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# Clean gas technologies — towards zero-emission repowering of Pomerania

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## Abstract

In the paper a brief review of processes for energy conversion developed recently within the framework of clean gas technologies is presented, including projects for future repowering of Pomerania. The main aim of this work is to show the different thermodynamic cycles which improve efficiency and/or decrease emissions such as: Cheng cycle, Szewalski cycle, LOTHECO cycle, hybrid pSOFC/GT cycle, inverse Brayton cycle, low-emission cycle and zero-emission cycle. For that purpose, cycles and results of analysis are first studied and, secondly, new concept is presented. The role of coupled analysis 0D and 3D in cycle's devices is also discussed.

**Keywords:** Clean gas technologies; Clean coal technologies; Zero-emission; Thermodynamic cycle; Steam-gas turbine; Oxy combustion; Carbon capture storage

## Nomenclature

| | | |
|---|---|---|
| $eCO_2$ | – | emission level of carbon dioxide, kg/MWh |
| $eNO_x$ | – | emission level of nitrogen oxides, kg/MWh |
| $k$ | – | steam-to-air ratio |
| $\dot{m}$ | – | mass flow rate, mass flux, kg/s |
| $N$ | – | power, kW |
| $p$ | – | pressure, Pa |
| $T$ | – | temperature, K |

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**Greek symbols**

$\beta$ – steam injection mass ratio
$\Delta T_{pp}$ – temperature difference in pinch point in heat recovery steam generator, K
$\Delta T_{RE}$ – temperature difference in regenerative exchanger, $\Delta T_{RE} = T_9 - T_{7'}$, K
$\Pi$ – compression/expansion ratio
$\eta$ – efficiency, %

**Subscripts**

1,2 ,... – points of cycle
$a$ – air
$amb$ – ambient
$C$ – compressor
$CC$ – combustion chamber
$Comb$ – steam and gas combined cycle
$CON$ – condensation
$el$ – electrical
$exh$ – exhaust gas
$ext$ – extraction of exhaust gas
$f$ – fuel
$g$ – generator
$GT$ – gas turbine outlet, gas turbine
$he$ – heat exchanger
$high$ – highest parameter in cycle
$i$ – internal
$low$ – lowest parameter in cycle
$m$ – mechanical
$max$ – maximum
$o$ – oxygen
$p$ – pump
$pp$ – pinch point
$RE$ – regenerative exchanger
$s$ – steam
$ST$ – steam turbine
$T$ – turbine
$TIT$ – turbine inlet temperature

**Superscripts**

$amb$ – ambient
$in$ – point in inversed Brayton cycle

# 1 Repowering by clean technologies — motivation of study

Facing the current clean coal technologies, the coal-fired power plants using carbon capture equipment become an expensive source of electricity in Pomerania. Other power sources, such as wind, nuclear, and geothermal are in a starting level, and cannot be taken to be a serious candidate for repowering of Pomeranian energy market. One of counter-candidate to the clean coal power plants are, at the moment, the plants based on natural and shale gas, i.e., the combined cycle with $CO_2$ capture technology that currently appear to be more economical [34]. Additionally combined gas-steam cycles are the most efficient power plant. The new units with Mitsubishi gas turbine (M501J) and a steam turbine have efficiency of 61.5% (LHV basis) [45]. On the other hand supercritical and ultracritical steam power plant with only a steam turbine achieve about 50 and 55% efficiency, respectively [58]. Let recall that the net capacity of Polish power stations based on steam turbines, dated on 31th December, 2008 have approached approx. 35.5 GW. On the contrary, electric power produced from natural gas approaches only 840 MW – this situation is due to financial barriers — electric power from steam is seven times cheaper than from gas. Therefore, looking for novel, highly efficient, clean technologies of electricity production it should be remembered that in Poland nearly whole electricity is coming from professional steam turbines. Nowadays, it is a common knowledge that steam turbines are one of the most versatile and oldest prime mover technologies. Let us notice, that steam turbines have been in use in Poland for about 105 years [4,18], when, especially after II Word War, they replaced reciprocating steam engines due to their higher efficiencies and lower costs. In years from 1960 to 1985 the Polish market of energy production has been dominated by 200 or 360 MW turbosets [28]. After 1991, repowering of the power plants turns into supercritical steam turbines working in turbosets of 460 and 860 MW, based on the heat energy obtained by utilization of lignite and hard coal in the high efficiency boilers. In the light of great achievement and experience in maintenance of the conventional steam turbines, thinking about repowering of Polish energy market must assume a some kind of steam turbine serving as a base for a new zero-emission turbo set design.

Due to recent possibility, appearing after the discovery of Polish natural and unconventional gas sources, a new opportunity of cleaner then coal heat energy source will be obtainable in a few years. In this light a new chance for modern repowering of Polish energy system, as we assume, the steam turbines technologies should be widely used, first of all, for combined heat and power plants [14,33]. The first candidates to repowering are the plants based on the gas-steam combined cycles. However, in a combined cycle the role a steam turbine changes radically. Now, unlike a gas turbine, in a steam turbine heat is only a byproduct of power, therefore in cogeneration, steam turbines generate reduced the amount of electricity as a byproduct of heat contained in flue gases. Recall, that usually about 10 kg/s of flue gases flux is needed for generating of 1 kg/s of fresh steam [29,31]. In conventional combined cycle, a steam turbine is captive only to a separate heat source and does not directly convert fuel to electric energy. The energy is transferred from flue gasses to the turbine through a multipressure steam generator –

a heat recovery steam generator (HRSG) [5], and steam produced in such way, in turn, powers the turbine and generator. This separation of functions enables steam turbines to operate independently from the gas turbine. Therefore some advantages of steam turbines, as for instance unusually high drop of pressure from 22 MPa to 500 Pa cannot be fully explored in the whole conversion process.

Now, the basic question is — does clean gas technologies shall be economically viable? The leading factor is the gas price, due to the Gasprom monopole. The hope on cheaper gas comes from the recent searches of the shale gas in the Pomerania region. The ability to economically produce natural gas from unconventional shale gas reservoirs in Pomerania has been made possible recently through the application of horizontal drilling technology and so-called 'hydraulic fracturing' (e.g., fracking, fracturing). Concerns like Chevron, Exxon Mobil, ConocoPhillips, CheSapeake XTO, and Marathon Oil, have already developed such drilling technique which has revolutionized gas production in the United States [39]. However, yet another serious obstacle exists – that is the commonly raised fact, that shale gas generates more greenhouse gas emissions than does the coal. Note that combustion of shale gas generates less $CO_2$ than combustion of coal, therefore it is a main reason for use of shale gas in place of coal. Nevertheless, it should be taken into account that shale gas production emits more methane (4–8% of shale gas rate) than does the conventional gas production, and the overall equivalent $CO_2$ emission could therefore be more than that of coal. Methane emissions consist of emissions during the well completion, routine venting and equipment leaks at well sites, emissions during liquid unloading, emissions during gas processing, emissions during transport, storage and distribution [22,40].

The unconventional Polish gas can be supplied also from coal gasification units [5]. Let recall, that hard coal undergoes complete conversion into syngas in a gasifier reactor with steam and oxygen-enriched air from air separation unit (ASU). The syngas consist primarily of CO (50–60% volume) and $H_2$ (27–30%) [29].

Combined gas-steam plants based on coal syngas are called integrated coal gasification cycle (IGCC) plant. But combustion of syngas with oxygen leads to the necessity of great $CO_2$ recirculation, and since there are commercially unavailable gas turbines for a carbon dioxide-reach working fluid, there are some practical needs for changing of mass ratio in flue gasses by employing the water gas shift. It means that the CO-rich syngas should be converted to $CO_2$ and $H_2$ in a heterogeneous catalytic reactor with steam [23].

In the paper we are interested in exploring further novelty of combined cycles which turns the combined heat and power plants only to power plants dedicated to clean electricity production. It can be done by removing of the HRSG from the role of heat exchanger between the flue gases and working water, and to replace the combustion chamber on a internal direct steam producer. Such combined gas and steam turbines can sustain their main performance specifications such as the high pressure (22 MPa) and high temperature (1100 °C) – it is quite a new situation – up to now there is no steam turbine with such high temperature of working medium, and vice versa, there is no gas turbines with such extremely high pressure. Since this concept can be simply and naturally connected with concept of oxy combustion there is a thermodynamical base for a zero-emission gas-steam turboset.

Especially, in Section 8 the particular analysis of gas-steam turbine within a low-emission cycle will be presented. In Section 9 the high-efficient gas-steam turbine with a zero-emission cycle will be presented. In these cases, the working fluid contains only the mixture of $CO_2$ and $H_2O$ what leads to direct separation of $CO_2$.

## 2 The classical gas-steam cycle

Power plants use several methods to convert gas to electricity. One method is to burn the gas in a boiler to produce steam, which is then used by a steam turbine to generate electricity. A more common approach is to burn gas in a combustion turbine to generate electricity. Yet another technology, that is growing in popularity is to burn the natural gas in a combustion turbine and use the hot combustion turbine exhaust to produce steam to drive the steam turbine. This technology is called 'combined cycle, combined heat and power (CHP)' and achieves a higher efficiency by using the same fuel source nearly twice.

In Poland there is the trend in power plants design towards a combined cycle, which incorporates an old steam turbine in a bottoming cycle with a gas turbine [21]. Steam generated in the heat recovery steam generator of the gas turbine is used to drive a steam turbine to yield additional electricity and improve the cycle efficiency. Combined cycle applications use an extraction-condensing type of steam turbine. Usually we need 10 kg/s of flue gases to produce 1 kg/s of steam. Steam turbines offer a wide array of designs and complexity to match the desired application and/or performance specifications [28], and for utility service may have several pressure casings and elaborate design features, all designed to maximize the efficiency of the power plant. For industrial applications, steam turbines are generally of simpler single casing design and less complicated for reliability and cost reasons. The CHP technologies can be adapted to both utility and industrial steam turbine designs [19].

### 2.1 Classification — unmixed versus mixed cycles

Within gas-steam cycles two working fluids are separated or mixed; from this point of view the combined units can be classified to be [21,26]:

a) Standard combined cycles — where flue gases and steam are secluded from each other all along the cycle. Due to strict separation between gas and steam the standard combined cycles are also called to be ‘unmixed cycles’.

b) Steam-injected cycles — where steam produced in HRSG is expanded directly into the gas turbine together with combustion gases.

c) Water-injected, regenerative cycles — where heat recovery from gas turbine exhaust is accomplished by heating a mixture of air and water to be fed a combustor chamber. Both last kinds can be grouped under the name ‘mixed cycle’.

## 2.2 Efficiency of gas-steam combined cycles

The electrical generating efficiency of standard combined turbine power plants varies from a high and of 55% HHV (higher heating value) for large, electric utility plants designed for the highest practical annual capacity factor, to under 30% HHV for small, simple plants which make electricity as a byproduct of delivering steam to processes or district heating systems. The example of the last case is the Gorzów Wielkopolski CHP [33,36].

Let recall that in classical gas and steam cycles combined together by the HRSG the net efficiency depends on both cycles in the following manner [21]:

$$\eta_{Comb} = \eta_{ST} + \eta_{GT} - \eta_{ST}\eta_{GT} \ , \tag{1}$$

where $\eta_{ST}$, $\eta_{GT}$ denote the efficiency of steam and gas turbine, respectively.

As usual, steam turbine thermodynamic efficiency (isentropic or politropic efficiency) refers to the ratio of power actually generated from the turbine to what would be generated by a perfect (ideal) turbine with no internal losses using steam at the same inlet conditions and discharging to the same downstream pressure. The actual (real) enthalpy drop is divided by the isentropic enthalpy drop the kind such obtained efficiency is called the isentropic one. If the isentropic enthalpy drop is replaced by the politropic (also ideal and perfect) drop of enthalpy such efficiency is called as the politropic one. This politropic efficiency, from the definition, is a little higher than isentropic efficiency and as it is shown by Nastałek *et al.* [19], it is more comfortable for marketing and computational fluid dynamics (CFD) purposes.

The turbine thermodynamic efficiency is not to be confused with electrical generating efficiency, which is the ratio of net power generated to the total fuel input to the cycle. Steam turbine thermodynamic efficiency measures how efficiently the turbine extracts power from the steam itself. Nowadays, multistage steam turbines have isentropic efficiencies that vary from 65% for small units to over 90% for large industrial and utility sized units [14,18]. Small, single stage steam turbines can have efficiencies as low as 50% [4,6]. When a steam turbine exhausts are used to a CHP application, the turbine efficiency is not as critical as in a power only condensing mode [29].

In practice, there is a simple manner to find an influence of a single apparatus to the net efficiency of combined cycle if one uses the computational fluid mechanics (CFM) tools. The exemplar of such analysis is to be found in Topolski and Badur [31] (Fig .1).

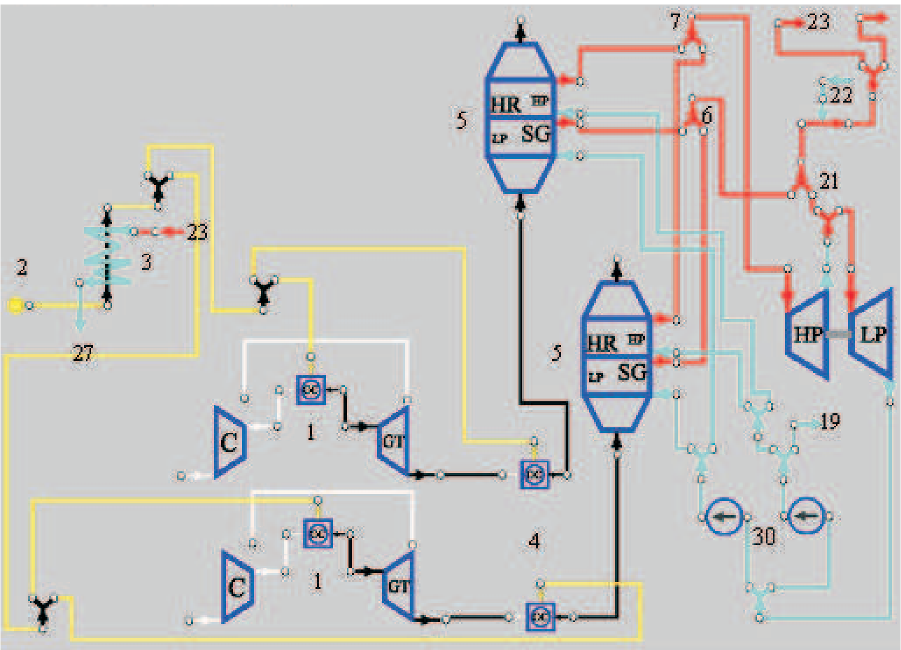

Figure 1. Nowa Sarzyna Combined Power Plant – modeled via COM-GAS [31], where: CC – combustion chamber, C – compressor, GT – gas turbine, HRSG – dual pressure heat recovery steam generator with HP – high pressure and LP – low pressure part, HP – high pressure steam turbine, LP – low pressure steam turbine.

To analyse the thermodynamic cycles the CFM codes are commonly used. Mathematical models in CFM (included in COM-GAS[1], DIAGAR, Gate Cycle,

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[1] Program COM-GAS makes it possible to calculate any thermodynamic cycle on the socalled design level. It has been developed at the Thermo-Chemical Power Department IFFM PASci in Gdańsk by Topolski and Badur [29-31,43,44], and has been developed by Karcz, Lemański, Wiśniewski, Kaczmarczyk and Ziółkowski [15,17,38,43,44,48,50]. All numerical procedures are written in the Fortran Language and visualization has been made in Delphi Language. The procedure was tested out in the works [2,3,15,16,29-31,35-38,43,44,48,50,51]. Computational procedures for each component COM-GAS belongs to zero dimensional models (0D), because it contains an algebralical integral formulation of typical balances: mass, momentum and energy

Alpro, and Aspen Plus codes) employ mass, momentum and energy balance equations in the integrated form (also called 0D or engineering form) [15,35,41–44,46–49,57].

## 3 Repowering by the Cheng cycle

Yet additional technical manner of increases of net efficiency of a gas turbine is steam injection which leads finally to a gas-steam turbine. This technology is known to be steam-injected gas turbine (STIG) [11,13]. Originally, by injecting steam or water into combustion chamber for decreasing NOx emissions and some increase of mass flow and, therefore, increase power output has been observed [8]. Generally, the amount of water is limited to the amount required to meet the $NO_x$ requirement in order to minimize operating cost and the impact on inspection intervals. When steam is injected for power augmentation, it can be introduced into compressor discharge casing of the gas turbine as well as into the combustion chamber [24]. The injection increases not only the mass flow but and the specific heat of the working fluid, what means that the higher enthalpy drop can be converted in the same blade stages with cooling the blades more effectively than air. Depending on the manner of water or steam injection different technologies have been developed [7,9]. These include the Cheng cycle [8,9], the dual-recuperated intercolled after cooled steam-injected cycle (DRIASI cycle), the humidified air cycle (HAT cycle), the low temperature heat combined cycle (LOTHECO cycle presented in section 5), etc. The advantages and disadvantages of these cycles are still an open scientific question [13,22,24,25].

The STIG system has promising energy properties. Its main advantages are: a widely variable association factor, high electrical efficiency at no demand for thermal power, and low emissions of nitric oxides into the atmosphere. The basic STIG system has been quite well investigated, but no data on an extended version of the system are available [7,].

The Cheng cycles [8,9] are applied in more complicated cycle like DRIASI and LOTHECO cycle or turbocharged steam injected gas turbine [22,49]. Wang and Chiou [32] have showed that the STIG modification of a simple cycle gas turbine

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and equilibrium models of reactions. Additionally, COM-GAS employs mathematical tables for the fluid properties [15,43,44,48]. At present, the in-house code COM-GAS is developed in the Department of Energy Conversion IFFM PASci at 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 examples are presented in works [17,19,52,57].

General Electric Frame 6B [59] can boost its efficiency from about 30 to 40% and the power output from 38 to 50 MWe. In particular, Carapellucci and Milazzo [7] have analyzed the existing gas-steam power plant repowered by an additional gas turbine and heat recovery steam generator. These new equipments have been integrated within the original thermal cycle for producing steam injected next into the gas turbine combustion chamber. In this way, the existing gas turbine can be converted into a steam injected gas turbine. As a result, the power of such repowered power plant has increased by 50%, simultaneously decreasing the overall efficiency by 3–6% [7]. Srinivas *et al.* [25] have reviewed STIG cycle with a dual pressure HRSG. They have investigated the effect of steam injection mass ratio (ratio of mass flow rate of steam, $\dot{m}_s$, to mass flow rate of fuel, $\dot{m}_f$; $\beta = \dot{m}_s/\dot{m}_f$) and concluded that for complete combustion in the gas turbine combustion chamber the maximum limit for the steam injection mass ratio is identified as $\beta = 6.0$. However, to get a stable combustion, steam is only injected up to the ratio of $\beta = 3.0$. Additionally, they noted that with the steam injection increase, the gas and steam cycle efficiency changes accordingly [25]. These results have independently been obtained by Ziółkowski [35].

The Cheng cycle is, in conclusion, beneficial from the thermodynamic, economic and ecological points of view available for traditional CHP systems. In Fig. 2 it is shown the main scheme of the Cheng cycle operating in a CHP system like PGE EC Gorzów Combined Heat and Power Plant [33].

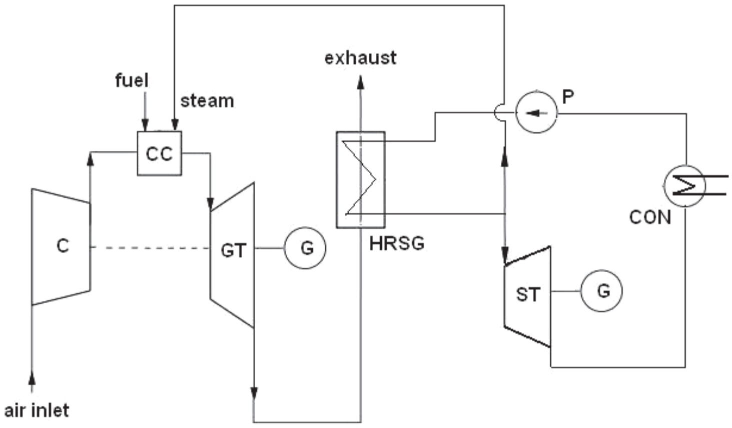

Figure 2. Scheme of Cheng system: GT – gas turbine, C – compressor, CC – combustion chamber, G – electric generator, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON – condenser [35,36].

In the paper by Ziółkowski *et al.* [36] the practical repowering of PGE EC Gorzów, by means of the Cheng cycle, has been considered. It should be emphasized that this method of modification has already been successfully implemented in a gas-steam power plant in ECK Generating Kladno, Czech Republic. Both ECK Generating Kladno and PGE EC Gorzów CHP plant operate on a 54.5 MWe-class gas turbine (GT8C) manufactured by ALSTOM Power.

The basic problem considered by Ziółkowski *et al.* [36] was to determine the optimal value of the steam injection mass ratio, $\beta$, that should be calculated for the constants cycle parameters. It has been assumed that steam produced by HRSG has the following thermodynamic parameters: steam pressure $p_s = 4$ MPa, steam temperature $T_s = 623.15$ K, and maximum steam mass flow rate $\dot{m}_{s,max} = 23.2$ kg/s ($\beta_{max} = 2.77$).

Generally, the discussion involving the influence of steam injection mass ratio $\beta$ on the electrical efficiency and power of gas turbine was conducted under the assumption that the fuel mass flow rate, $\dot{m}_f = 8.36$ kg/s, remains constant. Thus, mathematically speaking, the problem was in determination of optimal steam injection mass ratio $\beta$ within the range of 0–2.77 due to the fact that the heat recovery steam generator could only produce steam at maximum steam mass flow rate of about 23.2 kg/s. In the original Cheng cycle, steam injection mass ratio $\beta$ is usually close to the value of 3.0 to prevent unstable combustion conditions in the combustion chamber[2] [26].

From CFM numerical simulations it follows that $\beta$ influences both the drop of the exhaust gas temperature and an increase of gas turbine power. The results of calculations of the thermal cycle with various $\beta$ values are shown in Fig. 3. The analysis shows that, despite of the steam turbine (ST) power decrease, the overall power of combined power plant increases. Analyzing the results (Fig. 3), it can be seen that electrical efficiency of combined power plant averages $\eta_{el} = 41.21\%$ (without steam injection, $\beta = 0$), while employing the repowering of this cycle to Cheng cycle increases in both power and efficiency. The electrical power and efficiency of the repowered cycle averages $\eta_{el} = 42.05\%$ and $N_{el} = 66.92$ MWe, respectively, for the maximum steam injection [35]. In this extremely case, it means that the power generated by steam turbines is zero, because the entire steam mass flow rate, $\dot{m}_s$, is injected into the gas turbine (GT). This result is fully comparable with the performance of the real Cheng cycle in Kladno, in Czech Republic ($N_{el} = 66.90$ MWe) [33]. Note that this significant increase of both power output and electrical efficiency is due to new duty of the gas turbine GT8C, whose power output has increased by approximately $\Delta N_{el} = 12.40$ MWe. This implies that the power output of gas turbine is approximately of 23% higher than the initial power output of the gas turbine without the steam injection –

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[2] To estimate stable combustion conditions it is necessary to use a full 3D model of combustion chamber and burners in CFD framework. Examples of such calculations are presented in works [52,53].

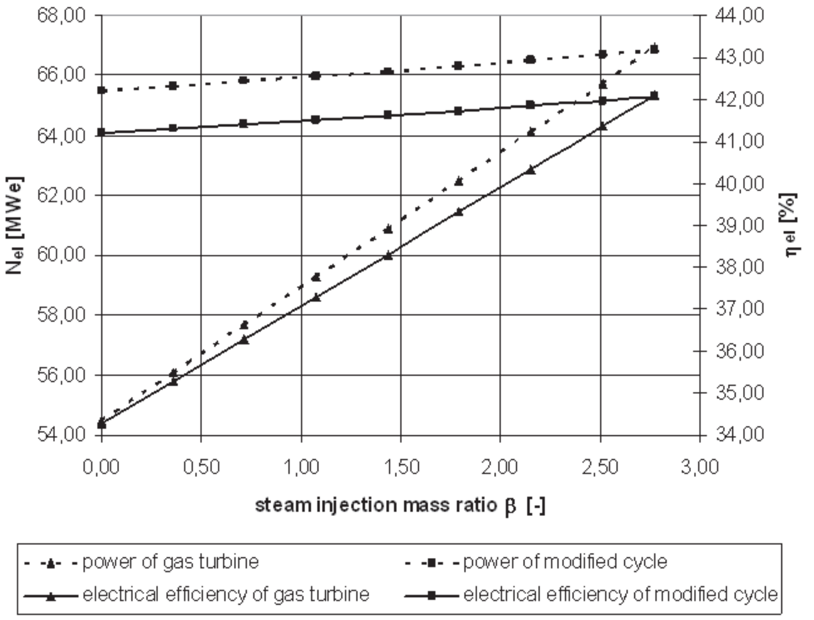

Figure 3. Electrical power, $N_{el}$, and electrical efficiency, $\eta_{el}$, of the repowered cycle versus steam injection mass ratio ($\beta$) [35,36].

these data agrees with recent literature [12].

Let's recall that the maximum load of the heat recovery steam generator, or the maximum mass flow rate of steam, $\dot{m}_{s,max} = 23.2$ kg/s, has been achieved for the assumption of the minimum temperature difference between produced steam and exhaust gases. According to literature the value of pinch point should be at least $\Delta T_{pp} = 8$ K [24,31]. The minimal pinch point in the HRSG has been assumed to $\Delta T_{pp} = 8$ K. This value is acceptable, at this level, in case of the heat recovery steam generators used in Polish combined power plants. For instance, Zielona Góra Combined Power Plant operates under the similar thermal conditions ($\Delta T_{pp} = 8.9$ K) [35].

Finally, note that apart from thermodynamic benefits, the steam injection also leads to decrease nitrogen and carbon oxides of harmful emissions [1,32,36].

## 4 Repowering by the Szewalski cycle

Standing on the base of general concept of cycle carnotization, Szewalski [27] has proposed a cycle with the internal heat regeneration that is realized by extraction of exhaust gas, $\dot{m}_7 = \dot{m}_{ext}$, with pressure $p_7$ and temperature $T_7 = T_{ext}$ (Fig. 4, point 7, GT). This extraction from the gas turbine is used for air heating before the combustion chamber (CC) in regenerative exchangers RE2 and RE1, and next, after cooling, is compressed in C' and added to air in the point 9 where pressure and temperature of both exhaust gases and air are similar. A gas turbine has a little different construction since in the part denoted by numbers 3–7 in Fig. 4 the mass flow rate is greater as usually, however, in the part 7–4 is lower than normal.

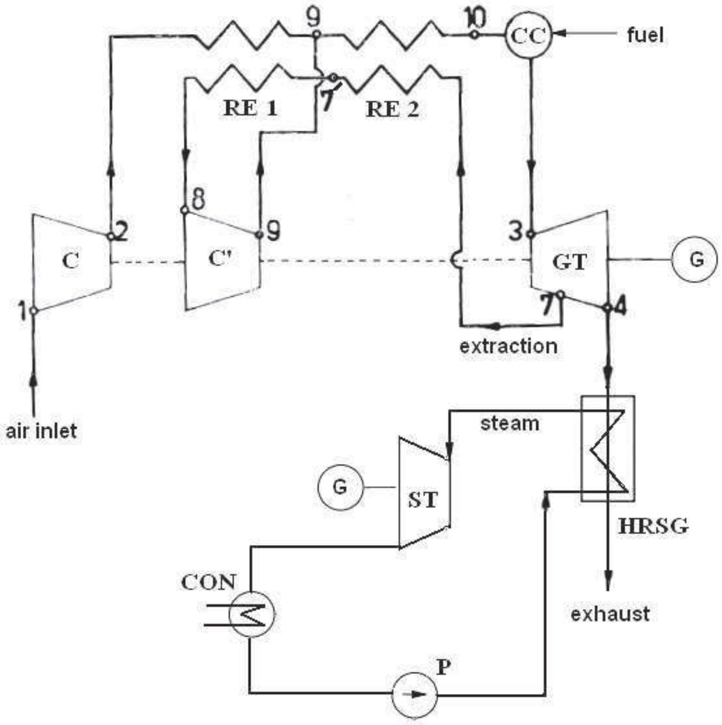

Figure 4. Scheme of the Szewalski cycle with the GT8C gas turbine [27,37]: CC – combustion chamber, C – compressor, C' – additional compressor, GT – gas turbine, RE – regenerative exchanger, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON – condenser.

Szewalski idea of the Brayton cycle modification causes an increase of the cycle efficiency, whisls it decreases the amount of fuel burned in the turbine. Ziółkowski *et al.* [37] have verified the Szewalski idea making parameterization of the extraction: mass flow $\dot{m}_{ext}$ and pressure $p_{ext}$. We will describe here the main results of thermodynamic analysis on the example of GT8C. The profitability of the Szewalski cycle will be clarified by comparison of PGE EC Gorzów CHP plant, before and after modernization into the Szewalski cycle.

The increase of the mass flux flowing through the extraction in the turbine causes simultaneous decrease of the exhaust flux flowing through the last gas turbine stages and then through the heat recovery steam generator (HRSG). *Ipso facto*, the power of the last gas turbine stages decrease and so does the mass flow rate of steam generated in the HRSG. In both turbines, gas and steam ones, the produced electrical power is lower. We keep the same temperature drop, $\Delta T_{pp}$, between working fluids in the HRSG – the modifications deal only with mass fluxes. We have found that together with the fall of the value of extraction pressure, $p_{ext}$, the possible range of $\dot{m}_{ext}$ becames smaller. Finally, in the case of small $\dot{m}_{ext}$ there is no possibility of sufficient heating of air and there is a serious dependence of temperature in the mixing point no. 9 (Fig. 4). This case we call that the node does not fulfill the Szewalski idea (Fig. 5). In turn, with greater value of $\dot{m}_{ext}$ the temperature difference, $\Delta T_{RE} = T_9 - T_{7'}$, at the node no. 9 becomes lower what leads to termination of heat transport process. For instance, for $p_{ext} = 0.7345$ MPa and $\dot{m}_{ext} = 100$ kg/s there is practically no heat exchange in RE1 and RE2 (Fig. 4), since the heating fluid possesses less temperature than the cooled fluid. Then, for large mass flow of flue gases there was found a limit value $\dot{m}_{ext} = 82$ kg/s, whereas $\Delta T_{RE} = 30$ K is taken as a constant pinch point at node no. 9.

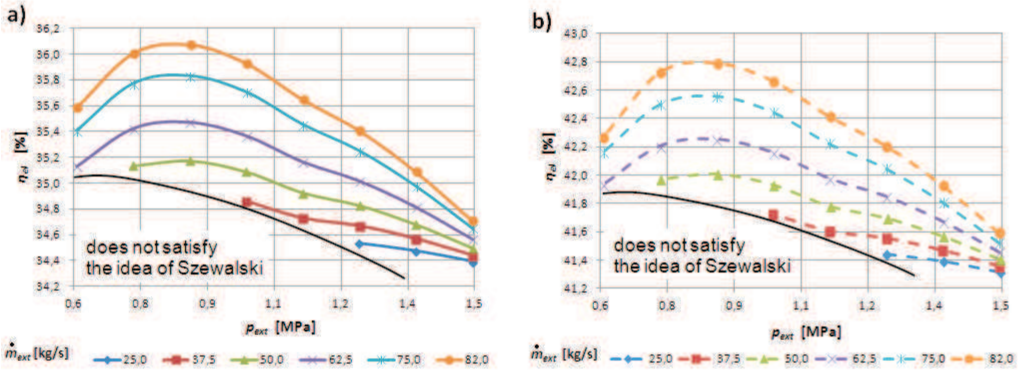

Figure 5. Efficiency of GT8C modification and efficiency of PGE EC Gorzów CHP Plant modification in function of extraction pressure, $p_{ext}$, and the extraction mass flow rate, $\dot{m}_{ext}$: a) only for GT8C, b) for whole system.

Figure 5 presents the change of the turbine GT8C efficiency as well as the entire power plant PGE EC Gorzów CHP plant after modernization into the Szewalski cycle. These changes depend on extraction pressure and the extraction mass flow rate. Maximum electrical efficiency, both for the gas turbine and the combined gas-steam power plant has been found for the following parameters: $\dot{m}_{ext} = 82$ kg/s and range of pressure $p_{ext}$ =0.75–0.9 MPa.

Concluding this research, one can say that conversion of a Brayton into a Szewalski cycle causes the increase of the cycle efficiency, decreases the amount of fuel burned in the turbine, and, unfortunately, lowers the net power of the cycle.

## 5 Repowering by the LOTHECO cycle

The LOw Temperature HEat COmbined cycle (LOTHECO cycle) employs external renewable heat source to heat up and evaporate water condensed from the exhaust gases (Fig. 6) [11,22]. Since the water-in-air evaporation takes place at the evaporator (EV), the saturation temperature is below 443.15 K. Temperature range of evaporation is from 373.15 up to 443.15 K, depending on the amount of injected water and the compressor pressure ratio, Π. For this temperature range, low-quality heat sources are favourably integrated, which, under other circumstances, cannot be utilized for electric power generation, such as: geothermal, solar, etc. Such arrangements are beneficial in terms of enhanced fuel-to-electricity efficiency compared to the efficiency of an equivalent conventional combined cycle [22]. Efficiencies above 60% have been recently reported [11,22].

Some alternatives of the cycle diagram are possible [11]. In first modification of the LOTHECO cycle, the excess air is not discharged but expanded in the air expander (EA) (Fig. 7). The excess air is branched off upstream of the evaporator, subsequently heated in the recuperator (REC) by the gas turbine (GT) exhaust gas and then expanded in the air expander. The exhaust air is fed into the heat recovery steam generator (HRSG) at a thermodynamically favorable point of the cycle.

The second cycle modification differs from the first one in that the excess air is enriched with steam in the evaporator [11]. However, the third one is the most complex one (Fig. 8) of the systems. It requires raising of the relative amounts of injected water, which is limited in the first and in the second modification by the maximum allowable steam-to-air ratio $k$ of the combustion air[3]. Since the composition of the working medium in the expander is irrelevant, its steam-to-air ratio can assume the value attainable in the evaporator. By modifying the air paths it becomes possible to maintain the maximum allowable steam-to-air ratio of the combustor air and simultaneously adapt the steam-to-air ratio of the wet expander air (WEA) to the value that can be realized in the evaporator [11].

## 6 Repowering by a SOFC

Within the context of mounting pressures on zero emission technologies fuel cell systems will play a major role. Since the fuel cell, like a solid oxide fuel cell (SOFC) is operating without the Carnot limit of efficiency, the systems build on the fuel cells can approach more effective, then conventional, energy conversion

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[3] In the literature, very often one may experience two definitions of the relative steam injection ratio: first, related to the air mass flow rate, $k = \dot{m}_s/\dot{m}_a$ (according to [11,12,25,49,54]) and second, related to the fuel mass flow rate, $\beta = \dot{m}_s/\dot{m}_f$ (according to [32,35,36]).

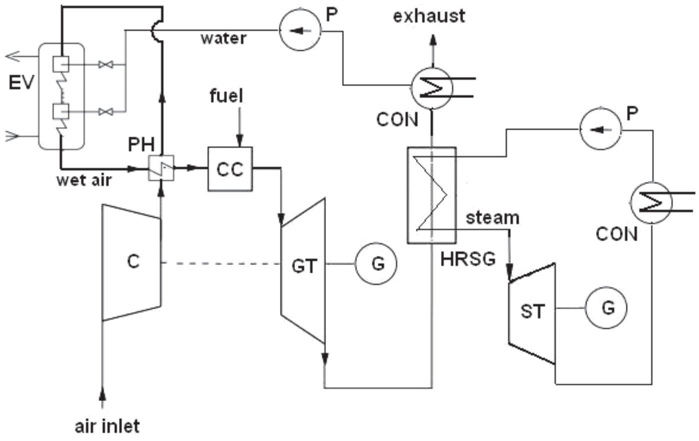

Figure 6. The LOTHECO cycle [11]: CC – combustion chamber, C – compressor, G – electric generator, GT – gas turbine, PH – preheater, EV – evaporator, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON – condenser.

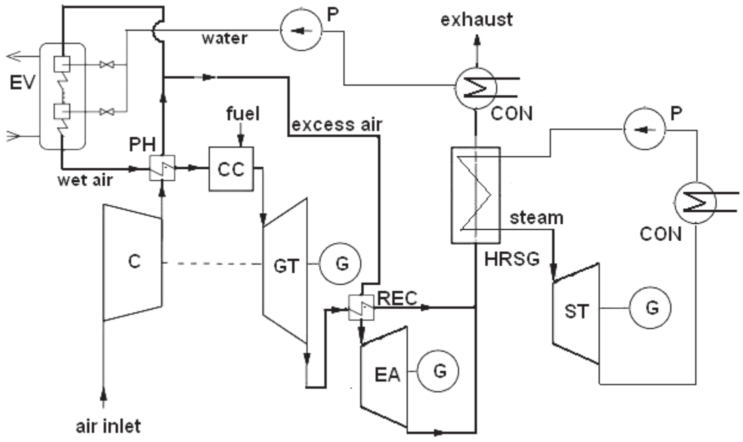

Figure 7. The LOTHECO basic combination of standard gas turbine and air expander [11]: CC – combustion chamber, C – compressor, G – electric generator, GT – gas turbine, PH – preheater, EV – evaporator, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON – condenser, EA – air expander, REC – recuperator.

processes [3,30]. Well known system combining SOFC and gas turbine working with the capacity of 300 kW, has been developed by Siemens Westinghouse (Fig. 9) [55,56]. Yet another way, leading to advance zero emission power plants has been prepared and tested by Lemański [15]. His concept is to combine high-temperature SOFC with gas shifting and post-combustion of a rest fuel within one chamber. Since the efficiency of pSOFC/GT (pressurised solid oxide fuel

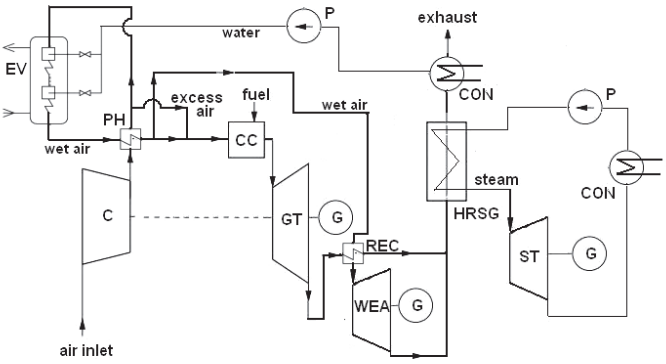

Figure 8. The LOTHECO basic combination of standard gas turbine and wet air expander (CC – combustion chamber, C – compressor, GT – gas turbine, PH – preheater, EV – evaporator, HRSG – heat recovery steam generator, ST – steam turbine, P – pump, CON – condenser, WEA – wet air expander, REC – recuperator) [11].

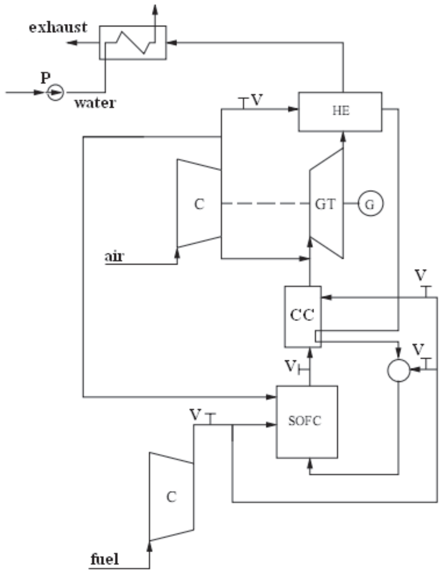

Figure 9. The scheme of hybrid pSOFC/GT used in commercial application by Siemens Westinghouse [15]: SOFC – solid oxide fuel cells, CC – combustion chamber, C – compressor, GT – gas turbine, HE – heat exchanger, V – valve, P – pump.

cell with gas turbine) hybrid system depends mainly form a level used pressure. Lemański also has investigated a system with double levels of pressure and double SOFC; one working in the higher and one in the clower pressure (Fig. 10a).

The compression ratio in the first fuel cell (FC1) is equal $\Pi_{FC1} = \frac{p_8}{p_7} = 3.8$. The pressure in the second fuel cell (FC2) is changed and it depends on the expan-

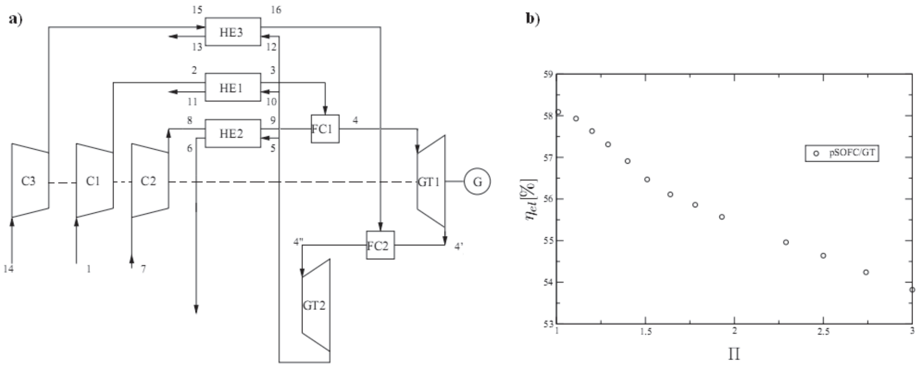

Figure 10. a) The scheme of the double pressure hybrid pSOFC/GT [15]: FC – fuel cells, C – compressor, GT – gas turbine, HE – heat exchanger, G – generator; b) the electrical efficiency, $\eta_{el}$, of double pressure pSOFC/GT system in function of expansion ration Π [15].

sion ratio $\Pi = \frac{p_4}{p_4'}$ in first gas turbine (GT1) which is investigated from Π = 1-3. In Fig. 10b the electrical efficiency, $\eta_{el}$, of double pressure pSOFC/GT system is presented – with increase of expansion ratio, Π, the electrical efficiency both second pSOFC (FC2) and gas turbine (GT) becomes smaller. It ought to be added that for low expansion ratio ($\Pi = 1$) the electrical efficiency, $\eta_{el} = 58.09\%$, of double pressure pSOFC/GT system is higher than the efficiency of traditional configuration with one pressurised fuel cell, which is equal $\eta_{el} = 57.40\%$ [15].

Additionally, the impact of fuel cell-$CO_2$ capture technology from hydrocarbon sources in terms of energy economics, as well as on our evolving energy resource mix and renewable energy development is presented in [3,16,17].

Yet other novel concept is to use instead of air some oxygen enriched mixture. It can be done with one exchange of air separation unit (ASU) (for oxygen separation) by the modern device called the mixed conducting membranes (MCM) which produces pure oxygen from air [5]. These membranes act similarly to the cathode in a SOFC which can conduct ions via nonporous, metallic oxides that operate at high temperatures, i.e., greater than 973.15 K. In comparison to cathode material it has a high oxygen flux and selectivity. The use of the MCM reactor instance of ASU means its integration into conventional gas turbine combustion. Similarly to SOFC [60–63], the classical chamber in an ordinary gas turbine is replaced by the MCM reactor, which includes a combustion chamber, a ‘low’ temperature heat exchanger, an MCM membrane and a high temperature heat exchanger. The concept allows for 100% $CO_2$ capture, increase of cycle efficiency up to 50%, and will, in this case, have less than 1 ppm v/v NOx in the oxygen depleted outlet air [5].

# 7 Repowering by the inverse Brayton cycle

By many means new energy conversion processes turn our attention onto original ideas connected with thermodynamical concept and thermodynamical cycles. Some concept, discovered by pioneers, did not reemerge until the work on $CO_2$ sequestration and storage. One of more important is an inverse Brayton cycle combined with a system where recycled exhaust gases contain a great amount of $CO_2$ and $H_2O$. In Fig. 11 there is shown a scheme of combination of the Brayton and inverse Brayton cycles. Inversion means that the role of a compressor and a gas turbine is changed and firstly we have expansion and next compression [2]. From the $H_2O$ steam point of view the system looks like a conventional steam turbine with condensation in low pressure, therefore the ratio of pressure $\Pi = p_{high}/p_{low} = p_2/p_{3^{in}}$, which determines the highest and lowest pressures in the system, is a one main parameter determining its efficiency. The second basic parameter is the temperature of condensation $T_{CON} = T_{3^{in}} = 303.15$ K that is determining the point where cooling of flue gases should be finished and wet steam from flue gases is condensed in condenser (CON). Next, the $CO_2$ and rest of steam go to the compressor ($C^{in}$), where it is compressed and, after that, it is cooled in the heat exchanger (HE). Moreover, the efficiencies of the elements system were all set up as it is written below:turbine: internal $\eta_{iT} = 88\%$, mechanical $\eta_{mT} = 99\%$; compresor: internal $\eta_{iC} = 87\%$, mechanical $\eta_{mC} = 99\%$; pump: internal $\eta_{iP} = 75\%$, mechanical $\eta_{mP} = 98\%$; generator $\eta_g = 97\%$, combustion chamber $\eta_{CC} = 99\%$ and heat exchanger $\eta_{he} = 98\%$. Oxygen, which purity varies in range (95-99.8)% is compressed in the compressor (C) and next is combusted with $CH_4$ (Fig. 11) in the combustion chamber (CC). This is hypothetical cycle with extremely high temperature in the combustion chamber (about $T_{TIT} = 5000$ K). In Fig. 12 there is presented the result of numerical simulation of the combined BC/IBC (Brayton cycle/inverse Brayton cycle). The exhaust (flue) gases contain $H_2O$ (66.6%) and $CO_2$ (33.3%) only. The optimal value of the efficiency of the BC/IBC cycle reached at the ratio of pressure are equal $\Pi = p_{high}/p_{low} = 190$.

# 8 Low-emission cycle via a steam-gas turbine

In principle the pre-combustion capture of $CO_2$ and oxy combustion can be accomplished more easily and cheaply than post-combustion removal of $CO_2$ from the exhaust gases emitted by a conventional coal power plant. The promise of a more efficient carbon capture is one of the primary rationales for clean gas technology (CGT).

Ziółkowski [38] has recently been considering a concept of repowering of GT8C

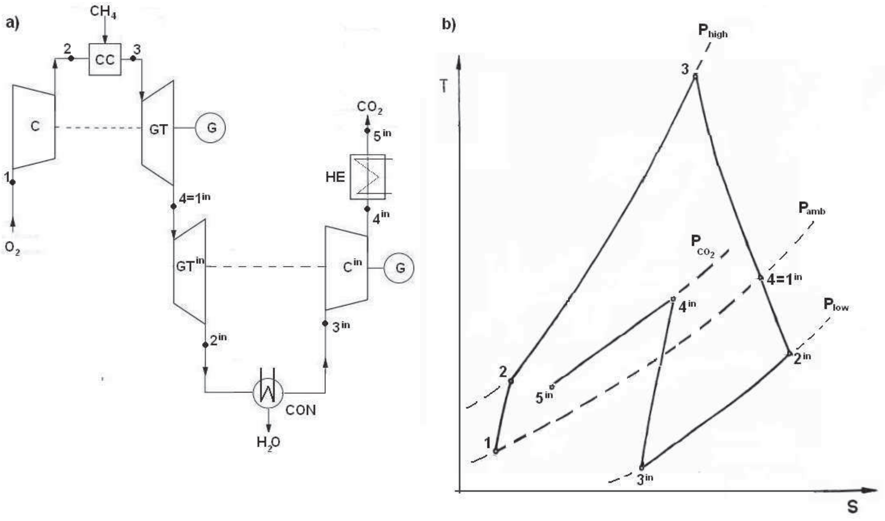

Figure 11. a) Scheme of Brayton/inverse Brayton cycle: CC – combustion chamber, C – compressor, GT – gas turbine, HE – heat exchanger, G – electric generator, CON – condenser; and b) its representation in temperature entropy diagram.

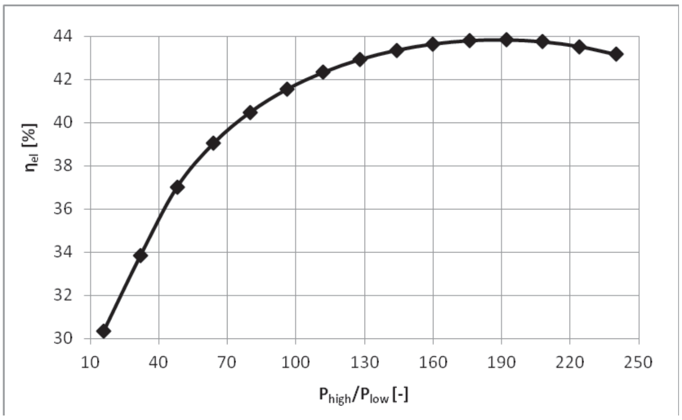

Figure 12. The electrical efficiency, $\eta_{el}$, of Brayton/inverse Brayton cycle depending on pressure ratio $p_{high}/p_{low}$.

turbine cycle. The concept is to exchange the combustion mode from the classical one into another one based on the oxygen enriched air. There is the removing steam turbine in comparison to a typical gas-steam cycle. Removing of heat from oxygen-hydrocarbon combustion process is governed by adequate steam injection into the combustion zone. It leads directly to flue gases that contain only $CO_2$ and steam.

A starting point for conversion of GT8C into low-emision GT8C has been taken as normal operating condition for PGE EC Gorzów CHP Plant. Data for numerical analysis of traditional GT8C have been taken from literature [33,35,36]. Two different modernizations of GT8C have been considered, both based on oxy combustion, namely the steam injection and flue gas recirculation (Fig. 13). The basic cycle contains: 12-stage compressor (C); silo-combustion chamber (CC), and three stage gas turbine (GT). Modernized cycle contains also; an air separation unit (ASU); heat recovery steam generator (HRSG), condenser of water vapour (CON), additional compressor (C') for recirculation flue gases, a pomp (P) and dividing elements (D).

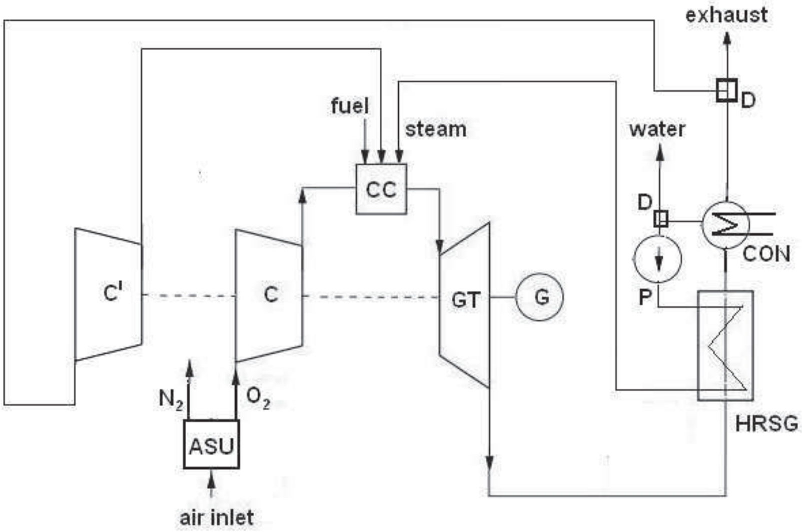

Figure 13. PGE EC Gorzów CHP Plant scheme of a gas-steam turbosets modification into oxy combustion, steam injection and flue gases recirculation cycle, where: ASU – air separation unit, C – compressor, C' – additional compressor, CC – combustion chamber, GT – gas-steam turbine, HRSG – heat recovery steam generator, P – pump, CON – condenser, D – mass flux divider.

The station ASU produces approx. $\dot{m}_o = 14.8$ kgO$_2$/s and its power consuming ratio is 0.248 kWh/kgO$_2$ [38]. From HRSG one obtains the mass flux of steam $\dot{m}_s = 28$ kg/s, what together with the combustion steam flux (approx. 8 kg/s) gives 0.2 mass fraction of $H_2O$ in steam-gas working fluid. This $H_2O$–$CO_2$ mixture expands in a gas turbine from pressure 1.62 MPa to ambient pressure, and next, after cooling down in HRSG, goes into a steam condenser. The excess of $CO_2$ after compressing to the pressure of liquefication is removed from the cycle.

The principal differences in both modifications are based on the treatment of the amount of injected fuel. In the first case the fuel mass flow rates, $\dot{m}_f$, become the same like in original GT8C (Tab. 1). In the second one, the fuel mass rate becomes higher since the power $N_{el}$ is stated at the same original level. We noted that together with increasing of $\dot{m}_f$ the temperature at combustion chamber, $T_{TIT}$, and exhaust gas temperature at turbine outlet, $T_{GT}$, grow respectively. It gives the possibility of growing of produced steam temperature up to $T_s = 853.15$ K. Table 1 summarizes the essential results of thermodynamic calculations in comparison to data included in available literature [33,35] for the thermal cycle of GT8C and modification presented in Fig. 13. The thermodynamic parameters of components of the medium in the characteristic points of the cycle were presented in Tab. 1, where: $T_{GT}$ – temperature at the turbine outlet; $T_{TIT}$ – temperature at inlet turbine; $T_a$ – temperature of air inlet; $T_f$ – temperature of fuel at the inlet; $T_s$ – temperature of steam produced in HRSG, $\dot{m}_{exh}$ – exhaust mass flow rate in gas turbine; $\dot{m}_{exh}^{amb}$ – exhaust mass flow rate outlet to the ambient air, $\dot{m}_f$ – fuel mass flow rate; $\dot{m}_s$ – steam mass flow rate, $p_a$ – pressure of air inlet; $p_f$ – pressure of fuel at the inlet; $eCO_2$ – emission level of carbon dioxide at the cycle outlet; $eNO_x$ – emission level of nitrogen oxides at the cycle outlet[4].

The electrical efficiency of the conventional GT8C was calculated to be $\eta_{el} = 34.8\%$. The conversion GT8C to the gas-steam turbine with oxy combustion results in decreasing of efficiency to the level of $\eta_{el} = 32.6\%$, however the significant decrease of $eNO_x$ emission to the level of 8 g/MWh has been observed simultaneously. The other advantage is reduction of the flue gases rates that are rejected into ambient air; $\dot{m}_{exh}^{amb} = 11$ kg/s, since the flux of 136 kg/s of flue gases undergoes recirculation connected with condensation of 35 kg/s of steam water. The increase of $eCO_2$ emission to approx. 18 kg/MWh was also observed in comparison to the original cycle GT8C. On the other hand, the grow of the mole fraction of $CO_2$ from 0.03 to 0.70 (I modification) and to 0.78 (II modification) was observed simultaneously. It appears that the first one obtains the better conditions to carbon capture from the exhaust gases.

The original electrical power of GT8C is estimated to be $N_{el} = 54.5$ MWe. In comparison to this the conversion into steam-gas turbine with oxy combustion and flue gases recirculation has reduced the power to 50.9 MWe (I modification) and 54.1 MWe (II modification), respectively. It appears that temperature of injected steam plays a crucial role. In Fig. 14 the results of parametric simulations are presented for the second modification. The level of anticipated emissions of $eCO_2$ and $eNO_x$ (Fig. 15) gas-steam turbine with oxy combustion in function of the

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[4] In order to find the emission level of carbon monoxide (CO) at the outlet of cycle some additional calculations of the combustion chamber by means of CFD software are needed. It should be noticed that the mass flow rate of recirculation $CO_2$ and the mass flow rate of injected steam are the variable factor that influences both the exhaust gas temperature and the increase of CO mole fraction [64]. The CFD and CFM codes have to be used for numerical predictions of optimal parameters of oxy-combustion.

Table 1. Comparison of two modifications of GT8C into the gas-steam turbine.

| Parameters | Unit | Oryginal GT8C | I modification | II modification |
|---|---|---|---|---|
| $T_a$ | K | 288.15 | 288.15 | 288.15 |
| $p_a$ | MPa | 0.1013 | 0.1013 | 0.1013 |
| $T_f$ | K | 288.15 | 288.15 | 288.15 |
| $p_f$ | MPa | 4.05 | 4.05 | 4.05 |
| $\dot{m}_f$ | kg/s | 3.21 | 3.21 | 3.41 |
| $\dot{m}_{exh}$ | kg/s | 182.3 | 182.3 | 182.5 |
| $\dot{m}_s$ | kg/s | – | 28 | 28 |
| $T_s$ | K | – | 823.15 | 853.15 |
| $T_{GT}$ | K | 793.15 | 833.15 | 864.15 |
| $T_{TIT}$ | K | 1373.15 | 1272.15 | 1305.15 |
| $\eta_{el}$ | % | 34.8 | 32.54 | 32.55 |
| $N_{el}$ | MWe | 54.5 | 50.9 | 54.1 |
| $\dot{m}_{exh}^{amb}$ | kg/s | 182.3 | 11.3 | 11.0 |
| $eCO_2$ | kg/MWh | 592 | 611 | 610 |
| $eNO_X$ | kg/MWh | 0.3670 | 0.0081 | 0.0079 |
| $O_2$ | – | 0.139 | 0.260 | 0.181 |
| $H_2O$ | – | 0.076 | 0.017 | 0.017 |
| $N_2$ | – | 0.745 | 0.025 | 0.027 |
| $CO_2$ | – | 0.031 | 0.698 | 0.775 |
| $NO_X$ | ppm | 30 | 14 | 15 |

injected steam temperature are shown. It means that together with the increase of electric power $N_{el}$ (temperature of injected steam, $T_s$) we observe the decrease of main emissions in Fig. 15. In presented cycle, steam temperature is desired to be as high as possible, since the higher thermal energy is then extracted from the exhaust gases. It ought to be emphasized that the higher steam temperature is so important similarly as higher steam mass flow rate. The advantage of the presented cycle with oxy combustion and $CO_2$ recirculation is low level of $NO_x$ and high level of $CO_2$ in exhaust gases, which next can be captured in an easy way.

## 9 Zero-emission cycle via a steam-gas turbine

Huge perspectives are ahead of the new concept of cycle featuring the combined Brayton cycle, inverse Brayton cycle, recuperation, oxy-combustion and capture $CO_2$ presented in Fig. 16.

Wet and hot exhaust gases at the atmospheric pressure from the gas turbine

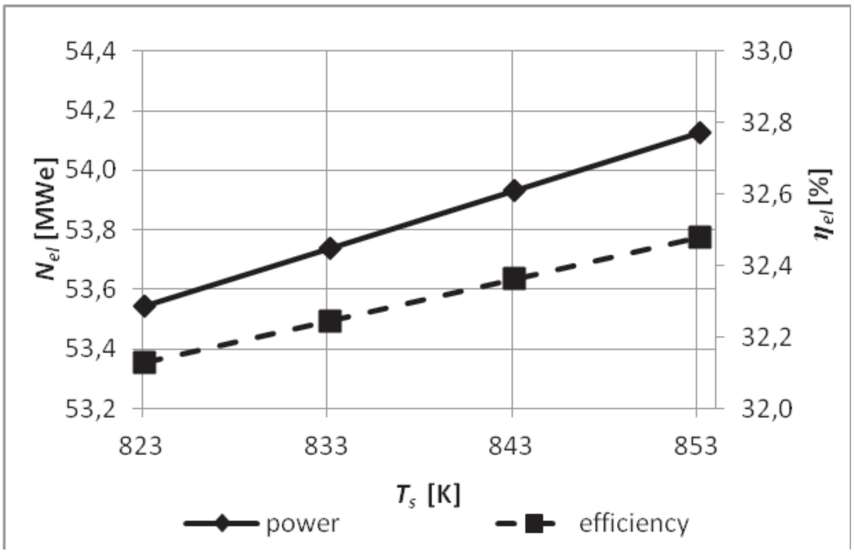

Figure 14. Dependence of the net electrical power, $N_{el}$, and the electrical efficiency, $\eta_{el}$, of the turbine with oxy combustion against temperature of injected steam, $T_s$ (for the second modification).

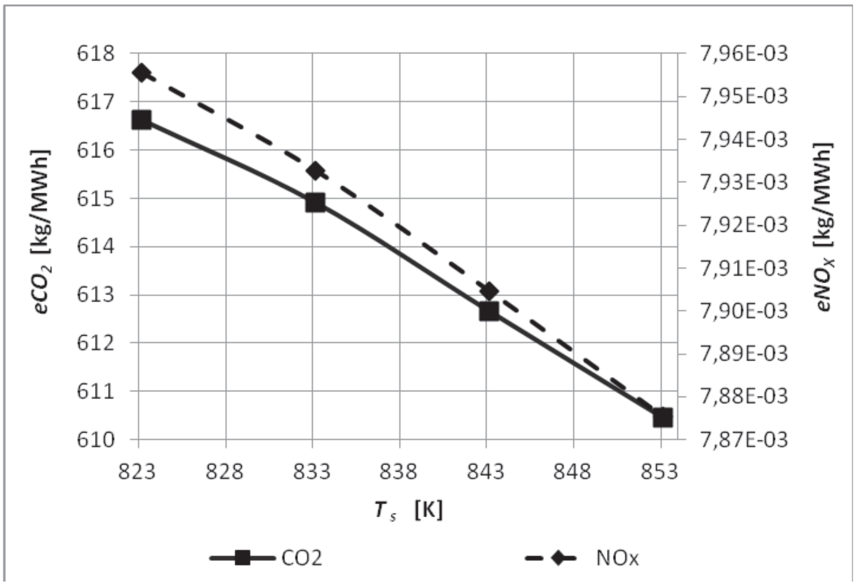

Figure 15. Simulated level of $eCO_2$ and $eNO_x$ emissions from the gas-steam turbine with oxy combustion versus the temperature of injected steam $T_s$, (for the second modification).

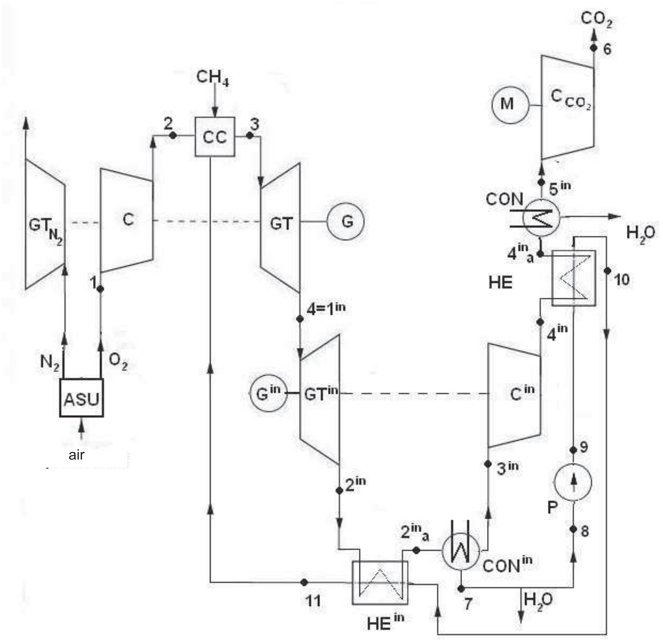

Figure 16. Schematic diagram of the double Brayton cycle with the use of oxy combustion, water injection and capture $CO_2$ (ASU – air separation unit, CC – combustion chamber, C – compressor, GT – gas turbine, HE – heat exchanger, G – electric generator, M – motor, CON – condenser, P – pump, $GT_{N2}$ – additional gas turbine of $N_2$, $C_{CO2}$ – compressor of $CO_2$).

(GT) are able to generate extra turbine power, by expanding in the negative pressure gas turbine ($GT^{in}$). The gas mixture, at that level, is still a high temperature fluid and it needs to be chilled. It is all done by using a special regenerative heat exchanger ($HE^{in}$). After pre-cooling, steam ought to condensate (or just a part of it) in the condenser ($CON^{in}$). The pressure of gases is lower than the atmospheric one and that is why the exhaust needs to be compressed in the compressor ($C^{in}$). Compressed gases are directed to the second pre-cooling heat exchanger (HE) and second condenser (CON), where dehumidification takes place. Additionally, the array of devices realizing the cycle consists of the negative pressure turbine ($GT^{in}$), negative pressure heat exchanger ($HE^{in}$) and compressor ($C^{in}$) (Fig. 16). The cycle described above is the inversed Brayton cycle (IBC) indeed.

As it was mentioned, in the condenser (CON), working fluid is separated into water and $CO_2$. Next, the $CO_2$ is directed to the compressor of $CO_2$ ($C_{CO2}$), where it is compressed and, after that, it is cooled and condensed. The main disadvantage of the whole system is the necessity of the air separating station (ASU), to supply the combustion chamber with the pure oxygen. Moreover the 95%-oxy-combustion eliminates almost entirely the problem of $NO_x$ emission. Additionally, the nitrogen turbine ($\mathrm{GT}_{N2}$) might be used and would be fueled from the oxygen and nitrogen separating station.

Thermodynamic analysis of such cycle with the use of oxy combustion and $CO_2$ capture is prepared in work [65].

## 10 Summary

In order to made the clean gas technologies perspective, and especially dedicated to Pomerania, a brief review of recently developed systems has been presented and selected system have been analyzed. The principal types of cycles based on gas fuel, which have a chance to be adopted for zero-emission systems, has been discussed, mainly with respect to thermodynamical efficiency of energy conversion. Firstly, the Cheng cycle has been described showing that: the gain of power, $N_{el}$, and electrical efficiency, $\eta_{el}$, by steam injection is about 23%; application of Cheng cycle reduces carbon dioxide and nitrogen oxide emissions in comparison to pre- modernization of the heat and power plant. Secondly, repowering by the Szewalski cycle has been presented showing that conversion of a Brayton into a Szewalski cycle causes the increase of the cycle efficiency, $\eta_{el}$, decreases the amount of fuel burned in the turbine, and, unfortunately, lowers the net power of a cycle, $N_{el}$. Thirdly, LOTHECO cycle has been described showing that: for low temperature range and for low-quality heat sources this cycle is favorably integrated. Fourthly, repowering by the dual pressure pSOFC and gas turbine has been presented showing that the concept of dual pressure hybrid pSOFC/GT gives electrical efficiency, $\eta_{el} = 58.09\%$, which is higher than efficiency of traditional configuration with one pressure fuel cell. Fifthly, the inverse Brayton cycle has been described showing that the optimal value of the efficiency, $\eta_{el}$, of the BC/IBC cycle reached at the ratio of pressure equal $\Pi = p_{high}/p_{low} = 190$. Sixthly, low-emission cycle via a steam-gas turbine has been presented showing that conversion of a Brayton cycle (GT8C) into a cycle with oxy combustion and recirculation exhaust gases causes a decrease of the cycle efficiency, $\eta_{el}$, decreases the amount of emissions and, unfortunately, lowers the net power of a cycle, $N_{el}$. Finally, zero-emission cycle via a steam-gas turbine has been described showing that the concept of power plant with the use of oxy combustion and $CO_2$ capture is the lack of pollution emissions such as $NO_x$ and $CO_2$. Further coupled analysis using 0D and 3D models in cycle's devices is required to establish the form and value of the model parameters.

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