THE INFLUENCE OF SOME SELECTED PARAMETERS ON THE THERMODYNAMIC CHARACTERISTICS OF A CHP PLANT WITH A GAS TURBINE

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Abstract

The paper presents ways of determining the thermodynamic characteristics of a CHP plant with a gas turbine. Special attention has been paid to the average annual efficiency of transforming the chemical energy of fuel into electricity and heat (the so-called Energy Utility Factor – EUF) and Primary Energy Savings (PES). Two different systems were analyzed, viz. the installation of a gas turbine with a waste-heat boiler and a heat and power generating plant. Concerning the former one the influence of the parameters of steam (temperature and pressure) generated in the waste-heat boiler on the values of PES and EUF was determined. Concerning the latter one, the methods of investigating the effect of the thermal power of heat exchangers on the characteristics have been presented. This will facilitate the proper choice of heat exchangers.

Keywords: *CHP plant, heat and power generating plants, thermodynamic characteristics, primary energy saving*

1. Introduction

The application of gas turbines in CHP plants contributes to:

a) a decreased noxious effect of these installations on the environment, mainly thanks to
- the elimination of the emission of SO_2 and dust;
- an essential restriction of CO_2 and NO_X emission;
- reduced losses of water,

b) a more effective consumption of the chemical energy of fuel associated with an increased efficiency of generating electricity and heat,

c) decreased capital costs and a shorter time of constructing the required installations,

d) a shorter starting time from all states (cold, warm and hot).

Small and middle-sized installations of CHP plants with gas turbines are nowadays an important alternative for bigger units to meet the demand for heat and electricity. Thanks to the advantages mentioned above the number of such systems increases constantly, particularly in developed countries. New installations replace mainly old coal-fired boiler houses. Their final technological structures as well as the way and scope of automation are determined basing on the kind and number of consumers, and also on special requirements of the users.

More and more often, particularly in the course of the recent ten years, in Poland disused heat-generating units of medium and large power, which do not come up to the standards of environment protection, are being replaced by gas-and-steam systems. Similarly as in the case of smaller installations, the aforesaid advantages and economical analyses are of decisive importance.

2. Fundamental work indices in the thermodynamic assessment of CHP plants

An important index in the assessment of the work of the analyzed systems is the instantaneous gross watt-hour efficiency of the CHP plant, determined also as the unit capacity of the chemical energy of fuel, expressed by the formula:

$$\eta_C = \frac{N_{el} + Q}{m_{1p} W_d} \tag{1}$$

where:

N_{el} – power rating of the system, Q – flux of heat emitted by the system, m_p – flux of fuel, W_d – net calorific value of the fuel.

Having at our disposal the relations N_{el}, Q = f(τ) (where τ - time), and knowing the annual time of generating electrical energy (T_{el}) and heat (T_c), we can determine the production of electricity (E_{el}) and heat $(Q_g)_r$ per year:

$$E_{el} = \int_0^{T_{el}} N_{el} d\tau \tag{2}$$

$$(Q)_r = \int_0^{T_C} Q d\tau \tag{3}$$

These quantities permit to determine the average efficiency of transformation of the chemical energy of fuel into electricity and heat:

$$\eta_{ecr} = \frac{(Q_g)_r + E_{el}}{B \cdot W_d} \tag{4}$$

The quantity $B \cdot W_d$ in this equation determines the yearly consumption of chemical energy of the fuel, basing on the known fuel flux (m_{1p})

$$B \cdot W_d = \int_0^{T_{el}} (m_{1p} W_d) d\tau \tag{5}$$

The quantity determined by means of equation (4) is also called Energy Utility Factor or energy efficiency of CHP. It may also be expressed as

$$\eta_{ecr} = \eta_{er} + \eta_{cr} \tag{6}$$

where:

$\eta_{er} = \frac{E_{el}}{BW_d}$ - average ampere-hour efficiency per year (the efficiency of the cogeneration of electric energy)

$\eta_{cr} = \frac{(Q_g)_r}{BW_d}$ - average annual efficiency of cogeneration of heat

The decree of the Minister of Economy, Labour and Social Policy of May 30th, 2003concernicg the scope of duties connected with the purchase of electricity and heat from renewable sources and its amendments of December 19th, 2004 obligate power stations to buy electrical energy cogenerated in compliance with $\eta_{ecr} \geq 70\,\%$ [6].

The energy efficiency of the considered installations is assessed taking also into account the instantaneous cogeneration coefficient

$$\sigma = \frac{N_{el}}{Q} \tag{7}$$

Another coefficient is the Primary Energy Savings

$$PES = 1 - \frac{1}{\dfrac{\eta_{er}}{\eta_e^{Ref}} - \dfrac{\eta_{cr}}{\eta_c^{Ref}}} \tag{8}$$

where:

$\eta_e^{Ref}, \eta_c^{Ref}$ - efficiency reference value for separate electricity and heat production.

Equation (7) is based on the definition of relative savings of the chemical energy of fuel, attained by cogeneration, compared with the consumption of the chemical energy of the same kind of fuel in separate establishments (power station and heating plant).

The Directive 2004/8/WE EU introduces the term "high-efficiency cogeneration". If the value of PES is not lower than 10%, this term can be used.

It still remains an open question which part of the produced electrical energy will be considered to belong to the technology of high-efficiency cogeneration. The conditions of the classification of the total production of electricity as high-efficiency cogenerations have been quoted in the Directive 2004/8/WE: PES≥10% and η_{ecr}≥75% (80%).

3. The influence of the parameters of steam on the thermodynamic characteristics of a gas turbine installation with a waste-heat boiler

There are many possible structures of power stations (heat and power generating plants) with a gas turbine, e.g. [1], [2], [3]. The considerations discussed in the present paper concern merely installations of a gas turbine with a waste heat boiler and a gas and steam power station.

The installation of a gas turbine with a waste heat boiler presented in Fig.1 consists of a compressor (SP), the combustion chamber (KS), the gas turbine (TG), generator (G), air filter (F) and the waste heat boiler.

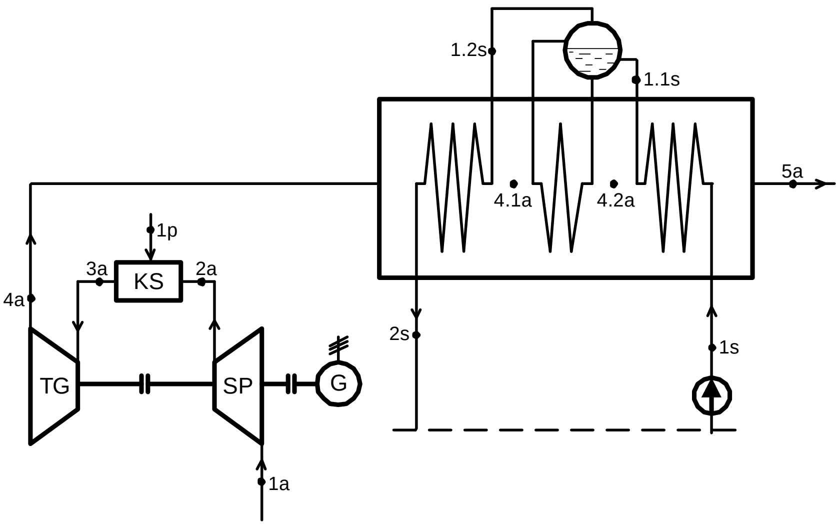

Fig.1. Thermal diagram of the investigated system

Under nominal conditions (i.e. ambient temperatures, respectively equal to t_{1a}=15°C and p_{1a}=101,325kPa) the parameters of the gas turbine are known, viz. power rating – 15,42 MW, pressure ratio – 11,11, flux fuel and combustion gases m_{1p}=0,968 kg/sec, m4a=51,74 kg/sec (the applied indices concern Fig.1) and temperatures of the combustion gases t_{3a}=1060°C and t_{4a}=544,7°C. In the waste heat boiler steam is generated. The temperature of the water feeding the boiler is known (t_{1s}=80°C), the pinch point (Δt_{pp}) and approach point (Δt_{ap}) are equal to 10K. The temperatures of the generated steam (t_{2s}, p_{2s}) affecting the fundamental operation characteristics of the power station have been determined. The results of calculations $\eta_c = f(t_{2s}, p_{2s})$ have been presented in Fig.2, the characteristics of $PES = f(t_{2s}, p_{2s})$ in Fig.3 (assuming that $\eta_e^{Ref} = 55\%$, $\eta_c^{Ref} = 90\%$). The flux of steam generated in the boiler results from the balances of the evaporator and superheater:

$$m_{2s} = \frac{c_p \{T_{4a} - [T_n(p_{2s}) + \Delta t_{pp}]\}}{h_{2s} + c_w \Delta t_{ap} - h'_{2s}} \tag{9}$$

where:

c_p, c_w - specific heat capacity of combustion gases and water, $T_n(p_{2s})$ - saturation temperature at the pressure p_{2s}, h_{2s}, h'_{2s} - enthalpy of steam and enthalpy of saturation.

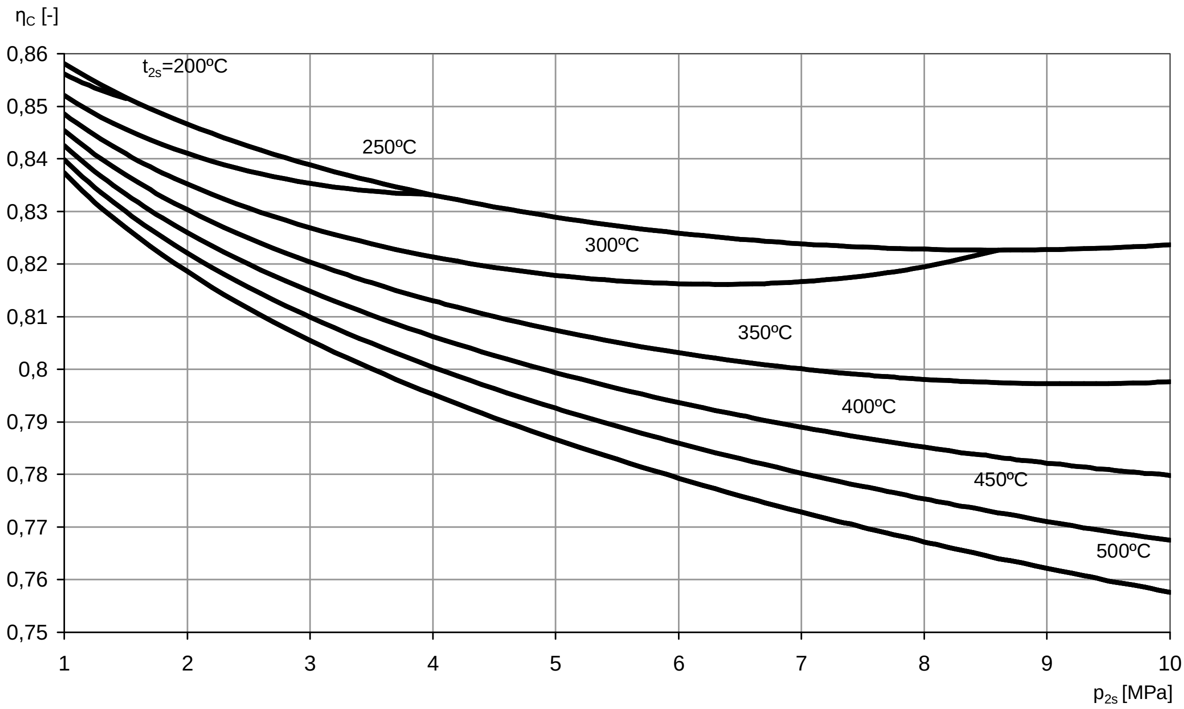

Fig.2. The relation $\eta_C = f(t_{2s}, p_{2s})$

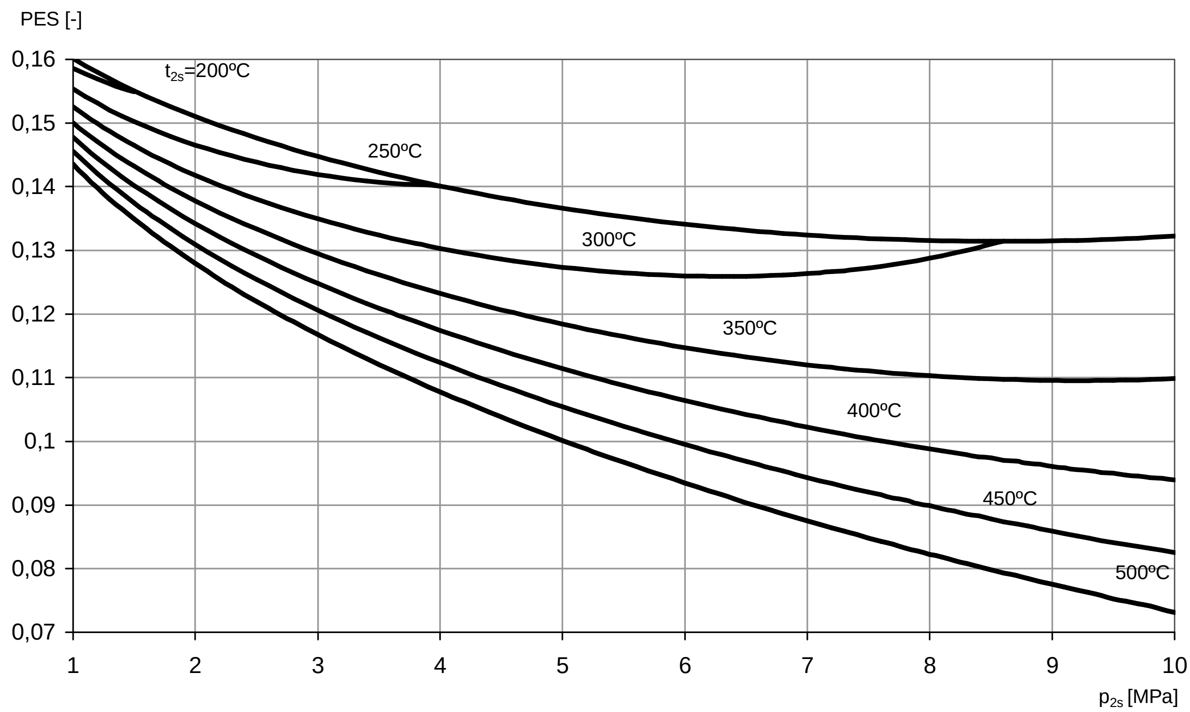

Fig.3. The relation $PES = f\left(t_{2s}, p_{2s}\right)$

4. The influence of the thermal power of heat exchangers on the thermodynamic characteristics of gas and steam power stations

The analyzed system of a heat and power generating plant is to be seen in Fig.4, where also its characteristic points have been denoted. This system consists of a gas-steam unit and water boilers [8]. The gas-steam unit consists of a simple gas turbine and a double-pressure waste heat boiler with a condensing-extraction turbine. The thermodynamic parameters of the system are known in all its characteristic points under nominal conditions. The power rating of the system amounts to 235 MW (in compliance with ISO at t_{1a}=15°C). The ambient temperature (t_{1a}) influences substantially the fundamental parameters of the gas-steam unit and also the demand for heat. Hence, the necessity of modeling the operation of the system and its analysis in the entire range of changes of the ambient temperature. The gas turbine was modeled basing on the parameters of the industrial turbine Siemens V.94.2. The characteristic quantities describing this turbine are: the temperature at the outlet of the combustion chamber t_{3a}=1226°C, the compression ratio 11, the maximum isentropic efficiency of the air compressor 0,913 and the isentropic efficiency of the turbine 0,936, the efficiency of the combustion chamber 0,992, the ratio of the flux of air cooling the turbine to the flux at the inlet to the compressor 0,205. The double-pressure waste heat boiler generates steam with the following parameters: : $p_{3s(h)} = 8078\,kPa$, $t_{3s(h)} = 525^{o}C$ in the high-pressure part, $p_{3s(l)} = 540kPa$, $t_{3s(l)} = 217^{o}C$ in the low-pressure part (the indices coincide with the denotations of the points in Fig.4). The minimum pinches of temperature (Δt_{pp}) in the boiler amount to 9 and 11K, respectively, for the high-pressure and low-pressure part. The steam generated in the boiler is passed to the turbine. The isentropic efficiencies of the high-pressure and low-pressure part of the steam engine amount to 0,905 and 0,8, respectively. The pressure in the condenser amounts to $p_{4s(l)} = 5\,kPa$, and in the deaerator to $p_{13s} = 130\,kPa$. The temperature of the feed water is $t_{WZ} = 76\,^{o}C$, thanks to the application of a preheater of the condensate. The applied heat exchanger consisted of two parts, one of which was being fed with steam from the controlled bleeder valve. The other part constitutes the last heating surface in the waste heat boiler. This permits to cool the combustion gases leaving the waste heat boiler to a temperature of $t_{5a} = 85\,^{o}C$. When the demand for heat exceeds the capacity of the heat exchanger, the connected water boilers are fired with coal.

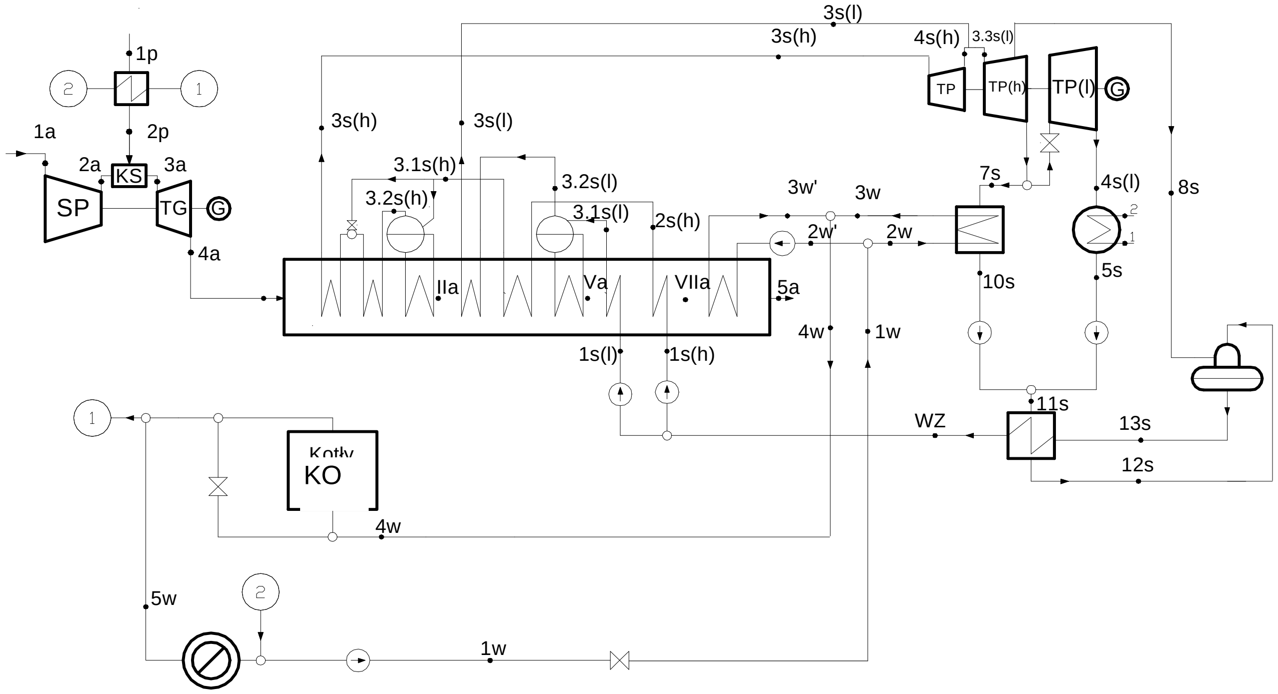

Fig.4. Thermal diagram of the investigated system (SP – compresor, KS – combustion chamber, G – generator, TP – steam turbine, KO – waste heat boiler)

The maximum demand for heat (at an ambient temperature of $t_{1a} = -20\,^{o}C$) has been assumed as $(Q_{ec})_{\max} = 400\,MW$, as is to be seen in Fig.5. Generally, a linear dependence of the flux of heat (Q_{ec}) on the ambient temperature is assumed during the heating season [9]. In other seasons the system operates with a constant thermal power of 10% of the maximum demand, producing warm water for household purposes. The gas-steam unit of the considered heat and power generating plant provides heat (Q_g) in compliance with the diagram of demand. Peak loads $(Q_{ec} - Q_g)$ are taken over by the water boiler with a power rating of $Q_{KW} = (Q_{ec})_{\max} - (Q_g)_{\max}$. Fig.5 presents the relation $Q_{ec} = f(t_{1a})$ based on the data quoted above. It also shows the relation $Q_g = f(t_{1a})$, assuming that most of the heating season the heat exchangers of the gas-steam unit operate with a constant calorific effect of $Q_g = (Q_g)_{\max} = 120\,MW$ (which may also be called design power rating of these heat exchangers). For the assumed process $Q_g = f(t_{1a})$ the power rating of the gas-steam unit is determined as $N_{el} = f\left[Q_g(t_{1a}), t_{1a}\right]$ [10].

Thermodynamic calculations were carried out making use of the programme GateCycle, changing the ambient temperature within the range -20°C ≤ t_{1a} ≤ 28°C in steps of 1°C. The thermodynamic parameters were determined in all the points denoted in the diagram, basing on which the quantities quoted below have been determined. Making use of the systematic diagram of the ambient temperature as a function of time, Fig.6 presents the determined dependences of the power rating of the unit and thermal power on time (τ). Knowing the time of operation of the gas-steam unit (T_{el}=T_c), the relation $N_{el}, Q_{ec}, Q_g = f(\tau)$ permits to determine the annual production of electricity (E_{el}) and heat $(Q_g)_r$.

The annual consumption of gaseous fuels is determined by means of the known or calculated characteristics of the flux of fuel depending on the ambient temperature $m_p = f(t_{1a})$ [8,10] and the systematic diagram of temperature, applying the relation [5].

These quantities will make it possible to determine also $\eta_{ecr,}\eta_{er,}\eta_{cr,} PES$, making use of the relations (4), (5) and (8). For this purpose the time of operation of the unit in the course of one year has been assumed to amount to 8200 hours.

Further on the same calculations were applied for changing values of $(Q_g)_{max}$.

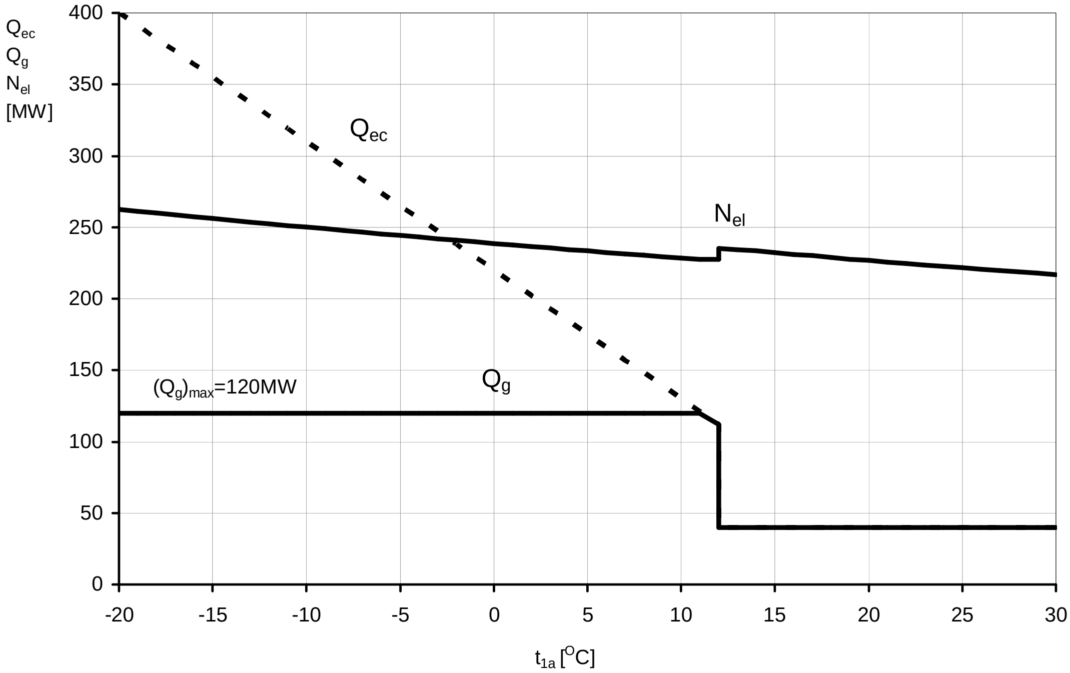

Fig.5. Dependence of the flux of heat generated in the heat and power generating plant (Q_{ec}) and the gas-steam unit (Q_g) and of the power rating of the unit (N_{el}) on t_{1a}

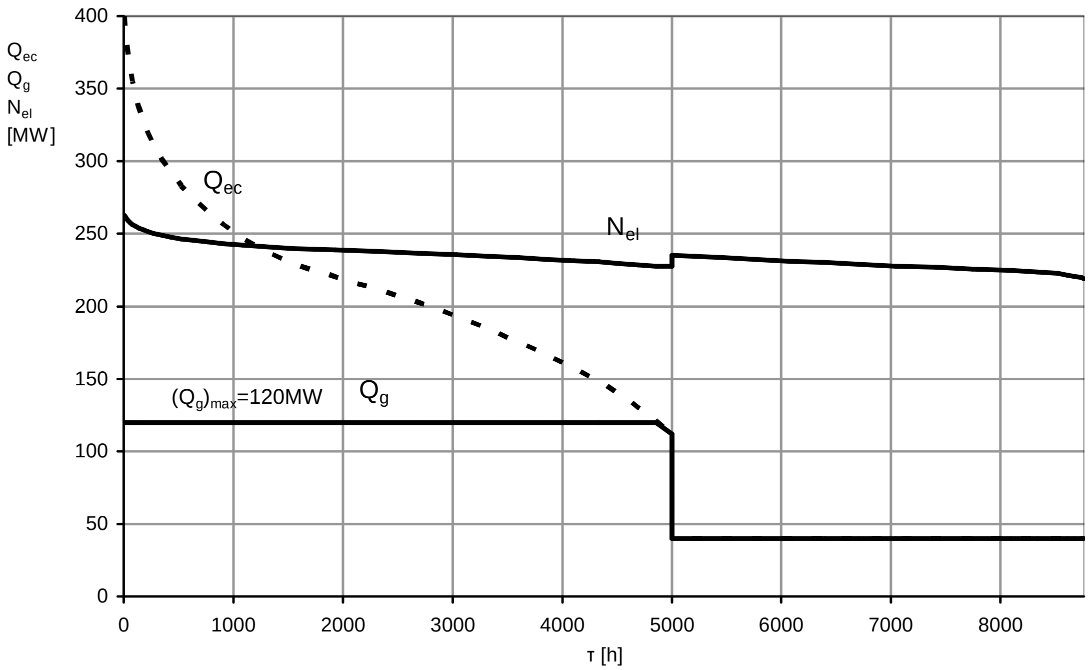

Fig.6. Dependence of the flux of heat generated in the heat and power generating plant (Q_{ec}) and the gas-steam unit (Q_g) and of the power rating of the unit (N_{el}) on the time of its occurrence τ

Calculations were carried out for $110\ MW \leq (Q_g)_{\max} \leq 180\ MW$ in steps of 10 MW. The upper limit results from the required minimum value of the flux of steam flowing through the last stages of the turbine. It has been assumed that this flux constitutes 7% of the flux of steam flowing through the condenser in the case of carrying out merely condensation operations. The most important results of calculations have been gathered in the form of characteristics $\eta_{ecr,}\eta_{er,} PES = f\lfloor (Q_g)_{\max} \rfloor$, as presented in Fig.7. In order to render it more perspicuous in the value of η_{er} were multiplied by 1,5.

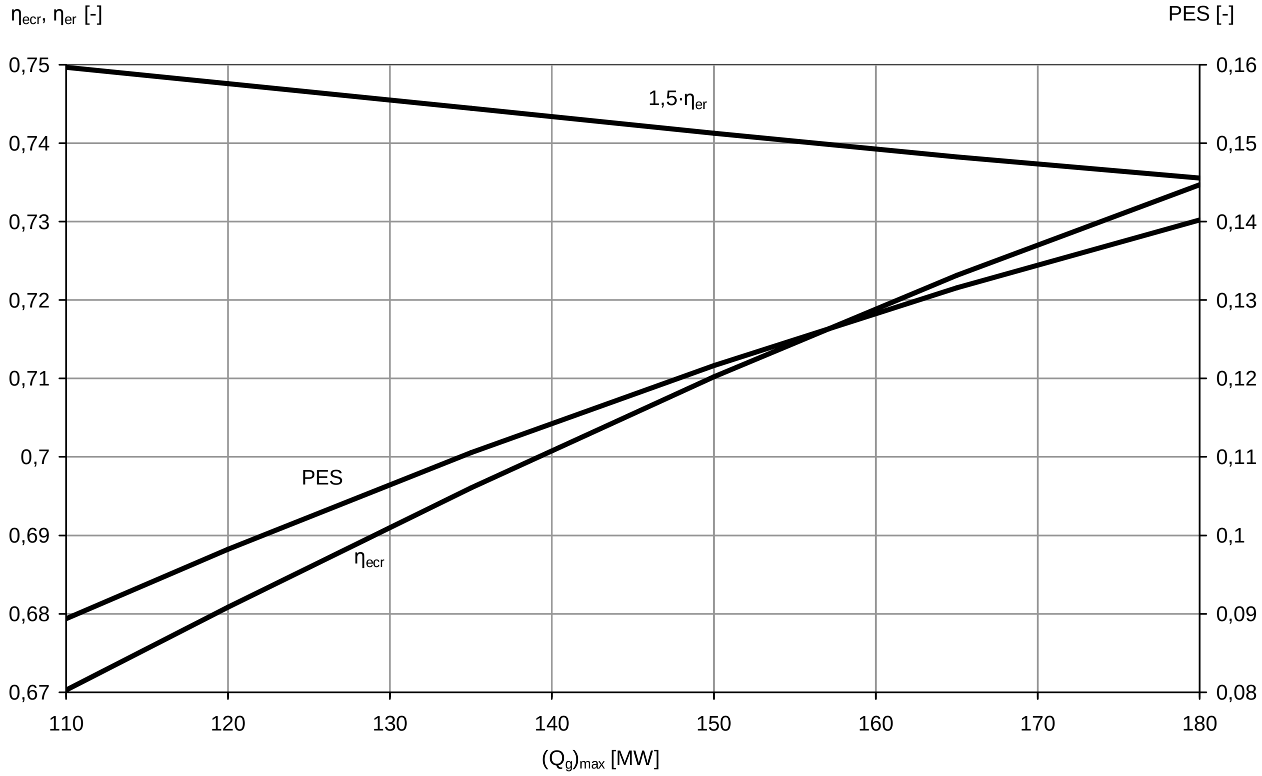

Fig.7. The relation $\eta_{c,} PES = f\lfloor (Q_g)_{\max} \rfloor$

Summary

- The installation of a gas turbine with a waste heat boiler generating steam even with very high parameters attains an Energy Utility Factor (EUF) exceeding 80%. In the case of high parameters of the steam (e.g. at t_{2s}=500°C, p_{2s}=5MPa) the primary energy savings (PES) are lower than 10%.
- The indices EUF and PES concerning the investigated gas and steam power station increase with the growth of the thermal power of the heat exchangers $(Q_g)_{max}$. It is possible to choose such a $(Q_g)_{max}$ which would permit to attain PES≥10% and simultaneously also EUF>70%. In compliance with the Directive 2004/8/WE EU this is high-efficiency generation. Not all the generation of electricity, however, can be classified as high-efficiency generation, because EUF<80%. Therefore, attempts should be made to reduce the time of operation of the heat and power generating plant (T_{el}<8200 h/year) during the summer season.

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