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Enhancement of the Brayton cycle efficiency by water or steam utilization

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Abstract

In the paper, thermodynamic analysis of two modifications of the Brayton cycle into enhancement cycles with water or steam injection are discussed. The first one deals with a water injected gas turbine modified system with both interstage compressed air cooling and air heating (heat regeneration) before combustion chamber. The second one, mainly devoted to higher steam to air ratio, is connected with separate steam production in the heat recovery steam generator. Hence, steam utilization for the gas turbine propulsion in the Cheng cycle has been analysed. The analysis has been based on the computational flow mechanics (CFM) models of these advanced humidified systems, thanks to which, the influence of the main thermodynamic and design parameters have been examined.

Keywords: Brayton cycle; Regeneration; Cheng cycle; Thermodynamic analysis; Gas steam turbine; Computational flow mechanics; Steam injection, Water injection, Interstage cooling

Nomenclature

h	–	specific enthalpy, kJ/kg
k	–	relative steam flux coefficient

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l	–	specific work, kJ/kg
N	–	power, kW
$\dot{m}$	–	mass flow rate, kg/s
P	–	pressure, Pa
$\dot{Q}_{chem}$	–	chemical energy flux, kW
$\dot{Q}_{he}$	–	heat flux, kW
q	–	specific heat, kJ/kg
r	–	degree of regeneration
t	–	temperature, °C
T	–	temperature, K
W_d	–	fuel calorific value, kJ/kg
s	–	entropy, kJ/(kgK)

Greek symbols

β	–	related to the fuel mass flux
κ	–	isentropic coefficient
φ	–	relative air humidity after humidification
ξ	–	pressure loss coefficient
η	–	efficiency, %
π	–	compression/expansion ratio

Subscripts

a	–	air
ah	–	humid air
C	–	compressor
CC	–	combustion chamber
CF	–	fuel compressor
Cw	–	compressor with interstage water cooling
c	–	compression
el	–	electrical
ex	–	exhaust
f	–	fuel
g	–	generator
GT	–	gas turbine outlet, gas turbine
he	–	heat exchanger
i	–	internal
LP	–	low pressure
m	–	mechanical
P	–	pump
s	–	isentropic, steam
$s5b$	–	steam injected into combustion chamber
ST	–	steam turbine
TIT	–	turbine inlet temperature
w	–	water
$1s, 2s, ...$	–	isentropic points of cycle
$1, 2, ...$	–	real points of cycle
$HRSG$	–	heat recovery steam generator
HC	–	humidification chamber

1 Introduction

The thermodynamic cycle upon which all gas turbines operate is called the Brayton cycle (or the Joule cycle). In practice this cycle admits to obtain electric power with the net efficiency ranging from 28% for small turbines to 38% for large output turbines. Modern gas clean technologies require a higher efficiency and zero-emission work, therefore different modifications of the Brayton cycle are considered in the literature [5,16,18].

The low efficiency of the Brayton cycle is especially evidenced since a gas turbine does not perform well under part-load operation. For instance, at 50% load, the gas turbine achieves around 75% of the full-load efficiency, and at 30% load this drops to 50% of the nominal efficiency [20]. It is a great disadvantage compared to the Rankine cycle. Therefore, some additional arrangements, such as the controlled inlet guide vanes and multi-shaft designs, are employed to improve the part-load performance. Other modifications of the cycle include intercooling and recuperation [5]. More developed cycle with internal heat regeneration (recuperation) is the idea due to Szewalski . This concept of the Brayton cycle modification causes an increase of the cycle efficiency, and decreases the amount of fuel burned in the turbine. The internal heat regeneration is realized by extraction of exhaust gases [23,29]. Another manner of increasing the expansion work can be obtained by means of reheating. Moreover, it enables to provide the full-load efficiency within a broader load range by varying reheat fuel flow [5].

Yet additional technical manner of increasing of net efficiency of a gas turbine is steam injection into the gas turbine's combustion chamber, usually realized in the Cheng cycle or steam injected gas turbine (STIG) cycle [7,11,12,14,21,28]. Steam injection also improves output power of the gas turbine and then reduces harmful emissions into the atmosphere of such compounds as nitrogen oxides and carbon oxides [7,28]. Cheng cycle is the gas-steam cycle, in which gas Brayton cycle is combined with the steam cycle through the combustion chamber. In the gas turbine expansion of both exhaust gases and steam from heat recovery steam generator (HRSG) occurs, concluding in the Brayton cycle power increase [6,7,21]. Particular benefits are achieved with the nitrogen oxides emission reduction, that are generated in the gas turbines chiefly according to the thermal mechanism (the so-called Zeldowicz mechanism) [10]. Combustion temperature is decreased as a consequence of steam injection, and nitrogen oxides emission reduction occurs [15,21].

Additionally, using hydrogen as a future fuel whose exhaust gases in a gas turbine will be a mixture of exhaust gases and steam has great prospects. The turbine will be a hybrid combining the advantages of both gas and steam turbines for this work flux [4]. The idea of gas-steam combined turbine has been presented in [8,17]. The gas and steam cycles receive the high temperature flux from the gas turbine, reaching up to 1400 °C. However, steam temperature in steam cycle is about 600 °C [17,19]. The hybrid turbine is to receive the flux pressure from the steam turbine. The maximum pressure value is 30 MPa (for a gas turbine it is (4 MPa) [17] and the minimum are 10 kPa, so as to enable steam condensation and the most effective use of the flux energy [8]. On the other hand, the maximum pressure value in a gas turbine is 4 MPa and the minimum one – 100 kPa (which is the pressure of the surrounding) [19]. The gas-steam research results have been presented [1]. Gas-steam is the work flux thereof. The gas-steam generator into which the fuel, pure oxygen and water are injected, is a very important element of the cycle [1]. In the presence of oxygen the fuel burns at a much higher temperature than in traditional burning chamber, however, the temperature is decreased as the result of water evaporations [1,27]. So far, the GEJ79 (General Electric gas turbine class) has been modified, it works at 1.16 MPa and 760 °C [1]. In the second generation power plant cycle, with the use the SGT900 (Siemens Gas Turbine), the temperature before the first turbine stage is expected to increase up to 1080–1260 °C [1]. In the third generation gas-steam turbine, the operation parameters are expected to reach approximately 1650–1760 °C and 4 MPa [1].

This paper presents some results of a numerical analysis of two humidified systems: first one is with both interstage compressed air cooling by water injection and air heating (heat regeneration) before combustion chamber. The second one is with HRSG which produces steam. This working fluid is partially injected into the combustion chamber. The analysis was based on numerical models of computational flow mechanics (CFM) [2,3] of an advanced humidified system, thanks to which the influence of the main thermodynamic and design parameters could be examined. Calculations have indicated that the humidified system's efficiency and unit power can be considerably increased for these examples [11].

2 The modified Brayton cycle

In this paper some analysis and comparisons regarding two modified Brayton systems are considered. The first one, a modified cycle with interstage cooling of compressed air is shown in Fig. 1. and its thermodynamic cycle state parameters are depicted in Fig. 2. Ambient air (1) is sucked into the low-pressure part of the compressor (C_1) and compressed to pressure P_{21}. Compression is connected with the increase in temperature and final temperature T_{21} depends on pressure P_{21} and the compressor internal efficiency η_{ic}. From the compressor's low–pressure part the air is directed to a humidification chamber where water is injected. On the surface of water the droplets evaporation takes place resulting in heat removal from the air and a reduction in its temperature. Evaporation from the surface of water droplets is possible if a relative air humidity is below 100%.

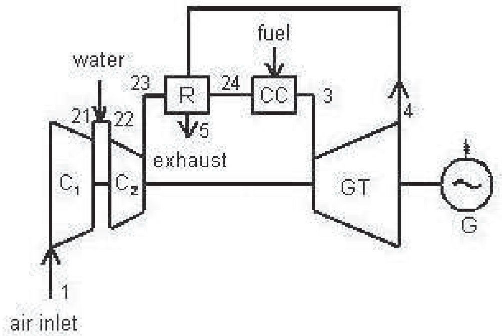

Figure 1. Diagram of a system with heat regeneration and interstage cooling by water injection: C_1, C_2 – low- and high-pressure compressor, R – regenerative exchanger, CC – combustion chamber, GT – gas turbine, G – electric generator.

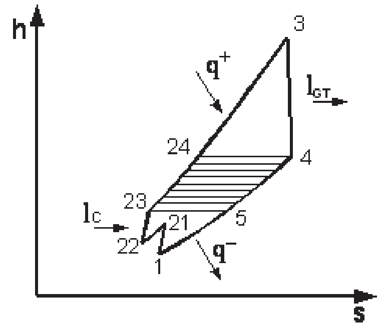

Figure 2. Thermodynamic cycle of gas system with heat regeneration and intercooling by water injection: q^+, q^- – positive and negative heat (heat added, and removed), l_C, l_{GT} – internal work of compressor and turbine.

Surface evaporation is not possible once saturation is reached and so water will then remain in the liquid state. From the humidification chamber at point (22) the air is directed to the compressor's high-pressure part (C_2) where it is compressed to pressure P_{23}. The compressed air passes through the regenerative exchanger R in which it is heated up to temperature T_{24}. Further heating and the increase in enthalpy takes place in the combustion chamber CC. Then the medium flows through the GT turbine where some of the total heat is converted into mechanical work which drives both the compressor C and the electric generator G.

The second modified Brayton cycle is based on the novelized Cheng cycle with a great amount of steam production at HRSG (Fig. 3). In the Cheng cycle, the steam created in the HRSG (5) is partially directed into the gas turbine's combustion chamber CC and partially into a steam turbine ST. Thus the steam flux directed into the extraction-counter-pressure steam turbine is reduced. This method increases the overall electric power and electrical efficiency of both the gas GT turbine and the entire gas-steam unit. Concurrently the rate of heat given to customers is reduced. Application of the Cheng cycle is particularly appropriate for the period with decreased heat consumption and in the period of the steam turbine refurbishment. Hence, the Cheng system provides flexible operation of the combined heat and power (CHP) station with the optimum utilization of the generated steam. Let's recall that, additional merit of the presented solution is the decrease in carbon dioxide and nitrous oxide emissions. Utilization of the steam for increase in the mass flux of the working medium in the Brayton cycle is, in conclusion, beneficial from the thermodynamic, economic and ecological points of view available for traditional CHP systems [30]. Temperature/entropy diagram of gas-steam power unit (CHP system) is presented in Fig. 4.

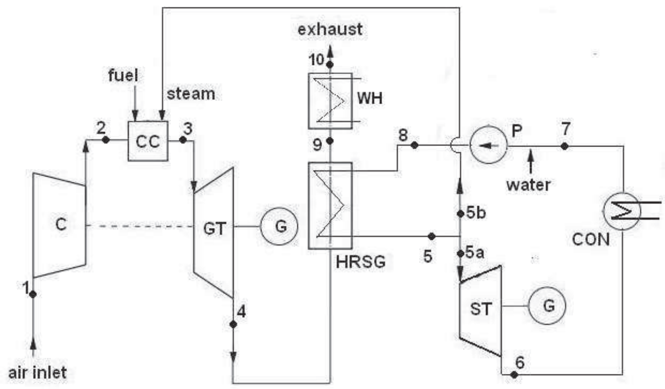

Figure 3. Diagram of the gas-steam power unit with the possible utilisation of steam in the Cheng cycle, (GT – gas turbine, C – compressor, CC – combustion chamber, G – electric generator, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON – condenser, WH – water heater).

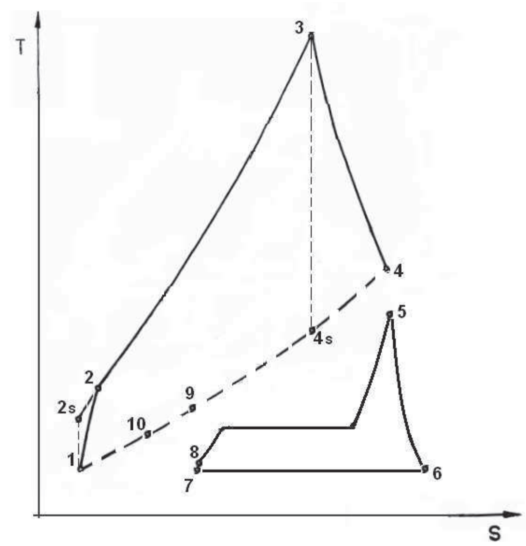

Figure 4. Temperature-entropy diagram of combined-cycle gas turbine/steam turbine.

3 Main governing equations

All computations of the mentioned cycles have been performed using the basic principles of gas systems and thermodynamic phenomena modelling [11,14,18], and algorithms for computing the properties of air, combustion gas and steam [9,12,14]. Analysis of the thermodynamic cycle was performed with the use of CFM numerical codes. Mathematical models in CFM employ mass, momentum and energy balance equations in the integrated form (also called 0D or engineering form) [2,3,24].

Input data required to compute the compressor power (N_C) are: internal and mechanical efficiency, η_{ic} and η_{mc}, respectively, compression π and the air mass flux $\dot{m}_a$ [kg/s]. When the air parameters at the compressor inlet temperature and pressure (T_1, P_1) are known, then pressure P_2 [Pa] after the compression process is computed as [12,24]

$$P_2 = \pi P_1 \ . \tag{1}$$

The medium undergoing the process is air with assumed value of the isentropic coefficient $\kappa = 1.4$. Perfect compression is described with the isentropic equation

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\kappa-1}{\kappa}} \ . \tag{2}$$

where $s = idem$.

Equation (2) allows for computing 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. The temperatures at the exits from the compressor's low- and high-pressure parts and the turbine are calculated using reversible adiabate equations and internal efficiency equations. Adiabate exponent, κ, and the medium's enthalpy, h, are a function of temperature and mixture composition. Therefore they are computed using special algorithms [12,24].

Compressor internal efficiency may be expressed by the formula [24]

$$\eta_{ic} = \frac{l_{t1-2s}}{l_{t1-2}} = \frac{h_1 - h_{2s}}{h_1 - h_2} = \frac{l_{Cs}}{l_C}, \tag{3}$$

where:

$l_{t1-2s}, (l_{Cs})$ – unit isentropic compression work,
$l_{t1-2}, (l_C)$ – real compression work,
h_1, h_2, h_{2s} – medium enthalpy determined at characteristic points.

To derive the compressor power and determine the medium enthalpy at the characteristic points, thermodynamic tables of properties are indispensible. Power required for driving of the compressor is determined with the formula [24,26]:

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

Furthermore, the interstage cooling compressor's power is [11]

$$N_{Cw} = \dot{m}_a \eta_{mc}(h_1 - h_{21}) + \dot{m}_{ah} \eta_{mc}(h_{22} - h_{23}) \ , \tag{5}$$

where:

$h_1, h_{21}, h_{22}, h_{23}$ – enthalpy of the medium at characteristic points of compression process,

$\dot{m}_a, \dot{m}_{ah}$ – medium mass flux through the low and high pressure part, respectively.

Computation of the combustion chamber requires the energy balance to be performed including all input and output energy fluxes. Heat losses in the combustion chamber (into the surrounding) are specified with the use of combustion chamber efficiency, η_{CC}, [24]. 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 $\dot{m}_f$ is the fuel mass flux, and W_d is the fuel calorific value. Heat energy balance for the combustion chamber may be expressed as [9,13]

$$\eta_{CC}(\dot{Q}_{chem} + \dot{m}_a h_2 + \dot{m}_f h_f + \dot{m}_{s5b} h_5) = \dot{m}_{ex} h_3 \ , \tag{7}$$

$$\text{or} \ \ \eta_{CC}(\dot{Q}_{chem} + \dot{m}_{ah} h_{24} + \dot{m}_f h_f) = \dot{m}_{ex} h_3 \ , \tag{8}$$

where:

$\dot{m}_{ex}$ – mass flow rate of exhaust gases flowing through the turbine,

$\dot{m}_{s5b}$ – mass flow rate of steam injected into the combustion chamber,

$\dot{m}_{ah}$ – mass flow rate of humid air heated the combustion chamber,

h_f – enthalpy of the fuel at the entry to the combustion chamber,

h_3 – enthalpy of the mixture at the combustion chamber's exit,

h_5 – heat recovery generated steam enthalpy,

h_{24} – enthalpy of the humid air at the entry to the combustion chamber.

Moreover, it requires mentioning, that the following mass balances need to be satisfied [3]:

$$\dot{m}_{ex} = \dot{m}_f + \dot{m}_a + \dot{m}_{s5b} \tag{9}$$

for the interstage cooling of the gas turbine

$$\dot{m}_{ex} = \dot{m}_f + \dot{m}_{ah} = \dot{m}_f + \dot{m}_a + \dot{m}_w \ , \tag{10}$$

and for the steam turbine

$$\dot{m}_s = \dot{m}_{s5a} + \dot{m}_{s5b} = \dot{m}_{s5a} + \dot{m}_w \ , \tag{11}$$

where:
$\dot{m}_s$ – heat recovery generated steam mass flux,
$\dot{m}_{s5a}$ – mass flow rate of steam expanding in the steam turbine,
$\dot{m}_w$ – mass flow rate of water admitted into the steam cycle Eq. (11) or mass flux of water injected to the humidification chamber Eq. (10).

For simplicity, fuel (gas) is assumed to undergo perfect combustion. General chemical reactions are shown below [24,26]

$$CH_4 + 2O_2 = CO_2 + 2H_2O \ , \tag{12}$$

$$C_2H_6 + 3,5O_2 = 2CO_2 + 3H_2O \ , \tag{13}$$

$$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O \ . \tag{14}$$

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

Expansion process in the gas turbine is characterised by means of a defined turbine internal η_{it} as follows [24,26]:

$$\eta_{it} = \frac{l_{t3-4}}{l_{t3-4s}} = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{l_{GT}}{l_{GTs}} \ , \tag{15}$$

where:
$l_{t3-4s}, (l_{GTs})$ – unit isentropic work of expansion,
$l_{t1-2}, (l_{GT})$ – real work of expansion,
h_3, h_4, h_{4s} – medium enthalpy at characteristic points.

The power generated in the turbine is expressed to be

$$N_{GT} = \dot{m}_{ex}\eta_{mt}(h_3 - h_4) \ , \tag{16}$$

where η_{mt} is the turbine mechanical efficiency. The power of the steam turbine is expressed to be [24]

$$N_{ST} = \dot{m}_{s5a}\eta_{mST}(h_5 - h_6),$$

where η_{mST} is the steam turbine mechanical efficiency, and h_5, h_6 are the medium (steam) enthalpy at characteristic points (5), (6).

The power of pump was calculated using the formula

$$N_P = \dot{m}_8\eta_{mP}(h_7 - h_8) \ ,$$

where $\dot{m} = \dot{m}_s = \dot{m}_5$ is the water mass flow rate on the pump outlet, η_{mP} is the pump mechanical efficiency, and h_7, h_8 are water enthalpies at characteristic points (7), (8). 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 steam turbine, N_{ST}, compressor, N_C, the water pump, N_P, the fuel compressor, N_{CF}, and the generator efficiency, η_g. Power of the gas-steam cycle, N_{el}, is the difference between the devices generating and consuming power according to the following expression [12,24,26]

$$N_{el} = \eta_g(N_{GT} + N_{ST} - N_C - N_P - N_{CF}) \ . \tag{17}$$

The electric power efficiency of the system is defined as a quotient of the electric power, N_{el}, generated in the unit, and the chemical energy flux of the fuel, $\dot{Q}_{chem}$, [12,24,26]

$$\eta_{el} = \frac{N_{el}}{\dot{Q}_{chem}} = \frac{N_{el}}{\dot{m}_f W_d} \tag{18}$$

and its internal unit power

$$l_i = l_{GT} - l_C \ . \tag{19}$$

The degree of regeneration used in calculations, is given [11]

$$r = \frac{T_{24} - T_{23}}{T_4 - T_{23}} \ , \tag{20}$$

where T_{23}, T_{24}, T_4 are the temperatures of the medium at the characteristic points of the system, respectively (Figs. 1 and 2).

Heat flux exchange in regenerative exchanger, with the assumed heat exchanger efficiency η_{he} is expressed to be:

$$\dot{Q}_{he} = \dot{m}_{ex}\eta_{he}(h_4 - h_5) = \dot{m}_{ah}(h_{24} - h_{23}) \ . \tag{21}$$

Additionally, heat flux exchange in heat recovery steam generator (HRSG) to produce steam may be expressed by the formula [24]:

$$\dot{Q}_{HRSG} = \dot{m}_{ex}\eta_{he}(h_4 - h_9) = \dot{m}_s(h_5 - h_8) \ , \tag{22}$$

where h_4, h_5, h_8, h_9 are the medium (working fluid) enthalpy determined at characteristic points (4), (5), (8), (9) (Figs. 3 and 4). Furthermore, the relative steam flux is defined as a quotient of the steam mass flow rate, $\dot{m}_{s5b}$, injected into the combustion chamber, and air mass flow rate $\dot{m}_a$

$$k = \frac{\dot{m}_{s5b}}{\dot{m}_a} \ . \tag{23}$$

In the literature, very often one may experience two definitions of the relative steam injection coefficient: first, related to the air mass flow rate, $k = \dot{m}_{s5b}/\dot{m}_a$ [9,13,22] and second, related to the fuel mass flow rate, $\beta = \dot{m}_s/\dot{m}_f$ [25,26,28,30].

4 Analysis of the first modification

In order to find the effect of water injection (the humidification cooling of the compressed air) calculations of the system (in Fig. 1) have been carried out. The position of the humidification chamber in the compressor was expressed by the low-pressure part's of the compression ratio π_{LP}. The following ranges of the variables were assumed for computations: the compressor's total compression $2.0 < \pi_c < 35.0$; temperature of the medium at the entry to the turbine $773.15 < T_3 < 1773.15$ K; relative air humidity after humidification $0.0 < \varphi_{22} < 1.0$; low–pressure part's compression $1.0 < \pi_{LP} < \pi_c$; degree of regeneration $0.0 < r < 1.0$. The following fixed parameter values were used: ambient pressure $P_1 = 0.1$ MPa; temperature of air at the entry to the system $T_1 = 283$ K; internal efficiency of the compressor $\eta_{ic} = 0.86$; internal efficiency of the turbine $\eta_{it} = 0.90$; mechanical efficiency of the turbine $\eta_{mt} = 0.99$, and mechanical efficiency of compressor $\eta_{mc} = 0.99$; electrical efficiency of the generator $\eta_g = 0.995$; pressure loss coefficient in the compressor, humidification chamber, combustion chamber, turbine and regenerative exchanger: $\xi_C = 0.007$, $\xi_{HC} = 0.02$, $\xi_{CC} = 0.03$, $\xi_{GT} = 0.03$, $\xi_{he} = 0.035$, respectively; heat loss in the combustion chamber $\xi_{CC} = 0.01$.

Some results of the computations for the total compression $\pi_c = 15.0$, temperature of the medium at the entry to the turbine $T_3 = 1473.15$ K and regeneration degree $r = 0.7$ are illustrated. The system's electrical efficiency η_{el} and internal unit work l_i versus compression ratio of the compressor's low-pressure part π_{LP} and relative humidity of the humidified air φ_{22} are shown in Figs. 5 and 6, respectively. The effect of humidification cooling of the compressed air is clearly visible but it is not identical for efficiency and internal unit power. The largest increase in efficiency – about 4.5% – occurs for the compression ratio $\pi_{LP} = 5$. The humidification of air at higher compression ratios π_{LP} results in smaller efficiency increments.

In contrast to the above, the system internal unit work curves have an ascending character for any value of π_{LP} due to an increase in specific heat of the medium. Increments in both efficiency and unit work are the largest after the air has been humidified to a relative humidity $\varphi_{22} = 100\%$. As the figures show, the lower the relative humidity, the weaker the effect of humidification. Considering that the rate of evaporation of water from the surface of droplets decreases as relative humidity increases, in order to keep the humidification chamber's size small, humidification is terminated once suitably lower values of relative humidity φ_{22} are obtained. To estimate an accurate level of relative air humudity after humidification, φ_{22}, it is necessary to use a 3D model of humidification chamber in CFD (computational fluid dynamics) framework [4,32,33].

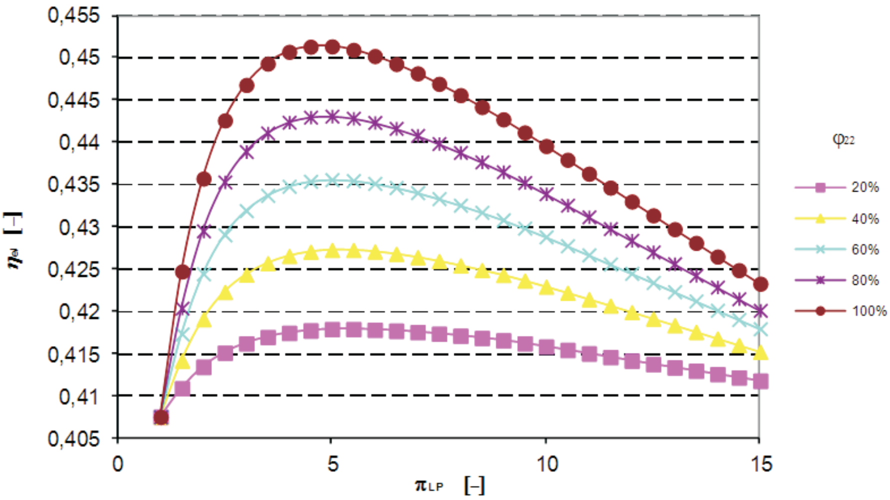

Figure 5. System electrical efficiency η_{el} versus compressor low-pressure part compression ratio π_{LP} and relative air humidity after humidification φ_{22}.

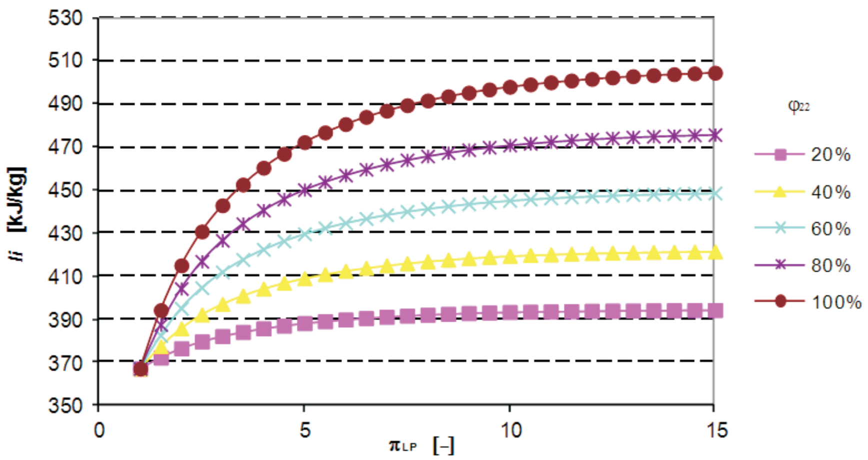

Figure 6. System internal unit work l_i versus compressor low-pressure part compression ratio π_{LP} and relative air humidity after humidification φ_{22}.

5 Analysis of the second modification

In the numerical analysis of the second modified cycle (Fig. 3) it has been assumed that steam produced by HRSG has the following thermodynamic parameters: steam pressure $P_s = 4$ MPa, steam temperature $T_s = 723.15$ K, maximum $P_s = P_5$, $T_s = T_5$ steam mass flow rate $\dot{m}_{s5b} = 23.2$ kg/s ($k = 1.55$ which is defined by). Another thermodynamical data are taken from [26,28], where the combined cycle based on GT8C turbine has been modeled [31]. Generally, the aim of the discussion is to compare the influence of steam injection mass ratio, k, on the electrical efficiency and power of gas turbine. Analysis of the modification shows

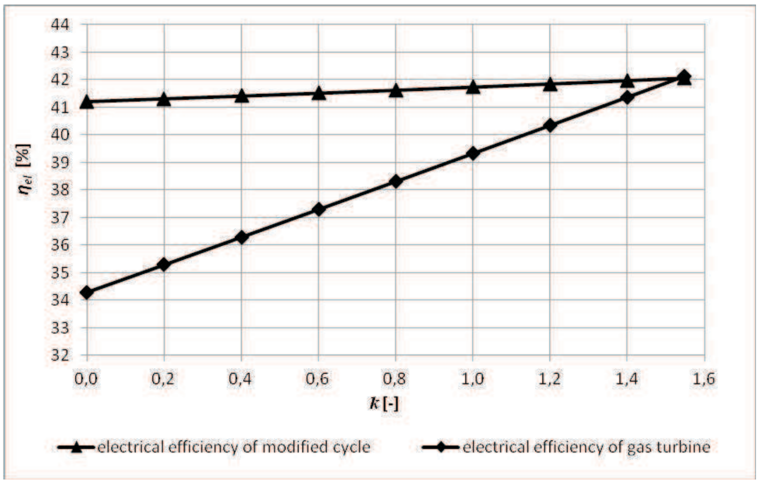

Figure 7. The electrical efficiency η_{el} of the second modified cycle versus k coefficient [30].

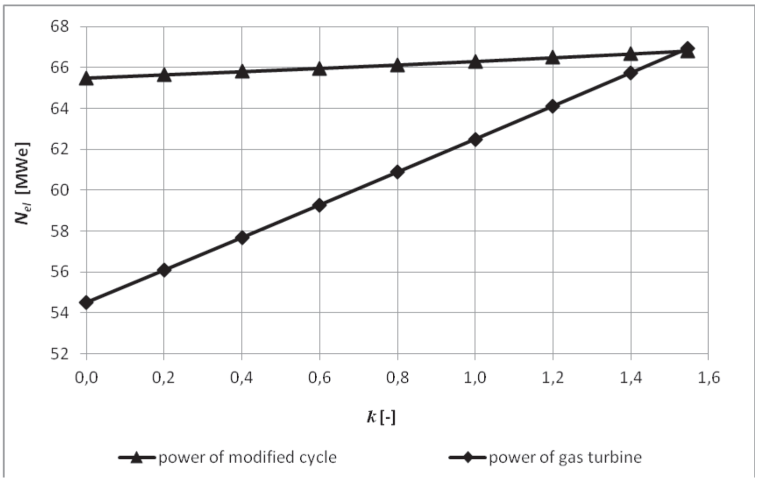

Figure 8. The power N_{el} of the second modified cycle versus k coefficient [30].

that despite the decrease of the steam turbine power, N_{ST}, aggregative electrical power of the combined system increases to the value $N_{el} = 66.9$ (Fig. 8). Chart in Fig. 7 shows that the electrical efficiency of the gas-steam system in the classical layout is $\eta_{el} = 41.21\%$, while employing the Cheng cycle increases to the value

42.05%. It requires further notice that for coefficient $k = 1.55$ ($\eta_{el} = 42.05\%$) and steam turbine power, N_{ST}, is equal zero, since the entire steam mass flow rate is injected into the gas turbine combustion chamber. Gas turbine electric power increases to the value $N_{el} = 66.9$ MWe. It shows that the steam injection is the favourable solution, which should be used during the breakdown of steam turbines or during lower heat demand in the summer period, when the turbine would be oriented either on the electricity generation or heat production [28,30].

6 Conclusion

Obtained results show that the intercooling of compressed air by water injection is a promising way to improve the properties of gas systems with regenerative heating. Water injection helps to obtain larger efficiency, η_{el}, increments and greater internal unit work l_i. The effect of cooling by water injection depends on many factors. One of the key factors is the compression ratio. The higher it is, the greater is the power consumed by the compressor and much can be gained by reducing the power of the compressor and increasing the power of the turbine. Through CFM (computational flow mechanics) numerical analysis, one can determine the optimum compressor low-pressure frequencies π_{LP} for a given system application. Another of the key factors is relative humidity of humidified air φ_{22}. In order to find the optimal φ some additional calculations of the humidification chamber by means of CFD software (computational fluid dynamics) are needed.

From the second modification under consideration it follows that by introducing steam injection in a gas turbine, the Brayton cycle efficiency, η_{el}, may be increased from 41.21 to 42.05%. Owing to this electrical power, N_{el}, and efficiency, η_{el}, increase both for the gas turbine and the entire gas-steam unit. Additional advantage of the modification is the decrease of the volatile emission such as carbon dioxide and nitrogen oxides. Enhanced mass flux in the Brayton cycle by means of redirected steam may be considered as a beneficial procedure from the thermodynamic, economic and ecological standpoint in the traditional cogenerative systems. The power of classical combined power plant has been estimated at $N_{el} = 65.49$ MWe. Modification of thermal cycle into the Cheng cycle ($k = 1.55$) increases the power to $N_{el} = 66.92$ MWe. The power gain of gas turbine and electrical efficiency of gas turbine by steam injection is about 23%.

Using a steam turbine to expand the steam, i.e., applying a conventional combined cycle instead of a Cheng cycle, gives higher efficiency gains. Accepted efficiency for the combined cycle is nowadays 50–58%, with a power rise of about 30–50% with respect to the simple cycle.

A practical concern with steam injection or water injection deals with the water consumption. For second modification it has been assumed that there is at HRSG some condensing system, therefore there is no total loss of water ($\dot{m}_{s5b} = 23.2$ kg/s, $k = 1.55$). It means that steam consumption is up to 1.6 kg of high purity water per kWh of electrical output. The necessary of water purification system for a large-scale plant would represent about 5% of the total capital expenditure and running costs would add about 5% to the fuel cost.

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