Selection of the air heat exchanger operating in a gas turbine air bottoming cycle

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Abstract   A gas turbine air bottoming cycle consists of a gas turbine unit
and the air turbine part. The air part includes a compressor, air expander
and air heat exchanger. The air heat exchanger couples the gas turbine to
the air cycle. Due to the low specific heat of air and of the gas turbine
exhaust gases, the air heat exchanger features a considerable size. The
bigger the air heat exchanger, the higher its effectiveness, which results in
the improvement of the efficiency of the gas turbine air bottoming cycle.
On the other hand, a device with large dimensions weighs more, which may
limit its use in specific locations, such as oil platforms. The thermodynamic
calculations of the air heat exchanger and a preliminary selection of the
device are presented. The installation used in the calculation process is
a plate heat exchanger, which is characterized by a smaller size and lower
values of the pressure drop compared to the shell and tube heat exchanger.
Structurally, this type of the heat exchanger is quite similar to the gas
turbine regenerator. The method on which the calculation procedure may
be based for real installations is also presented, which have to satisfy the
economic criteria of financial profitability and cost-effectiveness apart from
the thermodynamic criteria.

Keywords: Air heat exchanger; Air bottoming cycle; Gas turbine air bottoming cycle;
Gas turbine

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1   Introduction

The gas-air system is composed of a simple gas turbine unit and the air turbine part. The gas-air system is coupled together by means of the air heat exchanger which plays an important part in the entire installation. In order to improve the gas-air system efficiency, a highly-efficient air heat exchanger (featuring a high thermal effectiveness) should be used. Due to the fact that the gas-air system is conceptually similar to the gas-steam system (combined cycle power plant), the air heat exchanger that couples the gas and air parts together can be treated as a heat recovery air generator. The kind of the air heat exchanger selected for the system has an impact on the energy efficiency of both the gas turbine and the air part. In the case of the system analyzed in this paper, a decision was made to implement a plate heat exchanger PHE, as it features a smaller value of the total heat transfer surface area compared to the shell and tube installations [1]. Another important factor in the use of heat exchangers in power systems is the pressure drop value. Thermal calculations indicate that if the pressure losses of the order of 5–6% arise either on the gas or air side, the system energy efficiency may decrease by about 1% [2].

A counter flow heat exchanger case is assumed for the calculations. From the thermodynamic point of view, efforts should be made to achieve small temperature changes in the heat exchanger (the temperature difference between exhaust gases at the outlet and air at the inlet and between exhaust gases at the inlet and air at the outlet). This improves the energy efficiency considerably. However, it also determines large pressure drops and significant values of the total heat exchange area.

For the given polytropic values occurring in the air part turbomachinery, it is possible to define the minimum value of the air heat exchanger effectiveness for which the gas-air system is justified thermodynamically, i.e., its power efficiency is better than that of a standalone gas turbine unit. This, however, is sometimes not a good solution in terms of the system economy. Knowing the heat transfer surface area and assuming a certain material that could be used to make this kind of devices, the value of effectiveness at which the gas-air system equipped with such an heat exchanger is justified economically can be found based on the material unit prices, using basic cost-effectiveness indices, as well as the prices of the air part turbomachinery and of the gas turbine unit.

The modeling of plate exchangers that could be applied in the system under consideration was also discussed in [3], where a mathematical model

is developed in algorithmic form for the steady-state simulation of gasketed plate heat exchangers with generalized configurations. The configuration is defined by the number of channels, number of passes at each side, fluid locations, feed connection locations and the type of channel-flow. The main purposes of this model are to study the configuration influence on the exchanger performance and to further develop a method for configuration optimization. Plate exchangers were also analyzed in [4]. Authors present a methodology for the design of compact plate heat exchangers where full pressure drop utilization is taken as a design objective. The methodology is based on the development of a thermohydraulic model that represents the relationship between the pressure drop, heat transfer coefficient and exchanger volume.

2 System under analysis

A simple gas-air system, with a schematic presented in Fig. 1 was chosen for the analysis. This system features no interstage cooler. It is assumed that depending on the type of operation it has to be possible for the gas turbine exhaust gases to bypass the air heat exchanger AHX and to be carried away to the stack. A system of shut off dampers is used in exhaust gas ducts. The shut off dampers pass the exhaust gases to the stack or to the AHX (in the case of the air bottoming cycle operation). The structure under consideration is a two-shaft system.

Depending on the type of the powered machine, several configurations of the gas-air system can be distinguished: This cycle can be operated as power generation unit or installation itended to power working machinery and hybrid system featuring both the generator and another working machine.

3 Calculations

The calculations were carried out based on authors own algorithm [5]. The computational model was the model of ideal gases, where specific heat value is the function of temperature. Developed methodology is useful for the selection of air heat exchangers for the gas-air systems without interstage coolers. It should be noted that economic criteria are taken into account.

In order to perform thermodynamic calculations, it is assumed that the ambient conditions are as follows: temperature: $T_{amb} = 288$ K, pressure: $P_{amb} = 101.3$ kPa, and relative air humidity about $\varphi = 60\%$. The exhaust
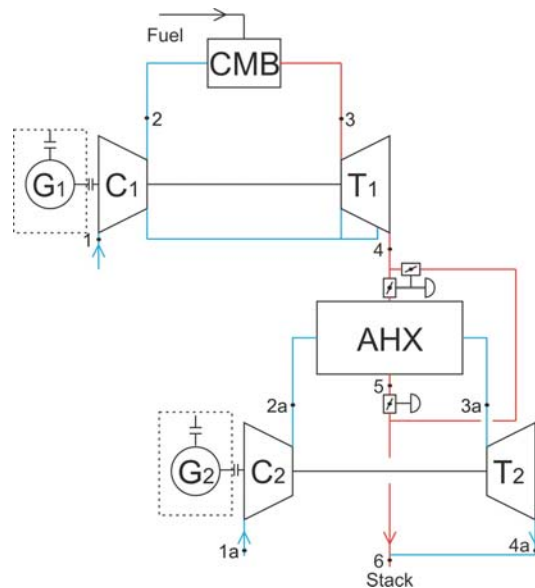
Figure 1. Reference gas turbine air bottoming cycle: AHX – air heat exchanger, CMB – cpmbustor, C_1 and C_2 – gas and air part compressors, G_1 and G_2 – gas and air part generators, T_1 and T_2 – gas and air turbines.

gas molar composition is as follows: nitrogen N_2 – 75.76%, oxygen O_2 – 13.56%, carbon dioxide CO_2 – 3.28%, and water (g) H_2O – 7.4%.

A decision was made to carry out calculations for unit mass flows of exhaust gases and air. Research results indicate that, in terms of power efficiency, the optimum value of the air mass flow at a known rate of flow of the gas turbine exhaust gases is more or less equal to the exhaust gas mass flow [5,6].

One of the objectives of calculations is to determine the overall heat transfer coefficient. This parameter is very important due to its relationship with the heat transfer surface area of AHX. In order to find the heat transfer coefficient and, consequently, to determine the heat exchange surface area, it is necessary to define a series of physical properties of the media

$$\alpha = f\left(\lambda, c, \rho, \eta, a, \Delta t, ...\right) \ . \tag{1}$$

The most important of these are: thermal conductivity (λ), specific heat (c), density (ρ), dynamic viscosity (η), thermal diffusivity (a), temperature difference (Δt), and other parameters [7-8]. Specific heat was determined based on the procedure in [9].

Nusselt number for the plate heat exchangers is expressed in the following form:

$$\mathrm{Nu} = 0.022\sqrt{\xi_0}\,\beta\,\beta_t\,\mathrm{Re}^{0.825}\mathrm{Pr}^{0.54}\ , \tag{2}$$

where ξ_0 is the flow resistance ratio, β and β_t are the turbulence damping ratio due to the proximity of the duct walls, and forced turbulence ratio (function of Re), respectively. Prandtl number results from kinematic viscosity and the thermal diffusivity, $\mathrm{Pr} = \frac{\nu}{a}$, and Reynolds number is defined as $\mathrm{Re} = \frac{wd_e}{\nu}$, where d_e is the equivalent diameter, w is the velocity.

Heat transfer and overall heat transfer coefficient are defined as

$$\alpha = \frac{Nu\,\lambda}{d_e}\ , \tag{3}$$

$$k = \frac{1}{\frac{1}{\alpha_{exh.-gas}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{air}}}\ , \tag{4}$$

where λ is the thermal conductivity of the baffle, δ is the baffle thickness, $\alpha_{exh.-gas}$, α_{air} are the exhaust gas and air heat transfer coefficient, respectively.

The heat exchange surface area is derived from the Peclet equation

$$A = \dot{Q}/\left(k\Delta t_m\right)\ , \tag{5}$$

where $\dot{Q}$ is the heat flux absorbed by the working medium, and Δt is the log mean temperature difference.

4   Results and discussion

In order to develop a method for selecting appropriate air heat exchanger to gas-air system, a few assumptions were made. In calculations the 312.5 kW gas turbine has been assumed, of which the exhaust mass flow rate from the gas turbine equals $m_{mexh.-gas} = 1$ kg/s. Figure 2 shows the relationship between air heat exchanger (AHX) effectiveness and air bottoming cycle (ABC) efficiency (energy as well cycle efficiency). Calculations were conducted for different values of pressure drop (on the air as well as on the exhaust gas side). It should be noted that polytropic efficiency of air part compressor and expander is equal $\eta_{C2} = \eta_{T2} = 86\%$. Gas turbine (GT) energy efficiency is assumed at the level of 38%

ABC energy efficiency is defined as

$$\eta_{e\_ABC} = \frac{N_{mT2} - N_{mC2}}{m_{exh-gas}\, c_{exh-gas}\left(T_{exh-gas\_IN} - T_{ref}\right)} \ , \tag{6}$$

where $m_{exh-gas}$ is the exhaust gas mass flow, $c_{exh-gas}$ is the ehaust gas specific heat, N_{mT} is the turbine mechanical power output, N_{mT} is the compressor mechanical power, T_{ref} is the reference temperature (288 K), $T_{exh-gas\_IN}$ is the inlet exhaust gas temperature to AHX, and $T_{exh-gas\_OUT}$ is the outlet exhaust gas temperature from AHX.

Then ABC cycle efficieny is defined as follows:

$$\eta_{c\_ABC} = \frac{N_{mT2} - N_{mC2}}{m_{exh-gas}c_{exh-gas}\Delta T_{exh-gas}} \ , \tag{7}$$

where

$$\Delta T_{exh-gas} = \left(T_{exh-gas\_IN} - T_{exh-gas\_OUT}\right) \ . \tag{8}$$
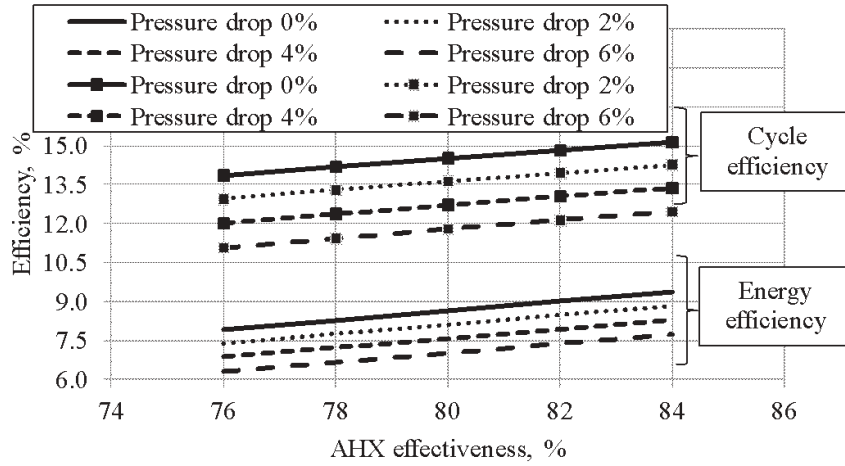
Figure 2. Increase in efficiency as a function of AHX effectiveness for various pressure drops.

The computed temperatures of exhaust gas and air are shown in Figs. 3 and 4, respectively. There are no big differences between temperatures of air in AHX inlet. Inlet air temperature results from system the optimization in terms of energy efficiency where the most important issue is to find a correct value of pressure ratio for each analyzed cycle(compressor C2). The higher
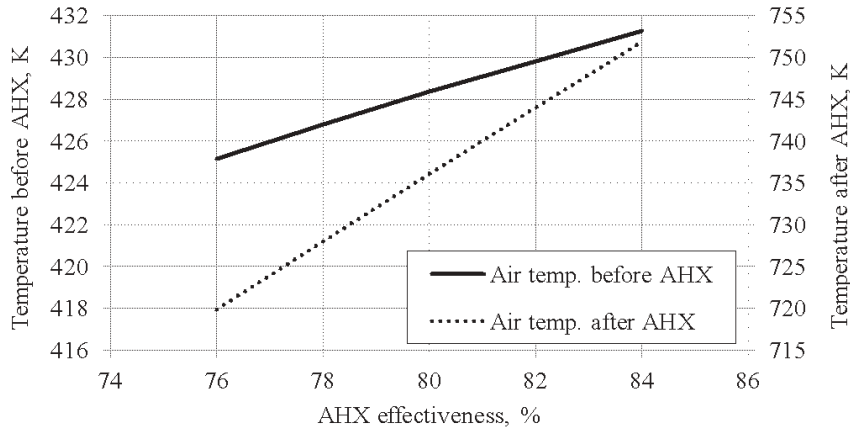
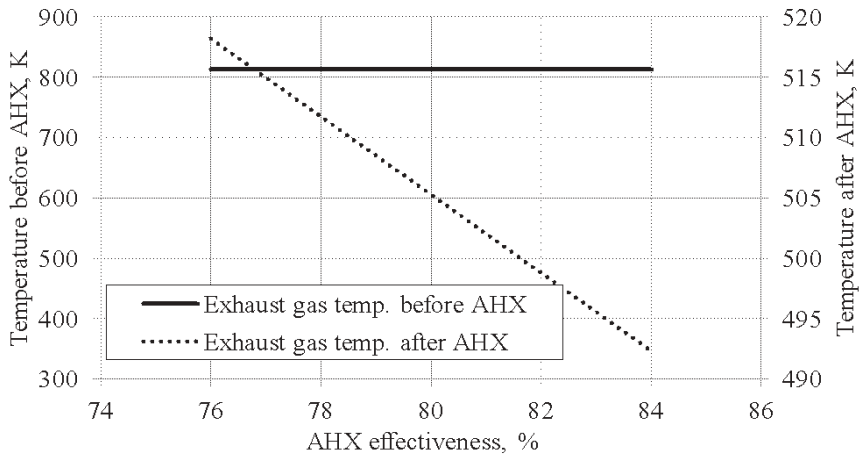Figure 3. Temperature of air for different values of AHX effectiveness.

Figure 4. Temperature of exhaust gas for different values of AHX effectiveness.

AHX effectiveness, the higher value of outlet air temperature and more power can be obtained in air bottoming cycle expander T2.

In full load of GT there is a constant AHX inlet exhaust gas temperature. The higher AHX effectiveness, the lower outlet exhaust gas temperature can be achieved. There is also no big difference between individual physical properties of each medium for analyzed range of AHX effectiveness. Exam-

ple plots present distribution of average density (between inlet and outlet), dynamic viscosity, specific heat and thermal conductivity for different AHX effectiveness (Figs. 5 and 6).
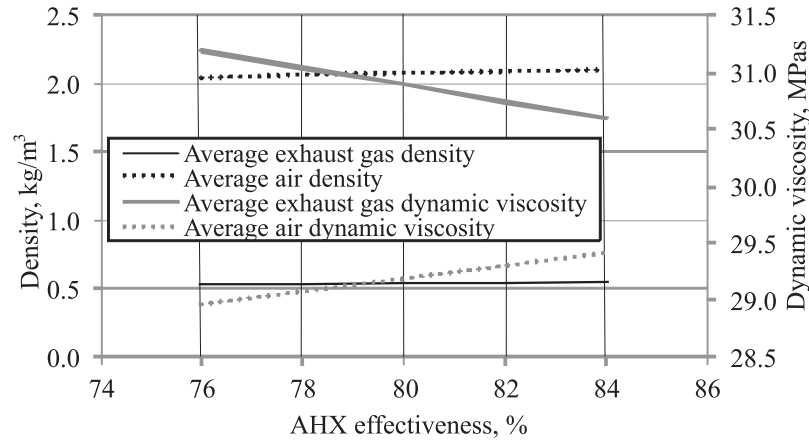

Figure 5. Average density and dynamic viscosity of exhaust gas and air for different values of AHX effectiveness.
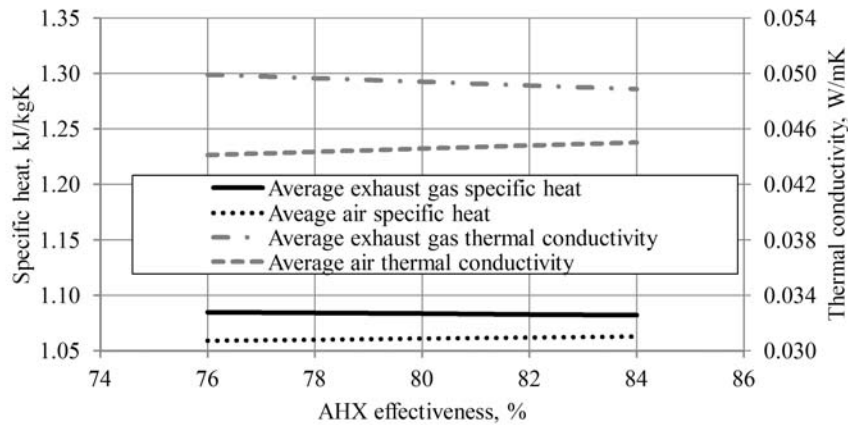

Figure 6. Average specific heat value and thermal conductivity of exhaust gas and air for different values of AHX effectiveness.

Slightly better heat transfer conditions can be achieved with higher values of the Reynolds number. In calculations process, the Reynolds number reaches a value of more than ten thousands (exhaust gas as well as air side). The

AHX effectiveness for different values of Reynolds number is presented in Fig. 7. Figure 8 shows the overall heat transfer coefficient for different values of AHX effectiveness.
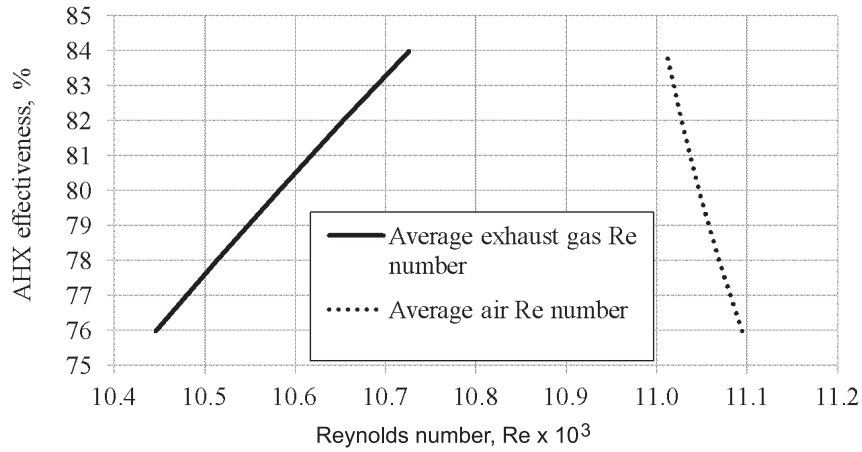

Figure 7. AHX effectiveness for different values of Reynolds number (exhaust gas as well air side).
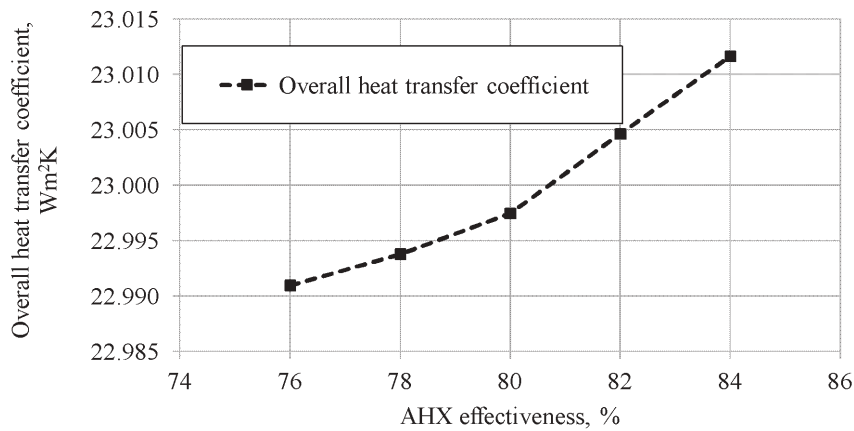

Figure 8. Overall heat transfer coefficient for different values of AHX effectiveness.

It is obvious that the less effective the AHX is the smaller is the device. But on the other hand there is a relationship between AHX effec-

tiveness value, the GT-ABC power output, energy efficiency and the net present value (NPV) of analyzed installation. Considering the GT-ABC as a power unit and assuming the price of 1 MWh produced from gas-air cycle equal 250 PLN/MWh (76.45 USD/MWh) and 240 PLN/MWh (73.39 USD/MWh) respectively, NPV, the internal rate of return (IRR) and the the simple pay back (SPB) values were determined. A chart with marked individual net present values (Fig. 9), internal rate of return (Fig. 10) and simple pay back period (Fig. 11) for a given system was developed for the adopted assumptions concerning the air part turbomachinery (the polytropic efficiency value).

Additional data and assumptions for calculations:

- The annual operation time under a rated load have been determined at 7884 hours per year. The current price of gas has been taken from the PGNiG tariffs [10].

- 25 years of service life of the plant.

- One year needed for the plant to be constructed.

- Fuel low heating value – 36 MJ/Nm^3.

- System auxiliary needs – 1%.

- Fees for using the environment are taken from [11].

- The total investment expenditures for the GT-ABC have been determined based on [13–15], and information obtained from the power industry (T. Chmielniak and D. Czaja: Personal communication, 2012),

- In order to perform the economic analysis, the current value of the discount rate have been calculated from the following dependence:

$$\alpha_d = r_1 i_{r1} + r_2 \left(1 - I_{tax}\right) i_{r2} + r_3 \left(1 - I_{tax}\right) i_{r3} \, , \tag{9}$$

where $r_1 = 0.2$ is the share of equity capital, $r_2 = 0.8$ is the share of commercial credit, $r_3 = 0$ is the share of soft loan (preferential mortgage), i_{r1}, i_{r2} and i_{r3} are credit interest rates, respectively, I_{tax} is the income tax.

The AHX effectiveness border can be determined according to the electricity price per MWh. This is the point where NPV for each case achieved zero. In the example presented (Fig. 9) for electricity price equal to 73.39 USD/MWh the justified value for AHX effectiveness (from energy as well economy point of view) is between 78–80%. Bigger values of AHX effectiveness results, of course, in bigger NPV value, but installation heat exchanger is characterized by much bigger heat transfer surface area and weight.
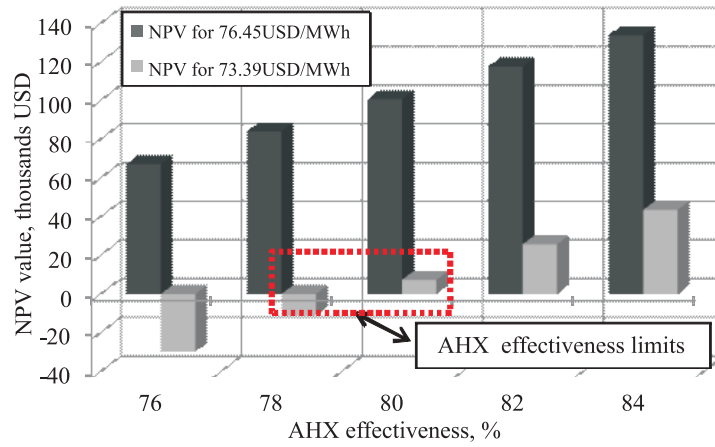
Figure 9. NPV values as the function of AHX effectiveness.

Three materials (steel grades) in appropriate proportions were choosen for the construction of the air heat exchanger. Considering the density of individual materials, it is possible to estimate the mass of the entire installation. Table 1 shows comparison between heat duty, weight, energy efficiency and power output for different values of AHX effectiveness.
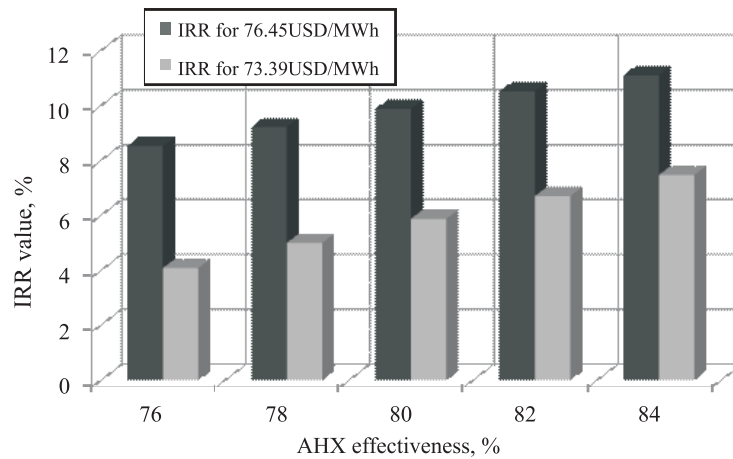

Figure 10. IRR values for different AHX effectiveness.

It has been seen the 80% effectiveness AHX features with weigh almost 2.8 Mg. This device can ensure, for the analyzed cycle with assumed poly-
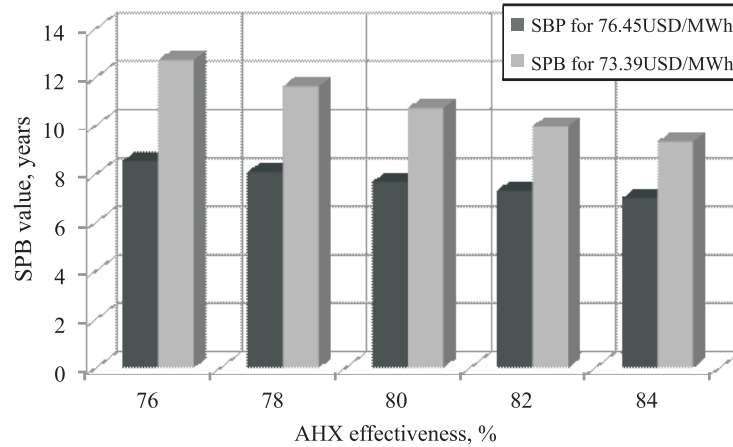
Figure 11. SPB values as the function of AHX effectiveness.

tropic efficiency of turbomachinery, the increase in power output which reaches almost 15.5% more than the stand-alone GT unit. This also results in increase in energy efficiency (about 8.7%).

Table 1. Weight and heat duty of AHX.

AHX effectiveness [%]	Heat duty [kW]	Weight, [Mg]	Increase in energy efficiency [%]	Increase in power output [%]
76	319.38	2.22	7.93	14.18
78	326.19	2.48	8.29	14.83
80	333.00	2.79	8.66	15.48
82	339.84	3.17	9.03	16.14
84	346.61	3.64	9.40	16.80

5   Conclusions

The selection of heat exchangers must not be based solely on purely thermo-dynamic criteria, and this is so despite the fact that due to them the system can reach higher values of both power efficiency and additional mechanical power (via a higher temperature at the air expander inlet). The method has

been presented, on which the calculation procedure may be based for real installations which, apart from the thermodynamic criteria, has to satisfy the economic criteria of financial profitability and cost-effectiveness. The calculations are focused on the plate exchanger, which features smaller dimensions and smaller pressure drops compared to shell and tube exchangers.

The selection of the gas-air system air heat exchanger must also result from the potential application for which the installation is intended. The entire gas-air system (both the gas turbine and the air components) can work as a power unit, as a system intended to power working machinery or as a combination of these two options. It can also be intended as a propelling unit for vessels and ships. In these cases, the criterion of the appropriately low value of the system mass becomes especially important. However, an excessive reduction in mass may have a dramatical adverse impact on the energy efficiency of the entire system.

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