Power enhancement of the Brayton cycle by steam utilization

KRZYSZTOF JESIONEK[1]
ANDRZEJ CHRZCZONOWSKI[1]
PAWEŁ ZIÓŁKOWSKI[2*]
JANUSZ BADUR[2]

[1] Wrocław University of Technology, Faculty of Mechanical and Power Engineering, Wybrzeże Wyspiańskiego 27, 50-370 Wrocław, Poland

[2] The Szewalski Institute of Fluid-Flow Machinery of Polish Academy of Sciences, Fiszera 14, 80-231 Gdansk, Poland

Abstract   The paper presents thermodynamic analysis of the gas-steam unit of the 65 MWe combined heat and power station. Numerical analyses of the station was performed for the nominal operation conditions determining the Brayton and combined cycle. Furthermore, steam utilization for the gas turbine propulsion in the Cheng cycle was analysed. In the considered modernization, steam generated in the heat recovery steam generator unit is directed into the gas turbine combustion chamber, resulting in the Brayton cycle power increase. Computational flow mechanics codes were used in the analysis of the thermodynamic and operational parameters of the unit.

Keywords: Brayton cycle; Gas-steam unit; Cheng cycle; Numerical analysis; CHP

Nomenclature

h	–	specific enthalpy, kJ/kg
k	–	relative steam flux coefficient
l_t	–	specific work, kJ/kg
N	–	power, kW

*Corresponding author. E-mail address: pawel.ziolkowski@imp.gda.pl

$\dot{m}$	–	mass flux, kg/s
P	–	pressure, Pa
$\dot{Q}_C$	–	heat flux, kWt
$\dot{Q}_{chem}$	–	chemical energy flux, kW
t	–	temperature, °C
T	–	temperature, K
W_d	–	fuel calorific value, kJ/kg

Greek symbols

η	–	efficiency, %
Π	–	compression

Subscripts

a	–	air
el	–	electrical
EC	–	CHP station
ex	–	exhaust
f	–	fuel
GT	–	at the gas turbine outlet
i	–	internal
m	–	mechanical
s	–	steam
TIT	–	turbine inlet temperature (average temperature in combustion chamber)
$1s, 2s, \ldots$	–	isentropic points of cycle
$1, 2, \ldots$	–	real points of cycle

1   Introduction

Contemporarily almost every developed country, including Poland, faces the problem of insufficient electric power generation [3]. An additional difficulty arises from the need for environment protection regulations and norms defined in the European Union Directive 2010/75/UE [19]. Construction and initialization of any new unit that satisfies the new restrictions takes several years. The fastest constructed gas-steam units, for example, take about 57 months to finish [22]. Much quicker, ecologic and providing electric power increase solution, is a modernisation of the existing combined heat and power (CHP) units into the Cheng cycle [18]. Using the Cheng cycle or the steam injection gas turbine (STIG) in a cogeneration system assures power increase of the Brayton cycle for the price of the heat power. Such a solution is particularly profitable in the period of decreased demand for heat or during the refurbishment of the steam turbine in the system. The Cheng cycle ensures flexible operation of the CHP unit with optimum

utilisation of the generated steam [18,21].

Cheng cycle is the gas-steam cycle, in which the 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 the heat recovery steam generator (HRSG) occurs, concluding in the Brayton cycle power increase [4,5,24]. The characteristics of the Cheng system causes that high electric efficiencies are achievable also for small systems [8,24]. Furthermore, exhaust gases leaving the combustion chamber contain little toxic components. Owing to this costly exhaust gas purification systems are not required [24]. 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) [7]. Combustion temperature is decreased as a consequence of steam injection, and nitrogen oxides emission reduction occurs [9,13].

The first modernization proposal, presented by Saad and Cheng, referred to the General Electric LM2500 gas turbine [13]. These days, steam injection into a combustion chamber is employed in turbines of the leading companies, such as: General Electric, Rolls-Royce, Kawasaki [9,12]. In the work [16] it is shown that the STIG modification of a simple cycle gas turbine, the General Electric Frame 6B, can boost its efficiency from about 30 % to 40%, while the power output may increase from 38 to 50 MWe. More sophisticated systems are known, e.g., employing saturation [10], humidified air (HAT – humidified air turbine, EvGT – evaporating gas turbine, CHAT – cascade humidified air turbine) [9], systems combining steam injection gas with regeneration (RSTIG) [10], systems combining steam and water injection DRIASI cycle (dual-recuperated intercooled-aftercooled steam injected) [12] and LOTHECO cycle (low temperature heat combined) employing external renewable heat source to heat up the water condensed from the exhaust gases [12].

The aim of this work is to compare thermodynamic and operational parameters of the combined heat and power (CHP) plant supplying power and heat to the city of Gorzow, Poland, (PGE EC Gorzów), before and after its modification. The comparison is based on the computational flow mechanics (CFM) codes [1,2], namely: the COM-GAS [15] and the Aspen Plus [25]. The analysis was based on data provided in literature [11,17,20] regarding the Gorzów Wielkopolski gas-steam unit.

2   Mathematical model in the CFM code

As it already has been noticed, analysis of the thermodynamic cycle was performed with the use of CFM codes. Mathematical models in CFM (included in COM-GAS and Aspen Plus codes) employ mass, momentum and energy balance equations in the integrated form (also called 0D or engineering form) [2,15,23]. Computational procedures of the components of the turbo is presented, i.e., compressor, combustion chamber and turbine. Formulations for power, efficiency and steam injection coefficient are also defined. Schematic layout of modeled combined gas-steam unit is shown in Fig. 1. Temperature/entropy diagram of combined-cycle gas turbine/steam turbine is presented in Fig. 2.
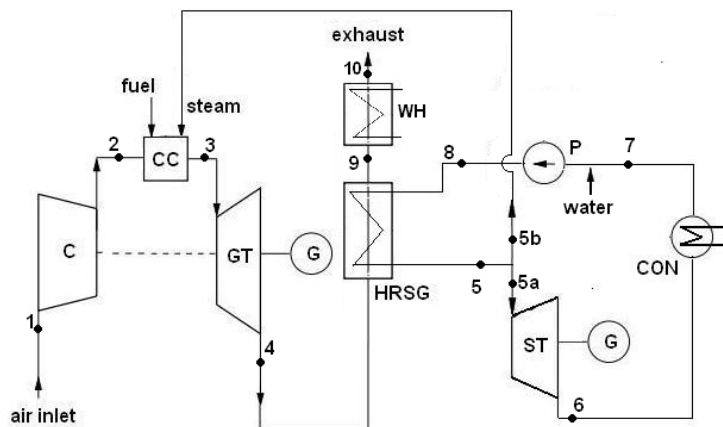

Figure 1. Scheme of the gas-steam power unit with the possible utilisation of steam in the Cheng cycle: GT – gas turbine, C – compressor, CC – combustion chamber, G – electric generator, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON –condenser, WH – water heater.

Input data required to compute the compressor power are: internal and mechanical efficiency, η_{is} and η_{ms}, respectively, compression Π and the air mass flux $\dot{m}_a$. When the air parameters (temperature and pressure) at the compressor inlet (T_1, P_1) are known, pressure P_2 after the compression process is computed [18,23]:

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

The medium undergoing the process is air with an assumed isentropic coefficient κ=1,4, so perfect compression is described with the isentropic equa-
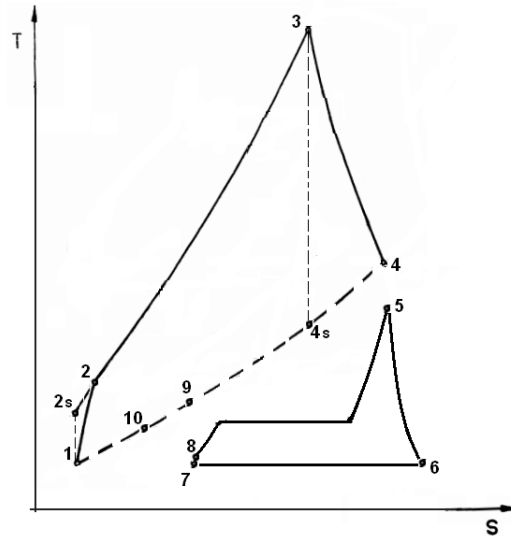
Figure 2. Temperature-entropy diagram of combined-cycle gas turbine/steam turbine.

tion, $s = idem$:

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

Equation (2) allows for computing of the theoretical temperature at the end of compression, T_{2s}. At the assumed internal efficiency of the compressor it is possible to determine the real temperature at the end of compression, T_2 (temperature of air at the combustion chamber inlet). Compressor efficiency may be expressed by the formula [18,23]

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

where: l_{t1-2s} – unit isentropic compression work, l_{t1-2} – real compression work, h_1, h_2, h_{2s} – medium enthalpy determined at characteristic points (1), (1), (2s) (Fig. 2).

To derive the compressor power and determine the medium enthalpy at the characteristic points, thermodynamic tables of properties are indispensible. Power required for compressor propulsion is determined with the following formula [18,23]:

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

where: $\dot{m}_a$ – air mass flux, η_{ms} – compressor mechanical efficiency.

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 the chamber efficiency, η_{CC}. Chemical energy flux of the fuel is defined to be [6,24]:

$$\dot{Q}_{chem} = \dot{m}_f W_d , \tag{5}$$

where: $\dot{m}_f$ – fuel mass flux ; a W_d – fuel calorific value.

Heat energy balance for the combustion chamber may be expressed as [6,24]:

$$\eta_{CC}\dot{Q}_{chem} + \dot{m}_a h_2 + \dot{m}_f h_f + \dot{m}_{s5b} h_5 = \dot{m}_{ex} h_3 , \tag{6}$$

where: $\dot{m}_{ex}$ – exhaust gas mass flux, $\dot{m}_{s5b}$ – mass flux of steam injected into the combustion chamber, h_f – fuel enthalpy , h_3 – exhaust gas enthalpy and h_5 – heat recovery generated steam enthalpy. Moreover, it requires mentioning, that the following mass balances need to be satisfied:

$$\dot{m}_{ex} = \dot{m}_f + \dot{m}_a + \dot{m}_{s5b} , \tag{7}$$

$$\dot{m}_s = \dot{m}_{s5a} + \dot{m}_{s5b} = \dot{m}_{s5a} + \dot{m}_w , \tag{8}$$

where: $\dot{m}_s$ – heat recovery generated steam mass flux, $\dot{m}_{s5a}$ – mass flux of steam expanding in the steam turbine, $\dot{m}_w$ – mass flux of water admitted into the steam cycle.

For simplicity, fuel (gas) is assumed to undergo perfect combustion. 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 defined turbine internal efficiency as follows [18,23]:

$$\eta_{it} = \frac{l_{t3-4}}{l_{t3-4s}} = \frac{h_3 - h_4}{h_3 - h_4 s} , \tag{9}$$

where: l_{t3-4s} – unit isentropic work of expansion, l_{t1-2} – real work of expansion, h_3, h_4, h_{4s} – medium enthalpy at characteristic points (3), (4), (4s) (Fig. 2). Power of the gas turbine is expressed to be:

$$N_{GT} = \dot{m}_{ex}\eta_{mt}(h_3 - h_4) , \tag{10}$$

where: η_{mt} – turbine mechanical efficiency.

Similarly as in the case of the compressor and the combustion chamber,

CFM numerical codes employ thermodynamic properties tables, improving the computation time. More detail on the mathematical models available in the CFM codes may be found in the work [23].

Electric power of the gas-steam turbine is defined upon the mechanical power of the individual components of the thermodynamic cycle, i.e., gas turbine N_{GT}, compressor N_C, water pump N_P, and generator efficiency η_g. Power of the gas-steam cycle is the difference between the devices generating and consuming power according to the following expression [18,23]:

$$N_{el} = \eta_g(N_{GT} + N_{ST} - N_C - N_P).$$ (11)

Electric efficiency of the system 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 [18,23]:

$$\eta_{el} = \frac{N_{el}}{\dot{Q}_{chem}} = \frac{N_{el}}{\dot{m}_f W_d} \ .$$ (12)

CHP station efficiency is defined as a quotient of a sum of the electric power, N_{el}, and heat flux, $\dot{Q}_C$, of the unit and the chemical energy flux of the fuel:

$$\eta_{EC} = \frac{N_{el} + \dot{Q}_C}{\dot{Q}_{chem}} = \frac{N_{el} + \dot{Q}_C}{\dot{m}_f W_d} \ ,$$ (13)

where heat flux for the consumer, with the assumed heat exchanger efficiency, η_{he}, is expressed to be

$$\dot{Q}_C = \dot{m}_{s5a}\eta_{he}(h_6 - h_7) + \dot{m}_{ex}\eta_{he}(h_9 - h_{10}) \ ,$$ (14)

where h_6, h_7, h_9, h_{10} are the medium enthalpy at characteristic points (6), (7), (8), (9) (Fig. 1 and Fig. 2). Furthermore, the relative steam flux, k, is defined as a quotient of the steam mass flux, $\dot{m}_{s5b}$, injected into the combustion chamber, and air mass flux:

$$k = \dot{m}_{s5b}/\dot{m}_a \ .$$ (15)

In the literature, very often one may experience two definitions of the relative steam injection coefficient: first, related to the air mass flux, $k = \dot{m}_{s5b}/\dot{m}_a$ (according to [6,14,24]) and the second, related to the fuel mass flux, $\beta = \dot{m}_s/\dot{m}_f$ (according to [16,18,21]).

3   Results

Schematic layout of the gas-steam unit presented in Fig. 1, showing the possibility to utilise steam in the Cheng cycle. Portion of the mass flux (specifically, $\dot{m}_{s5b}$), generated in the HRSG, is directed into the combustion chamber (CC). Thereby, steam flux $\dot{m}_{s5a}$, directed onto the steam turbine (extraction-backpressure type), is reduced. In conclusion, steam turbine power decreases, along with the heat flux decrease for the consumer $\dot{Q}_C$.

Table 1 gathers simulation results acquired in two separate CFM codes, i.e., COM-GAS and Aspen Plus. Simulation results are compared with measured nominal operation parameters of the Gorzów CHP station available in literature. The comparison confirmed a good agreement between measurement and numerical simulation, thereby proving correctness of the methodology adopted. Further detail on the methodology in modeling the gas-steam unit may be found in works [18,21].

Analysis of the modification shows, that despite the decrease of the steam turbine electric power aggregative power of the combined system increases to the value $N_{el} = 66.9$ MWe (Fig. 3). Analysing the chart in Fig. 4 it can be seen that the electric efficiency of the gas-steam system in the classical layout is $\eta_{el} = 41.21\%$ while employing the Cheng cycle the efficiency increases to the value $\eta_{el} = 42.05\%$. It requires further notice that for relative steam flux coefficient $k = 1.55$ ($\eta_{el} = 42.05\%$) steam turbine electric power, N_{el}, is equal to zero, since the entire steam mass flux, $\dot{m}_s$ is injected into the gas turbine combustion chamber. Gas turbine electric power increase of the gas to the value $N_{el}^* = 66.9$ MWe occurs also for the price of heat flux, $\dot{Q}_C$, available for the consumers. Heat power flux decreases from the initial value $\dot{Q}_C = 72.9$ MWt to an astonishing 20.6 MWt, which extracted solely from the water heat exchanger (WH) for relative steam flux coefficient $k = 1.55$ (Fig. 3). Also CHP station efficiency decrease occurs from the initial value $\eta_{EC} = 87.1\%$ down to 55.0, for the relative steam flux coefficient $k = 1.55$ (Fig. 4).

As shown in Fig. 5, the increase of steam injection, characterized by relative steam flux coefficient k, results in decrease of both the temperature of combustion, t_{TIT}, and the level of emission of nitrogen oxides, NO_x. In the analysed case, the carbon dioxide and nitrogen oxides emissions decrease approximately by $\Delta CO_2 = 10$ kg/MWh and $\Delta NO_x = 0,008$ kg/MWh, respectively. The trend of changes in emission of nitrogen oxides, NO_x, versus the combustion temperature has been kept linear, instead of exponential character reported in literature [7]. This is due to a constant value

Table 1. Results of the calculations for Gorzów combined power plant in Aspen Plus and
COM-GAS code [11,17,20], the superscript * refers to the gas turbine.

Parameter	Unit	Data [11,17,20]	COM-GAS	Aspen Plus
$t_a = t_1$	°C	15	15	15
$P_a = P_1$	Pa	1.013×10^5	1.013×10^5	1.013×10^5
Π	–	16	16	16
t_f	°C	15	15	15
P_f	Pa	40.5×10^5	40.5×10^5	40.5×10^5
$\dot{m}_f$	kg/s	–	8.51	8.36
$\dot{m}_{ex}$	kg/s	182.3	182.3	182.3
$t_{GT} = t_4$	°C	520	520	520
$t_{TIT} = t_3$	°C	1100	1100	1100
η_{el}^*	%	34.60	33.64	34.29
N_{el}^*	MWe	54.49	54.49	54.49
t_{10}	°C	100	100	100
$\dot{m}_s$	kg/s	23.2	23.2	23.2
$P_s = P_5$	Pa	40×10^5	40×10^5	40×10^5
$t_s = t_5$	°C	450	450	450
$P_6 = P_7$	Pa	4×10^5	4×10^5	4×10^5
N_{el}	MWe	65.49	65.49	65.49
η_{el}	%	–	40.43	41.21
CO_2	kg/MWh	313.01	505.48	492.65
NO_x	kg/MWh	0.324	–	0.324

of the conversion rate of nitrogen, as assumed in the calculations. This
assumption is acceptable since the real emissions should be much lower in
a range of analysed combustion temperatures [18]. To estimate an accurate
emission level of nitrogen oxides, it is necessary to use a full 3D model of
combustion chamber and burners in CFD framework [1].

4　Conclusion

Numerical analysis has shown, that with steam utilization, the Brayton
cycle power may be increased from $N_{el} = 54.5$ MW to 66.9 MW. Owing
to this electrical power and efficiency increase both for the gas turbine and
the entire gas-steam unit. An additional advantage of the modification
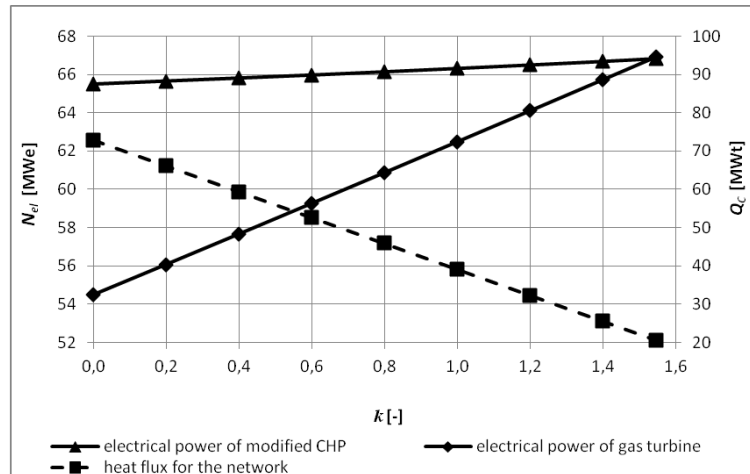is a decrease of the volatile emission such as carbon dioxide and nitrogen

Figure 3. Electrical power, N_{el}, and heat flux for the network, $\dot{Q}_C$, of the gas-steam unit working in the Cheng cycle versus relative steam flux coefficient k.
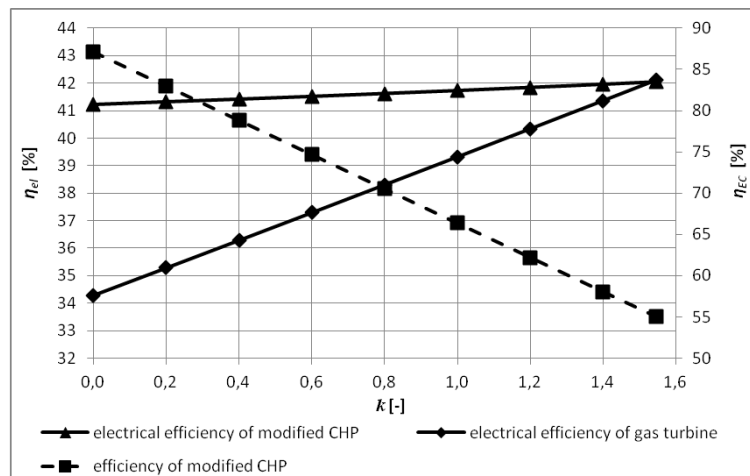
Figure 4. Electrical efficiency, η_{el}, and entire CHP unit efficiency, η_{EC}, versus relative steam flux coefficient k.

oxides by $\Delta CO_2 = 10$ kg/MWh and $\Delta NO_x = 0.008$ kg/MWh, respectively. Despite the CHP heat power and efficiency decrease, 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 cogeneration systems.
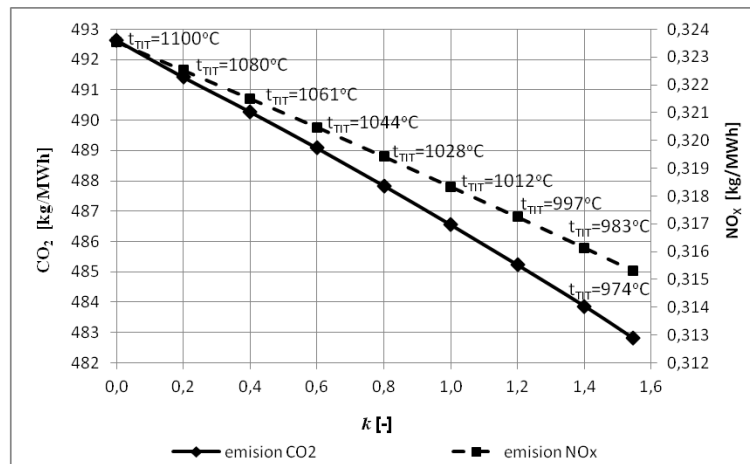
Figure 5. Emission of carbon dioxide, CO_2, nitrogen oxide, NO_x, and combustion temperature, t_{TIT}, versus relative steam flux coefficient k.

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