. Results

In order to verify the proposed relations 42..44 for the optimal compression ratio, the following assumptions were made in the calculations: ambient temperature (T_1) is 288.15 K (15℃), isentropic exponent (k) is 1.4, specific heat (C_p) is 1.005 . In the first case constant values of internal compressor (η_{is}) and turbine efficiency (η_{iT}) were assumed as 0.89 and 0.92, respectively.~~,~~ Inlet temperature to the gas turbine T3 varied according to the data given in Table 1. The data presented in Table 1 for the year, inlet temperature and compression ratio was taken from [4]. For such assumptions the compression ratios for maximum unit

Table 1: The compression ratios for maximum unit power, for proposed functions and for maximum system efficiency (constant internal compressor and turbine efficiency)

Year	1967	1972	1979	1990	1998
Inlet temperature to the gas turbine ℃	900	1010	1120	1260	1425
Compression ratio	10.5	11	14	14.5	19..23
Optimal compression ratio for maximum actual unit power π_{optN}	8.22	9.62	11.11	13.14	15.71
Optimal compression ratio for proposed functions $\pi_{opt,F}$,$\pi_{opt,f}$	9.78	11.50	13.34	15.86	19.05
Optimal compression ratio for maximum thermal efficiency $\pi_{opt\eta}$	20.62	26.33	32.95	42.77	56.47

Table 2: The compression ratios for maximum unit power, for proposed functions and for maximum system efficiency (constant polytropic compressor and turbine efficiency)

Year	1967	1972	1979	1990	1998
Inlet temperature to the gas turbine ℃	900	1010	1120	1260	1425
Compression ratio	10.5	11	14	14.5	19..23
Optimal compression ratio for maximum actual unit power π_{optN}	7.34	8.57	9.89	11.68	13.95
Optimal compression ratio for proposed functions $\pi_{opt,F}$,$\pi_{opt,f}$	9.34	11.04	12.86	15.35	18.53

power, for proposed functions and for maximum system efficiency were calculated (Table 2).

The calculations were repeated for a polytropic efficiency of 0.88 [7] for both the compressor and the turbine. The results are shown in Table 2.

For this case the optimal compression ration for maximum thermal efficiency was not~~n't~~ analyzed because real compression ratios are closer to the optimal compression ratio for maximum actual unit power (Table 1).

4. Conclusions

In the paper, the relation 12 was shown between current and maximum enthalpy and entropy changes, obtained for small pressure changes between the inlet and outlet of a gas turbine stage. Based on the proposed relation, three 21, 22, 23 other relations were provided, applicable to a gas turbine stage: between the internal efficiency, current and maximum enthalpy and entropy changes, and inlet temperature. The relation 29 for stage outlet temperature downstream of the gas turbine stage under changed working conditions was also proposed, which takes into account reference conditions. The optimal compression ratio for maximum efficiency can be determined iteratively from the relations 31, 32. In the paper, the analytical form of the optimal compression ratio for maximum thermal efficiency of a simple-cycle gas turbine 37 is provided. Even following major simplification, the relation has a quite complex form. Most often, the optimal compression ratio for maximum actual unit power or efficiency is provided for the purpose of determining the optimal working conditions of a turbine. Even in the case of the simple-cycle gas turbine, choosing the optimal compression ratio is not an easy task, and it also depends on the role of the turbine: whether it should work with maximum unit power or efficiency. In the paper, a functions were proposed to determine the real optimal compression ratio, involving, besides the enthalpy changes, the entropy changes in the compressor and turbine. The proposed relations 42, 44 are the~~a~~ difference between the theoretical unit power and the sum of entropy increments in the compressor and turbine. To enable the comparison of these two quantities, they were transformed into a dimensionless form for relation 42 and expressed in J/kg units for relation 44. The optimal compression for both proposed functions has the same form of the relation 43. As practice shows, the real compression ratio for a gas turbine is between the optimum pressure ratio for maximum power and thermal efficiency. The optimal compression ratio for the proposed functions (F, f) is larger than for the actual unit power and lower for maximum thermal efficiency. ~~What is the m~~Most importantly, the values of the optimal compression ratios as determined from the proposed functions (F, f) are the closest to the actual compression ratios of simple-cycle gas turbines (Tables 1, 2).

References

[1] A. Miller, Gas turbines and gas-steam systems (in Polish)., WPW Warszawa, 1982.

[2] T. Chmielniak, Thermal turbines - theoretical basis (in Polish)., WPŚ, 1998.

[3] A. Miller, J. Lewandowski, Gas-steam systems on solid fuel (in Polish)., WNT Warszawa, 1993.

[4] K. Badyda, A. Miller, Energy gas turbines and systems with their application (in Polish)., Kaprint, 2011.

[5] T. Chmielniak, Energy technologies (in Polish)., WNT Warszawa, 2008.

[6] S. Perycz, Steam and gas turbines (in Polish)., WPG, 1988.

[7] K. Badyda, Characteristics of advanced gas turbine cycles (in Polish). (2010).

[8] V. A. Prisyazhniuk, The turbine plant efficiency: Maximum efficiency attained and the share of individual stages, Applied Thermal Engineering 28 (2008) 1318–1323.

[9] Y. Haseli, Performance of irreversible heat engines at minimum entropy generation, Applied Mathematical Modelling 37 (2013) 9810–9817.
[10] M. Kowalski, K. Badyda, Performance analysis of a gas turbine air heat recovery unit using GateCycle software, Journal of Power Technologies 92 (1) (2012) 48–54.
[11] J. Kotowicz, Gas and steam power plants (in Polish), Kaprint, 2008.
[12] J. Kotowicz, Thermodynamic and Economic Analysis of the combined cycle gas turbine integrated with coal gasification and carbon dioxide separation membrane (in Polish), Silesian University of Technology, 2012.
[13] A. D. Sa, S. A. Zubaidy, Gas turbine performance at varying ambient temperature, Applied Thermal Engineering 31 (2011) 2735–2739.
[14] H. Erden, S. Sevelgen, Case study: effect of ambient temperature on the electricity production and fuel consumption of a simple cycle gas turbine in Turkey, Applied Thermal Engineering 26 (2-3) (2006) 320–326.
[15] H. Carniere, A. Willocx, E. Dick, M. D. Paepe, Raising cycle efficiency by inter cooling in air cooled gas turbine, Applied Thermal Engineering 26 (16) (2006) 1780–1787.
[16] R. Laskowski, A. Smyk, J. Lewandowski, An attempt to determine the compressor and turbine efficiency based on measurement data (in Polish), Rynek Energii 3 (2014) 63–68.
[17] R. Laskowski, A. Smyk, J. Lewandowski, A simplified approach to determining thermodynamic parameters and performance of a single-shaft gas turbine engine in off-design conditions, Rynek Energii 3 (118) (2015) 115–125.
[18] B. Staniszewski, Thermodynamics (in Polish)., PWN Warszawa, 1978.
[19] B. Staniszewski, Heat transfer (in Polish)., PWN Warszawa, 1979.

Nomenclature

C_p–specific heat at constant pressure, J/(kgK)
i–specific enthalpy, J/kg
k–isentropic exponent, -
n–polytropic exponent, -
p–pressure, Pa
R–gas constant, J/(kgK)
T–temperature, K
η–efficiency, -
θ–ratio between maximum and minimum temperatures in a system$\left(\frac{T_3}{T_1}\right)$, -
π–compression – a ration between pressures downstream and upstream of a compressor $\left(\frac{p_2}{p_1} = \frac{p_3}{p_4}\right)$, -

Indices

i–internal efficiency,
o–reference conditions,
r–actual enthalpy change,
s–relates to a compressor,
t–isentropic (theoretical) enthalpy change
T–relates to a turbine