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Accuracy problem of modeling in a gas turbine cycle with heat regeneration according to Szewalski's idea

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Abstract

In this paper, authors undertook a problem of accuracy and exactness verification of some assumptions and calculation methods that are important in the 0D modeling. The computational flow mechanics (CFM) tools and basic calculations of gas turbine cycle with heat regeneration according to Szewalski's idea have been done. Results and conclusions for this specific problem have been presented. Gas turbine cycle with heat regeneration according to Szewalski's idea includes regeneration as same as it is now commonly employed in steam turbine cycles. Mentioned regeneration is enabled by adding to the simple cycle devices configuration of two regenerative heat exchangers and auxiliary compressor. That modification allows to obtain working medium circulation in closed cycle inside a basic cycle. Efficiency gain is dependent on extraction pressure and extraction mass flow rate.

Keywords: Heat regeneration; Gas turbine; Thermodynamical analysis; Numerical analysis; Basic model vs numerical model; CFM (computational flow mechanics)

Nomenclature

c_p	–	specific heat at constant pressure, kJ/(kg K)
h	–	specific enthalpy, kJ/kg

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l	–	specific work, kJ/kg
N	–	power, kW
$\dot{m}$	–	mass flow rate, kg/s
p	–	pressure, MPa
R	–	gas constant, kJ/(kg·K)
$\dot{Q}$	–	rate of heat, heat energy flux, kW
$\dot{Q}_{chem}$	–	chemical energy flux, kW
T	–	temperature, K
W_d	–	fuel calorific value, kJ/kg

Greek symbols

γ	–	relative cooling air mass flow rate, –
δ	–	relative air mass flow rate blocking leakage losses, –
Δp	–	pressure losses, MPa
ΔT	–	the temperature difference in the heat exchanger, K
κ	–	isentropic coefficient, –
ζ	–	flow losses, –
η	–	efficiency, –
π	–	compression/expansion ratio, –
ρ	–	density, kg/m^3

Subscripts

a	–	air
bCC	–	before combustion chamber
C	–	compressor
C'	–	additional (auxiliary) compressor
CC	–	combustion chamber
el	–	electrical
f	–	fuel
g	–	generator
GT	–	gas turbine (expander)
i	–	internal
$inRE$	–	between RE1 and RE2
$loss$	–	loss
m	–	mechanical
RE	–	regenerative heat exchanger
s	–	isentropic
TIT	–	turbine inlet temperature
1s, 2s,...	–	isentropic points of process
1,2...	–	real points of process

1 Introduction

Nowadays activity of energy companies is focused on the production of cheap electricity with the highest efficiency of energy conversion and with the lowest air pollution. This is due to the requirements of the European Directive

2010/75/EU [1]. For several years turbine producers have used different operations to increase the efficiency. There are known methods such as the interstage cooling in compressor, the divided combustion chamber, the use of the regeneration in the gas turbine cycle and numerous methods with using steam or water injection. It should be pointed out, that particular advantages are achieved by injecting steam into the gas turbine, what contribute to increase power and efficiency and to reduce emissions of carbon dioxide and oxides of nitrogen [2–6].

Equally important problem to increasing efficiency is correct determination of this parameter and examination calculation method inaccuracy magnitude. Mentioned issue is worth to consider on the gas turbine cycle with heat regeneration according to Szewalski's idea example, presented in patent [7].

It should be added that this way to increase the efficiency of the gas turbine is the idea of using exhaust gases extraction in gas turbine, proposed by Polish scientist and engineer Professor Robert Szewalski (director of the Institute of Fluid-Flow Machinery, Polish Academy of Sciences in Gdansk), who has constructed first Polish industrial steam turbine. To this day this proposal has not been implemented and applied. However, in the near future the development of hydrogen technology may lead to the development of this method. As a result of using hydrogen as a fuel, the exhaust gas from the gas turbine will be a mixture of exhaust gases and steam (steam-gas). For such working medium, the turbine will be a hybrid, which combines the advantages of gas and steam turbines [8, 9].

The thermodynamic analysis of a gas cycle was performed by using programs type CFM (computational flow mechanics) and the basic calculation of ideal gas. There was performed a numerical simulations of cycle GT8C, based on data available in the literature [10–13]. The results of the numerical analysis of gas turbine cycle with heat regeneration according to Szewalski's idea are available at the works of Ziółkowski *et al.* [14, 15, 19]. In turn in this article the differences between the two tools and results of thermodynamic and performances calculation was analyzed and that will be described later.

2 Literature review

This section is devoted to the literature reviews in methods of improving the efficiency of gas turbines and ways to modeling the Brayton-Joule cycle.

2.1 Increase efficiency

Gas turbines use the Brayton-Joule cycle and have a relatively low electric efficiency at the level of 0.2–0.35 – depending on the turbine power [10, 16]. To increase the efficiency of the system, various modifications to the cycle are entered such as interstage cooling, heat recovery, etc. About 0.6 of power generated by the turbine (expander) is used by the compressor, so the power and efficiency can be improved by increasing the expansion work or decreasing compression work. The use of interstage cooling in a compressor results in a reduction of compression work. On the other hand, the use of a combustion chamber, divided to fundamental and after-burning parts, provides increased expansion work and increased gas temperature behind the turbine [16–18].

Another way to improve the efficiency of the gas turbine is heat regeneration. Regeneration is carried out by heating up the air by exhaust gases, outlet from the turbine in the regeneration exchanger, prior to the combustion chamber. The use of heat regeneration requires a change in the construction of the gas turbine, in order to derive the air from the compressor to the heat exchanger and deliver the air to the combustion chamber. Major benefits of regeneration are achieved at low compression ratio, because it increases the difference between the temperature of the exhaust gases and the air [16, 19].

The solution, leading to increased power and efficiency of the gas turbine, is the steam injection into the combustion chamber called STIG turbine (steam injection gas turbine). Injecting steam led to increased exhaust mass flow rate, thus increasing turbine power at the virtually unchanged compressor power. The turbine adjusting to operate with steam injection does not require significant changes to the construction of the turbine. STIG turbines are suitable for work in combined heat and power plants in Cheng system, because they should cooperate with heat recovery steam generator (HRSG) producing steam for injection [3, 5, 20]. An important advantage of the STIG turbines is its almost the same price as the turbines without injection of steam, in regard to the investment outlays per unit of power [19].

Construction of humidified air turbine (HAT) is based on concept combination of the regenerative air preheating for combustion and injects the water mass flow rate between compressor and heat exchanger (without changing the flow of compressed air). Injection of water into the air, before the regenerative heat exchanger, causes lowering of the temperature of compressed medium and therefore increases the efficiency of heat regeneration.

Because of the use of regenerative heat exchanger in HAT, the optimum compression ratios π are lower than in the STIG, and thereby resulting in a higher temperature after the turbine the order of 800 °C. In HAT systems, the temperature distribution mediums in the regenerative heat exchanger is significantly better adjusted, than in the heat recovery steam generator (HRSG) systems with steam injection or gas – steam cycles, therefore, an increase in efficiency, even above 0.5. But the disadvantage of these systems is the necessity to changes in construction and, related with that, increased investment outlays [9, 16, 21].

Another way to increase efficiency and unit power of the turbine is wet compression technique. The advantage of this method is the ability to inject this amount of water, so the process of evaporating and, thereby air cooling, extend continuously during compression. This results in independence from temperature and humidity of atmospheric air, and a reduction of the driving work of the compressor. This method does not require significant changes in the structure of the gas turbine. A disadvantage of wet compression techniques, similar like in the case of STIG and HAT, is the consumption of large amounts of demineralized water. Some of the water could be recovered, if the system acts as a combined heat and power plant with a condensing heat exchangers [14, 16].

Another method, that would increase the efficiency and power of the gas turbine, is the process of spraying water between compressor stages. As a result of the spraying, in contact with the hot and compressed air mass flow rate the water evaporates, causing lowering the air temperature and the work required to drive the compressor. This system with interstage cooling of air can be successfully used in high-pressure turbines [6, 14, 16].

Additionally, well known system combining SOFC and gas turbine working with power 300 kW, has been developed already [22]. Within the context of mounting pressures on high efficient, zero emission technologies fuel cell systems will play a major role. Since a fuel cell, like SOFC is working without the Carnot limit of efficiency, the systems build on the fuel cells can approach more effective, then conventional, energy conversion processes.

Perspective way, leading to high efficient, advance low emission combined gas turbine/fuel cells systems had been prepared and tested by Lemański [23, 24]. Main concept is to combine high-temperature solid oxide fuel cells (SOFC) with gas shifting and post-combustion of a rest fuel within a one chamber. It ought to be added that double pressure pSOFC/GT system efficiency is higher than the efficiency of traditional configuration with one pressurized fuel cell.

Huge perspectives are ahead of the new concept of cycle featuring combined Brayton cycle, inverse Brayton cycle, recuperation, oxy-combustion and capture CO_2 [14, 25]. Wet and hot exhaust gases at the atmospheric pressure from the gas turbine are able to generate extra turbine power, by expanding in the negative pressure gas turbine (GT^{in}). The pressure of gases is lower than the atmospheric one and that is why the exhaust needs to be compressed in the compressor (C^{in}), but flow mass rate is lower than in turbine because steam is condensed in condenser. The cycle described above is the inversed Brayton cycle (IBC) indeed.

Another good way to increase the efficiency of the gas turbine is the application of the Szewalski's ideas, discussed thoroughly in the next section.

2.2 Ways of modeling

To analyze the thermodynamic cycles (e.g. gas cycle, steam cycle, combined gas–steam cycle, ORC, etc.) the CFM code are commonly used. Mathematical models in CFM (included in COM-GAS, DIAGAR, Gate Cycle and Aspen Plus codes) employ mass, momentum and energy balance equations in the integrated form (also called 0D or engineering form) [9,25–31].

For example: COM-GAS program makes it possible to calculate any thermodynamic cycle on the so called design level. It has been written in the Center of Flow and Combustion IFFM PASci in Gdańsk by Topolski and Badur [26, 32–35], and has been used and developed by Karcz, Lemański, Wiśniewski, Kaczmarczyk, Ziółkowski and Kowalczyk [15, 23–25, 27, 35–38]. Computational procedures for each devices of cycle – numerical component of COM-GAS program belongs to zero-dimensional models (0D), because it contains an algebraical integral formulation of typical balances: mass, momentum and energy and equilibrium models of reactions. Additionally, COM-GAS employs thermodynamic tables for the fluid properties [26, 27, 35].

At present, the in-house code COM-GAS is developed in the Department of Energy Conversion IFFM PASci in Gdańsk that can improve the description of complex zero-emission systems. Since, in the literature there is no general consensus how to describe mathematically the mass, momentum, entropy and heat within real devices, the several convenient simplifications and idealizations are used. However, our efforts involved in this discipline seem to be more optimistic, since these are multidisciplinary within each of the processes and both thee- and zero-dimensional in mathematical description. The example is presented in works [14, 39, 40].

On the other hand, to analyse the thermodynamic cycles the basis calculation are used. The assumptions of these calculations correspond to published industrial experience and standard assumptions. They started from thermodynamic scheme and next in each indicated callout both temperature and pressure are modeled in the calculation. Pressure drops are introduced for the compressor inlet, between compressor blading outlet and turbine blading inlet (includes the combustor pressure drop) and between turbine blading exit and ambient. The state change in the compressor and in the turbine blading is assumed with polytropic (or isentropic) efficiencies. The corresponding calculation has been described in works [18, 41–45].

In the 4th Section used models was shown and compared. In the 5th Section the results of calculation was presented and discussed.

3 Gas turbine cycle with heat regeneration according to Szewalski's idea

In this section, the gas turbine cycle with heat regeneration according to Szewalski's idea is presented. Firstly, Fig. 1 shows scheme of thermal cycle of gas turbine, according to the Szewalski's patent [7] (with marked characteristic points of cycle). The idea of this solution is the exhaust gases extracting from the gas turbine (point 7 in Figs. 1 and 2), and then directing the exhaust gases mass flow rate from extraction to the regenerative heat exchanger (RE1–RE2). Extracting gases mass flow rate, $\dot{m}_{ext}$, after transfer heat flux in this regenerative heat exchanger, goes into the auxiliary (additionally) compressor (C'), where its compression process leads to the state defined in point 9. At this point, the compressed air mass flow rate, $\dot{m}_a$, combines with extracting gases mass flow rate, $\dot{m}_{ext}$, from the bleeder valve in a regenerative heat exchanger, and further the medium, going directly into the combustion chamber (CC), heats up in second part of regenerative heat exchanger (RE2). Hence, combined mass flow rate $\dot{m}_{ext} + \dot{m}_a$ heats up beginning in the temperature at the point 9 and finishing the temperature at point 10. The purpose of using modification is improvement of efficiency of cycle and reduction in the size of regenerative heat exchanger, in comparison to a solution without extraction with conventional regeneration [7].

Secondly, thermal cycle of gas turbine with heat regeneration according to Szewalski's patent is presented on T-s diagram (Fig. 2). It consist of two overlapping cycles 1-2-3-4-1 and 3-7-8-9-3. Air of initial parameter in point

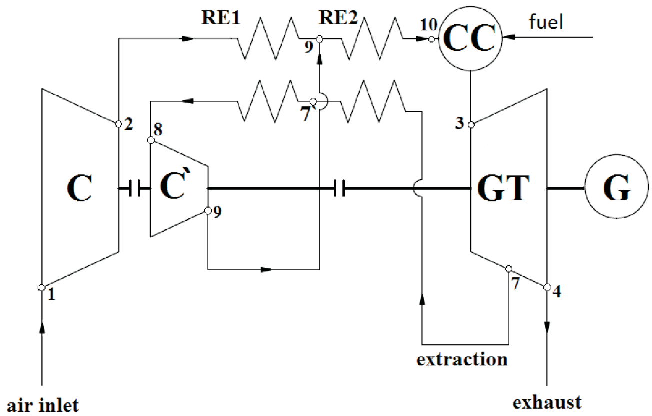

Figure 1: Scheme of gas turbine cycle with heat regeneration according to Szewalski's patent [7]: CC – combustion chamber, C – compressor, C' – auxiliary compressor, GT – gas turbine, RE – regenerative exchanger, G – electric generator.

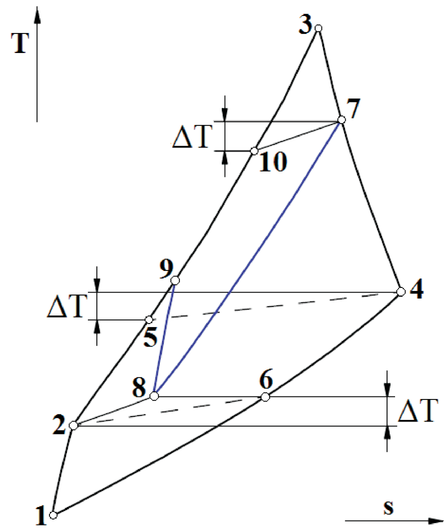

Figure 2: Thermal cycle of the gas turbine according to Szewalski patent on T-s graph [7].

1 is compressed to the state 2. Along lines 2-9 the compressed air in the regenerative heat exchanger RE1 heats up, along lines 9-10 the mixture of exhaust gases and air heats up in a heat exchanger RE2 and along lines 10-3 there is isobaric transformation into the combustion chamber.

This is followed by expansion of exhaust gases in the turbine to a state 7, where the exhaust gases mass flow rate $\dot{m}_{ext}$ separates into: the first one (extracting gases mass flow rate, $\dot{m}_{ext}$) directed to the heat exchanger as the heating medium and the second one working in the following part of turbine to point 4. Along lines 7-8 extracting gases mass flow rate, $\dot{m}_{ext}$, is isobaric cooled. That leads to the isentropic compression of the exhaust gases in the additional(auxiliary) compressor C' to parameters at point 9 and its connection with the air mass flow rate, $\dot{m}_a$ [7].

Finally, the modification of the gas turbine cycle based on the professor R. Szewalski's idea appears to be a variant, that significantly increases the efficiency of the gas turbine cycle in comparison not only to the simple cycle, but also to the regeneration cycle with maintaining the same heat exchange surface [7]. In contrast to the regeneration cycle, in modification mentioned above a medium, which gives up the heat, could be much higher pressure, what leads to reduction in the dimensions of the heat exchanger. The efficiency of the cycle improves by using a specific regeneration inside the cycle, what is currently done in the steam turbine cycles. In this case, the extracting part of the working medium stays still in the closed cycle loop (3-7-8-9-3), and thereby heating up the medium, which flows to the combustion chamber, and without any influence of external heat stream, the heat supplied to the cycle decreases. The increase in cycle efficiency depends largely on the efficiency of the compression process, which in the case of the additional compressor C' is relatively low due to the very high temperature of the working medium at this compressor inlet [7].

4 Governing equations

In this section, the procedure for calculating the various operating and thermodynamic parameters of gas turbine cycle with heat regeneration according Szewalski's idea using two models, namely basic model and CFM, will be presented.

4.1 Computational flow mechanics (CFM) model

All CFM computations of the gas turbine cycles have been performed using the basic principles of gas systems and thermodynamic phenomena modeling [26, 46, 47]. As was mentioned before mathematical models in CFM employ mass, momentum, energy balance equationsand equilibrium models of reactions in the integrated form (also called 0D or engineering form) [26,46].

Input data required to compute the compressor power are: internal and mechanical efficiency, η_{ic} and η_{mc}, respectively, compression ratio π and the air mass flow rate $\dot{m}_a$ [kg/s]. When the air parameters at the compressor inlet (T_1, p_1) are known, pressure p_2 [Pa] after the compression process is computed [26]:

$$p_2 = \pi p_1 \,. \tag{1}$$

The medium undergoing the process is air with an isentropic coefficient κ taken from thermodynamic tables for the fluid properties. Ideal compression process is described with the isentropic equation, $s = idem$:

$$T_{2s} = T_1 \, (\frac{p_2}{p_1})^{\frac{\kappa-1}{\kappa}} \,. \tag{2}$$

The Eq. (2) allows to compute the theoretical temperature at the end of compression, T_{2s}. At the assumed internal efficiency of the compressor, η_{ic}, it is possible to determine the real temperature at the end of compression, T_2. Adiabatic exponent κ and the medium's enthalpy, h, are a function of temperature and mixture composition [34]. Compressor internal efficiency may be expressed by the formula [26]

$$\eta_{ic} = \frac{h_1 - h_{2s}}{h_1 - h_2} = \frac{l_{Cs}}{l_C} \,, \tag{3}$$

where:
l_{Cs} – unit isentropic compression work [kJ/kg],
l_C – unit real compression work [kJ/kg],
h_1, h_2, h_{2s} – specific enthalpy of the medium determined at characteristic points [kJ/kg].

Power N_C required for compressor propulsion is determined with the following formula:

$$N_C = \dot{m}_a \, \eta_{mc} \, (h_2 - h_1) \,. \tag{4}$$

Furthermore, the additional compressor's internal power, $N_{C'}$, is:

$$N_{C'} = \dot{m}_{ext} \, \eta_{mc} \, (h_9 - h_8) \,, \tag{5}$$

where:
h_8, h_9 – the specific enthalpy of the medium at characteristic points [kJ/kg];
$\dot{m}_{ext}$ – the extracting gases mass flow rate [kg/s].

It should be added that efficiency of auxiliary compressor C' value was $\eta_{iC'} = 0.87\%$.

Computation of the combustion chamber requires an energy balance to be performed including all input and output energy fluxes. Heat losses in the combustion chamber (into the surrounding) are specified with use of the combustion chamber efficiency, η_{CC} [23, 26]. The fuel chemical energy flux supplied to the combustion chamber is given by the formula

$$\dot{Q}_{chem} = \dot{m}_f \, W_d \,, \tag{6}$$

where:

$\dot{m}_f$ – fuel mass flow rate [kg/s],
W_d – lower heating value depending on composition of the fuel[kJ/kg].

Energy balance for the combustion chamber may be expressed as [26]

$$(\dot{m}_a - \dot{m}_\gamma + \dot{m}_{ext})h_{10} + \dot{m}_f(h_f + W_d \cdot \eta_{CC}) = (\dot{m}_a - \dot{m}_\gamma + \dot{m}_{ext} + \dot{m}_f)h_3 \,, \tag{7}$$

where:

$\dot{m}_\gamma$ – the cooling mass flow rate[kg/s],
h_{10} – the enthalpy of the working medium at the inlet of combustion chamber [kJ/kg],
h_f – the specific enthalpy of the fuel at the entry to the combustion chamber [kJ/kg],
$h_3 = h_{ex}$ – the specific exhaust gases enthalpy at the exit of combustion chamber [kJ/kg].

The preceding energy balance can be simplified by no taking account the fuel enthalpy, because it is negligibly small in comparison with lower heating value W_d.

Heat fluxes, $\dot{Q}_{RE1}$ and $\dot{Q}_{RE2}$ exchanged in regenerative exchanger RE1 and RE2, with the assumed heat exchanger efficiency, η_{RE}, can be expressed as

$$\dot{Q}_{RE1} = (\dot{m}_a - \dot{m}_\gamma)(h_9 - h_2) = \dot{m}_{ext}(h_{7'} - h_8)\,\eta_{RE} \,, \tag{8}$$

$$\dot{Q}_{RE2} = (\dot{m}_a - \dot{m}_\gamma + \dot{m}_{ext})(h_{10} - h_9) = \dot{m}_{ext}(h_7 - h_{7'})\,\eta_{RE} \,. \tag{9}$$

Moreover, it requires mentioning, that the following mass balances need to be satisfied [15]:

$$\dot{m}_2 = \dot{m}_a - \dot{m}_\gamma \,, \tag{10}$$

for the point between regenerative heat exchangers RE1 and RE2:

$$\dot{m}_9 = \dot{m}_a - \dot{m}_\gamma + \dot{m}_{ext}\,, \tag{11}$$

for the outlet of combustion chamber CC:

$$\dot{m}_3 = \dot{m}_a - \dot{m}_\gamma + \dot{m}_{ext} + \dot{m}_f\,, \tag{12}$$

and for turbine GT outlet:

$$\dot{m}_4 = \dot{m}_a - \dot{m}_\gamma + \dot{m}_f\,, \tag{13}$$

where: $\dot{m}_2$, $\dot{m}_9$, $\dot{m}_3$, $\dot{m}_4$ – mass flow rates at appropriate points of scheme from Fig. 1.

For simplicity, fuel (gas) is assumed to undergo perfect combustion. General the chemical reactions are [26]

$$\mathrm{CH_4} + 2\,\mathrm{O_2} \rightarrow \mathrm{CO_2} + 2\,\mathrm{H_2O}\,, \tag{14}$$

$$\mathrm{C_2H_6} + 3.5\,\mathrm{O_2} \rightarrow 2\,\mathrm{CO_2} + 3\,\mathrm{H_2O}\,, \tag{15}$$

$$\mathrm{C_3H_8} + 5\,\mathrm{O_2} \rightarrow 3\,\mathrm{CO_2} + 4\,\mathrm{H_2O}\,. \tag{16}$$

Based on the composition of the resulting exhaust gases and energy balance of the combustion chamber, temperature $T_{TIT} = T_3$ and enthalpy h_3 are computed from energy balance in iteration process.

Expansion process in the gas turbine is characterized by means of a defined turbine internal efficiency η_{it} as follows [32]:

$$\eta_{it} = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{l_{GT}}{l_{GTs}}\,, \tag{17}$$

where:

l_{GTs}	–	unit isentropic work of expansion [kJ/kg],
l_{GT}	–	real unit work of expansion [kJ/kg],
h_3, h_4, h_{4s}	–	specific exhaust gases enthalpy at characteristic points [kJ/kg].

The power generated in the turbine, N_{GT}, is expressed as

$$N_{GT} = \dot{m}_{ex}\,\eta_{mt}\,(h_3 - h_7) + (\dot{m}_{ex} - \dot{m}_{ext})\,\eta_{mt}\,(h_7 - h_4)\,, \tag{18}$$

where η_{mt} is the turbine machine efficiency.

Electric power of the generator terminals is defined upon the mechanical power of the individual components of the thermodynamic cycle, i.e., the gas turbine, N_{GT}, the compressor, N_C, the auxiliary compressor, $N_{C'}$, and the generator efficiency, η_g. Hence electrical power of the gas turbine with heat regeneration according to Szewalski's idea, N_{el}, is the difference between the devices generating and consuming the power with respecting generator according to the following expression [15]

$$N_{el} = (N_{GT} - N_C - N_{C'})\eta_g \,. \tag{19}$$

The electric power efficiency of the system, η_{el}, is defined as a quotient of the electric power, N_{el}, generated in the unit, and the chemical energy flux, $\dot{Q}_{chem}$, of the fuel [26]:

$$\eta_{el} = \frac{N_{el}}{\dot{Q}_{chem}} = \frac{N_{el}}{\dot{m}_f \, W_d} \,. \tag{20}$$

4.2 Basic model

A calculation algorithm to determine basic parameters of the gas turbine-has been presented below. The calculation methods with the most commonly adopted parameter values, such as the thermodynamic properties of the working medium, compression and expansion efficiency, were taken from [18, 42]. Scheme of analyzed cycle is shown on Fig. 1.

In the beginning, it should be mentioned that specification of thermodynamic properties of used working medium was based on the average values in such way that its values should fulfill relation

$$c_p = \frac{\kappa}{\kappa - 1} R \,, \tag{21}$$

where: c_p – specific heat at constant pressure, κ – adiabatic exponent, R – gas constant.
It should be added that we assume throttling before compressor, so transformation 0-1 took meaning of isenthalpic throttling where in point 0 there were assumed conditions $p_0 = 0.1013$ MPa (absolute pressure), $T_0 = 288.15$ K, so

$$h_0 = c_{pC} T_0 \,. \tag{22}$$

Relative pressure losses of transition 0-1 is equal $\frac{\Delta p_{0-1}}{p_0}$ =0.003 [–], similar like in other cases of pressure loss modeling [18, 42]

$$p_1 = p_0 - p_0 \frac{\Delta p_{0-1}}{p_0} \,, \tag{23}$$

for perfect gases the temperature at the end of the isenthalpic throttling process:

$$h_1 = h_0\,, \tag{24}$$

where is the same as the intial temperature:

$$T_1 = T_0\,. \tag{25}$$

Transformation 1–2 was modeled as an irreversible adiabatic compression of efficiency, η_C, and pressure ratio π:

$$p_2 = \pi\, p_1\,, \tag{26}$$

$$T_2 = T_1 \left[1 + \frac{1}{\eta_C}\left(\pi^{\frac{\kappa_C - 1}{\kappa_C}} - 1\right)\right], \tag{27}$$

where η_C is the isentropic compression efficiency and is equal $\eta_C = 0.865$. So specific enthalpy is defined by

$$h_2 = c_{pC}\, T_2\,. \tag{28}$$

On the one hand pressure p_9 is determined

$$p_9 = p_2 - p_2\,\frac{\Delta p_{2-9}}{p_2}\,, \tag{29}$$

on the other hand pressure p_9 and temperature T_9 was determined as the parameter of the end of the compression at efficiency $\eta_{C'}$ of auxiliary compressor C':

$$p_9 = \pi_{ext}\, p_8\,, \qquad T_9 = T_8 \left[1 + \frac{1}{\eta_{C'}}\left(\left(\frac{p_9}{p_8}\right)^{\frac{\kappa_C - 1}{\kappa_C}} - 1\right)\right], \tag{30}$$

where π_{ext} is the pressure ratio in additional compressor.

Temperature T_8 was calculated with the assumption of maximum utilization in exchanger with the available heat energy, by setting the minimum temperature difference $\Delta T_{RE} = 30$ K between the temperature of cooling medium at inlet and temperature of medium supplied heat at regenerative exchanger outlet:

$$T_8 = \Delta T_{RE} + T_2\,. \tag{31}$$

Based on the compression efficiency and temperature difference, ΔT_{RE}, in regenerative heat exchanger, temperature T_8 and enthalpy h_9 are computed in iteration process

$$h_9 = c_{pZ}\, T_9\,. \tag{32}$$

The replacement values of specific heat at constant pressure, c_{pZ}, and at gas constant R_Z of mixture of air and exhaust gases were calculated as follows:

$$g_a = \frac{\dot{m}_a - \dot{m}_\delta}{\dot{m}_a - \dot{m}_\delta + \dot{m}_{ext}} , \tag{33}$$

$$g_{ex} = \frac{\dot{m}_{ext}}{\dot{m}_a - \dot{m}_\delta + \dot{m}_{ext}} , \tag{34}$$

$$c_{pZ} = g_a \, c_{pC} + g_{ex} \, c_{pT} , \tag{35}$$

$$R_Z = \sum_{i=1}^{n} g_i \, R_i , \tag{36}$$

where $\dot{m}_\delta$ is the mass flow rate used by leakage blocking system in turbine and compressor seals [kg/s]. Pressure p_{10} is determined on the basis of exchangers momentum balance:

$$p_{10} = p_9 - p_9 \, \frac{\Delta p_{9-10}}{p_9} . \tag{37}$$

Temperatures T_{10} and $T_{7'}$ were determined on the basis of exchangers momentum and energy balances RE1 and RE2, the equations below are transformed equations of mentioned exchangers balances:

$$T_{10} = \frac{(\dot{m}_{ext} + \dot{m}_a - \dot{m}_\delta) \, c_{pZ} \, T_9 + \dot{m}_{ext} \, c_{pT} \, (T_7 - T_{7'})}{(\dot{m}_{ext} + \dot{m}_a - \dot{m}_\delta) \, c_{pZ}} , \tag{38}$$

for the temperature T_{10} specification were needed to calculate temperature of extraction stream T_7:

$$T_7 = T_3 \left\{ 1 - \eta_T \left[1 - \left(\frac{p_7}{p_3} \right)^{\frac{\kappa_T - 1}{\kappa_T}} \right] \right\} , \tag{39}$$

and RE2 heat exchanger hot side outlet temperature $T_{7'}$:

$$T_{7'} = \frac{c_{pC} \, (\dot{m}_a - \dot{m}_\delta)(T_{7'} - T_2) + T_8 \, c_{pT} \dot{m}_{ext}}{c_{pT} \, \dot{m}_{ext}} . \tag{40}$$

To make these calculations determinable it was necessary to made the assumption from Eq. (31) $T_8 = \Delta T_{RE} + T_2$. Then it was possible to specify enthalpy h_{10}:

$$h_{10} = T_{10} \, c_{pZ} . \tag{41}$$

Pressure loses in combustion chamber CC were modeled as previously as isenthalpic throttling

$$p_3 = p_{10} - p_{10}\,\frac{\Delta p_{10-3}}{p_{10}}\,. \tag{42}$$

Combustion chamber outlet temperature T_3 was adopted from [13], so specific enthalpy after combustion chamber is defined:

$$h_3 = T_3 \cdot c_{pT}\,. \tag{43}$$

Transformation 3–4 was arranged as an irreversible adiabatic expansion in the turbine of efficiency, η_C, so thermodynamic parameters in analogy are defined as

$$p_4 = p_5 - p_5\,\frac{\Delta p_{5-4}}{p_5}\,,\quad T_4 = T_3\left\{1-\eta_T\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{\chi_T-1}{\chi_T}}\right]\right\},\quad h_4 = T_4\,c_{pT}\,. \tag{44}$$

Transformation 4–5 was modeled as previously as isenthalpic throttling, what was denoted by Eqs. (24)–(25), so in analogy

$$T_5 = T_4\,, \qquad h_5 = h_4\,, \tag{45}$$

with assumption that $p_5 = 0.1013$ MPa.

Temperature of the gas received from the gas turbine bleeder is calculated as the temperature of the end of irreversible adiabatic expansion of efficiency η_T to the assumed bleeder pressure $p_{ext} = p_7 =$ various hence, temperature and specific enthalpy are equal:

$$T_7 = T_3\left\{1-\eta_T\left[1-\left(\frac{p_7}{p_3}\right)^{\frac{\kappa_T-1}{\kappa_T}}\right]\right\},\ h_7 = c_{pT}\,T_7\,. \tag{46}$$

Pressure loses in regenerative heat exchangers was also modeled:

$$p_{7'} = p_7 - p_7\,\frac{\Delta p_{7-7'}}{p_7}\,,\quad p_8 = p_8 - p_8\frac{\Delta p_{7-8}}{p_8}\,. \tag{47}$$

In the following expression there is shown the balance of the combustion chamber, without fuel enthalpy, h_f, as insignificant in comparison with the appropriate chemical energy of fuel [42]:

$$(\dot m_a - \dot m_\delta - \dot m_\gamma + \dot m_{ext})\,h_{10} + \eta_{CC}\,\dot m_f\,W_d = (\dot m_a - \dot m_\delta - \dot m_\gamma + \dot m_{ext} + \dot m_f)\,h_3\,, \tag{48}$$

where $\dot{m}_\delta$ is the mass flow rate used by leakage blocking system in turbine and compressor seals [kg/s], and $\dot{m}_\gamma$ is the cooling mass flow rate [kg/s]. Hence fuel mass flow rate is defined as

$$\dot{m}_f = \frac{(\dot{m}_a - \dot{m}_\delta - \dot{m}_\gamma + \dot{m}_{ext})(h_3 - h_{10})}{(W_d\,\eta_{CC} - h_3)}\,. \tag{49}$$

Fuel mass contribution in the mass of air and flue gas, supplied combustion chamber, was

$$\alpha = \frac{\dot{m}_f}{\dot{m}_a - \dot{m}_\delta - \dot{m}_\gamma + \dot{m}_{ext}} = \frac{h_3 - h_{10}}{W_d\,\eta_{CC} - h_3}\,, \tag{50}$$

while amount of heat supplied to the cycle in the form of fuel stood at:

$$\dot{Q}_{chem} = \dot{m}_f\,W_d\,. \tag{51}$$

In the following expressions (52–54) were placed calculations and corrections of specific works:

$$l_C^* = \Big(h_2 - h_1\Big) + \frac{\dot{m}_{ext}}{\dot{m}_a}\Big(h_9 - h_8\Big)\,, \tag{52}$$

where l_C^* is the specific compressor work without correction;

$$l_T^* = (h_3 - h_7) + \left(\frac{\dot{m}_a - \dot{m}_\delta + \dot{m}_f}{\dot{m}_a - \dot{m}_\delta + \dot{m}_f + \dot{m}_{ext}}\right)(h_7 - h_4)\,, \tag{53}$$

where l_T^* is the specific turbine work without correction;

$$l_n^* = \frac{\dot{m}_a - \dot{m}_\delta + \dot{m}_f + \dot{m}_{ext}}{\dot{m}_a} l_T^*\,\eta_m - l_C^*\,, \tag{54}$$

where l_n^* is the specific power output, and η_m is the mechanical efficiency.

It must be noted that authors in basic model of cycle assumed that mechanical losses are occurring only in turbine part of cycle as like in similar models in literature [18, 42].

$$N_C^* = l_C^*\,\dot{m}_a\,, \tag{55}$$

$$N_T^* = l_T^*\,(\dot{m}_a - \dot{m}_\delta + \dot{m}_f + \dot{m}_{ext})\,, \tag{56}$$

$$\eta_0^* = \frac{l_n^*}{\frac{\dot m_f}{\dot m_a}\, W_d} \,. \tag{57}$$

In the following expressions Egs. (58–59) there are presented corrections for efficiency η_0 and specific power output l_n, which results from gas turbine GT elements cooling:

$$\eta_0 = \eta_0^* \left(1 - \frac{\Delta\eta_0}{\eta_0} \cdot \frac{\dot m_\gamma}{\dot m_a}\right), \tag{58}$$

$$l_n = l_n^* \left(1 - \frac{\Delta l_n}{l_n}\, \frac{\dot m_\gamma}{\dot m_a}\right). \tag{59}$$

Finally, electrical efficiency, η_{el}, was appointed from relation

$$\eta_{el} = \eta_0\, \eta_g \,, \tag{60}$$

where η_g is the efficiency of electric generator G.

The temperature of the exhaust gases were determined in accordance with [18]. Firstly, there was found the efficiency of expansion in turbine, for the case without cooling, that gives the same efficiency cycle as in the case of cooling, which was $\eta_T^* = 0.89883$. Secondly, the correct temperature of the outlet exhaust gas from the turbine T_{EXH} were calculated:

$$T_{EXH} = T_5 = T_3 \left\{1 - \eta_T^* \left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\kappa_T - 1}{\kappa_T}}\right]\right\} . \tag{61}$$

Summarizing for basic calculations of the gas turbine cycle and for modification in accordance with the idea of Szewalski, there were made the following assumptions [18, 42, 44, 45, 47, 48]:

- relative pressure losses in inlet system $\frac{\Delta p_{0-1}}{p_0} = 0.003$;
- compression efficiency of the basic compressor $\eta_C = 0.865$;
- relative cooling air mass flow rate $\gamma = 0.0783$;
- relative air mass flow rate blocking leakage losses $\delta = 0.005$;
- relative pressure losses in combustion chamber $\frac{\Delta p_{2-3}}{p_2} = 0.003$;
- the efficiency of combustion chamber $\eta_{CC} = 0.99$;
- the efficiency of turbine $\eta_T = 0.91$;
- relative pressure losses in outlet system $\frac{\Delta p_{5-4}}{p_4} = 0.003$;

- the mechanical efficiency $\eta_m = 0.99$;
- thermodynamic properties of the working medium: $\kappa_C = 1.4$ (isentropic exponent for air) , $\kappa_T = 1.33$ (isentropic exponent for exhaust gases), $c_{pC} = 1.005$ (specific heat with air at constant pressure), $c_{pT} = 1.165$ (specific heat with exhaust gases at constant pressure);
- net calorific value of methane $W_d = 50.035$ MJ/kg;
- relative pressure losses on heat exchangers RE1 and RE2 on low-temperature (cold) side: $\frac{\Delta p_{2-9}}{p_2} = \frac{\Delta p_{9-10}}{p_9} = 0.006$;
- relative pressure losses on heat exchangers RE1 and RE2 on high-temperature (hot) side: $\frac{\Delta p_{7-7'}}{p_7} = \frac{\Delta p_{7'-8}}{p_{7'}} = 0.0075$;
- compression efficiency of the additional compressor C' $\eta_{C'} = 0.8$.

5 Results

5.1 Simple gas turbine cycle

Table 1 gathers simulation results acquired in two separate models, i.e., CFM and basic; simulation results are compared with measured nominal operation parameters of the GT8C gas turbine available in exploitation paper [10–12] and data of producent [13]. Data from exploitation experiences comes from PGE Gorzow heat and power plants operate on a 54.5 MWe-class gas turbine (GT8C) manufactured by Alstom Power. The comparison, presented in Tab. 1, confirmed good agreement between measurement and simulation, there by proving correctness of the adopted methodology. As it is shown in Tab. 1, the electrical efficiency of the only $N_{el} = 54.5$ MWe – class gas turbine averages between $\eta_{el} = 0.342$ and $\eta_{el} = 0.343$ depending on the model used during modeling process. It should be added that GT8C combusting natural gas, with high content of nitrogen [5, 11], and because that fuel mass flow rate $\dot{m}_f = 8.36$ kg/s is higher than CH_4 mass flow rate $\dot{m}_f = 3.397$ kg/s estimated in basic model. The carbon dioxide emission (CO_2) and nitrogen oxides emission (NOx), averaged in CFM code, approaches nearly 505 kg/MWh and 0.324 kg/MWh, respectively. The emission of nitrogen oxides (NOx) versus the combustion temperature has been kept linear, instead of exponential character reported in literature [49], due to a constant value of the conversion rate of nitrogen assumed in the calculations. To estimate an accurate emission level of nitrogen oxides, it is necessary to use a 3D model of combustion chamber and burners in CFD

Table 1: The results of calculations of GT8C in basic model and CFM code.

Parameter	Unit	Producent data [13]	Data from exploitation experiences [10–12]	CFM code	Basic model
$T_a = T_1$	K	288.15	288.15	288.15	288.15
$P_a = P_1$	MPa	0.1013	0.1013	0.1013	0.1013
π	–	15.7	16	16	16
T_f	K	288.15	288.15	288.15	288.15
p_f	MPa	$\tilde{4}0.5$	40.5	40.5	40.5
$\dot{m}_f$	kg/s	–	–	8.36	3.397
$\dot{m}_{ex}$	kg/s	180	182.3	182.3	182.3
$T_{GT} = T_4$	K	793.15	793.15	793.15	799.55
$T_{ISOTIT} = T_3$	K	1373.15	1373.15	1373.15	1373.15
η_{el}	–	0.345	0.346	0.343	0.342
N_{el}	MWe	52	54.49	54.49	54.4
CO_2	kg/MWh	–	313	505	–
NO_x	kg/MWh	–	0.324	0.324	–

framework [46]. However, the obtained results should be regarded as satisfactory in spite of existing differences between results from both models and literature data.

5.2 Results modification into gas turbine with heat regeneration according to Szewalski's idea

We will describe here the main results of thermodynamic analysis on an example of GT8C obtained in CFM code and basis model. The profitability of the Szewalski cycle will be clarified by comparison of GT8C gas turbine, before and after modernization into the Szewalski cycle. However, Ziółkowski et al. [9,14] have verified the gas turbine with heat regeneration according to Szewalski's idea making parameterization of the extraction: mass flow rate $\dot{m}_{ext}$ and pressure p_{ext}. Szewalski idea of the Brayton cycle modification caused an increase of the cycle efficiency, decreased amount of fuel burned in the turbine.

In conducted numerical calculations the values compressor and turbine internal efficiencies, and other devices efficiencies, and most of volumes entered to the devices were maintained as same as in previous calculation examples held in simple gas turbine cycle and in papers [5,9]. On the other hand, assumption of basic model was presented in Section 4.2. and it was maintained, too. Before describing conducted analysis, it should be emphasized that temperatures in characteristic points of cycle were maintained, and were respectively in combustion chamber $T_{TIT} = T_3 = 1373.15$ K, and in outlet of turbine $T_{GT} = T_4 = 793.15$ K, in both cases – basis and CFM model. As was mention before in calculations of both models assumed that the minimum temperature difference in regenerative heat exchangers between working medium at high temperature and working medium at low temperature is $\Delta T_{RE} = 30$ K.

This minimum temperature refers to heat exchanger RE1 for all calculations cases, and to heat exchanger RE2 for case which exhaust gases mass flow rate from extraction $\dot{m}_{ext} = 82$ kg/s and $\dot{m}_{ext} = 80$ kg/s for CFM model and basic model respectively.

Air temperature after primary compressor C, so in the inlet of regenerative heat exchanger RE1 always is equal to $T_C = T_2 = 655.15$ K for CFM model and $T_C = T_2 = 690.62$ K for basic model. The air was heating until its temperature reached value equal to the working medium temperature after auxiliary compressor $T_{C'} = T_9$ (Figs. 1 and 2). Thus, at a constant temperature $T_C = T_2 = const$ also auxiliary compressor C' inlet temperature T_8 is constant, and equal to $T_8 = T_2 + \Delta T = 655.15\text{ K} + 30\text{ K} = 688.15$ K CFM model or 720.62 K basic model.

On the other hand, the value of temperature difference between working mediums at the side of auxiliary compressor, mainly $\Delta T_{in\,RE} = T_9 - T_{7'}$, so at the RE1 regenerative exchanger outlet or at RE2 regenerative heat exchanger inlet, changed. The value of $\Delta T_{in\,RE}$ is in the large range from several to several hundred degrees, but in accordance with the assumptions, to final analysis on the figures the temperature difference $\Delta T_{in\,RE} \geq 30$ K was taken into account. In calculations, various values of extraction pressure p_{ext} and extraction mass flow rate $\dot{m}_{ext}$ were set in order to find maximum cycle electrical efficiency η_{el}.

Figures 3 and 4 presented gas turbine GT8C electrical efficiency, η_{el}, change as a function of extraction pressure p_{ext} and extraction mass flow rate, $\dot{m}_{ext}$. Maximum electrical efficiency for gas turbine achieved for extraction mass flow rate equal to $\dot{m}_{ext} = 82$ kg/s and in extraction pressures values $p_{ext} = 0.85$ MPa in CFM model and for extraction mass flow rate equal to $\dot{m}_{ext} = 80$ kg/s and in extraction pressures values $p_{ext} = 0.6$ MPa in basic model. So the gain of electrical efficiency by realization gas tur-

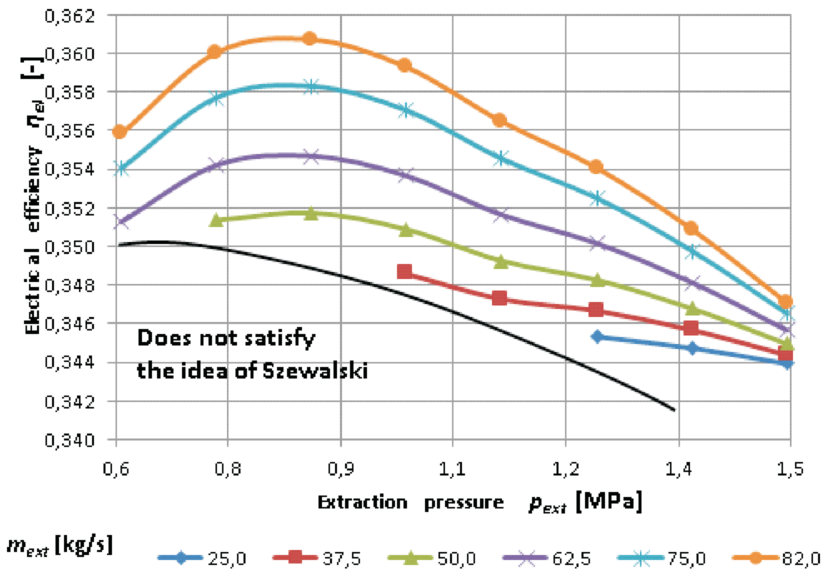

Figure 3: Efficiency of GT8C appointed by CFM code as a function of extraction pressure, p_{ext}, and the extraction mass flow rate, $\dot{m}_{ext}$.

bine cycle with heat regeneration according to Szewalski's idea is about 0.23 in comparison to the simple cycle (in the case of basic model calculations) or 0.0525 (in the case of calculation by CFM code). Further, it should be noted that turbine inlet pressure was equal 1.6 MPa.

Under the influence of the increasing extraction of exhaust gases mass flow rate $\dot{m}_{ext}$, the exhaust gases mass flow rate $\dot{m}_{ex}$, passing through the final stages turbine, changes. In both cases electrical power gas turbine N_{el}, CFM model and basic model, decreases (Figs. 5 and 6).

In Figs. 5 and 6 there are respectively presented dependences of gas turbine electrical power, N_{el}, in CFM code and basic model on extraction pressure p_{ext} and mass flow rate, $\dot{m}_{ext}$. Application of Szewalski's idea in gas turbine in this case causes turbo set electric power, N_{el}, decrease, however is possibility to increase electrical power. Analyzing figures leads to conclusion that extraction mass flow rate $\dot{m}_{ext}$ rise results in gas turbine electric power, N_{el}, loss. This happens, because of flue gases (exhaust gases) mass flow rate through turbine after extraction decrease. If mass flow rate $\dot{m}_{ex} = \dot{m}_4 = const$ through turbine after extraction is constant than mass

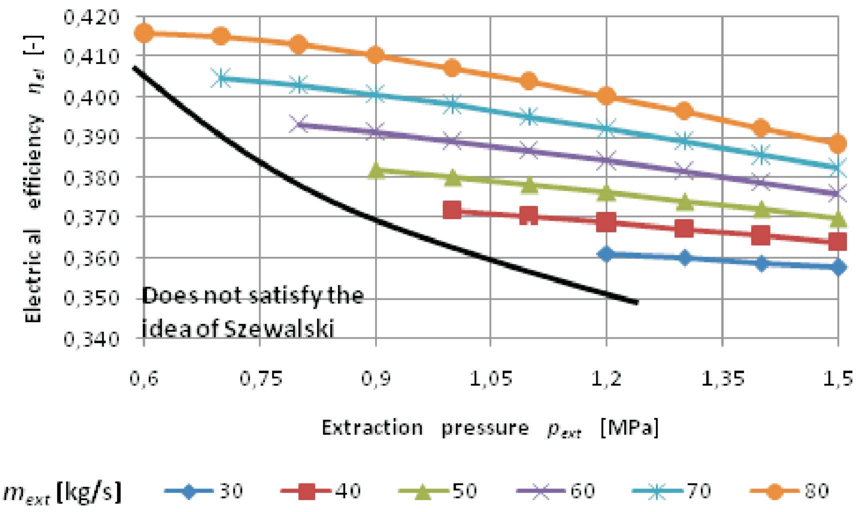

Figure 4: Efficiency of GT8C appointed by basic model as a function of extraction pressure, p_{ext}, and the extraction mass flow rate, $\dot{m}_{ext}$.

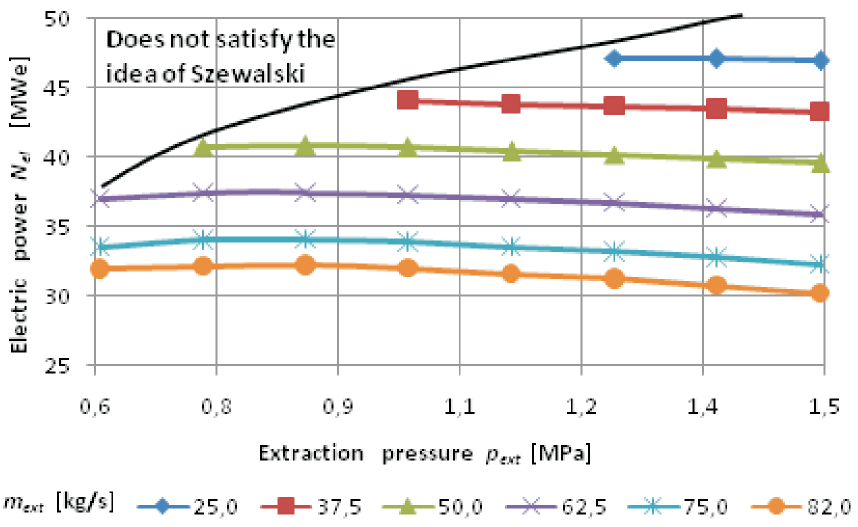

Figure 5: GT8C gas turbine electric power, N_{el}, appointed by CFM code as a function of extraction pressure, p_{ext}, and the extraction mass flow rate, $\dot{m}_{ex}$.

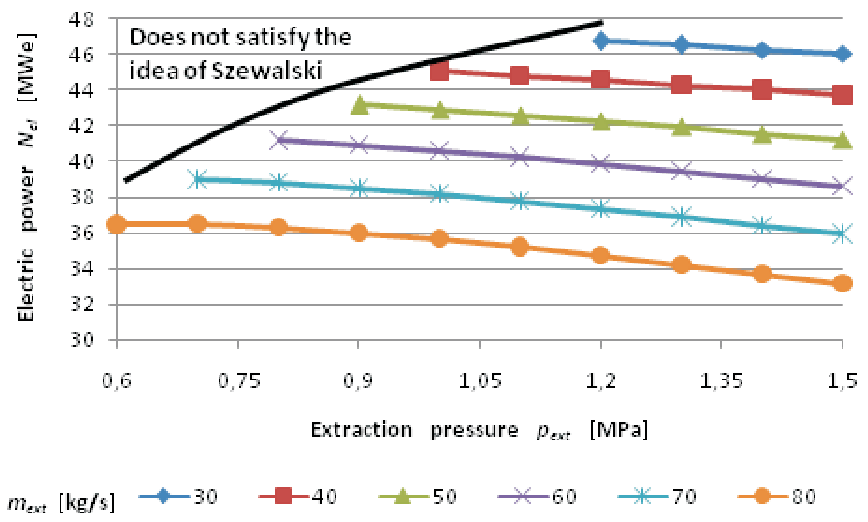

Figure 6: GT8C gas turbine electric power, N_{el}, appointed by basic model as a function of extraction pressure, p_{ext}, and the extraction mass flow rate, $\dot{m}_{ex}$.

flow rate in combustion chamber $\dot{m}_{ex} = \dot{m}_3$ have to increase and electric power, N_{el}, will be increased.

Comparing the results summarized in Figs. 3–6, it can be seen that with decreasing the extraction pressure p_{ext}, the capabilities of extraction mass flow $\dot{m}_{ext}$ usage are reduced. It happens in the case of low mass flow rate, because hot working medium (flue gas) will not heat up the cold working medium to temperature of air. The air, heated up in regenerative heat exchanger RE, has lower temperature than the outlet gases from auxiliary compressor. Therefore, it is not possible to meet Szewalski's idea, because of failure to meet the same thermodynamic parameters by gases, which mix in heat exchanger split point marked by number 9.

In turn, by increasing the extraction mass flow rate, $\dot{m}_{ext}$, the temperature difference $\Delta T_{in\,RE}$ between mediums, in split point of heat exchanger, decreases. For example in CFM calculations: for $p_{ext} = 0.7345$ MPa and $\dot{m}_{ext} = 87.5$ kg/s – $\Delta T_{in\,RE} = 7$ K, and for $p_{ext} = 1.4944$ MPa and $\dot{m}_{ext} = 87.5$ kg/s – $\Delta T_{in\,RE} = 25$ K. Thus the extraction mass flows rate were limited to value $\dot{m}_{ext} = 82$ kg/s, for which $\Delta T_{in\,RE} = 30$ K, as a minimal temperature difference value between heated up medium of point 9 thermodynamic parameters and exhaust gases, which gives the heat. For basic model the extraction mass flows rate were limited to value $\dot{m}_{ext} = 80$ kg/s.

The value of $\Delta T_{in\,RE}$ is strongly dependent on extraction pressure, p_{ext}, and extraction mass flow rate, $\dot{m}_{ext}$, and takes high values in range of low extraction pressures, p_{ext}, values and extraction mass flow rate, $\dot{m}_{ext}$. Another parameter taken into account is working medium temperature before combustion chamber $T_{bCC} = T_{10}$, that is in the Fig. 1 right behind RE2 regenerative heat exchanger. Exampled calculation results of GT8C with application the idea of Prof. Szewalski are summarized in Figs. 7 and 8. Hence, dependences temperature before combustion chamber on extraction mass flow rate $\dot{m}_{ext}$ and extraction pressures p_{ext} are presented for CFM model and basic model, respectively.

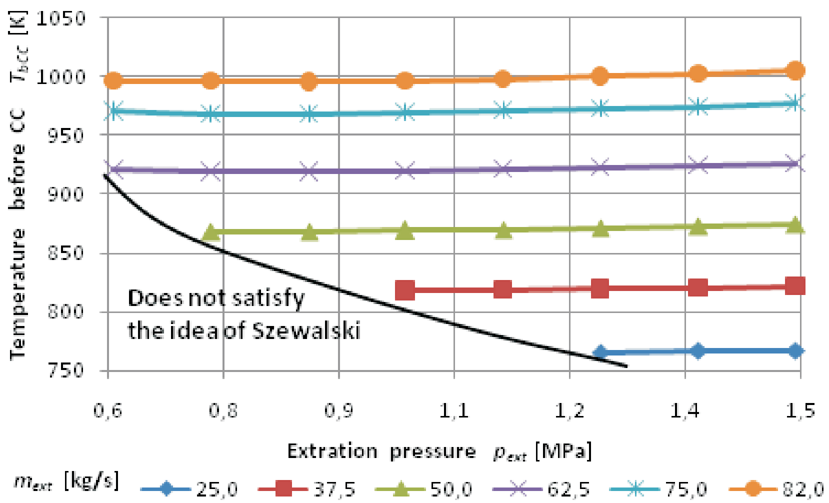

Figure 7: Temperature before combustion chamber T_{bCC} appointed by CFM code as a function of extraction mass flow rate, $\dot{m}_{ext}$, and extraction pressures p_{ext}.

Figure 9 shows change of fuel supplied to combustion chamber mass flow rate (natural gas, with high content of nitrogen) as a function of extraction mass flow rate, $\dot{m}_{ext}$, and extraction pressures, p_{ext}. Generally, increase of extraction mass flow rate, $\dot{m}_{ext}$, results in fuel mass flow rate decrease, $\dot{m}_f$, because temperature of working medium before combustion chamber is increasing.

In Fig. 10 there is presented comparison of both models, mainly basic model and CFM model, for extraction temperature, T_{ext}, and auxiliary compressor outlet temperature $T_{C'}$ under the influence of extraction pressure

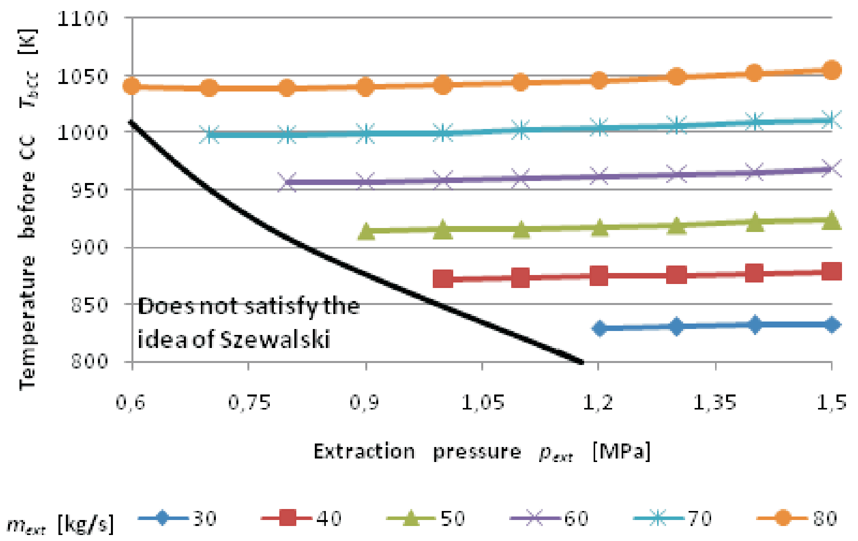

Figure 8: Temperature before combustion chamber T_{bCC} appointed by basic model as a function of extraction mass flow rate, $\dot{m}_{ext}$, and extraction pressures, p_{ext}.

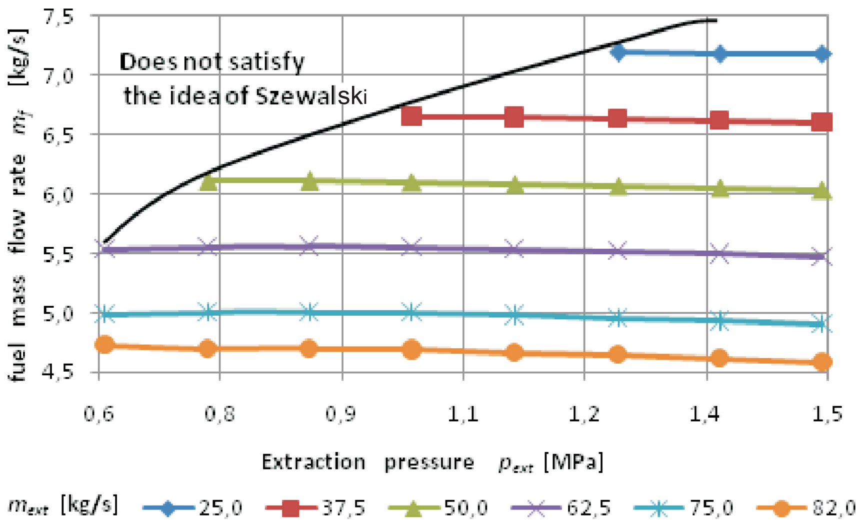

Figure 9: Mass flow rate of fuel $\dot{m}_f$ supplied to combustion chamber as a function of extraction pressure, p_{ext}, and extraction mass flow rate, $\dot{m}_{ext}$ – appointed by CFM code.

p_{ext}. Both temperatures are independent on extraction mass flow rate and they are functions only of expansion process for temperature T_{ext} and compression process for temperature at auxiliary compressor outlet $T_{C'}$. Both processes have fixed isentropic process beginning temperatures for expansion this is the temperature in combustion chamber $T_{TIT} = 1373.15$ K and for compression flue gas regenerative heat exchangers outlet temperature $T_8 = 685.15$ K in CFM case and $T_8 = 720.62$ K in basic model case. Additionally, estimated temperature in basic model gives higher values than in CFM code.

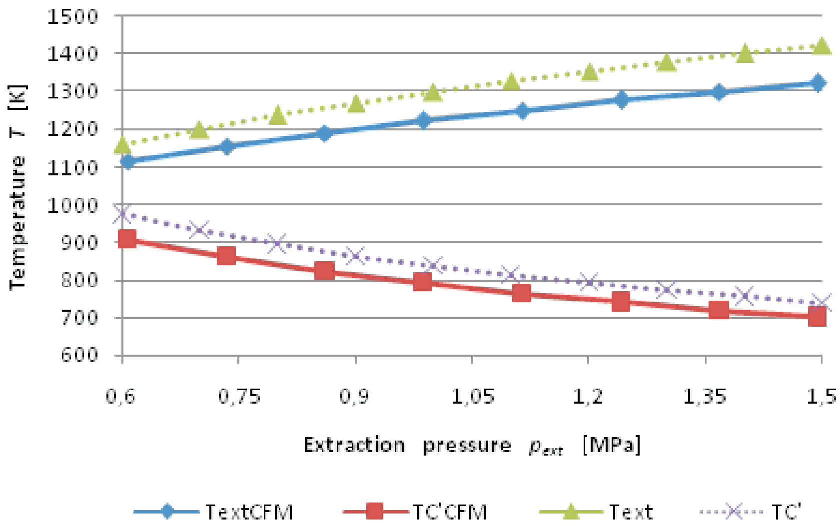

Figure 10: Extraction temperature T_{ext} and auxiliary compressor outlet temperature, $T_{C'}$, as a function of extraction pressure, p_{ext} – appointed by basic model and CFM code.

Results of calculations for different values of compression ratio, π, for temperatures behind combustion chamber T_3, extraction pressure, p_{ext}, and extraction mass flow rate, $\dot{m}_{ext}$, are shown in Fig. 11,which presents a comparison of this results for the following cycles: simple, and this one of the modification by Prof. Szewalski. Hence, figure presents the relation between the electrical efficiency, η_{el}, of the gas turbine cycle modified by Szewalski's idea and the simple cycle of gas turbine depending on specific work, l_n. Subsequent lines on the graph represent the dependences, designated during calculations, η_{el}= f (l_n), for subsequent mass flows rate from the turbine extraction $\dot{m}_{ext}$ = 30-80 kg/s and for various pressure ratio in basic compressor $\pi = 7$-33. As was already mentioned, the points are plotted on the graph only in cases, when the conditions for maintaining the assumed temperature differences in regenerative heat exchangers were satisfied. Additionally, the points were selected under the condition of the maximum cycle efficiency depending on the compression ratio, π, i.e., from among several points $\eta_{el} = f(l_n)$ that were possible for a given compression ratio, π, and for different pressures of extraction, p_{ext}, those with the maximum electrical efficiency, η_{el}, were chosen. In all cases, the lines suitable for Szewalski's modification have their beginning in the point where the compression ratio is $\pi = 7$ and end with the final compression ratio π still satisfied condition of temperature difference in the regenerative heat exchangers. Respectively, the maximum electrical efficiency η_{el} of the modified cycle, based on GT8C turbine, was marked on the graph. As was already shown in Fig. 4 maximum electrical efficiency η_{el} for the compression ratio $\pi = 16$ coincides with the point of extraction pressure $p_{ext} = 0.6$ MPa and the extraction mass flow rate $\dot{m}_{ext} = 80$ kg/s.

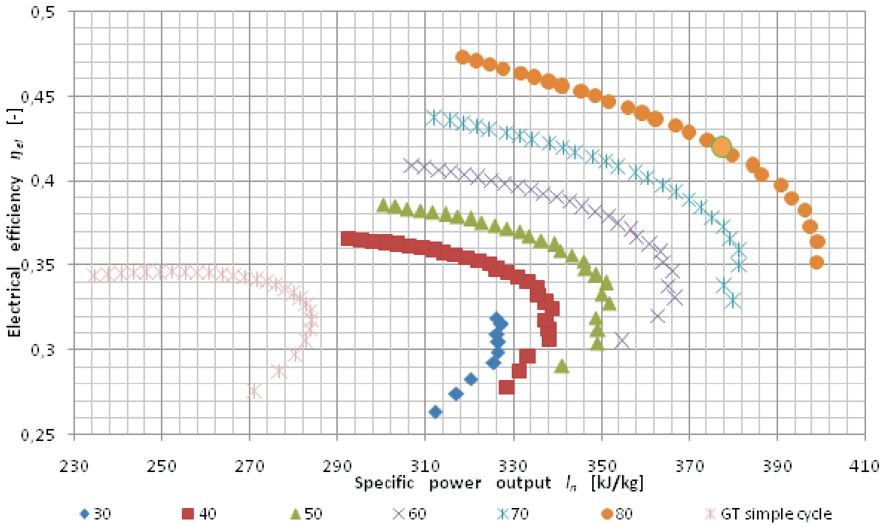

Figure 11: Electrical efficiency, η_{el}, as a function of specific work, l_n [kJ/kg].

6 Conclusions

The numerical and algebraical basic analysis of proposed modernization show that the efficiency augmentation by exhaust gases extraction from gas turbine's expander gives an increase of efficiency for both models. The main conclusions concerning the modification of GT8C gas turbine into the so-called gas turbine cycle with heat regeneration according to Szewalski's idea are listed below:

1. Electrical efficiency of classical GT8C approximates $\eta_{el} = 0.342$ and $\eta_{el} = 0.343$ appointed by basic model and CFM code, respectively. The exhaust gases extraction (gas turbine cycle with heat regeneration according to Szewalski's idea) increases the electrical efficiency to $\eta_{el} = 0.361$ for CFM code, and $\eta_{el} = 0.416$ basic model.
2. As the results of the calculations shows, the modification of the gas turbine cycle according to the idea of Szewalski could increase the cycle efficiency of up to 0.226 in comparison with the simple cycle (in the case of basic model calculations) or 0.0525 (in the case of calculation by CFM code).
3. The highest efficiency of the modified cycle is obtained for the highest value of the extraction mass flow rate from the turbine $\dot{m}_{ext} = 80$ kg/s, and $\dot{m}_{ext} = 82$ kg/s, basic model and CFM code, respectively, for which assumptions for temperature differences ΔT_{RE} on the regenerative heat exchangers are still met.

The differences in obtained values of efficiency could result from several uncertainties, that authors have faced during mathematical modeling of gas simple cycle, mainly relating to the internal parameters of the model such as the isentropic efficiency of basic and additional compressor, isentropic efficiency of the turbine, the exact value of the cooling mass flow and turbine components shaft or loss of pressure at the respective sites cycle, which because of the absence of experimental data, the authors adopted of the literature included in the bibliography.

Another explanation may be differences due to various models of gas as the working medium in both cases. Accuracy problem of modeling should be also investigated for another type of turbine, it is necessary to analyze it in next work.

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