Open Access Journal
Journal of Power Technologies 93 (5) (2013) 279–286
journal homepage:papers.itc.pw.edu.pl

Operation of a gas turbine air bottoming cycle at part load

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Abstract

The purpose of this study is to analyze the performance characteristics of a gas turbine air bottoming cycle operating under part-load conditions. In terms of energy efficiency, the most effective option for each installation is to operate under a nominal load. However, different applications have different needs. Marine gas turbines, for instance, should feature a high value of efficiency across a wide range of load. There are numerous other examples of installations which most of the time operate at generating power levels significantly lower than maximum capacity.

This paper presents a two-shaft gas turbine air bottoming cycle. The gas turbine is coupled to the air part by means of an air heat exchanger. This configuration allows the gas turbine to operate at a nominal load while the cycle power output is regulated by the air turbine partial load. However, due to the fact that the air turbine to gas turbine mechanical power output ratio is about 0.17...0.20, it is necessary to consider a variant where the gas turbine also operates under a partial load. Selected results are summarized and compared with those obtained for a standalone gas turbine unit.

Keywords: part-load operation, air bottoming cycle, gas turbine air bottoming cycle, gas turbine

1. Introduction

Current-generation combined cycles are usually gas-steam cycles where the gas turbine exhaust gases with outlet temperature in the range of 500...550°C give up heat in a waste heat boiler [1] (heat recovery steam generators) which generates water vapour which is then expanded in a steam turbine. Another way of recovering waste heat after the gas turbine is to add another Brayton cycle with air as the working agent. The main advantage of such an installation is the lack of complex structures that are inherent in a gas-steam system. This mainly concerns the fact that the installation is devoid of any condenser or a complicated cooling system. After expansion in the air cycle turbine, the air may be discharged into the atmosphere. Construction of this type of system may prove energy-efficient due to advances in turbomachinery construction, especially in the field of improvement to blade profiles and sealing.

The gas-air (air bottoming cycle) system was patented by W. Farrell of the General Electric Company [2]. At the same time Alderson presented an air system co-operating with a power plant integrated with a coal gasification installation [3]. In 1995 Kambanis found that the power efficiency of the GE LM2500 turbine rose from 36% to 47% after it is upgraded with an air system [4]. In 1996 Bolland recorded an improvement in the efficiency of the

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LM2500 PE turbine to 46.6% after an air turbine system was used [5]. Upgrading a gas turbine plant with an air turbine installation makes it possible to generate more power. Poullikkas estimated an 18...30% increase on average in the power output of the analyzed gas-air systems. The rise in power output is accompanied by an average increase in efficiency of ten percentage points [6]. In the case of more complicated technological structures of gas-air systems, which involve cooling of the working agent, it is necessary to determine the type of air coolant. In 1996 Najjar presented gas-air system calculations using interstage coolers [7]. The average rise in power output was in the region of 30%. Use of the gas turbine air bottoming cycle system in heat engineering was also studied by [8]. If a combined gas-air heat and power plant is to be used, the choice of structure depends on the information concerning the anticipated installation size. Air systems may also find application in carbon capture and storage installations [9].

A gas turbine air bottoming cycle features much more effective performance under varied loads compared to a standalone gas turbine unit [4]. This is especially important in the case of systems operating for considerable periods of time under loads lower than nominal, such as gas turbines propelling vessels, ships or installations powering working machinery.

Two options of the system of a gas turbine with an air part are analyzed in this paper. The power control range for both configurations is presented and the value of energy efficiency for individual variants is determined. In the case of the system power controlled by means of the air part only, the calculations are performed parametrically for various values of air temperature before the air expander.

2. System under analysis

A simple gas turbine air bottoming cycle is presented in Fig. 1. This systems features no interstage cooling of compressor C2. 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 exhaust gases to the stack or to the AHX (in the case of the

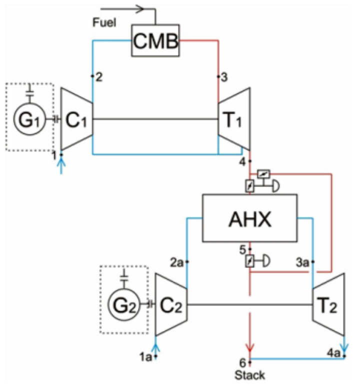

Figure 1: Reference gas turbine air bottoming cycle

air bottoming cycle operation). The structure under consideration is a two-shaft system. The air part expander drives the compressor C2 and, depending on the configuration, the generator or another working machine, e.g. a natural gas compressor. A decision was made to combine the exhaust gas system with the air system and to use a common stack. If the gas turbine operates with no air turbine, the AHX has to be separated from hot exhaust gases. For this purpose, an additional damper is used after the AHX. Both air and water can be used as coolant in the interstage cooler. In these calculations it is assumed that the cooling agent is air.
Depending on the type of machine powered, there are a number of gas-air system configurations:

- gas-air system for power generation (two generators powered by the gas turbine and air turbine),
- gas-air system intended to power working machinery only (Fig. 2),
- gas-air system as a hybrid system featuring both the generator and another working machine (Fig. 3).

The figures present an example configuration with a gas compressor as the working machine.

3. Calculations

An air turbine system may be evaluated by comparing values of appropriately defined efficiency. This definition may be determined by means of an energy analysis of the system illustrated in Fig. 4.

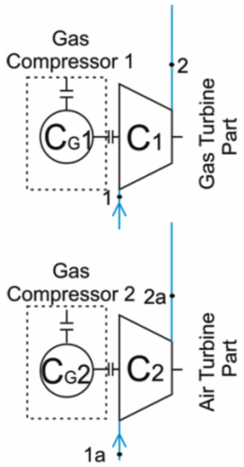

Figure 2: Gas-air system driving working machinery—a variant with two compressors

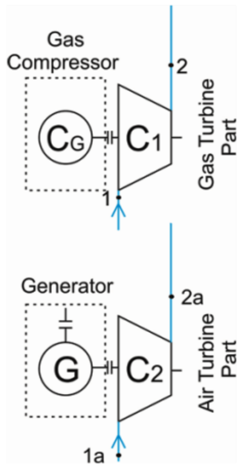

Figure 3: Gas-air system driving working machinery—a variant with a compressor and generator

In the case of a standalone gas turbine operation, the important parameter is energy efficiency defined by the following formula:

$$\eta_{elGT} = \frac{N_{elGT}}{\dot{m}_f LHV} \tag{1}$$

The definition of the gas turbine power efficiency can be determined analyzing the balance boundary BB_{GT}; in this case:

$$\eta_{eGT} = \frac{N_{mGT}}{\dot{m}_f LHV} \tag{2}$$

and:

$$\eta_{elGT} = \eta_{eGT}\eta_{gGT} \tag{3}$$

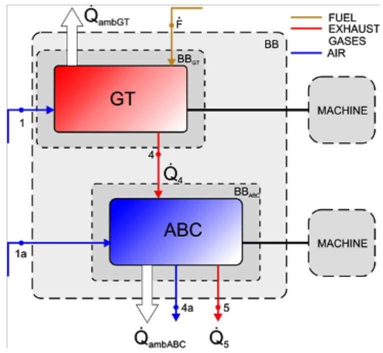

Figure 4: Gas-air system block diagram with balance boundaries

were:

$$\eta_g = \frac{N_{elGT}}{N_{mGT}} \tag{4}$$

The analysis of a system composed of an air turbine and a gas turbine should aim to maximize the generated mechanical (electrical) output compared to the chemical energy of fuel. For this reason, balance boundary BB is introduced, which defines the system power efficiency:

$$\eta_e = \frac{N_{mGT} + N_{mABC}}{\dot{m}_f LHV} \tag{5}$$

By adopting some simplifying assumptions it is possible to determine, in a very easy manner, the dependence of the energy efficiency of the entire system on the energy efficiency of the gas turbine and the air turbine systems. The dependence takes the following form:

$$\eta_e = \eta_{eGT} + \eta_{eABC}(1 - \eta_{eGT}) \tag{6}$$

The most important simplifying assumption is that there are no radiation heat losses from either the gas or the air turbine. The power efficiency of the air turbine system is defined as follows:

$$\eta_{eABC} = \frac{N_{mABC}}{\dot{Q}_4} \tag{7}$$

where: $\dot{Q}_4$—the heat of cooling exhaust gases to the reference temperature.

In order to simplify the analyses, it is assumed in the calculations presented in this paper that $\dot{Q}_4$ corresponds to the cooling of exhaust gases to the temperature of 15°C, and the water vapour contained in exhaust gases is not condensed.

When analyzing cogeneration systems, it may be essential to determine cycle efficiency. Cycle efficiency is defined as follows:

$$\eta_{cGT-ABC} = \frac{N_{mABC}}{\dot{Q}_4 - \dot{Q}_5} \tag{8}$$

The operation of the gas-air system under varied loads may be considered in two variants:

- gas turbine operates under a nominal load and the system is regulated by the air part,
- gas turbine operates under varied loads. The power output of the air system, and consequently of the entire gas-air unit, is then conditioned by the gas turbine expander outlet parameters (temperature, mass flow).

If the system is controlled only by the air part, there may be a problem if there is a need to lower the power output to below the gas turbine nominal value. If there is a need to lower the load further, an option should be considered of controlling the system by changing the performance characteristics of the gas turbine unit.

In the first case, the analyzed gas-air system features a gas turbine operating under a nominal load, which is regulated by the air part. The GT10 gas turbine was used for the calculations [10]. The turbine was fitted with an air part which was most beneficial in terms of power efficiency and which powered a working machine, e.g. an additional gas compressor in the natural gas compression station [11]. In order to carry out simulations under varied loads, the compressor universal characteristics were used [12–16]. In order to determine the temperature before the air expander, the value of inlet pressure had to be found according to the following dependence:

$$p_{3a} = \frac{m_a \sqrt{T_{3a}}}{\kappa_x \cdot A \cdot c} \tag{9}$$

where: T_{3a}—temperature at the AHX outlet; A—surface area of the turbine inlet stator flow; c—constant; κ_x—constant depending on the gas composition; m_a—air mass flow.

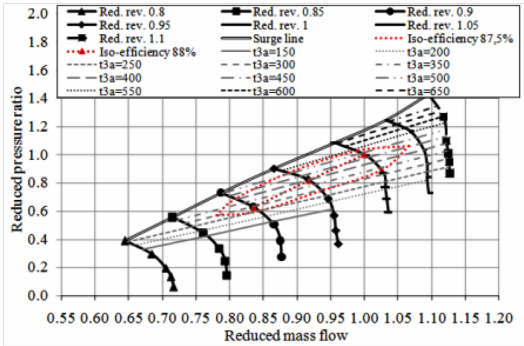

Figure 5: The compressor example characteristic with temperature values (in °C) before the expande

The temperature after the air expander can be determined using the following dependence:

$$T_{4a} = T_{3a} - \eta_{iT} T_{3a} \left[1 - \left(\frac{P_{4a}}{P_{3a}} \right)^{\frac{\kappa-1}{\kappa}} \right] \tag{10}$$

where: κ—adiabatic exponent; η_{iT}—isentropic expander efficiency.

Fig. 5 presents an example characteristic of a compressor operating at a nominal polytropic efficiency of 88% and 9,000 rpm. The isotherms which determine the air expander inlet temperature are also marked.

The reduced mass flow and reduced efficiency of the compressor are due to the reduced pressure ratio (β_{red}) and reduced value of rotational speed (n_{red}):

$$\frac{m_a \sqrt{T_{1a}}}{P_{1a}} = f(\beta_{red}, n_{red}) \tag{11}$$

$$n_{red} = f(\beta_{red}, n_{red}) \tag{12}$$

where the value of the reduced mass flow is illustrated by the following dependence:

$$\frac{m_a \sqrt{T_{1a}}}{P_{1a}} = m_{red} \tag{13}$$

In the equations above: m_a—mass flow through the compressor, kg/s; T_{1a}—air temperature at the compressor inlet, K; P_{1a}—air pressure at the compressor inlet, Pa.

The individual reduced values are defined as follows:

$$m_{red} = \frac{m}{m_{des}} \tag{14}$$

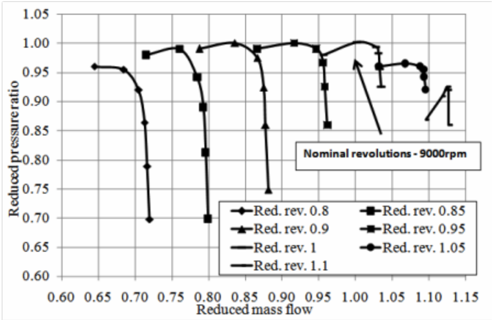

Figure 6: Example characteristic of reduced pressure ratio depending on reduced mass flow

$$\beta red = \frac{\beta - 1}{\beta_{des} - 1} \tag{15}$$

$$\eta_{red} = \frac{\eta}{\eta_{des}} \tag{16}$$

$$\eta_{red} = \frac{n}{n_{des}} \sqrt{\frac{RT_{des}}{RT}} \tag{17}$$

where subscript *des* refers to design parameters.

Compressor efficiency changes according to varying loads, which impacts the efficiency of the gas-air system under analysis. Fig. 6 shows the change in compressor efficiency depending on reduced values of the mass flow.

4. Results and discussion

As a result of the calculations performed, a set of the gas-air system characteristics of performance under varied loads was presented both for the case when the system was controlled exclusively by the air part and for the case when the gas turbine unit operation was regulated by a reduction in loads.

4.1. Air part control of the system power output

An example GT-ABC energy efficiency characteristic depending on the mass flow for different values of temperature before the expander is shown in Fig. 7. A similar characteristic was developed for the air system power output. It is presented in Fig. 8. Air heat exchanger efficiency is assumed at ε_{WSP}=0.84, which corresponds to the kA product of approximately 440 kW/K (k—overall heat transfer coefficient, W/m^{2}K; A—heat transfer surface area of the air AHX, m^2). Fig. 9 presents the dependence of the

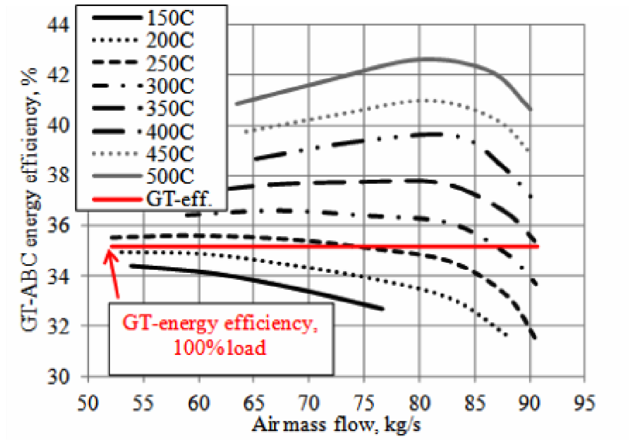

Figure 7: Gas-air system efficiency depending on the air mass flow

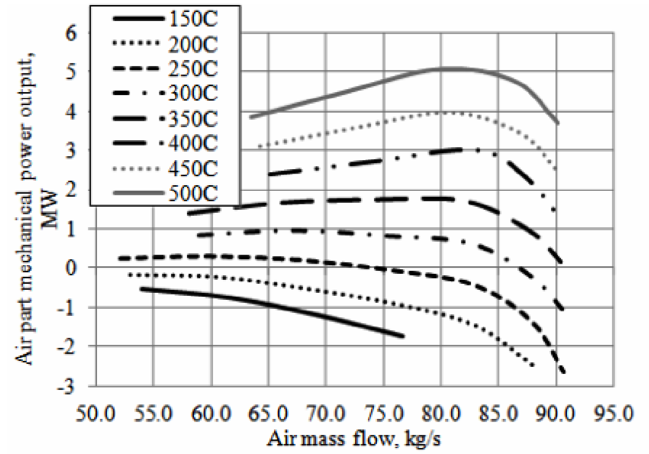

Figure 8: Gas-air system power output depending on the air mass flow

GT-ABC energy efficiency on the temperature before the air expander for different values of the pressure ratio (air bottoming cycle compressor). For temperature values at the expander inlet lower than 300°C, the GT-ABC energy efficiency decreases as the pressure ratio increases. At temperatures above 300°C the power efficiency function in the analyzed range of loads features an inflection point. System efficiency rises to a certain value, depending on the temperature at the air expander inlet. It should be emphasized, however, that for temperature values at the air expander inlet in the range of 300...350°C the system features practically constant efficiency, regardless of the pressure ratio applied.

The air bottoming cycle is controlled by rotational speed. Fig. 10 shows the air system power output for different values of rotational speed. The presented

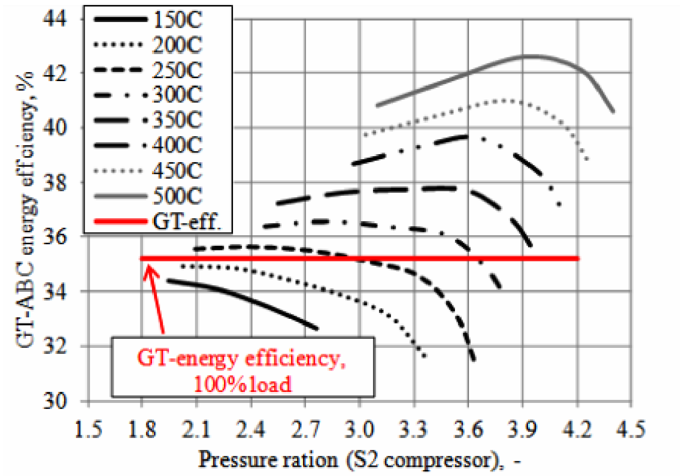

Figure 9: Gas-air system efficiency for different values of the pressure ratio of the air part compressor

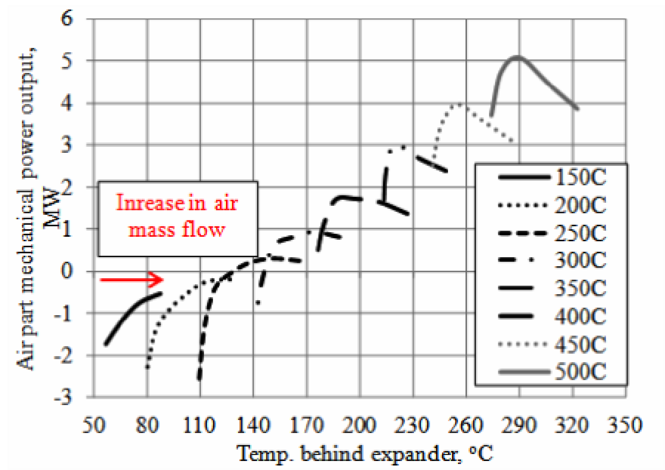

Figure 11: Power obtained by the air part for certain values of temperature after the air part expander

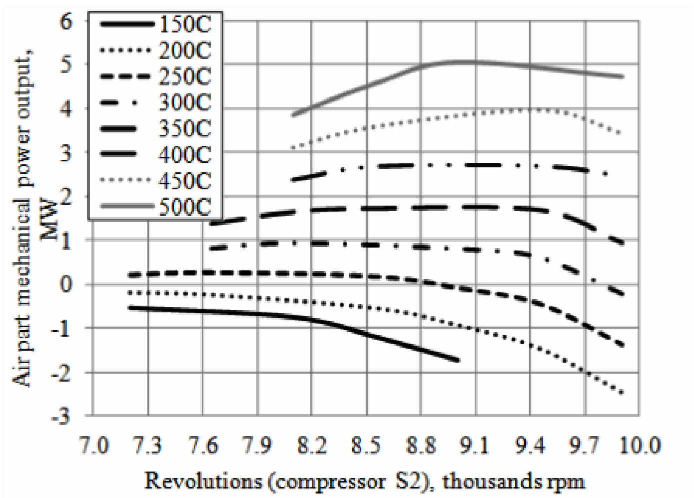

Figure 10: Power generated by the air part depending on the air turbine system rotational speed

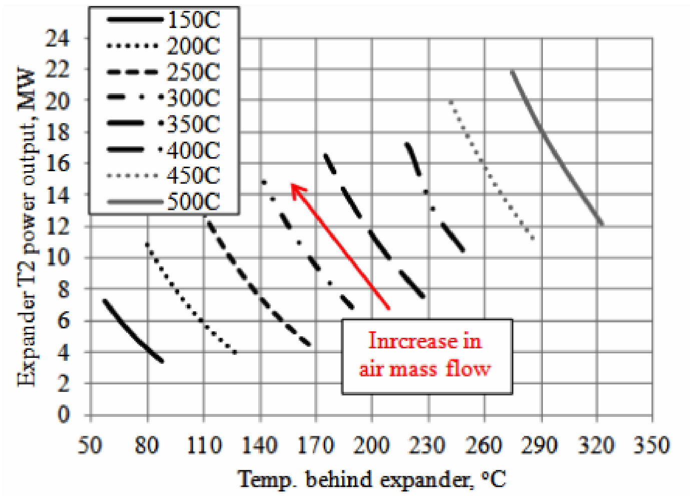

Figure 12: The dependence of the power output of the gas-air system air-part expander on temperature values after the air expander

values of power output are different for different values of temperature before the air expander. It can be seen that to achieve a positive effect of an increment in GT-ABC energy efficiency, the temperature of 300°C before the expander has to be reached and rotational speed must not exceed the nominal value (9,000 rpm).

The dependence of the power output of the gas-air system on the temperature at the expander outlet for different values of temperature before the air expander is shown in Fig. 11. If the temperature before the expander is higher than 500°C, the air bottoming cycle power output exceeds 5 MW. The dependence of power generated by the air expander on the outlet temperature (for different values of air temperature before the expander) is shown in Fig. 12. It can be seen that under part loads the temperature at the expander outlet rises for a given value of temperature before the air expander. For temperatures behind the AHX of the order of 500°C, the air outlet temperature can reach values of approximately 325°C, which corresponds (for a given temperature at the expander inlet) to the lowest value of power generated by the expander.

The drop in power output is due to the change in the compressor-turbine system working point on the compressor characteristic. For high values of exhaust gas temperature, a reduction in rotational speed re-

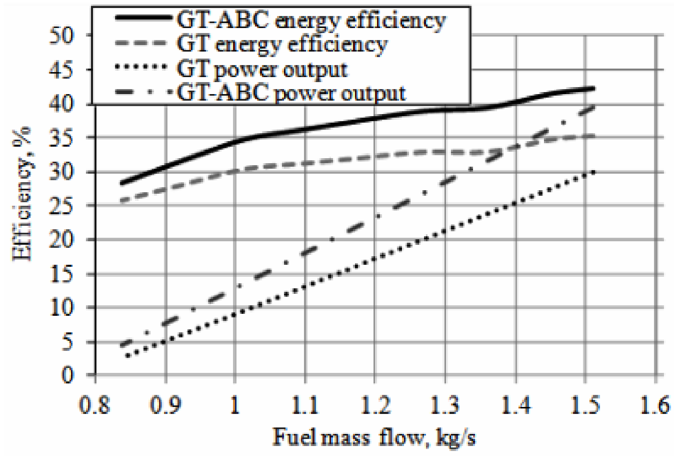

Figure 13: Increment in the gas-air system efficiency and mechanical power output

sults in a reduction in the air mass flow, which in turn translates into a drop in pressure before the expander.

Controlling the power output by means of changes in rotational speed is possible if load characteristics (of the power receiver, i.e. of the gas compressor) are changed (power output depending on rotations $N = f(n)$ and torque as a function of rotations $M = f(n)$). One regulation option is to throttle the gas compressor in pumping mode. However, the energy effects obtained by means of this control method are not analyzed in this paper. Another type of control which also involves energy losses is to use a bypass pipeline and discharge part of the exhaust gas mass flow directly to the stack, taking no account of the AHX. This action results in a change in air temperature before the expander and, consequently, in a reduction in power output and in machine rotational speed.

4.2. *Operation of the gas turbine air bottoming cycle under varied loads*

Using the GT 10 turbine characteristics [17], the operation of the GT-ABC system under varied loads was simulated. In this case the system was controlled by changes in the mass flow of fuel fed into the gas turbine combustor. Fig. 13 presents a comparison of the values of energy efficiency and mechanical power output between a standalone gas turbine unit and a GT-ABC system in the function of a varied mass flow of fuel.

The increment in cycle power efficiency and the increase in system mechanical power output were also determined, compared to a standalone gas turbine unit. The obtained results are presented in Fig. 14.

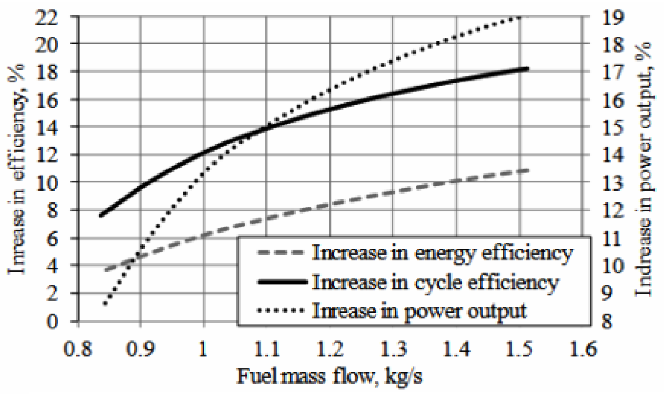

Figure 14: Increment in gas-air system efficiency and mechanical power output compared to a simple gas turbine unit for various values of fuel mass flow

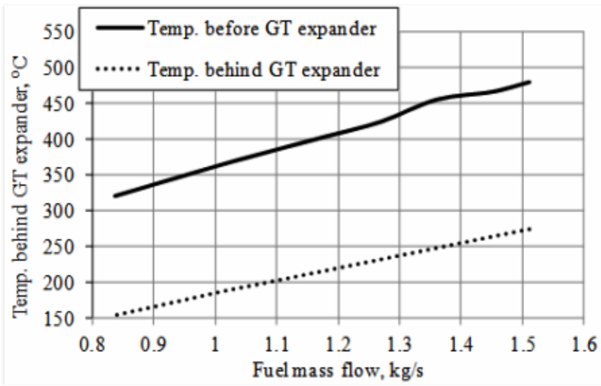

Figure 15: Temperature values before and after the air part expander for various values of fuel mass flow

Due to varying parameters at the gas turbine outlet, the temperature values after and before the air part expander also change Fig. 15.

5. Conclusions

It seems that if more frequent changes in the GT-ABC system load are required, the variant where the gas turbine operates under a varied load is better. The power generated by the air part is then conditioned by the parameters at the gas turbine expander outlet. The lower the exhaust gas outlet temperature, the lower the temperature before the air part expander and, consequently, the lower the resultant power output of the system. However, in the case under analysis, the drop in efficiency and power output should be considered for the entire system at a varied fuel mass flow. If the system is controlled by the gas turbine and the load is reduced to about 90% (which corresponds to a reduction in fed fuel mass flow to 1.36 kg/s), the system reaches a power output value which is slightly higher than that of a standalone gas turbine unit: about 25.8 MW. This corresponds to GT-ABC energy efficiency of 39%. A similar power output for the case controlled by the air part can be achieved for efficiency of approximately 36.5%.

The control range of the system total load in both cases selected for the calculations is incomparable. If the analyzed system is applied as a propelling unit (ships, vessels), the power output of the installation should be controlled by means of changes in fuel mass flow, which results in a change in the gas turbine load. The air part power output depends on the air temperature before the air part expander.

Acknowledgements

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

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