archives
of thermodynamics
Vol. **35**(2014), No. 4, 83–95
DOI: 10.2478/aoter-2014-0035

Thermo-economic comparative analysis of gas turbine GT10 integrated with air and steam bottoming cycle

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Abstract A thermodynamic and economic analysis of a GT10 gas turbine integrated with the air bottoming cycle is presented. The results are compared to commercially available combined cycle power plants based on the same gas turbine. The systems under analysis have a better chance of competing with steam bottoming cycle configurations in a small range of the power output capacity. The aim of the calculations is to determine the final cost of electricity generated by the gas turbine air bottoming cycle based on a 25 MW GT10 gas turbine with the exhaust gas mass flow rate of about 80 kg/s. The article shows the results of thermodynamic optimization of the selection of the technological structure of gas turbine air bottoming cycle and of a comparative economic analysis. Quantities are determined that have a decisive impact on the considered units profitability and competitiveness compared to the popular technology based on the steam bottoming cycle. The ultimate quantity that can be compared in the calculations is the cost of 1 MWh of electricity. It should be noted that the systems analyzed herein are power plants where electricity is the only generated product. The performed calculations do not take account of any other (potential) revenues from the sale of energy origin certificates.

Keywords: Gas turbine air bottoming cycle, Air bottoming cycle, Gas turbine, GT10

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1 Introduction

In their technological structure, the gas turbine air bottoming cycle (GT-ABC) systems use a gas turbine as the upper heat source. The situation is similar in the case of gas turbine steam bottoming cycles (GT-SBC) or combined cycle power plants (CCPP). The installations with the steam system usually achieve higher power efficiency, i.e., a higher power capacity, which results from the characteristics of the medium used in the bottoming cycle. The comparative analysis allows a rational assessment of gas-air and gas-steam systems. The most essential aspect of the analysis is the economic comparison. In order to make the comparison between gas-air and gas-steam systems, the same gas turbine – GT10 – was selected. Considering that the air bottoming cycle medium of the gas-air system is characterized by a low heat capacity (which results in considerable dimensions of the air heat exchanger, AHX, [1]), the systems under analysis have a better chance of competing with gas-steam configurations in the small range of the power output capacity of up to about 30 MW. The gas turbine, the data of which are used in the calculations, finds application in commercial gas-steam units and has a power capacity of 25 MW [1,2]. Thermodynamic and economic analysis of a gas-air system whose technological structure includes a GT10 gas turbine is introduced in [3]. The turbine operates in one of the Polish gas pumping stations. The use of the power potential of air systems may be very favourable in terms of a higher power efficiency of a stand-alone gas turbine unit, especially in the case of operation at locations without access to cooling water. Air-cooled gas-air systems will have a lower value of generated power due to the need to save some of the energy to drive the air fans cooling the compressor interstage cooler.

2 System under analysis

A simple gas-air system is presented in Fig. 1a (Variant A). It is a system with no interstage cooling of compressor C2. The border between the gas system and the air part is marked symbolically. The system with one interstage cooler is presented in Fig. 1b (Variant B) and the most sophisticated installation in terms of the compressed air cooling system is shown in Fig. 1c (Variant C). It is assumed that depending on the type of operation, it has to be possible for the gas turbine exhaust gases to bypass the flue gas-air exchanger and to be carried away to the stack. A system of con-

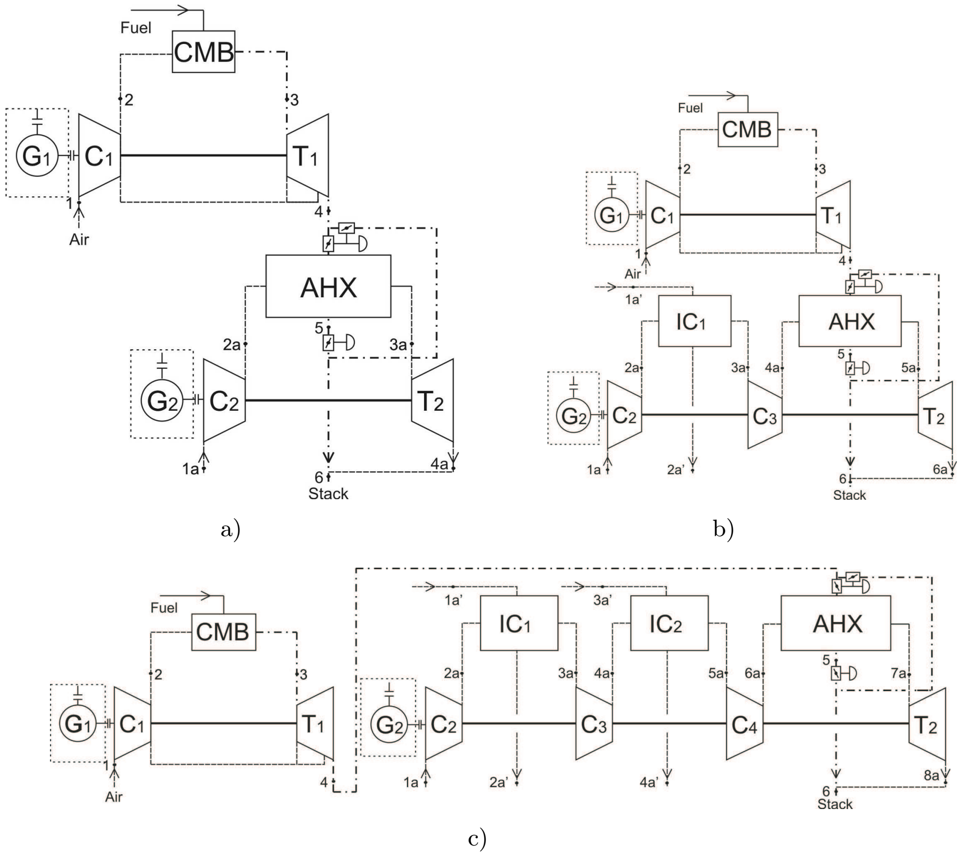


Figure 1: Basic technological structures of gas-air systems: a) system with no cooling (Variant A); b) system with one intercooler (Variant B); c) system with two interstage coolers (Variant C); C – compressor, T – turbine, G – generator, IC – intercooler, CMB – combustion chamber, AHX – air heat exchanger; 1,2...,a,a' – individual cycle characteristic points.

trol dampers is used in flue gas ducts. The control dampers direct the flue gases to the stack or to the heat exchanger (in the case of the air system operation). The analyzed structures are two-shaft systems. The air part expander drives the compressor and (depending on the configuration) the generator or another working machine, e.g., a gas compressor. A decision was made to combine the flue gas system with the air system and to use a common stack. If the gas system operates with no air turbine, the air heat exchanger has to be separated from hot exhaust gases. For this purpose, an additional damper is used after the heat exchanger. Both air and water can be used as coolant. The GT10 turbine has been manufactured since 1991. Its basic data are presented in Tab. [4]. The GT10 gas turbine unit is also used in the gas-steam system [2].

Table 1: GT10 gas turbine performance data.

Parameter	Value	Unit
Shaft power	25	MW
Energy efficiency	34	%
Rotor Speed	7700	rpm
Exhaust gas temperature	540	°C

3 Thermodynamic analysis

The focus of the thermodynamic analysis was to achieve the maximum power efficiency of the gas-air system for selected technological structures of the air part. The following parameters were considered as the most important variables: air heat exchanger effectiveness, ε_{WSP}, turbomachinery polytropic efficiency, η_{pCi}, η_{pTi} (where η_p – polytropic efficiency, C – compressor, T – turbine, i – individual number of compressor or turbine), and the interstage coolers effectiveness, ε_{CHi}. A decision was made not to change the pressure drop value in individual heat exchangers (a constant value was assumed of $\Delta P_{hot} = \Delta P_{cold} = 4\%$). The values of the effectiveness of individual heat exchangers and of the polytropic efficiency of turbomachinery were varied according to the data presented in Tab. 2.

Table 2: Data of individual air part cases.

Variant	1	2	3	4	5
AHX effectiveness, %	80	82.5	85	87.5	90
Polytropic efficiency, %	85	86	87	88	89
Intercollers effectiveness, %	70	74	78	82	86
Pressure drop in AHX and ICi, %	4	4	4	4	4

The calculations were carried out for the following variants:

- gas-air system, where the working medium of the air part is not cooled;
- gas-air system, where the compressed medium is cooled with air.

If air is used as the cooling medium, the heat exchanger is the gas-to-gas type where the value of specific heat for both fluids is very low. Therefore the heat transfer coefficient is relatively low, too. Another factor that should be taken into consideration is the need to bring cooling air to the heat exchanging unit. This requires the use of the air fan. It is assumed for the needs of the thermodynamic analysis that the value of the cooling air mass flow is the same as the mass flow of compressed air. In order to deliver an appropriate amount of air, the fan has to be used. The fan power can be determined from the following relation:

$$N_{fan} = \frac{\dot{m}_{air_{cool}} \cdot \Delta P_{air_{cool}}}{\rho_{air_{cool}} \cdot \eta_{fan_a ir_{cool}}} \,, \tag{1}$$

where: $\eta_{fan_a ir_{cool}}$ – cooling air fan efficiency, %; $\Delta P_{air_{cool}}$ – increase in air pressure, Pa $\rho_{air_{cool}}$ – air density, kg/m^3; $\dot{m}_{air_{cool}}$ – air mass flow rate, kg/s.

Optimum thermodynamic parameters (in terms of the system energy efficiency) were determined for individual cases. The calculations were performed for the structure with no cooling (Variant A), with one interstage cooler (Variant B) and with two interstage coolers (Variant C). As the parameters listed in Tab. 2 rise, the optimum value of the mass flow of air sucked in by the air decreases (Fig. 2). However, the differences are slight. Depending on the computational variant, they vary in the range from 80.8 to 81.6 kg/s. The highest mass flow value is characteristic of the system with two interstage coolers. This is also the case for the optimum total pressure ratio – cooled systems achieve the highest value. For Variant C it is 8.78 (Fig. 2b). The higher the system parameters (changing for individual cases according to Tab. 2), the higher the total pressure ratio value. However, in the case of the optimum value of the pressure ratio, a greater non-linearity can be noticed here, compared to changes in the air mass flow.

Figure 3 presents the temperature value in individual points in relation to subsequent computational cases. As the parameters rise, the temperature at the air heat exchanger outlet decreases. The temperature values are the lowest in cooled systems, especially those with highly effective heat exchangers and with a high value of polytropic efficiency. The air temperature at the AHX outlet rises as the system parameters increase. The system with no interstage cooling is characterized by the lowest temperature. The exhaust gas temperature at the AHX outlet decreases as the system parameters rise. This effect is best seen for the cooled systems. The AHX inlet temperature rises monotonically in the cases analyzed for Variant A.

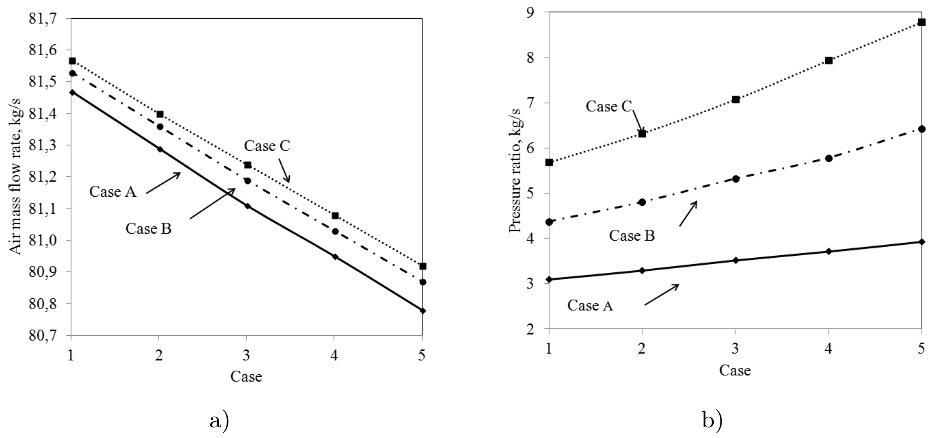
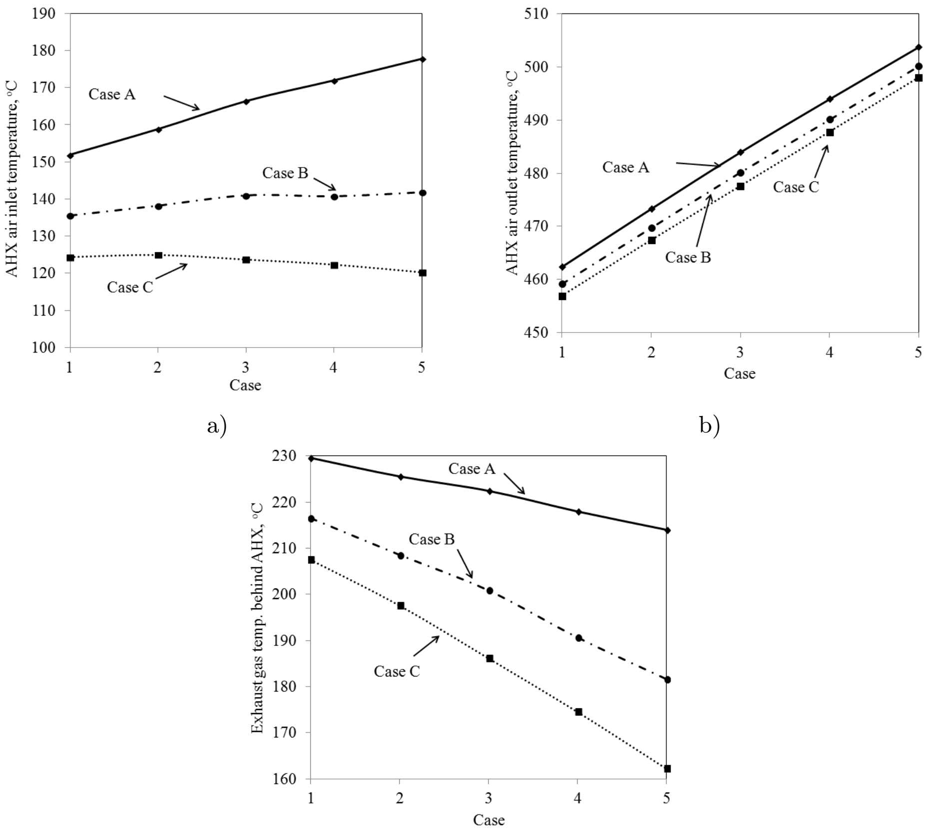


Figure 2: Power efficiency-wise optimum parameters of the air part: a) changes in mass flow; b) changes in total pressure ratio.

Another quantity that was determined was the variation of the temperature of compressed and cooling air for the analyzed interstage coolers. The results are shown in Fig. 4.

The highest value of power efficiency was achieved for the system with two interstage coolers. For case 5, it was higher than 44.12%, which means that the power efficiency of the air part exceeds 15.3% (Fig. 5). The mechanical power that can be achieved is 6.8 MW. In the case of systems with cooling, the calculations take account of the power needed to drive the cooling air fan.

The most important factor that affects the difference in generated power is the use of the fan transporting cooling air. For systems with two interstage coolers, two fans should be used, or one – but with power equal to that of the sum of two smaller units (assuming the same values of internal efficiency of the machines). The total power needed to drive the fans for the cases under analysis is presented in Tab. 3.

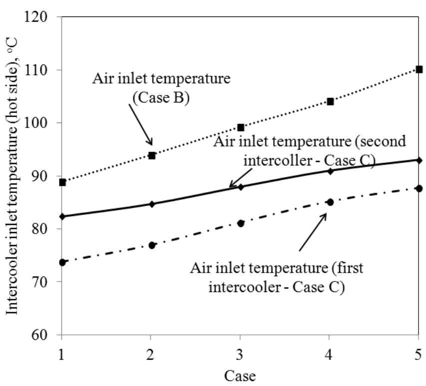


c)

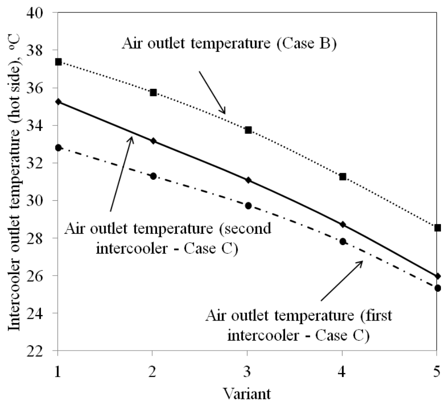
Figure 3: Temperature values at the AHX characteristic points: a) inlet temperature of air; b) outlet temperature of air; c) outlet temperature of exhaust gases.

Table 3: Power needed to drive fans in air-cooled systems.

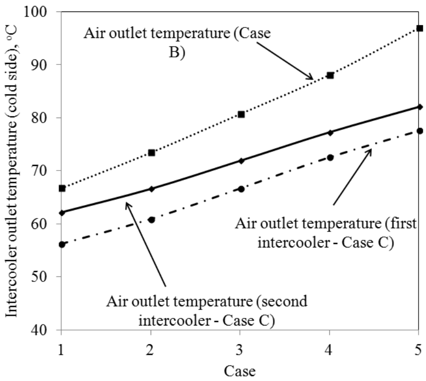
Variant		1	2	3	4	5
Power, kW	One cooler	344.2	343.5	342.8	342.1	341.4
	Two coolers	688.8	687.3	686.0	684.6	683.3



a)



b)



c)

Figure 4: Temperature values at the interstage cooler (CH) characteristic points: a) inlet temperature of compressed air; b) outlet temperature of compressed air; c) outlet temperature of cooling air.

4 Economic analysis

Discount methods, which belong to the most precise instruments of the financial analysis, were used to assess the project profitability. One of the basic economic indices is the net present value (NPV), defined as

$$NPV = \sum_{t=1}^{t=N} \frac{CF_t}{(1+r)_t} \ , \tag{2}$$

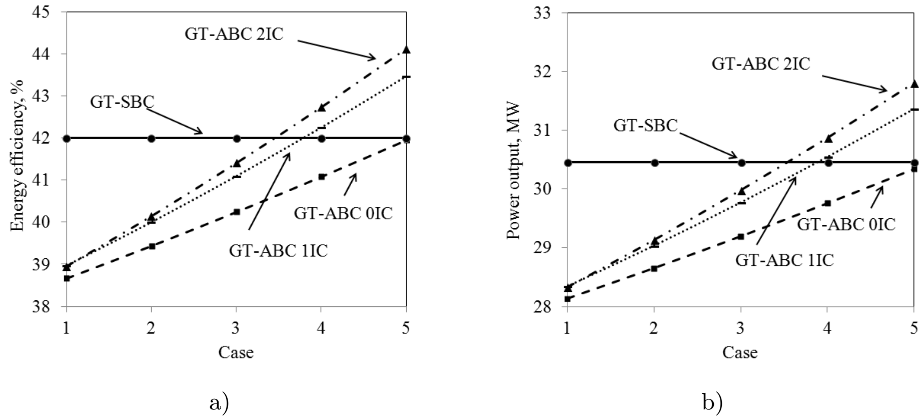


Figure 5: Comparison between gas-air and gas-steam systems: a) power efficiency; b) generated power.

where: CF_t – cash flows at time t, r – discount rate, t – subsequent year since the system construction was started, N – total number of periods, years.

Market prices of gas turbines which are currently in use may be found in the *Gas Turbine World Handbook* [5]. Beside the prices, the basic data concerning a specific machine may be found there too, e.g. power output, efficiency, working fluid mass flow or the outlet gas temperature. It is also possible (if not necessary) to convert the outlay values valid for previous years for the current year using relevant updating indices. The most popular is the chemical engineering plant cost index (CECPCI) [6], which is the weighted average of indices calculated separately for different groups of components (e.g. heat exchangers). The values of this index (based on the CEPCI = 100 in years 1957-59) and of the partial indices are published every month in the *Chemical Engineering* magazine.

In the case of heat exchangers, the purchase cost of the machines was determined based on the system mass. Following the recommendation obtained from [7], a decision was made to employ the indices that define the use of a given steel type expressed in USD/kg of steel. Due to the fact that the air heat exchanger is operated under high temperature conditions, its structure has to be made of steels with a high thermal strength. According to [8], the cost of ducts, control dampers and stack was taken into account in the estimation of the total purchase cost.

The investment expenditures on the air part include mainly the cost of purchase of the air compressor, the air turbine and the air heat exchanger. The turbomachinery purchase cost was determined based on the update of the relationships from [9,10]. The remaining assumptions for economic analysis were taken from [11,12]. Figure 6 presents investment expenditures on the purchase of the air system with one interstage cooler. The investment expenditures on a gas-steam system with a GT10 turbine obtained from [2] are also marked. It should be noted that the higher outlay value in the case of cooled systems results from a considerable rise in the expenditures on the compressor. The compressor purchase cost strongly depends on the pressure ratio.

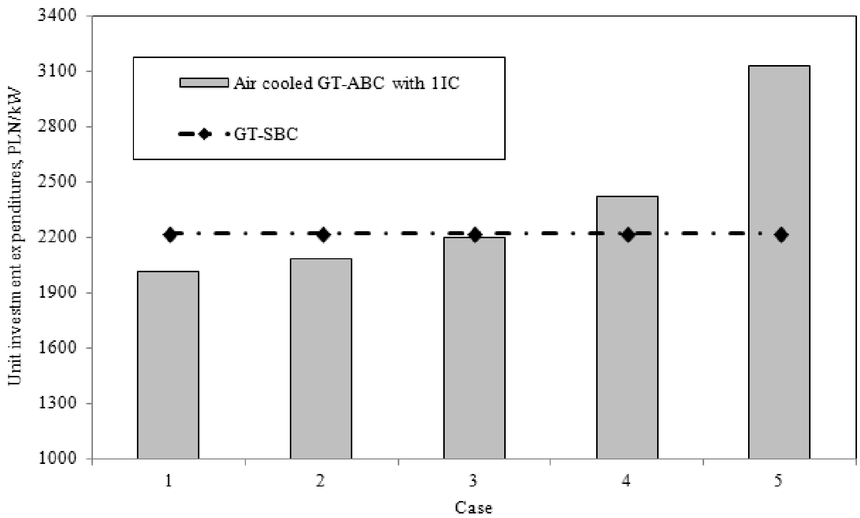


Figure 6: Investment expenditures on the individual gas-air and gas-steam systems.

The marginal cost of purchase of electricity for the analyzed gas-air systems cooled with air are presented in Fig. 7. The marginal purchase cost for the gas-steam system is also marked. It amounts to about 348 PLN/kW. The gas-air systems based on case 4 are characterized by a similar price. The use of a gas-air system from case 5 will give better results in terms of obtaining a lower marginal cost of purchase of electricity. However, it has to be remembered that, in the case of this system, the air heat exchanger and the compressor interstage cooler are characterized by relatively high values of effectiveness (90% and 86%, respectively). A system with such highly effective heat exchanging components and with turbomachinery with polytropic efficiency of 89% may, structurally, be difficult to realize.

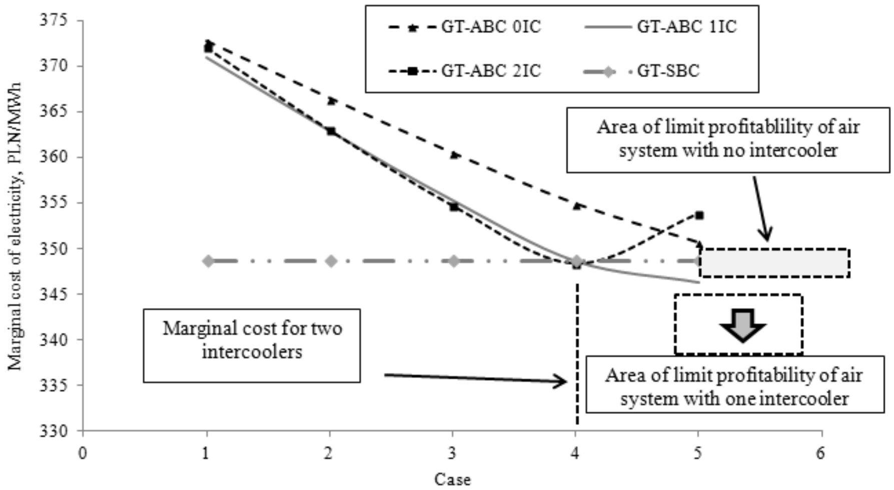


Figure 7: Marginal cost pf electricity for systems with air cooling.

5 Conclusions

The article presents a comparative analysis of gas-air and gas-steam systems. The comparison was thermodynamic and economic in nature. The most important effect is the obtained value of the final purchase cost of electricity. In the case of the analysis of systems based on the gas turbine with a power capacity of 25 MW, an area was found in which air-cooled gas-air systems could be competitive compared to commercially available gas-steam systems. Better results can be obtained for systems integrated with gas turbines with a lower power capacity [13]. This is mainly caused by two factors. Firstly, gas turbines in such installations are usually characterized by lower values of power efficiency. Secondly, in the case of systems with a higher power capacity (above 25–30 MW), it will be difficult for gas-air systems to be competitive due to the size of the air heat exchanger and intercoolers (especially when it comes to air-cooled systems). Due to the specific medium such as air, these components are characterized by a large heat transfer surface area which substantially impedes the design studies. Attention should also be drawn to the mass of the heat exchangers and the higher investment expenditures resulting thereof. Any extra expenditure has a substantial impact on the final marginal cost of electricity generated in a given system.

Acknowledgments The results presented in this paper were obtained from research work cofinanced by the National Centre of Research and Development in the framework of Contract SP/E/1/67484/10 – Strategic Research Program – Advanced Technologies for obtaining energy: Development of a technology for highly efficient zero emission coal-fired Power units integrated with CO_2 capture.

Received 18 June 2014

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