[in:] TECHNICAL, ECONOMIC, AND ENVIRONMENTAL ASPECTS OF COMBINED CYCLE POWER PLANTS
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MORE EFFICIENT GAS-STEAM POWER PLANT TOPPED BY A LiBr ABSORPTION CHILLER

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Abstract
During summer period a some part of the cooling water that is not used, can be employed for the electrical power and efficiency increasing within a gas-steam power plant combined with LiBr absorption chiller. In the present paper, the numerical analysis of power and net efficiencies of tri-generation plant are presented and discussed. The analysis is based on the mathematical chiller model that has been tested and implemented to the COM-GAS code

1. Motivation

It is commonly know that the decreasing of air temperature at gas turbine inlet and its inter cooling inherently increases the thermodynamic efficiency, but the problem is in which manner such a great amount of air could be cooled. Therefore, the objective of this paper has been to present a method of air cooling by an absorption chiller. The primary objective has been connected with the determination of the air cooling effectiveness on overall gas turbine performance.

This paper presents a mathematical model of single-stage LiBr/H_20 absorption refrigeration system, which consists of an evaporator, a low-pressure absorber, a high-pressure generator, a condenser, a heat exchanger, drives by a temperature hot source. The operation of absorption chiller within the combined cycle should be analyzed. The absorption chiller is a device which does not generally need electrical power to function[1], but consumes almost all of heat, so we shall analyze performance of a LiBr chiller fed from a steam turbine extraction. In the present paper some results of numerical analysis of a real combined gas-steam cycle in Gorzów Wielkopolski topped with an additional absorption chiller have been presented. The calculation are performed by means of our in-house code COM-GAS.

2. Mathematical model of the LiBr absorption chiller

As far as absorption devices are thermally activated, the high input power is not required. The LiBr absorbtion chiller employes water as a working fluid and lithium bromide as an absorbent. A single-stage LiBr absorption refrigerant system generally consists of an evaporator, absorber, generator condenser and solution heat exchanger (Fig.1). Schematic representation of a thermodynamical cycle on a pressure-temperature diagram is illustrated in Fig.2. The system requires a heat source that should have temperature at least 85ºC, in order to achieve a reasonable COP. The absorption technology provides a peak cooling performance coefficient approximately equal 0.7 and operates with temperatures up 168ºC. The multistage absorption chiller should be installed when higher temperature of heat source is available. In this case higher COP could be achieved. The pump forces the weak flow of LiBr solution from absorber {4} to desorber {7} through a heat exchanger {6}. The temperature of the solution rises in heat exchanger. In the desorber, thermal energy is added so the solution starts to boil and refrigerant is freed. Next, the refrigerant vapour flows from desorber {7} to the condenser {1}. The heat is rejected as the refrigerant condenses. The condensate flows to evaporator {3} through expansion valve {29} where the expansion

[1] A small amount of electrical power is consumed by gear pumps only.

take place. Then in the evaporator, the refrigerant evaporates and flows back to the absorber {4}. A small portion of the refrigerant leaves the evaporator as a liquid spillover. The strong LiBr solution exits the desorber {7} and flows to absorber {4} through heat exchanger where it is cooled. In the absorber {4} steam is absorbed by the concentrated solution of LiBr. Then the LiBr aqueous solution changes from strong to the weak solution and rejects heat.

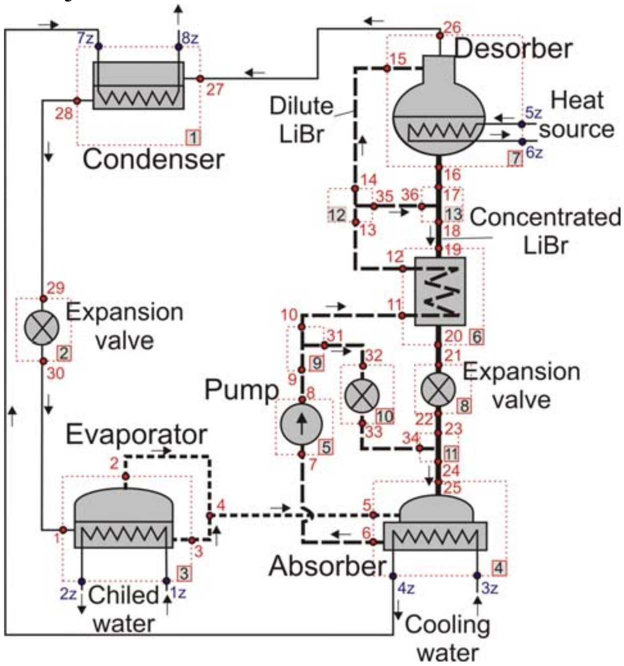

Fig. 1. LiBr-water absorption cycle diagram

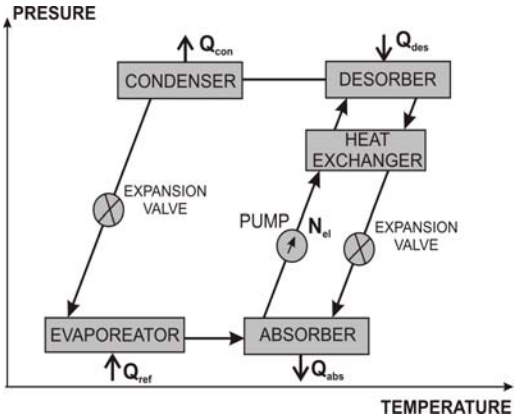

Fig. 2. Schematic presentation on a pressure temperature diagram

The mathematical model is proposed as follows (see Fig.1):

Evaporator

mass balance: $\dot{m}_2 + \dot{m}_3 = \dot{m}_4$, $\dot{m}_3 = (2.5 \div 3)\% \dot{m}_3$ (1)

energy balance: $\dot{Q}_{Ref} = \dot{m}_2 i_2 + \dot{m}_3 i_3 - \dot{m}_1 i_1$ (2)

Absorber

mass balance: $\dot{m}_6 = \dot{m}_5 + \dot{m}_{25}$ (3)

LiBr balance: $X_{25}\dot{m}_{25} = X_6 \dot{m}_6$ (4)

X_{25} -concentration of strong LiBr solution,

X_{26} -concentration of weak LiBr solution,

energy balance: $\dot{Q}_{Abs} = \dot{m}_5 i_5 + \dot{m}_{25} i_{25} - \dot{m}_6 i_6$ (5)

Pump of weak LiBr solution

mass balance: $\dot{m}_7 = \dot{m}_8$ (6)

energy balance: $\eta_{el} N_{El} = \dot{m}_7 \upsilon_7 (p_8 - p_7)$ (7)

Heat exchange

mass balance: $\dot{m}_{11} = \dot{m}_{12}$, $\dot{m}_{19} = \dot{m}_{20}$ (8)

energy balance: $\dot{Q}_{Exch} = \dot{m}_{11}(i_{12} - i_{11}) = \dot{m}_{19}(i_{19} - i_{20})$ (9)

Desorber

mass balance: $\dot{m}_{15} = \dot{m}_{16} + \dot{m}_{26}$ (10)

LiBr balance: $X_{15}\dot{m}_{15} = X_{16}\dot{m}_{16}$ (11)

energy balance: $\dot{Q}_{Des} = \dot{m}_{26}i_{26} + \dot{m}_{16}i_{16} - \dot{m}_{15}i_{15}$ (12)

Condenser

mass balance: $\dot{m}_{27} = \dot{m}_{28}$ (13)

energy balance: $\dot{Q}_{Con} = \dot{m}_{27}(i_{27} - i_{28})$ (14)

Coefficient of performance is defined to be equal: $COP = \dot{Q}_{Ref} /(\dot{Q}_{Des} + N_{El})$ (15)

In the evaporator, the refrigerant is saturated and the temperature t2z (chilled water outlet) is assumed. The saturation pressure at point 2 is calculated for temperature $t_2 = t_{2z} - \Delta t_{Evap}$. In the first step of numerical calculation the enthalpy is assumed at point 1 as equal 130 kJ/kg. At the point 3, the refrigerant is a saturated liquid, and its enthalpy is 23.5 kJ/kg. Then the program needs as an input heat in evaporator. The mass at point 1 is calculated in the next step. At point 16, the mass fraction and temperature of the solution must be given at the beginning of calculations. Temperature in desorber is set as the result of difference between temperature at the outlet (point 6z) and a temperature difference in desorber $t_{16} = t_{6z} - \Delta t_{Des}$. The solution mass fraction value is assumed, for example, as equal 60% LiBr. Then, a saturation pressure $p_{16} = p_{des}$ and enthalpy in desorber is known. The pressure at point 16 is the same as in point 29. Then the evaporator can be calculated. Mass flow rate of steam $\dot{m}_6$ is calculated with the equation (4) and mass in absorber is assumed as $\dot{m}_5 = \dot{m}_1$ for weak and strong concentration of solution. The temperature at point t_{26} is taken to be an arithmetic mean $t_{26} = (t_{5z} + t_{16})/2$ and mass $\dot{m}_{26}$ is given from equations (10) and (11). The calculations according to the above mathematical model have been made by computer program written in Fortran language, which involves the algorithm described above and mathematical tables for the fluid properties [3]. The model in the form of a subroutine has been implemented into the COM-GAS code.

3. Implementation in the COM-GAS code

Program COM-GAS makes it possible to calculate any thermodynamic cycle on the so-called design level. It has been written in IMP PAN Gdansk, Thermo-Chemical Power Department. All numerical procedures are written in the Fortran Language and visualization has been made in Delphi Language. A new part that is implemented to COM-GAS is the absorption chiller. The procedure was tested out in the works [1-4,8]. It makes it possible to calculate a thermodynamic cycle in the case of cogeneration and trigeneration (energy power production, production of heat and cold).

The following data are needed to perform a chiller calculation:

- weak and strong LiBr solution,
- inlet and outlet temperature in evaporator,
- temperature in inlet absorber (cooled absorber),
- heat efficiency of chiller parts,
- temperature difference in particular parts of chiller,

Finally, the temperature and heat flux which are generated somewhere outside, for instance, at the steam turbine extract points, are needed. As the result, the cooling heat flux and heat fluxes in each part of chiller and the coefficient of chilling performance (COP) are obtained.

4. Topping of EC Gorzów Wielkopolski by an absorption chiller

Let us consider gas and steam combined cycle similar to EC Gorzów Wielkopolski [9]. The 3 modifications (variants) of present cycle are assumed:

- I – before topping of combined gas and steam turbines (condensed and counter-pressure turbine),
- II – after topping of combined gas and steam turbines (condensed and counter-pressure turbine) with the absorption chiller which provides the cooling air at the compressor inlet,
- III – after topping of combined gas and steam turbines (condensed and counter-pressure turbine) with the absorption chiller which is employed as compressor intercooling.

4.1. Variant I

In this case the efficiency and power of combined gas and steam cycles have been calculated for air temperature changes. It initial data were assumed as follow: inlet air mass flow rate – 174.2 kg/s, stream of fuel – 8.09 kg/s, temperature at output to heat recovery steam generator - 160 ^{0}C, output pressure in condenser 0.1 MPa in the cause of counter-pressure turbine– variant Ia and 5kPa for the condensing turbine -variant Ib.

The theoretical diagram and calculation results for combined cycle with the counter-pressure turbine in the case of air inlet temperature equal 30 ^{0}C, are presented in Fig.3. Additionally the results of the combined cycles Ia and Ib computations are included in Table 1.

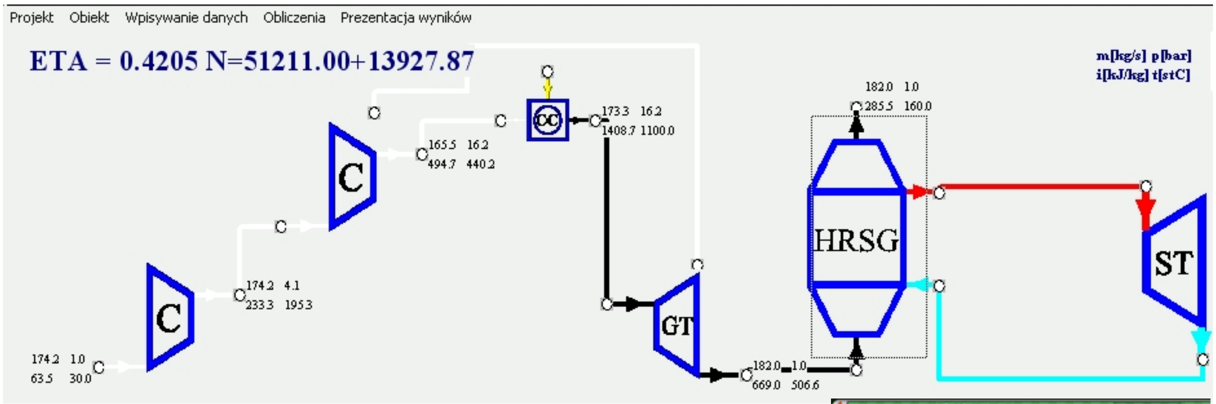

Fig. 3. Combined cycle

Initial data for calculation:
Mass flux of air mass - 174,2 kg/s
air pressure - 0,101325 MPa
charger compression - 4
turbine expand (decompression) - 16
temperature of exhaust steam-440 ºC
chargers and turbine efficency - 88%
heat exchenger in output pressure - 0,13 MPa

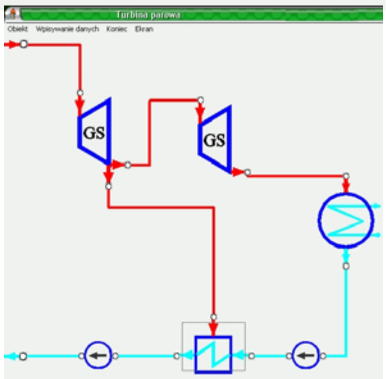

Fig. 4. Steam turbine

Table 1. Results for the design combined cycle, variant Ia –counter-pressure turbine 0,1 MPa, variant Ib- condensation turbine 5 kPa

	Variant Ia and Ib				Variant Ia			Variant Ib		
Temp air inlet [^{0}C]	t_{TIT} [^{0}C]	$t_{Out\ GT}$ [^{0}C]	$\dot{m}$ steam [kg/s]	N_{GT} [kW]	N_{ST} [kW]	N_{GT}+ N_{ST} [kW]	η [%]	N_{ST} [kW]	N_{GT}+ N_{ST} [kW]	η [%]
5	1086,6	496,9	22,85	55036,9	13395,5	68432.4	42,82	20828,5	75865,4	47,47
10	1094,1	501,7	23,2	54693,2	13602,5	68295,7	42,73	21147,5	75840,7	47,45
20	1107,0	510,2	23,9	54007,7	14016,6	68024,3	42,56	21785,6	75793,3	47,42
30	1118,6	517,9	24,6	53316,5	14431,0	67747,5	42,4	22423,9	75740,4	47,39
35	1123,7	521,3	24,95	52952,5	14638,2	67590,7	42,3	22735,4	75687,9	47,36

The results of combined cycle for the same input parameters but with the constant temperature equal 1100ºC after combustion chamber are presented in Table 2.

Table 2. Results for the design combined cycle, variant Ia –counter-pressure turbine 0.1 MPa, varianbt Ib- condensation turbine 5 kPa

	Variant Ia and Ib			Variant Ia			Variant Ib		
Temp air inlet [^{0}C]	$\dot{m}$ fuel [kg/s]	$\dot{m}$ steam [kg/s]	N_{GT} [kW]	N_{ST} [kW]	N_{GT}+ N_{ST} [kW]	η [%]	N_{ST} [kW]	N_{GT}+ N_{ST} [kW]	η [%]
5	8,27	23,45	56552,66	13750,4	70303,1	43,03	21375,7	77928,4	47,7
10	8,17	23,5	55365,0	13779,9	69144,9	42,84	21421,3	76786,3	47,58
20	8,00	23,6	53250,7	13839,1	67089,9	42,45	21512,1	74762,9	73,31
30	7,84	23,74	51202,8	13922,0	65124,8	42,05	21640,0	72842,8	47,0
35	7,78	23,86	50335,6	13992,9	64328,6	41,85	21749,1	72084,7	46,9

4.2. Variant II

In the following case a single-stage absorption chiller was introduced instead of heat exchanger. It allowed to decrease the temperature of air at compressor inlet from 30ºC to 10ºC. In Fig. 5 a theoretical diagram and numerical results for the combined cycle with the counter-pressure turbine and absorption chiller are shown.

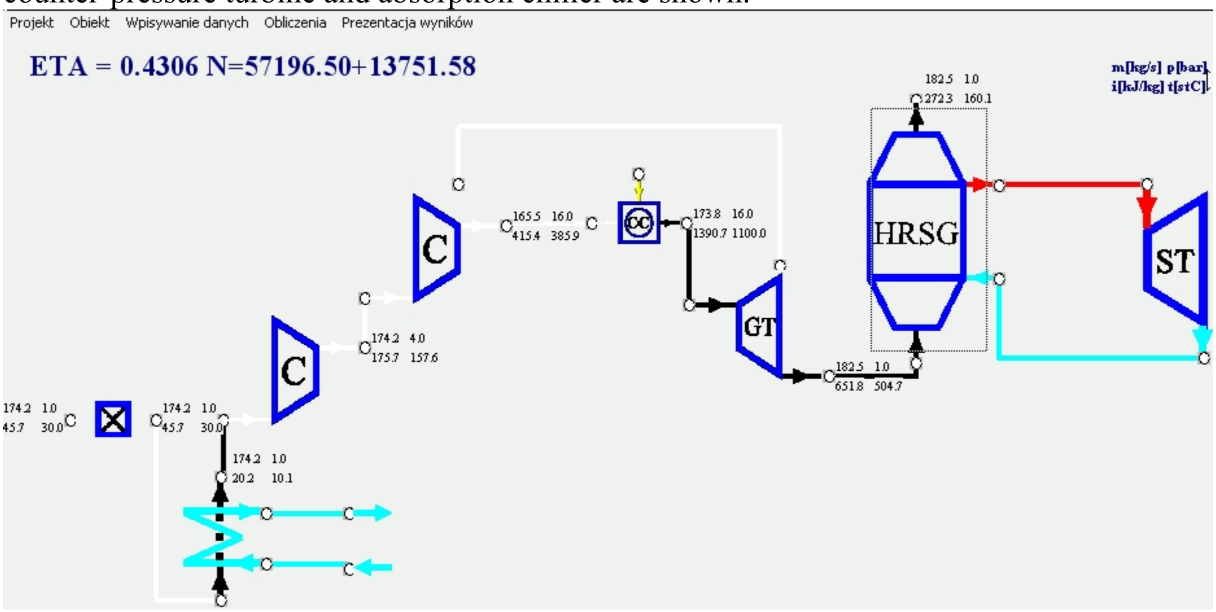

Fig. 5. Combined cycle

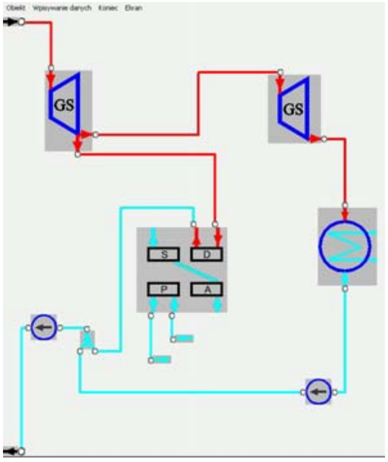

Fig. 6. Steam turbine with absorption chiller

Absorption chiller is fed by wet steam coming from steam turbine output (vent) under the pressure equal 0.13 MPa.

Results of calculations for cycle with LiBr chiller has been compared to the results of calculation for variant I – with the GT inlet temperature 30ºC (Table 3).

Table 3. Comparison between results obtained for variant I and II

Variant	Counter-pressure turbine 0.1 MPa		Condensing turbine 5 kPa	
	Combined cycle	Combined cycle with absorption chiller	Combined cycle	Combined cycle with absorption chiller
t_{air} [^{0}C]	30	30	30	30
$t_{air\ inlet}$[^{0}C]	30	10	30	10
$\dot{m}_{air}$ [kg/s]	174,2	174,2	174,2	174,2
$\dot{m}_{cool\ GT}$ [kg/s]	8.7	8.7	8.7	8.7
compression C_1	4	4	4	4
compression C_2	4	4	4	4
expand TG	16	16	16	16
$\dot{m}_{fuel}$ [kg/s]	7,841	8,34	7,841	8,34
$t_{out\ compressor}$ [^{0}C]	440,2	385,9	440,2	385,9
t_{tit} [^{0}C]	1100	1100	1100	1100
$t_{out\ GT}$ [^{0}C]	506,6	504,7	506,6	504,7
$t_{out\ HRSG}$[^{0}C]	160	160	160	160
$\dot{m}_{exhaust\ gas}$[kg/s]	182,0	182,5	182	182,5
N_{GT} [kW]	**51211,0**	**57196,5**	**51211,0**	**57196,5**
p_{steamp} [MPa]	4	4	4	4
t_{steam} [^{0}C]	440	440	440	440
$\dot{m}_{steam}$ [kg/s]	23.75	23,5	23,75	21,7
$p_{steam\ extraction}$ [kPa]	130	130	130	130
$t_{steam\ extraction}$ [^{0}C]	126,24	126,24	126,24	126,24
$\dot{m}_{steam\ extraction}$ [kg/s]	0,13	2,95	2,68	2,95
Heat desorber [kW]	-	6714,1	-	6714,1
Heat intercooling [kW]	-	4341,0	-	4341,0
Heat absorber+condenser[2]	-	10706,0	-	10706,0
$\eta_{absorp.\ chiller}$ [%]	-	64,6	-	64,6
$p_{condenser}$ [kPa]	100	100	5	5
N_{ST} [kW]	**13927,87**	**13751,58**	**21649,13**	**19556,8**
$N_{GT}+N_{ST}$ [kW]	**65138,87**	**70948,08**	**72860,13**	**76753,3**
η_{el} [%]	**42,05**	**43,06**	**47,04**	**46,6**

[2] To proper chiller operation heat released during absorber and condenser cooling has to be taken back, for axample in a cooling tower.

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Details of the chiller numerical results are presented in Fig. 7.

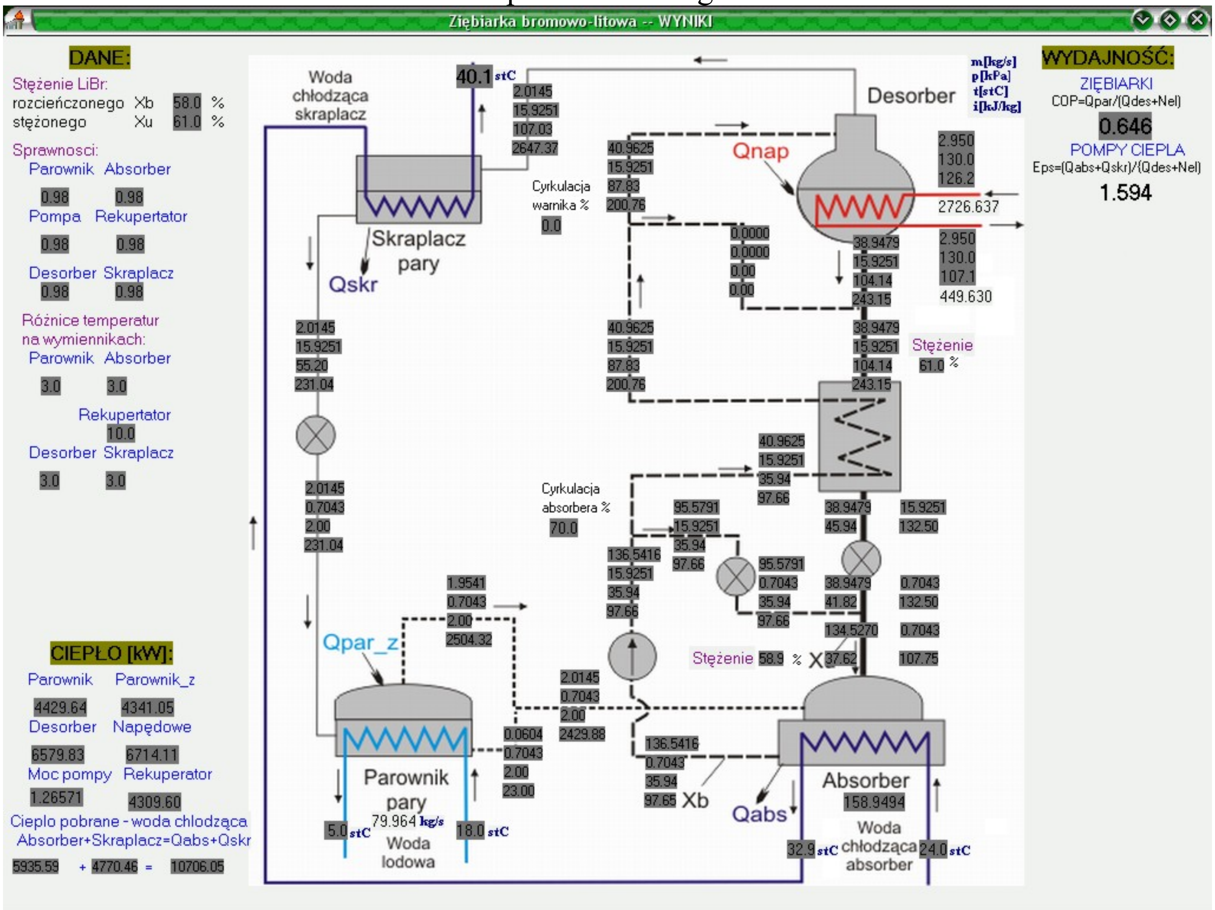

Fig. 7. Details of the chiller numerical results

4.3. Variant III

Let us consider the case when the main task of an absorption chiller is an air intercooling. The temperature change in intercooler was assumed equal 50ºC. An example gas and steam turbine cycle with condensing turbine and absorption chiller as intercooler is presented in Fig.8.

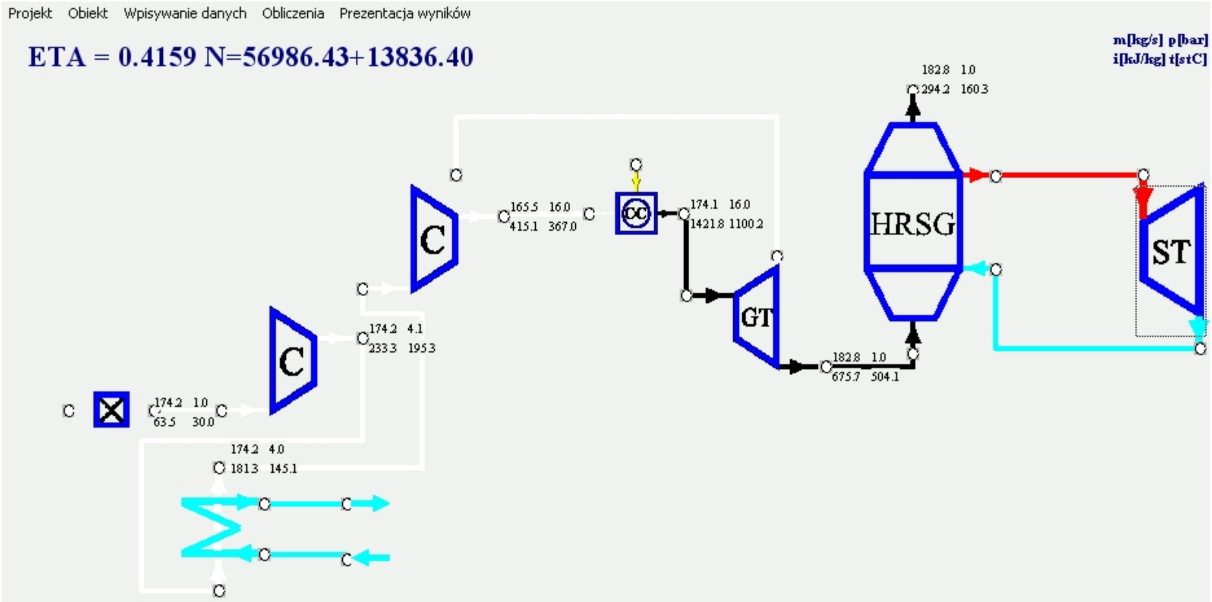

Fig. 8. Scheme of combined cycle (variant III)

Table 4. Comparison between results obtained for variant I and II

Variant	Counter-pressure turbine 0.1 MPa		Condensing turbine 5 kPa	
	Combined cycle	Combined cycle with absorption chiller	Combined cycle	Combined cycle with absorption chiller
t_{air} [^{0}C]	30	30	30	30
$t_{air\ inlet}$[^{0}C]	30	30	30	30
$\dot{m}_{air}$[kg/s]	174,2	174,2	174,2	174,2
$\dot{m}_{cool\ GT}$ [kg/s]	8.7	8.7	8.7	8.7
compression C_1	4	4	4	4
$t_{out\ C1}$ [^{0}C]	-	195,3	-	195,3
$t_{inlet\ C2}$ [^{0}C]	-	145,1	-	145,1
compression C_2	4	4	4	4
expand GT	16	16	16	16
$\dot{m}_{fuel}$ [kg/s]	7,841	8,53	7,841	8,53
t_{tit} [^{0}C]	1100	1100	1100	1100
$t_{out\ GT}$ [^{0}C]	506,6	504,1	506,6	504,1
$t_{out\ HRSG}$ [^{0}C]	160,0	160,0	160,0	160,0
$\dot{m}_{exhaust\ gas}$ [kg/s]	182,0	182,8	182,0	182,8
N_{GT} [kW]	**51211,0**	**56986,4**	**51211,0**	**56986,4**
p_{steam} [MPa]	4,0	4,0	4,0	4,0
t_{steam} [^{0}C]	440,0	440,0	440,0	440,0
$\dot{m}_{steam}$ [kg/s]	23.75	23,53	23,75	22
$p_{steam\ extraction}$ [kPa]	130	130	130	130
$t_{steam\ extraction}$ [^{0}C]	126,24	126,2	126,24	126,2
$\dot{m}_{steam\ extraction}$ [kg/s]	0,13	6	2,68	6
Heat desorber [kW]	-	13657,2	-	13657,2
Heat intercooling (from evaporator) [kW]	-	8850,99	-	8850,99
Heat absorber+condenser[3]	-	21799,46	-	21799,46
$\eta_{absorp.\ chiller}$ [%]	-	64,8	-	64,8
$p_{condenser}$ [kPa]	100,0	100,0	5,0	5,0

[3] To proper chiller operation heat released during absorber and condenser cooling has to be taken back, for axample in a cooling tower.

N_{ST} [kW]	**13927,87**	**13836,4**	**21649,13**	**18586,86**
$N_{GT}+N_{ST}$ [kW]	**65138,87**	**70822,8**	**72860,13**	**75573,26**
$\eta_{el[\%]}$	**42,05**	**41,59**	**47,04**	**44,38**

5. Conclusion

Results, in case of cooling by absorber chiller, show a significant increase of gas turbine power. Unfortunetely it does not follow increase of the unit efficiency. Only the cycle II with counter-pressure turbine tends to increase the overall cycle efficiency. In the condensing turbine an extraction steam flux does not work in the turbine but instead of it is a heat source to an absorber chiller, which cause a significant decrease of steam turbine power and the overall combined cycle power. The only gains caused by the inlet air decrease could be significant for a gas turbine alone. For example, in heat and power generation plant Władysławowo [10], without a steam turbines, the absorption chiller could be employed, because a significant increase of power and efficiency is expected. In the case of combined cycles with a counter-pressure turbine, the absorption chiller causes slight increase of cycle efficiency. For a heat and power generation plant it will bring significant profits beyond heating season. In this period turbines are not in use or even some blocks are shut off. Connecting such a configuration with the absorption chiller enables a unit for operate in all seasons with high efficiency, without shuting off or limitation. Chiller may be connected directly to heat exchanger and absorb a significant amount of heating changing its into a cool water which may be used for chilling air inlet or intercooling as well as selling or using for own needs (air conditioning).

A significant power decrease is noticed in the case of the condensing. Steam does not work in the turbine but is leading to the absorption chiller. The absorption chiller has some limitations as minimal working temperature of 85ºC and pressure of 0.1 MPa. This is a reason why the combined cycles with the condensing turbine have not possibilities to use an absorption chiller unless we a free source of heating like geothermal source is available. The efficiency of such a system will be greater.

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