Ryszard Bartnik · Zbigniew Buryn

Conversion of Coal-Fired Power Plants to Cogeneration and Combined-Cycle

Thermal and Economic Effectiveness

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Contents

Notation

b	Construction period, years
CF	Cash flow, PLN
DPBP	Discounted payback period, years
e_h, e_{el}, e_{coal}, e_g, e_{cert}, e_{co_2}	Specific price of heat, electricity, coal, gas, certificates of origin, CO_2 allowance, PLN/GJ, PLN/MWh
$\dot{E}_{ch}$	Stream of chemical energy of fuel, MW
$E_{ch,A}$	Annual use of chemical energy of fuel, MWh
$E_{el,A}$	Annual production of electricity, MWh
F	Heat transfer surface area, m^2
h	Specific enthalpy, J/kg
L_{pip}	Length of heat distribution pipeline, m
IRR	Internal rate of return, 1/a
j	Specific capital expenditure, PLN/unit
J	Capital expenditure, PLN
k	Heat transfer coefficient, kW/(m^2K)
k_h	Specific cost of heat generation in a power plant without electric capacity compensation, PLN/GJ
k_h^{com}	Specific cost of heat generation in a power plant with electric capacity compensation, PLN/GJ
K_e	Annual exploitation cost, PLN
$\dot{m}$	Stream of mass, kg/s
N	Nominal exploitation period
N_{el}	Electrical power output, MW
NCV	Net calorific value, J/kg
NPV	Net present value, PLN
p	Rate of profits tax
p	Pressure, Pa
$\dot{\mathrm{P}}$	Stream of fuel, kg/s
Q_A	Annual heat output, MWh
v_m	Relative value of heat and electricity market

r	Discounting rate of investment, %/a
t, T	Temperature, °C, K
$z\rho + \delta_{serv}$	Annual rate of investment service and remaining fixed cost relative to capital expenditure, %/a
z	Discount rate of assets
Z_A	Annual gross profit, PLN/a

Greek Symbols

Δ	Increase
η	Efficiency
ρ	Annual rate of progressive depreciation
ε	Relative decrease of electricity production per unit of the heat produced in cogeneration in a power plant
ε_{el}	Relative coefficient of power station internal load
τ	Time, s

Subscripts and Superscripts

A	Year
c	Heat
ch	Chemical variables
CHP	Combined heat and power
com	Compensation the decrease of electricity production
dhw	Domestic hot water
el	Electricity
El	Power plant
env	Environment
h	District heating
h	Hot water
n	Nominal conditions
r	Return water
s	Summer season (non-heating season)
u	Utility heat
w	Water
w	Winter season (heating season)

Chapter 1
Introduction

Combined heat and power forms the most effective means of reducing the consumption of primary fuels and, thus, decreasing the emission of hazardous products of combustion into the environment [1–4, 6]. In particular, it would be possible to considerably reduce the emission of greenhouse gases.

Schematic diagram of one of the existing thermal power stations with a capacity of 1,200 MW is presented in Fig. 1.1.

The potential of heat and power cogeneration as a means of serving the purposes of Primary Energy Saving (PES, Fig. 1.2) is insufficiently utilized in the EU member states.

The promotion of cogeneration constitutes the priority of the EU, as described in the preamble to directive 2004/8/EC of the European Parliament. This preamble introduces the notion of *high efficiency cogeneration* and indicates that the relative energy saving of PES of over 10% justifies the application of this term with regard to electric heat and power generation in a combined process.

The effective application of energy resulting from cogeneration is likely to increase the security of the energy supply across the EU. At present, the dependence of the EU on imported primary fuel supplies currently accounting for 50% of the requirements is projected to rise to 70% by 2030 if the current trends continue. Thus, it is necessary to take adequate action to ensure better utilization of the opportunities offered by cogeneration based on the demand of heat. The EU member states have been obliged to conduct an analysis of the potential for the application of combined heat and power production and analyze the framework for the development of CHP.

An interesting and relevant opportunity in terms of energy and ecology as well as for economic reasons is offered by the modernization of steam turbine power stations to operate for purposes of cogeneration and heat generation $\dot{Q}_c^{El}$ besides electric power production, (Fig. 1.3).

The modernization of power stations to combine heat and power will result in an improvement of the overall energy efficiency. Concurrently, on the global scale

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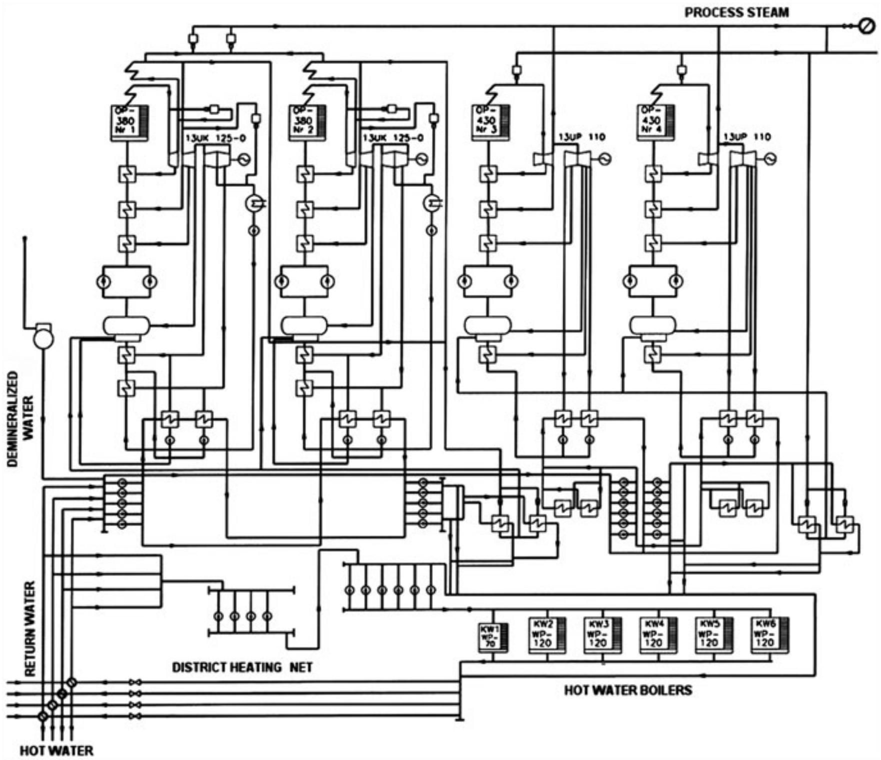

Fig. 1.1 Schematic diagram of thermal power station

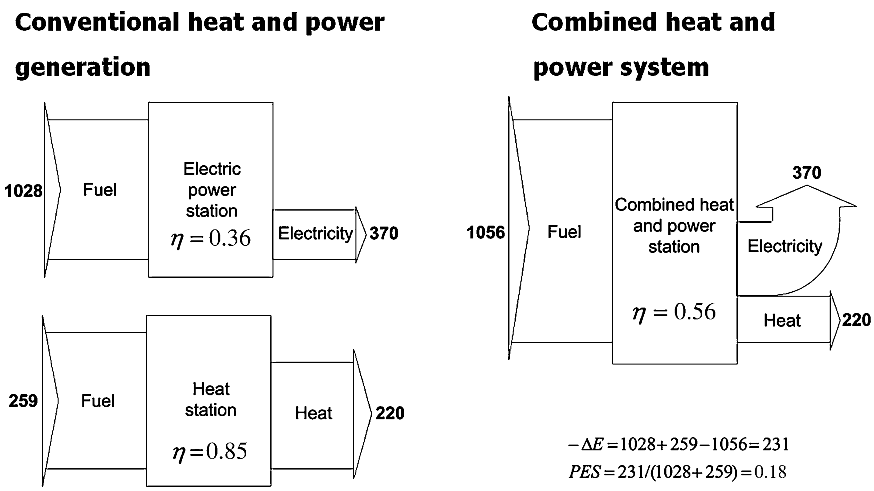

Fig. 1.2 Comparison between separate and combined heat and power systems

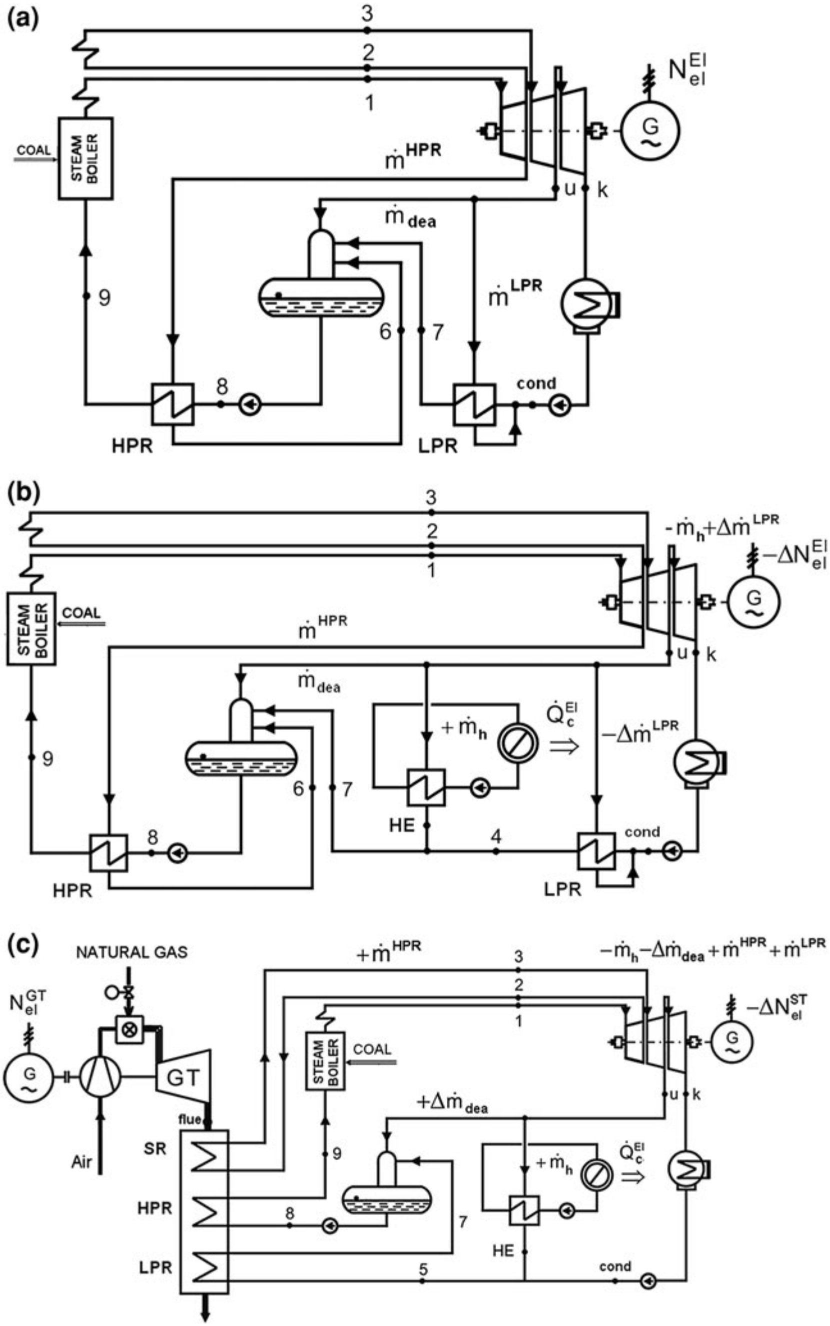

Fig. 1.3 Schematic diagram: **a** conventional coal fired condensing power plant with steam turbine; **b** steam turbine power plant adapted to cogeneration; **c** steam turbine power plant adapted to cogeneration and repowered by gas turbine in parallel system (*SR* steam reheating, *HPR* high pressure regeneration, *LPR* low pressure regeneration)

the process will lead to limiting the emission of pollutants into the natural environment, and this particularly concerns CO_2 emission. In addition, the economic parameters of the utilization of the power station will be considerably improved. However, this will be determined by the relations between the market prices of energy carriers, relation between the prices on heat and electrical energy, and the level of charges imposed on the pollution of the environment, in particular positions on the allocation of CO_2 allowances.

Among the possible ways of adapting a power station to cogeneration it is possible to distinguish the following alternatives:

(a) extraction of hot steam supplying a heater from (Fig. 1.3b):

- high pressure steam pipeline (point 1)
- cold reheat pipe or hot reheat pipe (point 2 or 3)
- turbine extractions or crossoverpipe (point u)

(b) removal of two, three final rows of blades from the low-pressure section of the steam turbine to increase the pressure and concurrently the saturation temperature in the condenser and application of the latter as a heater; for lower ambient temperatures the thermal power for heating purposes would have to be complemented from peak load water boilers

(c) modified power station for heat and power production with installation of gas turbine and heat recovery steam generator (Fig. 1.3c).

In alternative (a) (point 1) we do not deal with the effect of a combined system; however, depending on the above-mentioned relations between the market prices of energy carriers it is possible to obtain the effect of improvement of economic indicators; hence, a necessity to consider such an alternative follows.

In the above alternatives, it is necessary to analyze the viability of increasing coal batching into the boiler, thus restoring the power station to its initial power output.

The smallest exergy losses [4, 5] occur for the case of alternative (a) (point u), which results in the smallest decrease of electricity production in the power station. In addition, this alternative seems to be most reasonable with regard to the technology since it secures a constant yet flexible operation of the power station that meets the requirements of the combined system in accordance with a yearly schedule summarizing demand for heat. For these purposes, this alternative requires an in-depth technical and economic analysis. The results of such analyses will render it possible to undertake a rational decision regarding the selection of an investment program aimed at the modernization of a power station to cogeneration.

A very important modernization potential of the coal-fired power industry are the so-called clean coal technologies based on gas turbines and characterized by relatively high energy effectiveness and low emission of pollutants to the environment. They are, among others, dual fuel gas-steam combined-cycle technologies coupled in series or parallel systems (Fig. 1.4).

What is really important is that the systems, can be created on the basis of the presently existing coal structures by adding a gas turbine. In practice, the parallel

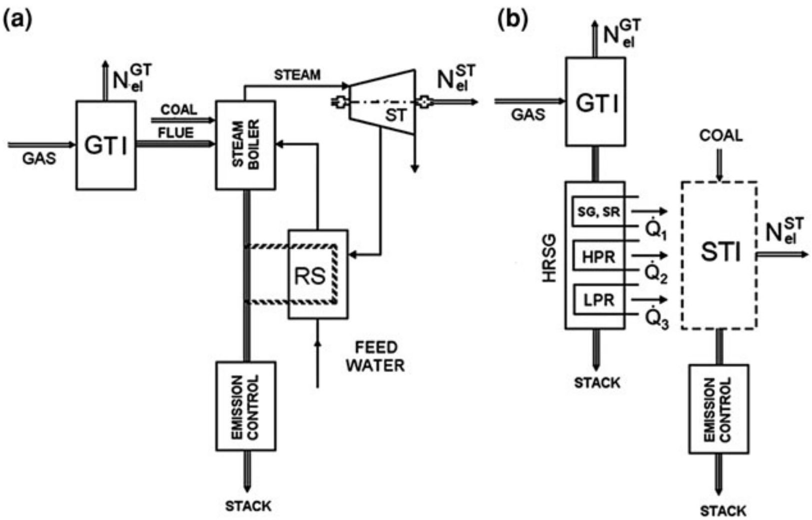

Fig. 1.4 Diagrams of dual fuel gas-steam combined cycle: **a** in series system (Hot Windbox), **b** in parallel system. *GTI* gas turbine installation, *HRSG* heat recovery steam generator, *ST* steam turbine, *RS* regeneration system, *STI* steam turbine installation, *SG* steam generation, *SR* steam reheating, *LPR*, *HPR* low pressure heaters and high pressure regeneration, N_{el}^{GT}, N_{el}^{ST} power of gas turbine and power of steam turbine

system is a more energetically and economically effective method of a power plant modernization [4]. That is why for such a system a mathematical model of a power unit was elaborated enabling the analysis of its energetic and economic modernization effectiveness and selection of a gas GT power unit optimum power.

The principal aim of this monograph is the presentation and analysis of the following:

- the economic and energetic efficiency of the process of adapting a coal-fired power station to cogeneration by application of low-pressure regenerative exhaust for feeding heaters using heating steam for various configurations
- the thermal and economical effectiveness of conventional coal-fired condensing power plants repowered by gas turbine in a parallel system.

The analysis involves a coal-fired power station with rated electrical power output of 370 MW.

In particular, the scope of the monograph involves the description of the following:

- mathematical model of the technology at a power station for the particular alternatives of the feeding heaters using heat with steam from turbine extractions,
- optimization algorithm,
- optimization calculations of various alternatives of feeding heaters,

- analysis of background-related factors, e.g. the effect of the distance from the power station to heat consumers as a parameter affecting energetic and economic justification for the selection of alternatives of heater feeding modes and selection of an optimum solution,
- methodology of thermal and economical analysis of modernization of existing coal-fired power plants repowered by gas turbine and heat recovery steam generator.

In summary, the presentation of the methodology, calculation procedures, and tools used to support the planning of enterprises aimed at adapting power stations to cogeneration and combined cycle forms the primary objective of the research presented in this monograph. It also undertakes a research problem that is very up-to-date and necessary to solve. It is relevant both in terms of offering more in-depth analysis in the area of power engineering and can also serve for the purpose of practical application of the results.

This methodology of conversion of coal-fired electric power stations to combined heat and power concerns nuclear power plants as well.

During optimization of technical and thermodynamic parameters, construction of machines and installations and their structures it is necessary to apply the economic criterion [3, 4]. Economic criterion is superior to the thermodynamic criteria. Thermodynamic analysis (exergy, entropy) enables one to search for the possibility of improving technological processes and technical solutions. But finally in the market economy it is the economical criteria and the income maximization that decide about the justification of each technological concept, and the cost-effectiveness decides about embarking on an investment. If the investor had been unsure that the return rate from an investment would not have been high enough prior to taking the decision about an investment, the risk of investing money would not have been taken. However, we have to bear in mind that economic analysis is possible after thermodynamic analysis has been performed. It is their results that offer input variables for economical analysis.

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Chapter 2
Selection of Optimal Heating Structures for Modernization of Coal-Fired Power Stations to Cogeneration

2.1 Methodology of Selection of Optimal Heating Structures for Modernization of Coal-Fired Power Stations to Cogeneration

The economic criterion should govern the process of optimization in the market economy in solving technical problems. The economic criterion is superior to the thermodynamic design aspects. Thermodynamic criteria may be adopted in the quest for an opportunity to improve thermal processes; however, the justification of the application of a specific technical solution should be decided on the basis of the economic evaluation criteria.

Maximization of the profits resulting from its application needs to be the criterion for deciding the selection of optimum heating structures for modernization of coal-fired power stations. In general cases, this criterion is expressed by the relation [1]:

$$\begin{aligned}\mathrm{NPV} = \Bigg[&\frac{S_{\mathrm{A}}^{\mathrm{M}} - K_{\mathrm{e}}^{\mathrm{M}}}{\rho_{\mathrm{M}}} + \frac{S_{\mathrm{A}}^{\mathrm{M}+1} - K_{\mathrm{e}}^{\mathrm{M}+1}}{(1+r)^{\mathrm{M}+1}} + \left(S_{\mathrm{A}}^{\mathrm{mod}} - K_{\mathrm{e}}^{\mathrm{mod}}\right)\left(\frac{1}{\rho_{\mathrm{N}}} - \frac{1}{\rho_{\mathrm{M}+1}}\right) \\ &- J_{\mathrm{O}} - \frac{J_{\mathrm{mod}}}{(1+r)^{\mathrm{M}}}\Bigg](1-p)(1-v_{\mathrm{m}}) \rightarrow \max\end{aligned} \tag{2.1}$$

where:

J_{mod} capital expenditure made in year M (Fig. 2.1) on the modernization of a power station to cogeneration; J_{mod} is the function of the means of modernization;

J_0 discounted capital expenditure on construction of a power station or the price paid for its purchase, e.g. from State Treasury by IPP (Independent Power Producer) (Eq. 2.2);

K_{e} annual cost of exploitation of a power station;

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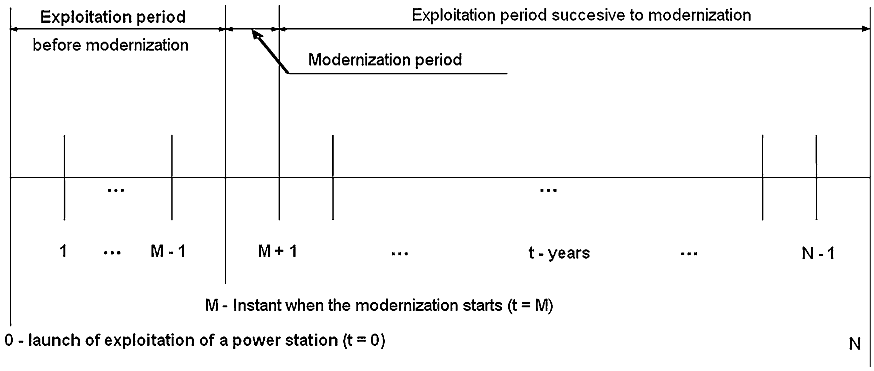

Fig. 2.1 Exploitation cycle of a power station

v_{m}	relative value of the market of heat and power; share of State Treasury in the total profit gained from exploitation of a power station after its purchase by an IPP;
p	gross corporate profits tax rate expressed by the relation: $Z_{\mathrm{A}} = S_{\mathrm{A}} - K_{\mathrm{e}} - \sum_i \rho_{\mathrm{X\,i}} J_{\mathrm{Y\,i}}$;
r	interest rate of capital expenditure;
S_{A}	annual revenues from power station;
ρ_{M}, $\rho_{\mathrm{M+1}}$, ρ_{N}, $\rho_{\mathrm{N-M}}$	annual depreciation rate expressed by interest rater to secure the payback of capital expenditure J_0 and J_{mod} along with interest after N and N−M years of exploitation.

If the modernization of a power station to cogeneration is undertaken by an IPP investor, the discounted capital expenditure J_0 (at a time $t = 0$; Fig. 2.1) associated with the construction of power station should be substituted with the price of purchase e.g. from State Treasury in the function of the objective (2.1). This price is expressed by Eq. 2.2 [1].

Figure 2.1 presents the diagram illustrating the exploitation period of a power station by an IPP, which identifies the periods prior to and successive to the modernization used in the methodology for the purposes of the current research (Eqs. 2.1, 2.2).

The arguments of the criterion for maximizing gain (2.1) involve the following functions (decision making variables):

- function of annual cost of the operation of the adapted power station which is formed by the sum of capital costs (depreciation) $K_{\mathrm{cap}} = \sum_i \rho_{\mathrm{X\,i}} J_{\mathrm{Y\,i}}$ and exploitation cost $K_{\mathrm{e}}^{\mathrm{mod}}$, $K_{\mathrm{A}}^{\mathrm{mod}} = K_{\mathrm{cap}}^{\mathrm{mod}} + K_{\mathrm{e}}^{\mathrm{mod}}$,
- function of annual revenues $S_{\mathrm{A}}^{\mathrm{mod}}$ which is the function of the price of energy carriers along with $K_{\mathrm{e}}^{\mathrm{mod}}$ among others,
- function of capital investment necessary for the modernization of the power station to cogeneration J_{mod} relative to the technology applied for the modernization,

- functions of annual depreciation rates ρ_{M}, $\rho_{\mathrm{M+1}}$, ρ_{N}, $\rho_{\mathrm{N-M}}$ and rates of $[\rho_{\mathrm{M}}]_{\mathrm{IRR_p^{IPP}}}$, $[\rho_{\mathrm{M+1}}]_{\mathrm{IRR_p^{IPP}}}$, $[\rho_{\mathrm{N}}]_{\mathrm{IRR_p^{IPP}}}$ Eq. 2.2, which are relative to the sources of funding of the investment, $\mathrm{IRR_p^{IPP}}$ rate, which is the minimum interest that an IPP should receive from investment J_0 and J_{mod} in the purchase and modernization of the power station and discount rate of capital investment r.

In the search for the highest possible value of the function of the objective (2.1) there are a number of constraints imposed on the decision making variables resulting from the details of technology and financial determinants.

They are:

- constraint equations.
  purchase price paid by IPP (this constraint does not pertain when the State Treasury offers the funding to cover the cost of modernization) [1]:

$$J_O = \frac{\left[\frac{(S_{\mathrm{A}}^{\mathrm{M}} - K_{\mathrm{e}}^{\mathrm{M}})}{[\rho_{\mathrm{M}}]_{\mathrm{IRR_p^{IPP}}}} + \frac{(S_{\mathrm{A}}^{\mathrm{M+1}} - K_{\mathrm{e}}^{\mathrm{M+1}})}{(1+\mathrm{IRR_p^{IPP}})^{\mathrm{M+1}}} + (S_{\mathrm{A}}^{\mathrm{mod}} - K_{\mathrm{e}}^{\mathrm{mod}})\left(\frac{1}{[\rho_{\mathrm{N}}]_{\mathrm{IRR_p^{IPP}}}} - \frac{1}{[\rho_{\mathrm{M+1}}]_{\mathrm{IRR_p^{IPP}}}}\right)\right](1-p)(1-v_{\mathrm{m}})}{1 - \frac{\rho_{\mathrm{N}}}{[\rho_{\mathrm{N}}]_{\mathrm{IRR_p^{IPP}}}}\left[1-(1-p)(1-v_{\mathrm{m}})\right]} + \frac{\rho_{\mathrm{N-M}} J_{\mathrm{mod}}\left[1-(1-p)(1-v_{\mathrm{m}})\right]\left(\frac{1}{[\rho_{\mathrm{N}}]_{\mathrm{IRR_p^{IPP}}}} - \frac{1}{[\rho_{\mathrm{M}}]_{\mathrm{IRR_p^{IPP}}}}\right) - \frac{J_{\mathrm{mod}}}{(1+\mathrm{IRR_p^{IPP}})^{\mathrm{M}}}}{1 - \frac{\rho_{\mathrm{N}}}{[\rho_{\mathrm{N}}]_{\mathrm{IRR_p^{IPP}}}}\left[1-(1-p)(1-v_{\mathrm{m}})\right]} \quad (2.2)$$

where:

$\mathrm{IRR}_p^{\mathrm{IPP}}$	minimum interest rate that an IPP should gain on the capital investment J_0 and J_{mod}; investors tend to determine the minimum, i.e. threshold value of the internal payback rate $\mathrm{IRR_p^{IPP}}$ to compensate the risk of investment (besides warrants in the form of sale agreements and energy price guarantees forming an incentive to invest) at a rate which is higher than investment in deposits in the capital market;
$[\rho_{\mathrm{M}}]_{\mathrm{IRR_p^{IPP}}}$, $[\rho_{\mathrm{M+1}}]_{\mathrm{IRR_p^{IPP}}}$, $[\rho_{\mathrm{N}}]_{\mathrm{IRR_p^{IPP}}}$	annual depreciation rates expressed by $\mathrm{IRR_p^{IPP}}$ rate; and

- non-equality constraints:

  (a) annual gross profit $Z_{\mathrm{A}}^{\mathrm{mod}}$ from the operation of a power station successive to modernization should be greater than or equal to zero:

$$Z_{\mathrm{A}}^{\mathrm{mod}} = S_{\mathrm{A}}^{\mathrm{mod}} - K_{\mathrm{e}}^{\mathrm{mod}} - \rho_{\mathrm{N}} J_{\mathrm{O}} - \rho_{\mathrm{N-M}} J_{\mathrm{mod}} \geq 0, \quad (2.3)$$

  (b) capital expenditure needed for the modernization can be constrained by the financial capacities of IPP or the limitation can result from the adopted technology

$$J_{\mathrm{mod}} \leq J_{\mathrm{mod}}^{\max} \tag{2.4}$$

and

$$J_{\mathrm{mod}} \geq J_{\mathrm{mod}}^{\min}, \tag{2.5}$$

(c) interest rate needs to be at least equal to the interest rate that can be offered to IPP by the capital investment J_0 and J_{mod} on the purchase and modernization of the power station:

$$r \leq \mathrm{IRR}_{\mathrm{p}}^{\mathrm{IPP}}. \tag{2.6}$$

Instead of constraint (2.3) it is possible to impose a stricter restriction on gross annual profit such that the latter should not be lower than the profit obtained prior to the modernization of the power station to cogeneration:

$$Z_{\mathrm{A}}^{\mathrm{mod}} \geq Z_{\mathrm{A}}^{\mathrm{M}} = S_{\mathrm{A}}^{\mathrm{M}} - K_{\mathrm{e}}^{\mathrm{M}} - \rho_{\mathrm{N}} J_{\mathrm{O}} \geq 0. \tag{2.7}$$

The restrictions (2.3)–(2.6) impose the range of admissible solutions (Fig. 2.2). The equation constraint in (2.2) makes it possible to eliminate from calculations one of the decision making variables.

In the generalized case, i.e. taking all decision making variables into consideration in the optimization criterion, it is non-linear and, as a consequence, it is possible that its extreme could be an intermediate point in the set of admissible solutions. The search for its maximum value may, consequently, involve *gradient methods* after complementing the functional using *penalty functions* [2, 3] that can considerably deteriorate its values for the case when constraints are not fulfilled (2.3)–(2.6).

Accounting in the relation (2.1) for instance as decision making variables only for linear quantities:

1. value of the difference between the revenues and cost of exploitation of the power station after modernization $S_{\mathrm{A}}^{\mathrm{mod}} - K_{\mathrm{e}}^{\mathrm{mod}}$
2. capital expenditure J_{mod}

and with the application of non-equality constraints:

1. capital investment J_{mod} is at least equal to the expenditure $J_{\mathrm{mod}}^{\min}$ necessary solely for the construction of inexpensive water boilers in the power station and is lower or equal to the financial capabilities $J_{\mathrm{mod}}^{\max}$ of the IPP (its credit rating) or it results from the technology adopted in modernization

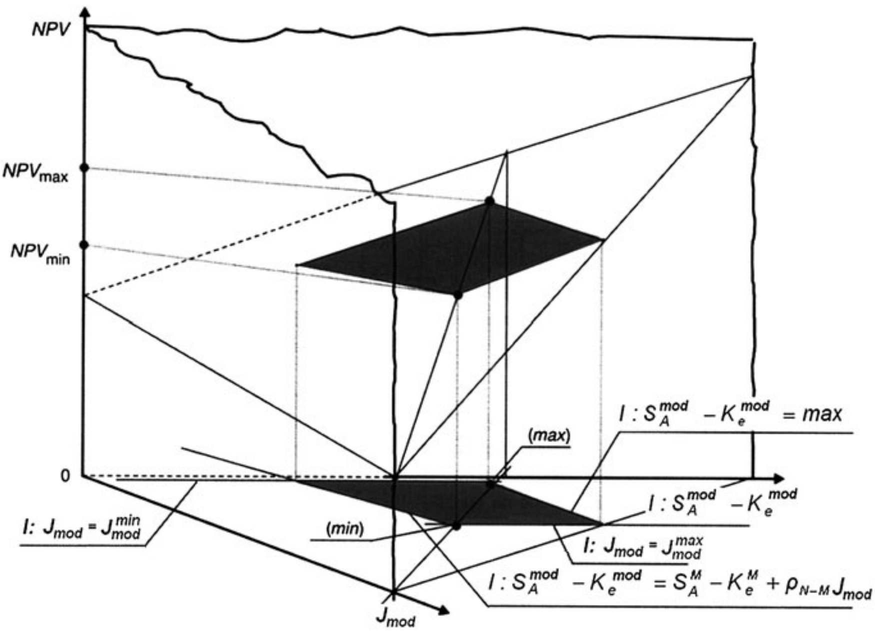

Fig. 2.2 Graphical representation of the extreme of a linear function

$$J_{\text{mod}}^{\min} \leq J_{\text{mod}} \leq J_{\text{mod}}^{\max} \tag{2.8}$$

and

2. constraint (2.7) in order to secure an improvement in the economic efficiency of the power station after modernization, i.e. annual gross profit $Z_{\text{A}}^{\text{mod}}$ should not be lower than the profit prior to the modernization process Z_{A}^{M}:

$$S_{\text{A}}^{\text{mod}} - K_{\text{e}}^{\text{mod}} \geq S_{\text{A}}^{\text{M}} - K_{\text{e}}^{\text{M}} + \rho_{\text{N-M}} J_{\text{mod}}. \tag{2.9}$$

Thus, we have to do with a model of linear programming since the criterion of optimization NPV is in the form of a linear function (Fig. 2.2). The analysis of the functional (2.1) is then considerably simplified. If, additionally, the range of the admissible solutions is limited by a system of linear relations, the functional gains a maximum at a point that is the extreme in this area (Fig. 2.2). If the extreme is gained at two different peaks, this value is also obtained in every point of the section which joins them. The optimal value of $S_{\text{A opt}}^{\text{mod}}$, $K_{\text{e opt}}^{\text{mod}}$ and $J_{\text{mod opt}}$, which secure that the optimization criterion gains the maximum value, can be obtained using *simplex method* [2]. If the constraints are non-linear, the extreme will lie on the border of the area of the admissible solutions.

At the point (max) of the intersection between two lines l: $J_{\text{mod}} = J_{\text{mod}}^{\min}$ and l: $S_{\text{A}}^{\text{mod}} - K_{\text{e}}^{\text{mod}} = \max$ the functional in (2.1) assumes its maximum value.

An optimum procedure to be applied during the modernization of the power station to cogeneration will be the alternative for which case the values of $S_{\mathrm{A}}^{\mathrm{mod}} - K_{\mathrm{e}}^{\mathrm{mod}}$ and J_{mod} will lie as close to this point as possible.

The search for an optimum way of modernization will make it possible to find not only a reply to the question regarding the optimum decision making strategy, i.e. the scope and technology to be adopted during the modernization thus enabling the most effective way of modernization from the economic perspective, but will also help to determine the scope of the economically acceptable price relations between the use of energy carriers and utilization of the natural environment, and the acceptable prices of CO_2 allowances. These are the fundamental questions that need to be answered during the modernization of coal-fired power stations to cogeneration.

2.2 Analysis of Energetic and Economic Efficiency Resulting from Modernization of Coal-Fired Power Station to Cogeneration Using Steam from Turbine Extraction

As noted in Chap. 1, the alternative that is most justified from the thermodynamic point of view involves the modernization of coal-fired power stations to cogeneration using steam from the turbine extraction. This is made sensible due to the lowest possible exergy losses accompanied by the smallest decrease in production of electric energy while maintaining a constant heat stream. This chapter will focus on the analysis of such a mode of operation from the thermodynamic and economic perspective. Due to the fact that the research involves the use of a single extraction point for steam bleeding to a single heater (Fig. 1.3b) the analysis is somewhat simplified. However, it does have a practical justification. This is so because the analytical economic and thermodynamic notations of cogeneration are general in nature. They present a number of regularities that are relevant also during the supply of a number of heaters using a number of turbine extractions.

An examination accounting for a number of exhausts and heaters and involving the identification of their optimum structure is associated with the need to consider a particular power unit and develop a mathematical model applying the energy characteristics of the particular facilities in a power unit and perform numerical calculations. All these parameters are considered in the latter chapters of this book (Chaps. 3–6). The results of analyses are therefore in the form of numerical data gained for specific input data obtained for a case of a particular power station; hence, the results cannot be generalized. This is not the case for analytical notations, which are general and thus, the results can serve for an extensive analysis. In addition, the need to repeat calculations a number of times as well as to perform large numbers of computations for various input data forms an impediment to the application of the mathematical model of the examined phenomenon in a programming environment. Moreover, the utilization of the model is practically possible only by their author—if a specific program interface is developed. This does not pertain to the analytical notation. Therefore, analytical notations are immensely valuable, as they form the expression of the all-time truth: *For the things of this world cannot be made known without a knowledge of mathematics* (Roger Bacon, 1214–1294). Besides being general, analytical forms offer the possibility of gaining some more information about an examined phenomenon. They make it possible to explicitly assess the effect of the particular input data on the output and easily and quickly establish an optimum solution together with a set of the solutions that are close to the optimum. Moreover, it is possible to indicate the direction of the established changes. They permit the discussion and analysis of the results, which plays an important role in engineering. Furthermore, analytical notations make it possible to derive general conclusions. As a result of the application of economic parameters, the solutions gained enable one to state conclusions regarding the economic determinants of the implementation of cogeneration in a power station. Moreover, it is possible to determine the economically justified relations and acceptable ranges of price ranges of energy carriers.

The fundamental inequality constraint during the search for the maximum of the functional (2.1) is the condition that the annual gross profit $Z_{\mathrm{A}}^{\mathrm{mod}}$ gained from the operation of the power station after modernization is at least not lower than the profit prior to the modernization—relation (2.7).

Hence, the prerequisite of the economic justification of the modernization of the power station to cogeneration—under the assumption that its technical condition enables its long further exploitation to be expressed by the relation:

$$\Delta Z_{\mathrm{A}} = Z_{\mathrm{A}}^{\mathrm{mod}} - Z_{\mathrm{A}}^{\mathrm{M}} = Q_{\mathrm{A}} \mathrm{e}_{\mathrm{h}} - \Delta E_{\mathrm{el,A}}^{\mathrm{El,net}} e_{\mathrm{el}} - \Delta K_{\mathrm{A}}^{\mathrm{El}} \geq 0. \tag{2.10}$$

If we were to additionally account for the likelihood of the change in the price of electricity by Δe_{el}, the Eq. 2.10 could take the following form:

$$\begin{aligned} Z_{\mathrm{A}}^{\mathrm{mod}} &= Q_{\mathrm{A}} \mathrm{e}_{\mathrm{h}} + \left(E_{\mathrm{el,A}}^{\mathrm{El,M}} - \Delta E_{\mathrm{el,A}}^{\mathrm{El,net}}\right)\left(e_{\mathrm{el}} + \Delta e_{\mathrm{el}}\right) \\ &\quad - \left(K_{\mathrm{A}}^{\mathrm{El,M}} + \Delta K_{\mathrm{A}}^{\mathrm{El}}\right) \geq Z_{\mathrm{A}}^{\mathrm{M}} = E_{\mathrm{el,A}}^{\mathrm{El,M}} e_{\mathrm{el}} - K_{\mathrm{A}}^{\mathrm{El,M}} \end{aligned} \tag{2.10a}$$

while $E_{\mathrm{el,A}}^{\mathrm{El,M}}$, $K_{\mathrm{el,A}}^{\mathrm{El,M}}$ constitute the net annual production of electricity and annual operating cost of the power plant before its modernization to combined heat and power.

Consequently, the revenues from the sales of heating $Q_{\mathrm{A}} e_{\mathrm{h}}$ have to at least compensate for the reduction of the revenues gained from the sales of energy $\Delta E_{\mathrm{el,A}}^{\mathrm{El,net}} e_{\mathrm{el}}$ and increase of the operating cost in the power station $\Delta K_{\mathrm{A}}^{\mathrm{El}}$. The increase in the annual cost $\Delta K_{\mathrm{A}}^{\mathrm{El}}$ is associated with the capital investment on the modernization of the power station to cogeneration $J_{\mathrm{mod}} = J_{\mathrm{bild}} + J_{\mathrm{switchgear}} + J_{\mathrm{aut}} + J_{\mathrm{turb}}^{\mathrm{mod}} + J_{\mathrm{pip}}^{\mathrm{dist}} + J_{\mathrm{water}}^{\mathrm{inst}} + J_{\mathrm{stat}}^{\mathrm{heat}}$ (investment in buildings, switchgear, systems of automatic control, modernization of turbines for combined heat and power, heat distribution pipeline with network pumps, installation to maintain adequate static pressure of water in the heat distribution pipeline and heat exchanger stations and an additional cost K_p of the power needed to supply pumps necessary to put in motion the pumps in the main heat distribution pipeline and the accessory ones

$$\Delta K_{A}^{El} = (z\rho + \delta_{serv})J_{mod} + K_{P} \tag{2.11}$$

where:

$z\rho J_{mod}$ depreciation;

$\delta_{serv}J_{mod}$ cost of maintenance and overhaul of equipment.

On condition that the consumers of heat are located in the vicinity of the power station, the investment J_{mod} will be relatively smaller, since the term J_{pip}^{dist} has a predominant effect on the value of J_{mod}.

Note that Eq. 2.11 does not account for the increase of the payroll in the power station. It has been assumed that this cost does not increase. The reduction of electric power in power station ΔN_{el}^{El} results from the hot steam $\dot{m}_h$ extraction from the turbine that is fed into a heater. Bleeding of the heating steam concurrently results in decrease in the volume of feedwater heated in the low-pressure regenerative preheater (if condensate from the exchanger with the specific enthalpy of h_{HE} is carried back into the system behind the regenerative preheater) and thus leads to the decrease of steam extraction to feed this heater. The decrease in the electric power of the turbogenerator whose electromechanical efficiency is equal to η_{me} is thus equal to:

$$\Delta N_{el}^{El} = (\dot{m}_h - \Delta\dot{m}^{LPR})(h_u - h_k)\eta_{me}. \tag{2.12}$$

With the application of the balance of energy in the low-pressure regenerator prior to the modernization of the power station to cogeneration (Fig. 1.3a):

$$\dot{m}^{LPR}h_u + (\dot{m}_1 - \dot{m}^{HPR} - \dot{m}_{dea} - \dot{m}^{HPR})h_{cond} = (\dot{m}_1 - \dot{m}_{HPR} - \dot{m}_{dea})h_7, \tag{2.13}$$

and energy balance in this economizer together with the heat distribution center after the modernization of the power station (Fig. 1.3b):

$$\begin{aligned}(\dot{m}^{LPR} - \Delta\dot{m}^{LPR})h_u + \dot{m}h_{HE} + (\dot{m}_1 - \dot{m}^{LPH} - \dot{m}_{dea}\\ - \dot{m}_h - \dot{m}^{LPN} + \Delta\dot{m}^{LPN})h_{cond} = (\dot{m}_1 - \dot{m}^{LPH} - \dot{m}_{dea})h_7\end{aligned} \tag{2.14}$$

we gain the following:

$$\frac{\Delta\dot{m}^{LPR}}{\dot{m}_h} = \frac{h_{HE} - h_{cond}}{h_u - h_{cond}} \tag{2.15}$$

The ratio of ΔN_{el}^{El} to the thermal power of the power station $\dot{Q}_c^{El} = \dot{m}_h(h_u - h_{HE})$ is expressed by the equation:

$$\varepsilon = \frac{\Delta N_{\mathrm{el}}^{\mathrm{El}}}{\dot{Q}_{\mathrm{c}}^{\mathrm{El}}} = \left(1 - \frac{\Delta \dot{m}^{\mathrm{LPR}}}{\dot{m}_{\mathrm{h}}}\right) \frac{(h_{\mathrm{u}} - h_{\mathrm{k}})\eta_{\mathrm{me}}}{h_{\mathrm{u}} - h_{\mathrm{HE}}} = \frac{(h_{\mathrm{u}} - h_{\mathrm{k}})\eta_{\mathrm{me}}}{h_{\mathrm{u}} - h_{\mathrm{cond}}} \tag{2.16}$$

The value of ε in the examined example assumes a constant value that is regardless of the power $\dot{Q}_{\mathrm{c}}^{\mathrm{El}}$, $\varepsilon = \mathrm{const}$. The value of ε is relative only to thermodynamic parameters of the medium that circulates in the system. The higher the pressure of the steam supplied to the heater, the higher the energy losses in the system [4, 5] and the greater the values of ε and $\Delta N_{\mathrm{el}}^{\mathrm{El}}$; hence, the greater the decrease in the production of energy $\Delta E_{\mathrm{el,A}}^{\mathrm{El}}$ for the same volume of heat Q_{A}. For these reasons the alternatives (a) with the bleeding of heating steam from extractions 1, 2 or 3 (Fig. 1.3) are not justified from the economic or thermodynamic perspective. The differences regarding capital expenditure for the case of alternative (a) are so small that their decrease for the case of alternative with heating steam extraction from points 1, 2 and 3 do not compensate for the losses associated with decrease in the revenue from the sale of electricity in comparison to the alternative with heating steam extraction at point u. Additionally, one has to bear in mind that intake of heating steam at extraction 1 and 2 results in the reduction of the volume of coal combustion in the boiler due to smaller steam bleed for interstage reheat purposes. The values of parameter ε for these alternatives are several times, i.e. 3 or 4 times higher than the parameter for the alternative for extraction at point u.

Due to the possibility of practically applying two or more extractions ($i = 1, 2, 3\ldots$) of the regeneration steam from the low-pressure section of the turbine for the purposes of heating (since it can contribute to the reduction of energy losses in the system and concurrently, decrease of the value of $\Delta E_{\mathrm{el,A}}^{\mathrm{El}}$), the value of the coefficient ε has to be calculated as the weighted mean [6]:

$$\varepsilon = \frac{Q_1}{\sum_i Q_i}\varepsilon_1 + \frac{Q_2}{\sum_i Q_i}\varepsilon_2 + \frac{Q_3}{\sum_i Q_i}\varepsilon_3 + \cdots \tag{2.16a}$$

where $\sum_i Q_i = Q_{\mathrm{A}}$ and the values of ε_1, ε_2, $\varepsilon_3\ldots$ refer to the heat generated from the first, second, etc. extraction in a series of connected heaters.

Using Eq. 2.16 to calculate the value of coefficients ε_1, ε_2, $\varepsilon_3\ldots$, in this equation the role of enthalpy h_{cond} is played by the subsequent enthalpy of the water fed into ith ($i = 1, 2, 3\ldots$) regenerative preheater in LPR (for the first regenerative preheater from the side of the condenser this enthalpy is equal to h_{cond}, Fig. 1.3). However, if this fact was not accounted for and h_{cond} was substituted in each of the extractions the effect on the value of coefficient ε would be inconsiderable [6].

The values of the proportions of the particular heat batches, i.e. from the first Q_1/Q_{A} and the following Q_2/Q_{A}, $Q_3/Q_{\mathrm{A}}\ldots$ have to be determined using a chart characterizing the demand for heat together with the chart of external temperatures and characteristics of district heating network [3, 4, 6].

From the analysis of the relation (2.10) it results that a considerable thermodynamic and economic problem is associated with the selection of heaters HE connected in a series from the successive extractions. An increase in their number leads to the improvement in the thermodynamic effect of cogeneration since the total electrical energy production $\Delta E^{El}_{el,A}$ in the power station decreases, which is accompanied by an increase of the capital expenditure necessary for the modernization to cogeneration. There is an optimum solution in terms of technology and economic efficiency whose finding is the aim of the analysis. Concurrently, the lower the pressure of the extracted steam, the greater the effects in terms of technology and economic efficiency. The pressures of saturation are, however, limited to the most remote extraction points by the temperatures corresponding to them. In the lowest extraction point, i.e. the one situated closest to the condenser, the low pressure limit corresponds to the temperature necessary to obtain hot water in the district distribution system in the non-heating season. Concurrently, the extraction point found most remote from the condenser has to secure the demand for heat in the peak season.

The net decrease of the output of electrical energy to gain useful heat from CHP plant is equal to:

$$\Delta E^{El}_{el,A} = \int_0^{\tau_A} \varepsilon \dot{Q}^{El}_{c} d\tau = \varepsilon Q_A. \tag{2.17}$$

The partial efficiency of the heat production in the power station is equal to [1, 3–8]:

$$\eta_{Ec} = \frac{\dot{Q}^{El}_{c}}{\dot{P}(\mathrm{NCV}) - \dfrac{N^{El}_{el} - \Delta N^{El}_{el}}{\eta_{Eel}}} = \frac{\eta_{Eel}}{\varepsilon}, \tag{2.18}$$

while the stream of chemical energy of the coal combustion in the boiler of the power station is equal to $\dot{P}(\mathrm{NCV}) = N^{El}_{el}/\eta_{Eel}$. For the energy efficiency of the station $\eta_{Eel} = 0.36$ and e.g. the value of $\varepsilon \cong 0.1$ this efficiency is equal to ca. 3.6, which is a very good result from the thermodynamics point of view.

By substitution of (2.11) and (2.17) into (2.10) we can obtain the minimum price of heat Q_A that forms a condition necessary to meet to ensure that the process of power station modernization to cogeneration is economically justified:

$$e_h \geq \frac{(z\rho + \delta_{serv})J_{mod} + K_P}{Q_A} + \varepsilon\, e_{el}. \tag{2.19}$$

When in the relation (2.19) we obtain an equality sign ($\Delta Z_A = 0$), the price e_h expresses the specific cost of heat generation in the power station (cost per energy unit):

$$e_h \equiv k_h. \tag{2.19a}$$

The specific cost of heat production can be reduced as a result of restoring the original electrical power output by burning an additional batch of coal whose net calorific value NCV is equal to:

$$\Delta\dot{P}(\mathrm{NCV}) = \frac{\Delta N_{\mathrm{el}}^{\mathrm{El}}}{\eta_{\mathrm{Eel}}}. \tag{2.20}$$

In most cases, the boiler has a certain efficiency surplus [9] and the system in the high-and middle-pressure section of the turbine makes it possible to increase the stream of steam flowing through it. The value of the stream of chemical energy of the fuel $\Delta\dot{P}(\mathrm{NCV})$ used for the compensation of electric energy stream is also the ratio of the chemical energy needed to produce heat and hence, the relation:

$$\frac{\dot{Q}_{\mathrm{c}}^{\mathrm{El}}}{\Delta\dot{P}(\mathrm{NCV})} = \frac{Q_{\mathrm{A}}\eta_{\mathrm{Eel}}}{\Delta E_{\mathrm{el,\,A}}^{\mathrm{El}}} = \frac{\eta_{\mathrm{Eel}}}{\varepsilon} \tag{2.21}$$

is the partial efficiency of heat production in the power station (Eq. 2.18). The specific cost of heat accounting for compensation is equal to:

$$k_{\mathrm{h}}^{\mathrm{com}} = \frac{(z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_{\mathrm{P}} + e_{\mathrm{coal}}\int_0^{\tau_{\mathrm{A}}} \Delta\dot{P}(\mathrm{NCV})d\tau + \Delta K_{\mathrm{env}}^{\mathrm{com}}}{Q_{\mathrm{A}}}. \tag{2.22}$$

If we were to additionally account for the possibility of a change in the price of electricity by Δe_{el}, Eq. 2.22 would take the following form (compare Eq. 2.10a):

$$k_{\mathrm{h}}^{\mathrm{com}} = \frac{(z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_{\mathrm{P}} + e_{\mathrm{coal}}\int_0^{\tau_{\mathrm{A}}} \Delta\dot{P}(\mathrm{NCV})d\tau + \Delta K_{\mathrm{env}}^{\mathrm{com}} - E_{\mathrm{el,\,A}}^{\mathrm{El,\,M}}\Delta e_{\mathrm{el}}}{Q_{\mathrm{A}}}. \tag{2.22a}$$

Equation 2.22a results from the relation

$$Z_{\mathrm{A}}^{\mathrm{mod}} = Q_{\mathrm{A}}k_{\mathrm{h}}^{\mathrm{com}} + E_{\mathrm{el,\,A}}^{\mathrm{El,\,M}}(e_{\mathrm{el}} + \Delta e_{\mathrm{el}}) - (K_{\mathrm{A}}^{\mathrm{El,\,M}} + \Delta K_{\mathrm{A}}^{\mathrm{El}}) = Z_{\mathrm{A}}^{\mathrm{M}} = E_{\mathrm{el,\,A}}^{\mathrm{El,\,M}}e_{\mathrm{el}} - K_{\mathrm{A}}^{\mathrm{El,\,M}} \tag{2.22b}$$

By subtracting from (2.19) the relation (2.22) we will gain the value of the decrease of the specific cost of heat generated after compensation:

$$\Delta k_{\mathrm{h}} = \varepsilon\left(e_{\mathrm{el}} - \frac{e_{\mathrm{coal}}}{\eta_{\mathrm{Eel}}}\right) - \frac{\Delta K_{\mathrm{env}}^{\mathrm{com}}}{Q_{\mathrm{A}}} \tag{2.23}$$

where e_{coal} means the specific coal price (price per energy unit).

The increase of the charges $\Delta K_{\mathrm{env}}^{\mathrm{com}}$ associated with the natural environment pollution is caused by the additional stream of chemical energy of the coal combustion in the boiler annually and is expressed with the equation:

$$\Delta P_{\mathrm{A}}(\mathrm{NCV}) = \int\limits_{0}^{\tau_A} \Delta \dot{P}(\mathrm{NCV}) \mathrm{d}\tau = \frac{\varepsilon Q_{\mathrm{A}}}{\eta_{\mathrm{Eel}}}. \tag{2.24}$$

In the search for an optimum alternative of feeding heaters during cogeneration Eq. 2.1 could be replaced with a relation that states the minimum specific cost of heat production. The value of NPV gains its maximum when the specific cost k_{h} keeps at its lowest. The criterion (2.1) can be written in the form of a relation:

- for cogeneration without compensation of electricity production:

$$k_{\mathrm{h}} \rightarrow \min. \tag{2.25}$$

- for cogeneration with compensation of electricity production:

$$k_{\mathrm{h}}^{\mathrm{com}} \rightarrow \min. \tag{2.26}$$

2.3 Technical and Economical Effectiveness of Conventional Coal-Fired Condensing Power Plants Repowered by Gas Turbine with their Modernization for Combined Heat and Power Production

Instead of combustion of additional volume of coal, thus increasing the production of electricity, it is possible to repower the existing coal-fired unit adapted to combined heat and power by an additional gas turbine and a heat recovery steam generator. The parallel coupling of the two systems could in this case for instance, involve superheating of inter-stage steam and heating of feedwater that supplies the steam boiler in the recovery boiler by exhaust gases from a gas turbine. The closed regeneration high-and low-pressure regenerative preheaters could then be excluded from the system and the only remaining part would be a deaerator (Fig. 1.3c). The superheating of inter-stage steam in the heat recovery steam generator thus enables one to avoid the problems associated with an additional load on the blading system in this section of the steam turbine that has an already increased steam flow through it due to the exclusion of high-pressure regenerative extractions from the system. Additionally, exergy losses in this section of the heat recovery steam generator are decreased due to the reduction of the difference between the temperatures of exhaust flue gas and steam, thus increasing the efficiency of electricity production in the system.

A prerequisite for the preservation of economic effectiveness of repowering of a power plant with a gas turbine and a heat recovery steam generator and its modernization to the combined heat and power is that the annual revenues from the sales of heat and additional electricity exceeds the increase of the annual operating costs of the power plant associated with its repowering $\Delta K_{\mathrm{A}}^{\mathrm{El}}$ (compare Eq. 8.7)

$$\Delta Z_{\mathrm{A}} = Q_{\mathrm{A}} e_{\mathrm{h}} + (E^{\mathrm{GT}}_{\mathrm{el,A}} - \Delta E^{\mathrm{ST}}_{\mathrm{el,A}}) e_{\mathrm{el}} - \Delta K^{\mathrm{El}}_{\mathrm{A}} \geq 0. \tag{2.27}$$

The decrease in the production of electricity in the steam turbogenerator $\Delta E^{\mathrm{ST}}_{\mathrm{el,A}} = \int_0^{\tau_A} \Delta N^{\mathrm{ST}}_{\mathrm{el}} d\tau$ results from heating steam extraction $\dot{m}_{\mathrm{h}}$ to supply the network heater and additional steam $\Delta \dot{m}_{\mathrm{dea}}$ for the purposes of deaerator (due to the exclusion of high-pressure regeneration). Concurrently, due to the complete exclusion of high-and low-pressure regenerative preheaters we deal with an increase of the output from the intermediate-and low-pressure section of the steam turbine. The increase of the power output from the steam turbogenerator is equal to

$$\Delta N^{ST}_{\mathrm{el}} = [(\dot{m}_{\mathrm{h}} + \Delta \dot{m}_{\mathrm{dea}})(h_{\mathrm{u}} - h_{\mathrm{k}}) - \dot{m}^{\mathrm{HPR}}(h_3 - h_{\mathrm{k}}) - \dot{m}^{\mathrm{LPR}}(h_{\mathrm{u}} - h_{\mathrm{k}})]\eta_{\mathrm{me}}. \tag{2.28}$$

From the energy balance of the deaerator prior to and after repowering it by a gas turbine in a power plant after its modernization to the combined heat and power (Fig. 1.3a and c) results in the increase in the deaerating steam stream due to the exclusion of high-pressure regeneration exhaust equal to

$$\Delta \dot{m}_{\mathrm{dea}} = \dot{m}^{\mathrm{HPR}} \frac{h_6 - h_7}{h_{\mathrm{u}} - h_7}. \tag{2.29}$$

The relative decrease of electricity production in the steam turbogenerator for the purposes of district heating $\Delta E^{\mathrm{ST}}_{\mathrm{el,A}} = \int_0^{\tau_{\mathrm{A}}} \varepsilon^{\mathrm{GT}} \dot{Q}^{\mathrm{El}}_{\mathrm{c}} d\tau$, which is expressed with thermodynamic parameters of the circulating medium, in this case is equal to

$$\begin{aligned}
\varepsilon^{\mathrm{GT}} = \frac{\Delta N^{\mathrm{ST}}_{\mathrm{el}}}{\dot{Q}^{\mathrm{El}}_{\mathrm{c}}} = \Bigg[& \left(1 + \frac{h_9 - h_8}{h_2 - h_6} \frac{h_{\mathrm{HE}} - h_{\mathrm{cond}}}{h_5 - h_{\mathrm{cond}}} \frac{h_6 - h_7}{h_u - h_8}\right)(h_{\mathrm{u}} - h_{\mathrm{k}}) \\
& - \frac{h_9 - h_8}{h_2 - h_6} \frac{h_{\mathrm{HE}} - h_{\mathrm{cond}}}{h_5 - h_{\mathrm{cond}}} \frac{h_{\mathrm{u}} - h_7}{h_{\mathrm{u}} - h_8}(h_3 - h_{\mathrm{k}}) \\
& - \frac{h_7 - h_{\mathrm{cond}}}{h_{\mathrm{u}} - h_{\mathrm{cond}}} \left(1 - \frac{h_9 - h_8}{h_2 - h_6} - \frac{h_9 - h_8}{h_2 - h_6} \frac{h_7 - h_6}{h_{\mathrm{u}} - h_7} - \frac{h_8 - h_7}{h_{\mathrm{u}} - h_7}\right) \\
& \times \frac{h_{\mathrm{HE}} - h_{\mathrm{cond}}}{h_5 - h_{\mathrm{cond}}} \frac{h_{\mathrm{u}} - h_7}{h_{\mathrm{u}} - h_8}(h_{\mathrm{u}} - h_{\mathrm{k}}) \Bigg] \frac{\eta_{\mathrm{me}}}{h_{\mathrm{u}} - h_{\mathrm{HE}}}.
\end{aligned} \tag{2.30}$$

Equation 2.30 is derived after obtaining the energy balances of: deaerators (Fig. 1.3a and c), systems of high- and low-pressure regeneration HPR and LPR (Fig. 1.3a), and energy balance for the section behind the heater (Fig. 1.3c). If $\dot{Q}^{\mathrm{El}}_{\mathrm{c}} =$ const, $h_5 =$ const and in this case $\varepsilon^{\mathrm{GT}} =$ const, and $\Delta E^{\mathrm{El}}_{\mathrm{el,A}} = \varepsilon^{GT} Q_{\mathrm{A}}$. If $\dot{Q}^{\mathrm{El}}_{\mathrm{c}} \neq$ const, it is possible to determine the mean value of h_5 for the examined range of fluctuation of the thermal power in the range $\dot{Q}^{\mathrm{El}}_{\mathrm{c}} \in \langle 0;\ \dot{Q}^{\mathrm{El}}_{\mathrm{c,max}} \rangle$. The maximum thermal power $\dot{Q}^{\mathrm{El}}_{\mathrm{c,max}}$ of the power plant after its modernization to combined heat and power results from the maximum accessible stream of extraction steam. The maximum extraction of heating steam has to secure that the remaining flow of the condensing stream guarantees a sufficient cooling of the low-pressure section of the turbine's rotor. This stream should be equal to around 10% of the fresh steam flowing into the turbine.

The decrease of the power of the steam turbogenerator in the case of repowering of the coal-fired system by a gas turbine and a heat recovery steam generator (Eq. 2.28) is smaller than for the case of a power plant that is only adapted to combined heat and power (Eq. 2.12). This results mostly from the total exclusion of the high-and low-pressures regeneration systems. The mass streams $\dot{m}^{\mathrm{HPR}}$ and $\dot{m}^{\mathrm{LPR}}$ form a considerable share of the fresh steam stream $\dot{m}_1$ produced in the steam boiler (in the above cases we assume a constant value for all analyzed cases (Fig. 1.3a, b, c). Practically speaking, the entire internal load of the power plant related to steam is equal to $\dot{m}^{\mathrm{HPR}} + \dot{m}_{\mathrm{dea}} + \dot{m}^{\mathrm{LPR}} \cong 0.3\dot{m}_1$ while $\dot{m}^{\mathrm{HPR}} \cong \dot{m}^{\mathrm{LPR}}$ and $\dot{m}_{\mathrm{dea}} \cong 0.03\dot{m}_1$. From Eq. 2.28 it stems that if the heating steam stream $\dot{m}_h$ fulfills the relation

$$\dot{m}_{\mathrm{h}} < \dot{m}^{\mathrm{HPR}} \frac{h_3 - h_{\mathrm{k}}}{h_{\mathrm{u}} - h_{\mathrm{k}}} + \dot{m}^{\mathrm{LPR}} - \Delta\dot{m}_{\mathrm{dea}} \tag{2.31}$$

the power of the steam turbine in a combined system repowered by a gas turbine (Fig. 1.3c) is larger than the power of the power plant $N_{\mathrm{el}}^{\mathrm{El}}$ (Fig. 1.3a) ($\Delta N_{\mathrm{el}}^{\mathrm{ST}} < 0$, Eq. 2.28). The variable $\varepsilon^{\mathrm{GT}}$ assumes a negative value and it means an increase in the electricity production in the steam turbogenerator.

An important quantity in the circumstances of a system repowered by a gas turbine is associated with the incremental efficiency of electricity production, which is defined as the ratio of the increase of electric power of the system to the chemical energy of the natural gas that fuels the turbine. This efficiency can be determined for the combined heat and power system in the form

$$\eta_{\Delta} = \frac{N_{\mathrm{el}}^{\mathrm{GT}} + (-\Delta N_{\mathrm{el}}^{\mathrm{ST}})}{(\dot{P}(\mathrm{NCV}))_{\mathrm{gas}}}. \tag{2.32}$$

It assumes its highest value for $\dot{m}_{\mathrm{h}} = 0$, which corresponds to the completely condensing operation of a unit. The incremental efficiency makes it possible to compare a system based on two fuels with that of a classical in-series gas-steam single-fuel system based on the same gas turbine. The minus sign in the numerator of Eq. 2.32 results from the necessity of accounting for the sign of power increase $\Delta N_{\mathrm{el}}^{\mathrm{ST}}$ (Eq. 2.28) following a change in the power $\dot{Q}_{\mathrm{c}}^{\mathrm{El}}$. The incremental efficiency can also be interpreted as the apparent efficiency of the gas turbogenerator operating in a gas-steam dual system. From Eq. 2.32 we can derive that $\eta_{\Delta} = \eta_{\mathrm{GT}} + \Delta\eta_{\mathrm{GT}}$ where $\Delta\eta_{\mathrm{GT}} = -\Delta N_{\mathrm{el}}^{\mathrm{ST}}/(\dot{P}(\mathrm{NCV}))_{\mathrm{gas}}$ denotes the apparent increase of the efficiency of the gas turbogenerator, while $(\dot{P}(\mathrm{NCV}))_{\mathrm{gas}} = N_{\mathrm{el}}^{\mathrm{GT}}/\eta_{\mathrm{GT}}$ is the chemical energy stream of the gas combusted in the gas turbine with efficiency η_{GT}.

For $\dot{m}_{\mathrm{h}} = 0$ the apparent efficiency of electricity production in the steam turbogenerator assumes its maximum value

$$\chi = \frac{N_{\mathrm{el}}^{\mathrm{El}} + (-\Delta N_{\mathrm{el}}^{\mathrm{ST}})}{(\dot{P}(NCV))_{\mathrm{coal}}} = \frac{[\dot{m}_1(h_1 - h_2 + h_3 - h_{\mathrm{k}}) - (\dot{m}_{\mathrm{dea}} + \Delta\dot{m}_{\mathrm{dea}} + \dot{m}_{\mathrm{h}})(h_{\mathrm{u}} - h_{\mathrm{k}})]\eta_{\mathrm{me}}\eta_{\mathrm{b}}}{\dot{m}_1(h_1 - h_9)} \tag{2.33}$$

where the chemical energy stream of the coal combustion in the steam boiler (Fig. 1.3c), whose energy efficiency is η_{b}, is equal to

$$(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}} = \frac{\dot{m}_1(h_1 - h_9)}{\eta_{\mathrm{b}}}. \tag{2.34}$$

The total energy efficiency of the power plant after its modernization to the combined heat and power and repowered by a gas turbine (Fig. 1.3c) is equal to

$$\eta_{\mathrm{Egs}} = \frac{N_{\mathrm{el}}^{\mathrm{El}} + (-\Delta N_{\mathrm{el}}^{\mathrm{ST}}) + N_{\mathrm{el}}^{\mathrm{GT}} + \dot{Q}_{\mathrm{c}}^{\mathrm{El}}}{(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}} + (\dot{P}(\mathrm{NCV}))_{\mathrm{gas}}} \tag{2.35}$$

and is higher than the energy efficiency of the power plant prior to the modernization (Fig. 1.3a)

$$\eta_{\mathrm{Eel}} = \frac{N_{\mathrm{el}}^{\mathrm{El}}}{(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}}^{\mathrm{El}}} = \frac{[\dot{m}_1(h_1 - h_2 + h_3 - h_{\mathrm{k}}) - \dot{m}^{\mathrm{HPR}}(h_3 - h_{\mathrm{k}}) - (\dot{m}_{\mathrm{dea}} + \dot{m}^{\mathrm{LPR}})(h_{\mathrm{u}} - h_{\mathrm{k}})]\eta_{\mathrm{me}}\eta_{\mathrm{b}}}{\dot{m}_1(h_1 - h_9) + (\dot{m}_1 - \dot{m}^{\mathrm{HPR}})(h_3 - h_2)} \tag{2.36}$$

while the chemical energy stream of the coal combustion in the steam boiler in the power plant is equal to

$$(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}}^{\mathrm{El}} = \frac{\dot{m}_1(h_1 - h_9) + (\dot{m}_1 - \dot{m}^{\mathrm{HPR}})(h_3 - h_2)}{\eta_{\mathrm{b}}}. \tag{2.37}$$

The annual net production of electricity in the gas turbogenerator with the gross power of $N_{\mathrm{el}}^{\mathrm{GT}}$ is equal to

$$E_{\mathrm{el,A}}^{\mathrm{GT}} = N_{\mathrm{el}}^{\mathrm{GT}}(1 - \varepsilon_{\mathrm{el}})\tau_{\mathrm{A}}, \tag{2.38}$$

where:

$\varepsilon_{\mathrm{el}}$ relative coefficient of power station internal load;
τ_{A} annual in-service time of the power plant.

The minimum power of the gas turbine $N_{\mathrm{el\,min}}^{\mathrm{GT}}$ for the adopted alternative for parallel coupling of the two systems results from the necessity of preserving the thermal power of the exhaust gas $\dot{H}_{fg}$ (Eq. 2.43), which subsequently replaces the thermal power of the inter-stage reheater and the thermal power of the completely excluded high- and low-pressure regeneration heaters $\dot{Q}^{SR} + \dot{Q}^{HPR} + \dot{Q}^{LPR}$

$$\dot{H}_{fg} = \frac{N_{el\,min}^{GT}}{\eta_{GT}} - N_{el\,min}^{GT} = \frac{(\dot{Q}^{SR} + \dot{Q}^{HPR} + \dot{Q}^{LPR})}{\eta_{HRSG}}, \tag{2.39}$$

while

$$\dot{Q}^{SR} = \dot{m}_1(h_3 - h_2), \tag{2.40}$$

$$\dot{Q}^{HPR} = \dot{m}_1(h_9 - h_8), \tag{2.41}$$

$$\dot{Q}^{LPR} = (\dot{m}_1 - \dot{m}_{dea} - \Delta\dot{m}_{dea})(h_7 - h_5), \tag{2.42}$$

where:

η_{HRSG} efficiency of heat recovery steam generator (η_{HRSG} is in particular relative to the temperature t_5 of feedwater into low-pressure section, i.e. the demand for low-temperature energy; a decrease of t_5 is accompanied by an increase of η_{HRSG}).

Concurrently, the temperatures of exhaust gases in the heat recovery steam generator have to be higher than the temperatures of interstage steam and feed water. An increase in the power of the gas turbine above $N_{el\,min}^{GT}$ would increase the temperature of gases at the exhaust of the heat recovery steam generator. Thus, this solution would be a drawback from the thermodynamics perspective and, presumably, economic one as well. In fact, this is relative to the structure of energy carrier prices.

2.4 Technical Effectiveness of Conventional Coal-Fired Condensing Power Plants Repowered by a Gas Turbine in Parallel Systems

The power of the gas turbine and stream of enthalpy of the gases at its exhaust that are fed into the heat recovery steam generator for the case of parallel method applied in the modernization of the power plant (Figs. 1.4, 2.3) can be expressed with an equation [4]

$$\dot{H}_{fg} = \frac{N_{el}^{GT}}{\eta_{GT}} - N_{el}^{GT} = \frac{\sum_i \dot{Q}_i}{\eta_{HRSG}}, \tag{2.43}$$

where:

$\dot{Q}_{\mathrm{i}}$ thermal power transferred via flue exhaust from the gas turbine to the steam, feed water and condensate in an *ith exhaust gas-steam* or *exhaust gas–water* heater situated in the heat recovery steam generator;

η_{HRSG} efficiency of the heat recovery steam generator.

From Eq. 2.43 it stems that the parallel system is characterized with a large freedom regarding the selection of the power of a gas turbine and a better possibility of exploitation of exhaust gases in comparison to the in-series system (Hot Windbox, Fig. 1.4) [4].

The power output of the turbine $N_{\mathrm{el}}^{\mathrm{GT}}$ can be arbitrarily large and is relative only to the volume and value of power $\dot{Q}_{\mathrm{i}}$. These parameters, in turn, can only be limited by the economic factors associated with the financial capabilities of an investor. For thermodynamic reasons, the higher the degree in which the gas turbine overtakes the role of the coal-fired boiler (which is the source of exergy losses in the system), the higher the energy efficiency of the production of electricity. In fact, the increase in the power output of the turbine can be limited by the possible load on the blading of the steam turbine or the admissible overload of the electric generator coupled with it. Thus, in a parallel system one can consider installation of the following heaters in the heat recovery steam generator: closed feedwater heaters for the production of high-, intermediate-and low-pressure steam, closed interstage reheater, closed regenerative high-and low-pressure heaters and closed condensate preheaters alternatively with deaeration evaporator [4]. The number of possible combinations of closed feedwater heaters and their distributions in the boiler is very large. Some of them can be immediately rejected since their realization would practically lead to overloading the steam turbine, i.e. closed feedwater heaters for the production of high-and intermediate-pressure steam along with exclusion of regenerative steam extractions in the turbine without a change in the load on the coal boiler. The thermodynamic criterion of the selection and distribution of closed feedwater heaters should then be the reduction of the total of exergy losses in the boiler: the newly designed heat recovery steam generator and the existing coal-fired one and accounting for such limitations as the technically admissible changes in the load of the coal fire boiler and steam turbine. As mentioned above, the coal-fired boiler has the largest exergy losses.

This chapter takes into consideration the following cases of repowering and distribution of closed feedwater heaters in the heat recovery steam generator (Fig. 2.3):

- inter-stage reheater + high-pressure regenerator + low-pressure regenerator—*alternative I* (Fig. 2.3a),
- high-and intermediate-pressure superheaters + high-and intermediate-pressure steam evaporators + high-and intermediate-pressure water pre-heaters + low-pressure regenerator—*alternative II* (Fig. 2.3b).

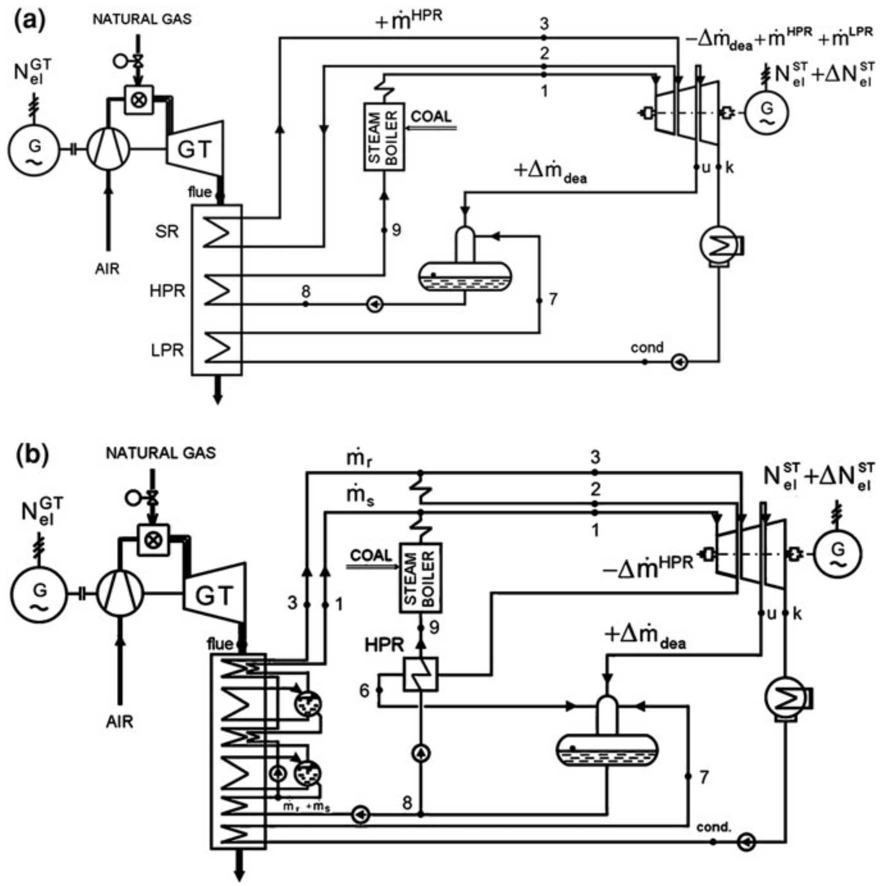

Fig. 2.3 Diagrams of: **a** Alternative I: conventional coal-fired condensing power plant repowered by gas turbine and heat recovery steam generator in a parallel system; SR, HPR, LPR—in turn steam reheating, high pressure regeneration, low pressure regeneration. **b** Alternative II; conventional coal-fired condensing power plant repowered by gas turbine and heat recovery steam generator with closed feedwater heaters for production of high and intermediate pressure steam and a low-pressure regeneration in a parallel system

The total energy efficiency of the power plant repowered with a gas turbine can be expressed with the equation

$$\eta_{\mathrm{Eel}}^{\mathrm{rep}} = \frac{N_{\mathrm{el}}^{\mathrm{ST}} + \Delta N_{\mathrm{el}}^{\mathrm{ST}} + N_{\mathrm{el}}^{\mathrm{GT}}}{(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}} + (\dot{P}(\mathrm{NCV}))_{\mathrm{gas}}}, \tag{2.44}$$

and is greater than the energy efficiency of the power plant priori to its repowering (Fig. 1.3a, Eq. 2.36).

The variable ΔN_{el}^{ST} is Eq. 2.44 which means the increase in the power of the steam turbine following the process of modernization involving repowering by a gas turbine.

For combined heat and power system (Eq. 2.32), the increase in the production of electricity in the situation of repowering a coal-fired power plant with a gas turbine can be expressed as

$$\eta_{\Delta} = \frac{N_{el}^{GT} + \Delta N_{el}^{ST}}{(\dot{P}(\mathrm{NCV}))_{gas}}. \tag{2.45}$$

In the literature in this field there is often an additional term $(\Delta\dot{P}(\mathrm{NCV}))_{coal}\,\eta_{Eel}$ added to the denominator of Eq. 2.45, which apparently represents electrical energy, which would have been produced from the coal that has been saved $(\Delta\dot{P}(\mathrm{NCV}))_{coal}$ (due to the modernization of the system in alternative I as the difference between fuel use in Eqs. 2.37 and 2.48, while in alternative IIa by Eqs. 2.37 and 2.54). Attributing apparent benefits to repowered power plants does not appear to be purposeful.

In an similar manner we can define the efficiency of generating electricity in the steam turbine

$$\chi = \frac{N_{el}^{ST} + \Delta N_{el}^{ST}}{(\dot{P}(\mathrm{NCV}))_{coal}}. \tag{2.46}$$

The thermodynamic analysis of alternatives I and II is presented below.

2.4.1 Alternative I

The increase of the power of the intermediate and low-pressure section of the steam turbogenerator following repowering of the power plant due to the complete exclusion of the high-and low-pressure regenerative heaters is equal to

$$\Delta N_{el}^{ST} = [\dot{m}^{HPR}(h_3 - h_k) + \dot{m}^{LPR}(h_u - h_k) - \Delta\dot{m}_{dea}(h_u - h_k)]\eta_{me}. \tag{2.47}$$

From the energy balance of deaerator prior to and following repowering (Figs. 1.3a and 2.3a) it results in the increase of the steam of deaerating steam $\Delta\dot{m}_{dea}$ expressed by Eq. 2.29.

The chemical energy stream of coal combustion in the steam boiler after its repowering (Fig. 2.3a) is equal to

$$(\dot{P}(\mathrm{NCV}))_{coal} = \frac{\dot{m}_1(h_1 - h_9)}{\eta_b}. \tag{2.48}$$

If we are not familiar with the characteristics of the efficiency of the boiler in the function of its operating load, then in the calculations we can assume the efficiency of the boiler prior to and following its repowering to be approximately the same.

The power of the gas turbine N_{el}^{GT} results from the necessary thermal power of the exhaust gas from $\dot{H}_{fg}$, which that substitutes the thermal power of inter-stage reheater $\dot{Q}^{SR}$, thermal power of the completely excluded high- and low-pressure regeneration reaheaters $\dot{Q}^{HPR}$ and $\dot{Q}^{LPR}$ (Eqs. 2.39–2.42).

2.4.2 Alternative II

In alternative II (Fig. 2.3b) we take into account the following conditions:

- coal-fired boiler operates with a smaller load (a decrease by the value of high-pressure steam produced in the heat recovery steam generator—*alternative IIa*), and
- operation of the coal-fired boiler under a constant load—*alternative IIb*.

The increase of the power of the steam turbine after repowering of the power plant is equal to:

- alternative IIa (steam stream produced in the coal-fired boiler is equal to $\dot{m}_9 = \dot{m}_1 - \dot{m}_s$)

$$\Delta N_{el}^{ST} = [(\Delta\dot{m}^{HPR} + \dot{m}_r)(h_3 - h_k) + (\dot{m}^{LPR} - \Delta\dot{m}_{dea})(h_u - h_k)]\eta_{me}, \tag{2.49}$$

- alternative IIb (steam stream produced in the coal-fired boiler is equal to $\dot{m}_9 = \dot{m}_1$)

$$\Delta N_{el}^{ST} = [\dot{m}_s(h_1 - h_2 + h_3 - h_k) + \dot{m}_r(h_3 - h_k) + (\dot{m}^{LPR} - \Delta\dot{m}_{dea})(h_u - h_k)]\eta_{me}. \tag{2.50}$$

From the energy balance HPR and for the deaerator prior to and following repowering of the power plant (Figs. 1.3a, 2.3b) it stems in the decrease of the stream of regeneration steam due to the decrease of the workload on the coal-fired boiler by the value of stream $\dot{m}_s$ equal to

$$\Delta\dot{m}^{HPR} = \dot{m}_s \frac{h_9 - h_8}{h_2 - h_6}, \tag{2.51}$$

and the necessary increase of deaerating steam due to the decrease of stream of high-pressure regeneration steam and the production of steam in heat recovery steam generator $\dot{m}_r$ equal to

$$\Delta\dot{m}_{dea} = \dot{m}_r \frac{h_8 - h_7}{h_u - h_7} + \Delta\dot{m}^{HPR} \frac{h_6 - h_7}{h_u - h_7}, \tag{2.52}$$

- alternative IIb

$$\Delta \dot{m}_{\mathrm{dea}} = (\dot{m}_{\mathrm{s}} + \dot{m}_{\mathrm{r}}) \frac{h_8 - h_7}{h_{\mathrm{u}} - h_7} \ , \tag{2.53}$$

where $\dot{m}_{\mathrm{s}}$ and $\dot{m}_{\mathrm{r}}$ denote high-pressure steam streams produced in the heat recovery steam generator.

In alternative IIb we do not deal with the decrease in the stream of high-pressure regenerative steam since the workload of the boiler does not change.

The chemical energy of the coal combustion in the coal-fired boiler after its repowering (Fig. 2.3b) is equal to:

- alternative IIa

$$(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}} = \frac{(\dot{m}_1 - \dot{m}_{\mathrm{s}})(h_1 - h_9) + (\dot{m}_1 - \dot{m}^{\mathrm{HPR}} + \Delta \dot{m}^{\mathrm{HPR}})(h_3 - h_2)}{\eta_{\mathrm{b}}}, \tag{2.54}$$

- alternative IIb

$$(\dot{P}(\mathrm{NCV}))_{\mathrm{coal}} = \frac{\dot{m}_1(h_1 - h_9) + (\dot{m}_1 + \dot{m}_{\mathrm{s}} - \dot{m}^{\mathrm{HPR}})(h_3 - h_2)}{\eta_{\mathrm{b}}}. \tag{2.55}$$

The streams of high- and intermediate-pressure steam $\dot{m}_{\mathrm{s}}$ and $\dot{m}_{\mathrm{r}}$ produced in the heat recovery steam generator fed with gas from the exhaust of the gas turbine with the power output $N_{\mathrm{el}}^{\mathrm{GT}}$result from the energy balances in the high- and intermediate-pressure boiler sections (Fig. 2.3b)

$$\dot{C}[t_{\mathrm{out}}^{\mathrm{GT}} - (t_{\mathrm{s}}^{\mathrm{HP}} + \Delta T_{\mathrm{min}}^{\mathrm{HP}})] = \dot{m}_{\mathrm{s}}(h_{\mathrm{s}} - h_{\mathrm{s}}^{'}) + \dot{m}_{\mathrm{r}}(h_{\mathrm{r}}^{\mathrm{p}} - h_{\mathrm{r}}) \tag{2.56}$$

$$\dot{C}[t_{\mathrm{s}}^{\mathrm{HP}} + \Delta T_{\mathrm{min}}^{\mathrm{HP}} - (t_{\mathrm{s}}^{\mathrm{IP}} + \Delta T_{\mathrm{min}}^{\mathrm{IP}})] = \dot{m}_{\mathrm{s}}(h_{\mathrm{s}}^{'} - h_{\mathrm{r}}^{'}) + \dot{m}_{\mathrm{r}}(h_{\mathrm{r}} - h_{\mathrm{r}}^{'}) \tag{2.57}$$

while the stream of the thermal capacity of the flue gas is expressed by the equation

$$\dot{C} = \dot{m}_{\mathrm{fg}}^{\mathrm{GT}} \bar{c}_{\mathrm{fg}}^{\mathrm{GT}} |_{t_{\mathrm{amb}}}^{t_{\mathrm{out}}^{\mathrm{GT}}} = \frac{(\dot{P}(\mathrm{NCV}))_{\mathrm{gas}}|_{t_{\mathrm{amb}}} - N_{\mathrm{el}}^{\mathrm{GT}}}{t_{\mathrm{out}}^{\mathrm{GT}} - t_{\mathrm{amb}}} = \frac{N_{\mathrm{el}}^{\mathrm{GT}}(1 - \eta_{GT})}{\eta_{GT}(t_{\mathrm{out}}^{\mathrm{GT}} - t_{\mathrm{amb}})} \tag{2.58}$$

and:

$h_{\mathrm{r}}^{\mathrm{p}}$, h_{r}, h_{s} — specific enthalpy of the intermediate-and high-pressure superheated steam (the specific enthalpy of the intermediate-pressure superheated steam $h_{\mathrm{r}}^{\mathrm{p}}$ is determined for the temperature of $t_{\mathrm{r}}^{\mathrm{p}} \equiv t_1 = t_3$and pressure $p_{\mathrm{r}}^{p} \equiv p_2$, enthalpy h_{r} for the temperature of $t_r \equiv t_s^{HP}$ and pressure of $p_r = p_r^p \equiv p_2$, enthalpy h_s for the temperature of $t_s \equiv t_1$ and pressure $p_s \equiv p_1$);

$h_s^{'}, h_r^{'}$ the specific enthalpy of water at the bubbly flow point (x = 0) in the intermediate-and high-pressure sections of the heat recovery steam generator;

t_s^{IP}, t_s^{HP} temperature of saturation in the condenser of the intermediate- and high-pressure section of the heat recovery steam generator (temperature t_s^{IP}corresponds to the pressure of $p_r = p_r^p \equiv p_2$, the temperature t_s^{HP} to the pressure $p_s \equiv p_1$);

ΔT_{min}^{IP}, ΔT_{min}^{HP} pinch point of the intermediate- and high-pressure sections of the heat recovery steam generator.

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Chapter 3
Mathematical Model of a Power Unit with Rated Capacity of 370 MW After its Modernization to Cogeneration and Combined-Cycle

The mathematical model of a power unit whose structure is presented in Fig. 3.4 is based on the balance of energy and mass, non-linear relations that characterize the operation of power system and non-linear equations that determine the state of the heat transfer medium [1]. The balance presented for the specific sections of the power unit are in the form of a system of algebraic linear equations since all values accounted for as specific enthalpy of steam and water h_1–h_{171} as well as the values of the energy efficiency of facilities η_b, η_{em}, η_{m1K12}, η_{mPZ1}, η_{ST} are the input values. The other values that are given include pressure drop $\Delta p_{2\text{–}4}$, electrical power output N_{el}^{ST} (for operation under constant electric power output) or chemical energy stream $\dot{E}_{ch}^{coal}$ (for operation under a constant stream of chemical energy), heating steam mass flow $\dot{m}_{39}$, $\dot{m}_{40}$, $\dot{m}_{41}$, bleed steam streams from diaphragm and turbine valves $\dot{m}_{45} - \dot{m}_{64}$, steam flux from under the balance piston $\dot{m}_{bp}$, cooling water stream of mass in the condensers $\dot{m}_{KQ1}$, $\dot{m}_{KQ2}$ and its temperature T_{w2KQ1}, T_{w1KQ1}, T_{w2KQ2}, T_{w1KQ2} and ambient temperature T_{amb} (temperatures T_{w2KQ1}, T_{w2KQ2} are calculated by means of iterations from energy balances with simultaneous calculations of their pressures and temperatures of saturation). The unknown variables include the remaining streams of mass and stream of chemical energy $\dot{E}_{ch}^{coal}$ of the coal combustion in the boiler (for the operation under a constant power output) and the natural gas in the gas turbine $\dot{E}_{ch}^{gas}$ or the electrical power output from a unit N_{el}^{ST} (for the operation with a constant stream of chemical energy). The invariable input quantities for each set of input data include: temperature of fresh steam and temperature of the superheated steam T_1, T_4, temperatures $t_{out,n}^{GT}$, t_{out}^{HRSG} as well as streams of mass $\dot{m}_1$, $\dot{m}_{KQ1}$, $\dot{m}_{KQ2}$. It is convenient to solve the system using high-precision methods, e.g. method of Gaussian elimination.

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3.1 General Characteristic of Power Unit with Rated Power Output of 370 MW

The power station which is the subject of research in this monograph is a condensing power station with interstage reheat cycle operating under closed circulation of cooling water. At present the power station consists of four power units with a total power output of 1,492 MW (and reaching 1,532 MW):

- 1 unit: 386 MW,
- 2 units: 383 MW,
- 1 unit: 380 MW.

The fuel applied is hard coal; all blocks have installations for flue gas desulphurization. The maximum rating capacity of the power station is limited to around 10 TWh of energy annually.

A power unit consists of:

- steam boiler BP-1150
- turbine 18K370
- generator GTHW 370
- atmosphere protection facilities

3.1.1 Steam Boiler BP-1150

Steam boiler BP-1150 (Fig. 3.1) is a superheated steam, once-through, drum-less type with induced feedwater flow across the boiling tubes operating under subcritical pressure. Hard coal, which after pulverizing is fed into the combustion chamber forms the basic fuel (Table 3.1).

3.1.2 18K370 Turbine

The 18K370 is a reaction, axial, three-casing, condensing turbine with unregulated steam extraction, designed to meet the requirements of qualitative and quantitative regulation of the supply and interstage steam reheater. The modernized models of the 18K370 turbine apply a new type of rotor design in the LP section (Fig. 3.2, highlighted section) and include engineering changes thus increasing the turbogenerator's output by around 10 MW (Table 3.2).

The turbine consists of three sections:

- high-pressure section (HP),
- intermediate pressure section (IP) with double exhaust,
- low-pressure section (LP) with double exhaust.

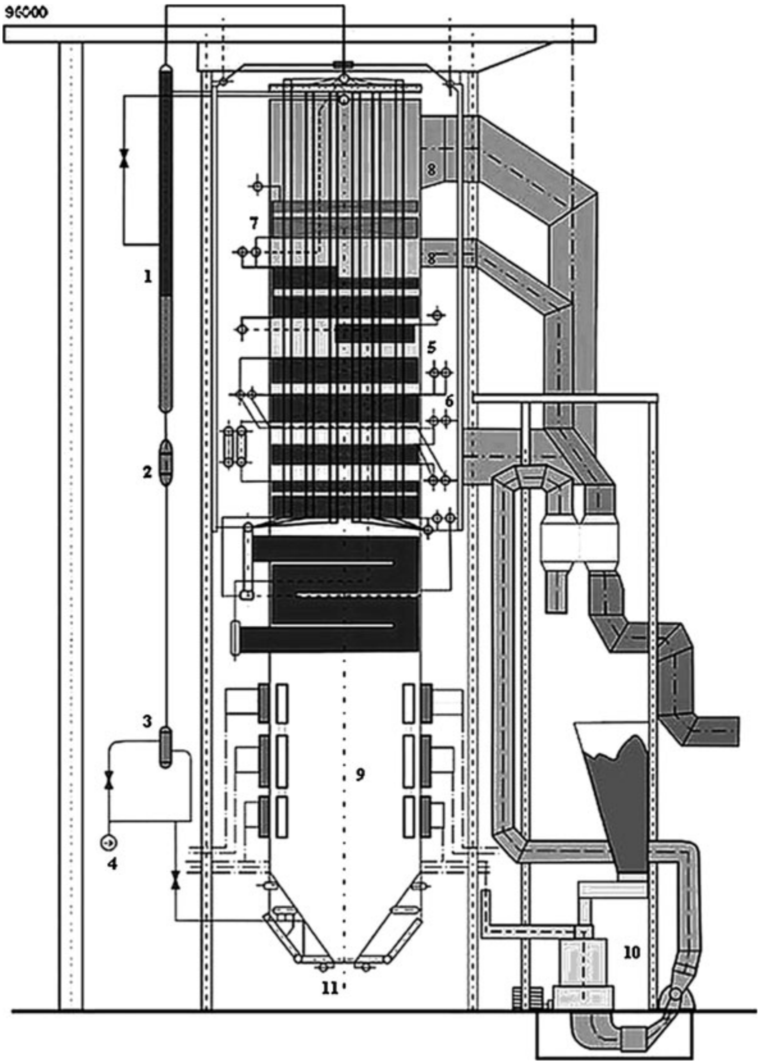


Fig. 3.1 Diagram of the BP-1150 steam boiler (*1* separator, *2* mixer, *3* filter, *4* circulation pump, *5* fresh steam exhaust, *6* secondary steam exhaust, *7* economizer, *8* flue gas exhaust, *9* combustion chamber, *10* pulverizer assembly, *11* slag drain)

3.1.3 GTHW 370 Electrical Generator

GTHW 370 electrical generator whose role is to produce electric power is designed for direct coupling with the steam turbine 18K370. It operates in the system of direct cooling of the stator and rotor windings. The winding of the stator is cooled with demineralized water, i.e. distillate and the rotor's winding, rotor

Table 3.1 Summary of basic technical details of the steam boiler BP-1150

Parameter	Unit	Value
Maximum continuous rating	kg/s	320
Temperature of fresh exhaust steam	°C	540
Pressure of fresh exhaust steam	MPa	18.3
Superheated steam temperature—inlet/exhaust	°C	335/540
Superheated steam pressure—inlet	MPa	4.2
Feedwater temperature	°C	255
Guaranteed efficiency	%	91.7
Fuel type	–	Hard coal
Net calorific value	MJ/kg	23

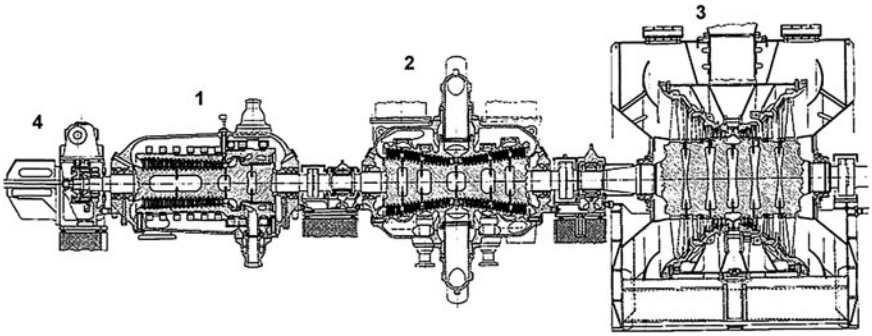


Fig. 3.2 Diagram of 18K370 turbine (*1* HP casing, *2* IP casing, *3* LP casing, *4* turning gear)

Table 3.2 Summary of basic technical details of 18K370 steam turbine

Parameter	Unit	Value
Rated capacity	MW	370
Maximum continuous capacity	MW	380
Temperature of fresh steam at the inlet of HP casing	°C	535
Pressure of fresh steam at the inlet of HP casing	MPa	17.65
Temperature of steam at the exhaust of HP casing	°C	335
Pressure of steam at the exhaust of HP casing	MPa	4.48
Temperature of superheated steam at the inlet of IP casing	°C	535
Pressure of superheated steam—inlet	MPa	4.2
Condenser pressure	kPa	6.8
Rated temperature of cooling water	°C	22
Heat consumption per unit of power	kJ/kWh	7,853

Table 3.3 Summary of basic technical data of GTHW 370 generator

Parameter	Unit	Value
Apparent power	MVA	426
Active power	MW	370
Stator voltage	kV	22
Stator current	kA	11.2
Power coefficient	–	0.85
Excitation current	kA	2.8
Excitation voltage	\V	533

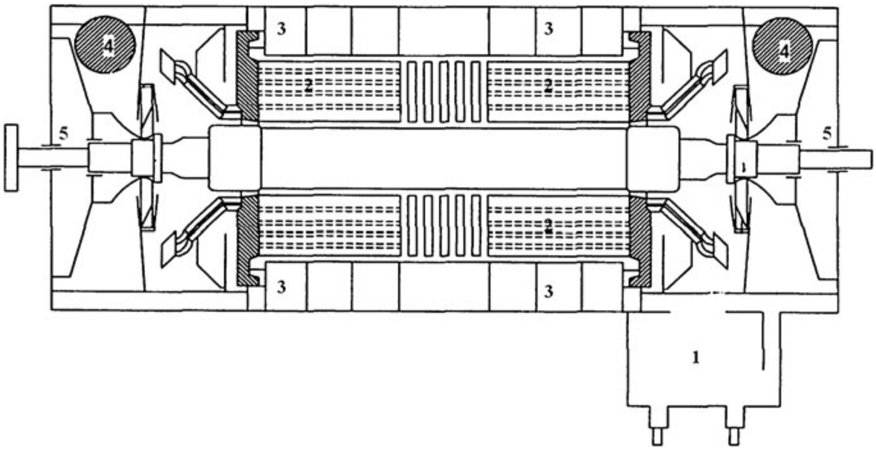


Fig. 3.3 Cross-section of the GTHW-370 generator (*1* clamping chamber, *2* rotor winding, *3* stator winding, *4* hydrogen cooler, *5* axial fan)

barrel and stator's core with hydrogen, which fills the entire empty space of the generator. The circulation of the water is induced by pumps situated outside the generator. The circulation of the hydrogen whose role is to cool the interior of the generator is ensured by two fans located on the generator's shaft on the two sides of the rotor (Table 3.3). The heated hydrogen passes through water coolers situated vertically in the most remote chamber in the stator (Fig. 3.3).

3.2 Power Unit

Figure 3.4 presents the thermal diagram of the unit that is adapted to the combined heat and power using heaters XC2, XC3 and XC4 supplied from extractions A2, A3 and from IP to LP crossoverpipe.

The diagram is based on an electric power unit with the rated power of 370 MW. In addition, a power unit is repowered by gas turbine in a parallel system.

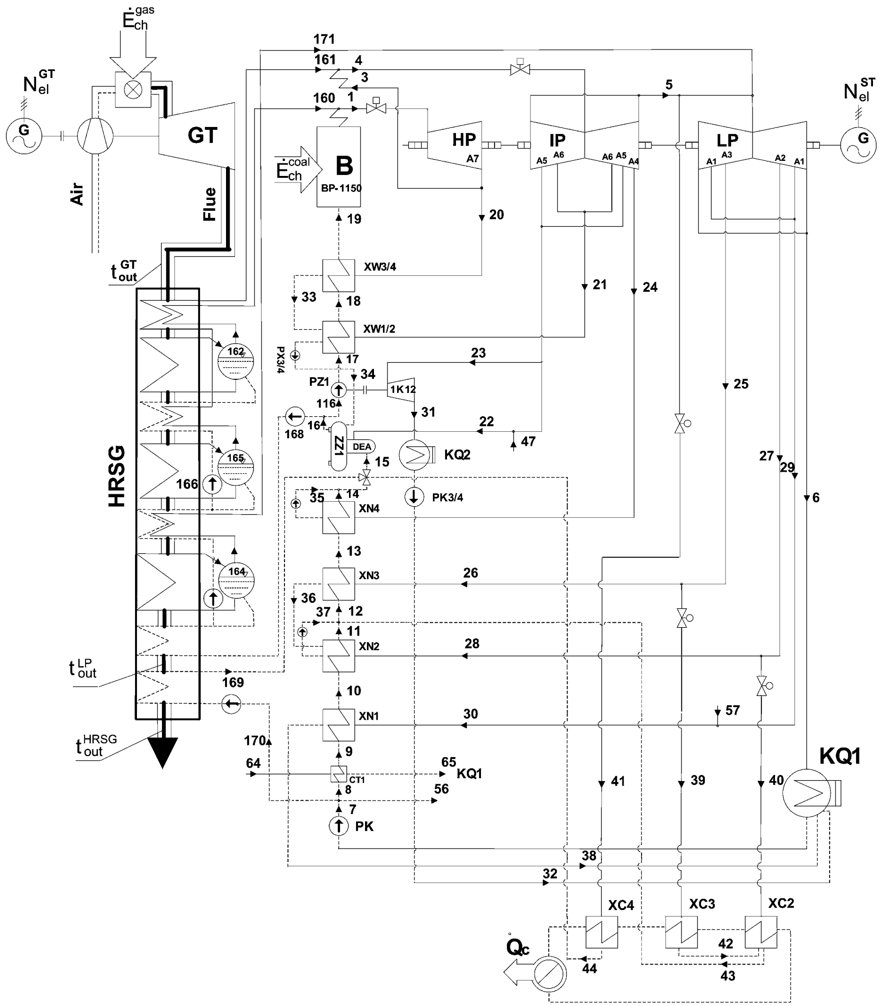


Fig. 3.4 Thermal diagram of 370 MW electric power unit repowered by gas turbine with triple pressure heat recovery steam generator and operating under combined heat and power with steam supply to XC2, XC3, XC4 heaters (*GT* gas turbine; *HRSG* heat recovery steam generator XC2, XC3, XC4 heaters; in the alternative of power unit modernization to combined heat and power GT and HRSG are excluded; in the alternative with repowering the system to a dual fuel combined-cycle without cogeneration XC2, XC3, XC4 heaters are excluded)

3.3 Steam Extraction from the Power Unit

The extraction of considerable extra amounts of steam (besides the steam required for the LP regenerative preheaters during condensing operation only) and feeding it into heaters is associated with overcoming numerous relevant problems. For instance, bleeding additional steam from asymmetric extractions A2 and A3

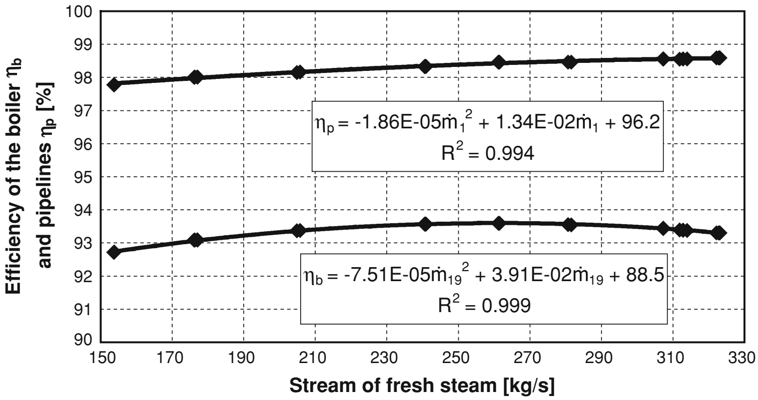


Fig. 3.5 Pressure drop in the interstage reheater in the function of stream of reheated steam in the IP section of the turbine

(Fig. 3.4) results in the origin of an additional axial force in the LP section of the turbine. One has to bear in mind that 18K370 turbine has been designed to operate for condensing work and, as a result, its modernization to combined system will require the introduction of significant modifications.

Thus, the scope of modifications in 18K370 turbine will involve:

- increasing the surface of the gaps in the blades at extraction points,
- aligning the blades in a symmetric way,
- considerable increase in the diameters of the extraction pipelines (this is due to two reasons: first, as a result of the need to increase the bleed steam stream; second, due to the larger specific volume of the steam as a result of the reduced values of the pressure at the extractions resulting from the increased stream of mass, in accordance with Stodola–Flügel turbine passage equations,
- exchange of the turbine stages preceding the extractions into new ones with considerably greater strength due to the increased bleed steam mass stream.

This scope of changes imposes a need to replace the internal assemblies in the LP section of the turbine with new ones. This also provides an opportunity to apply the latest generation blade assembly, which will ultimately result in the overall efficiency of the turbine and its power output. For this reason it is also necessary to replace the casing of the LP section with a new one with increased strength.

3.4 Operating Characteristics of Basic Facilities

3.4.1 Steam Boiler

For the case of a steam boiler the characteristic of its thermal efficiency, the efficiency of the primary steam and superheated steam pipelines coupling it with the turbine (Fig. 3.5) is used to define the energy losses in the system as well as the

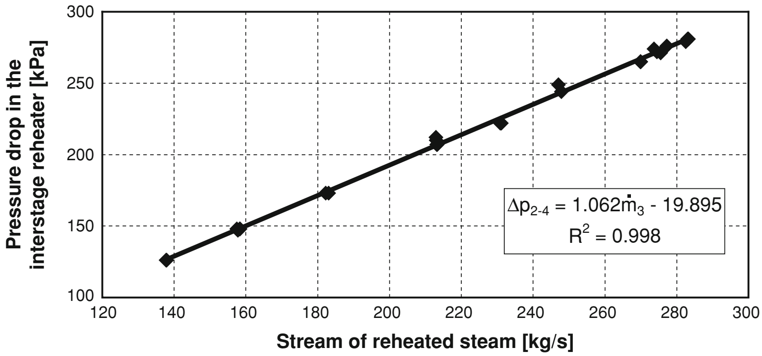


Fig. 3.6 Pressure drop in the interstage reheater in the function of stream of reheated steam in the IP section of the turbine

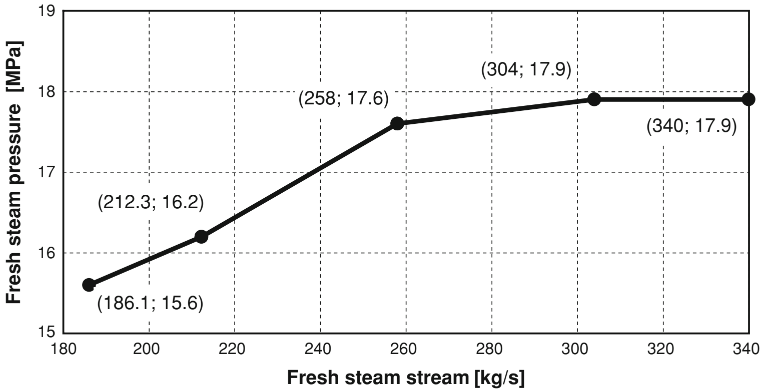


Fig. 3.7 Boiler exhaust fresh steam set value in the function of the steam stream

characteristics of the pressure drop in the interstage regenerative preheater (Fig. 3.6).

The pressure of the fresh steam changes along with the characteristic of the relation of its set value at the boiler exhaust from stream of fresh steam (Fig. 3.7).

3.4.2 Steam Turbogenerator

We need to determine the characteristics of the electromechanical efficiency and the efficiency of the HP, IP and LP sections of the steam turbine (Figs. 3.8, 3.9, 3.10, 3.11).

On the basis of the manufacturers' data the equations for the relation of mass stream of steam across the diaphragms in the turbine, valve stems along the

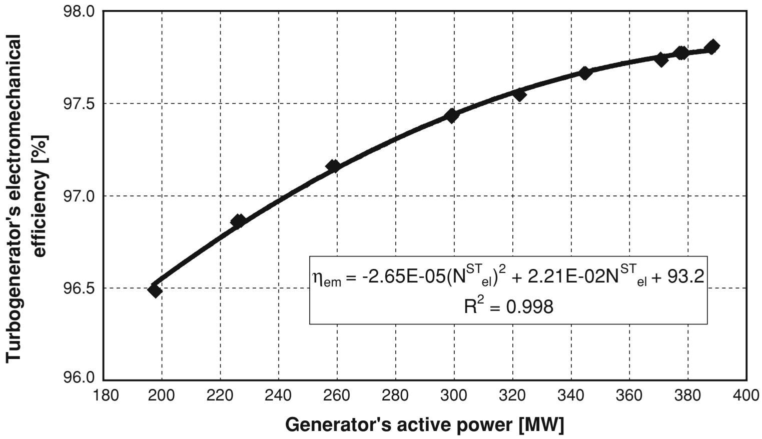


Fig. 3.8 Electromechanical efficiency of the turbogenerator in the function of electric power output

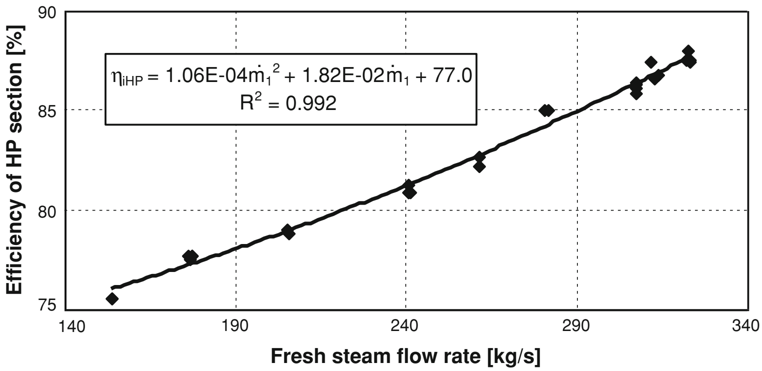


Fig. 3.9 Internal efficiency of the HP section of the turbine in the function of stream of fresh steam

pipelines supplying the HP and IP casings and steam flow from under the pressures relieve valve of the turbine. The equations are summarized in Table 3.4.

3.4.3 Low and High-Pressure Regenerative Feed Water Preheaters

The auxiliary devices include low- and high-pressure regenerative preheaters. These pipelines that lead to the turbine have a considerable length and therefore it is necessary to apply the characteristics accounting for the pressure losses in them in the overall model. For the low-pressure regeneration pipelines (XN2, XN3 and XN4)

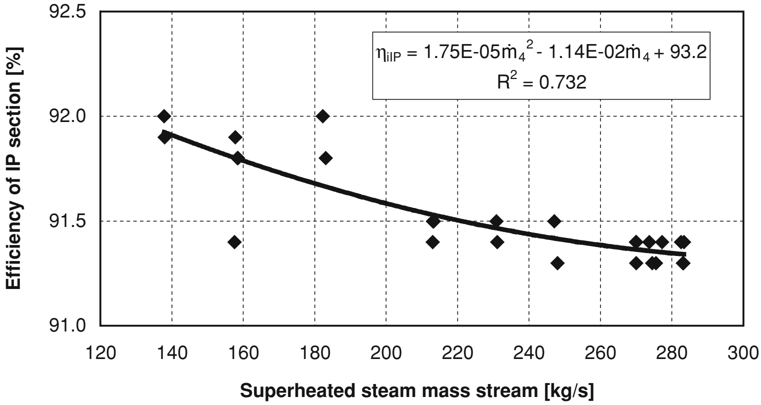


Fig. 3.10 Internal efficiency of the IP section of the turbine in the function of superheated steam stream

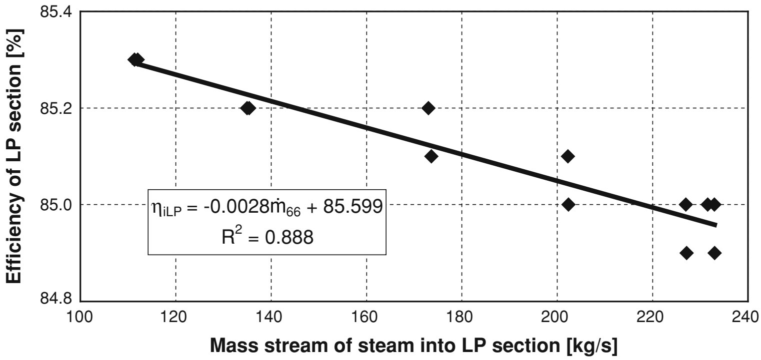


Fig. 3.11 Internal efficiency of the LP section of the turbine in the function of steam stream into LP casing

it was assumed that the pressure losses are constant in them. For the pipeline feeding the XN1 heater the loss of the pressure is disregarded (Table 3.5). For the pipelines joining the turbine with HP heaters the calculated pressure drops in the function of the mass stream of steam through them are presented in Figs. 3.12 and 3.13.

The product of heat transfer coefficient and heating surface (kF_n) has been determined for each of the heaters for the operation of the power unit with the rated capacity $N_{el}^{ST} = 370$ MW. They can be considered as constant over the entire range of the variability of water and steam mass flows. In addition, a constant value of condensate supercooling after the heaters. The summary of all variables is found in Table 3.5.

Table 3.4 Summary of equations defining steam mass flow across diaphragms of the turbine, mass stream of steam from seals of the turbine valve spindles, mass stream of steam from below balance piston and feedwater admission to sealing steam collector

	Mass flow (kg/s)
Mass stream of steam through HP diaphragms	$\dot{m}_{45} = 0.0032\ \dot{m}_1 + 0.0232$
	$\dot{m}_{46} = 0.0022\ \dot{m}_1 - 0.0026$
	$\dot{m}_{49} = 0.0017\ \dot{m}_1 - 0.0142$
	$\dot{m}_{50} = 0.0011\ \dot{m}_1 - 0.0065$
	$\dot{m}_{58} = 0.018$
	$\dot{m}_{59} = 0.012$
Mass stream of steam through IP diaphragms	$\dot{m}_{52} = 0.0005\ \dot{m}_1 - 0.0154$
	$\dot{m}_{53} = 0.0007\ \dot{m}_1 - 0.0182$
	$\dot{m}_{60} = 0.016$
	$\dot{m}_{61} = 0.024$
Mass stream of steam through LP diaphragms	$\dot{m}_{54} = 0.243$
	$\dot{m}_{55} = 0.243$
	$\dot{m}_{62} = 0.073$
	$\dot{m}_{63} = 0.073$
Mass stream of steam from under balance piston	$\dot{m}_{bp} = 0.0083\ \dot{m}_1 - 0.0232$
Mass stream of steam from the seals of HP and IP spindle valves	$\dot{m}_{48} = 0.0011\ \dot{m}_1 + 0.0198$
	$\dot{m}_{51} = 0$
Water feed into the steam diaphragm collector	$\dot{m}_{56} = 0.06$

Table 3.5 Summary of the operating parameters of the heaters

Heater	kF_n (F_n) (kW/K), (m^2)	Condensate subcooling (K)	Pressure drop in the pipeline (kPa)
XN1	1,735 (518)	1.8	0
XN2	1,907 (465)	0.2	8
XN3	2,428 (572)	2.9	18
XN4	2,752 (518)	4.1	8
XW1/2	2,869 (360)	0.3	Figure 3.12
XW3/4	3,800 (468)	0.6	Figure 3.13

3.4.4 Feedwater Tank

For the pipeline feeding the auxiliary turbine that drives the major feedwater pump and pipeline to deaerator it is important to note the pressure drops in them (Figs. 3.14, 3.15).

3.4.5 Auxiliary Turbine and Feedwater Pump

A characteristic of the water pressure in the feed water pump along with that of the auxiliary pump which drives the feed water pump is presented in Fig. 3.16. The calculations assume a constant value of temperature increase at the output of

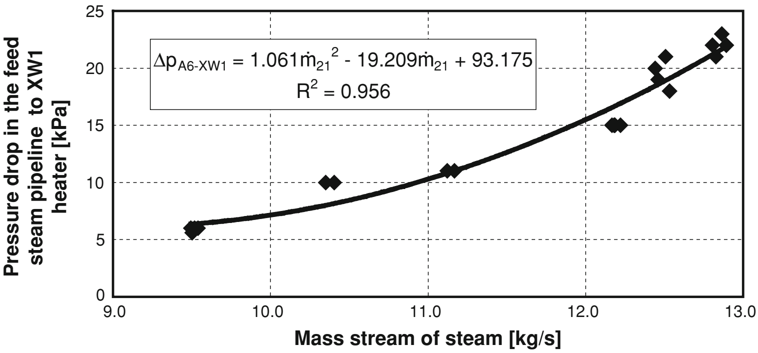


Fig. 3.12 Pressure drop in the steam pipeline feeding XW1/2 HP heater in the function of steam stream

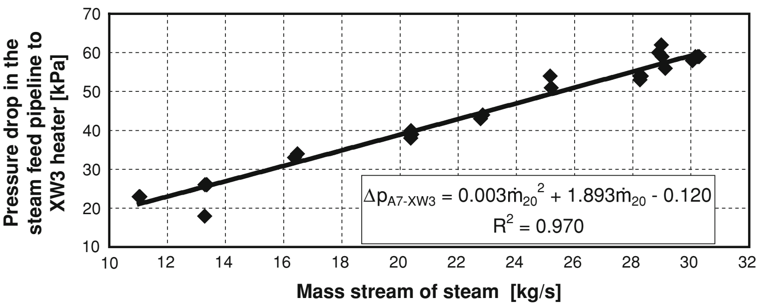


Fig. 3.13 Pressure drop in the steam pipeline feeding XW3/4 HP heater in the function of steam stream

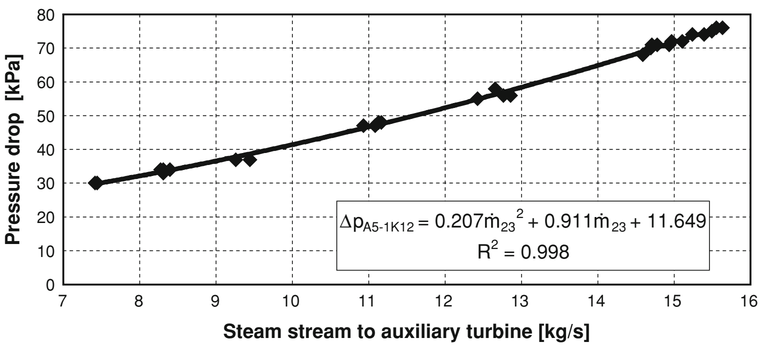


Fig. 3.14 Pressure drop in the steam pipeline feeding the auxiliary turbine in the function of steam stream

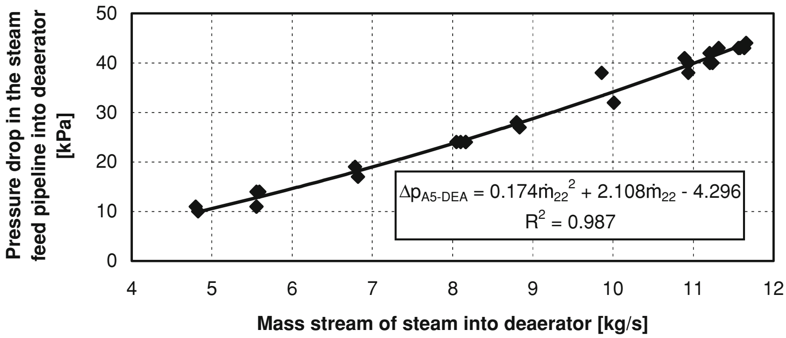


Fig. 3.15 Pressure drop in the steam feed pipeline to deaerator in the function of steam stream

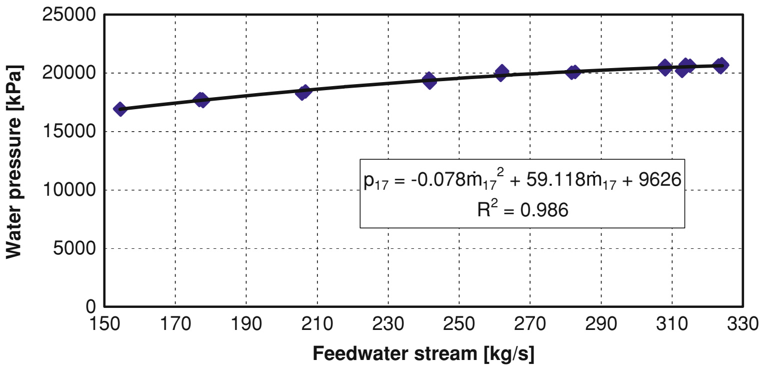


Fig. 3.16 Water pressure at the output of main feed water pump in the function of feedwater stream

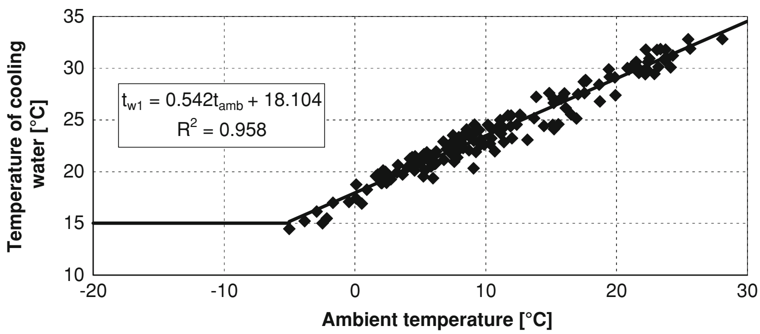


Fig. 3.17 Temperature of the cooling water in the function of the ambient temperature

Table 3.6 Summary of condenser parameters

Condenser	kF_n (F_n) (kW/K) (m^2)	Condensate subcooling (°C)	Cooling water stream (kg/s)
KQ1	40.511 (10.290)	0	10.500
KQ2	5.143 (729)	0	724

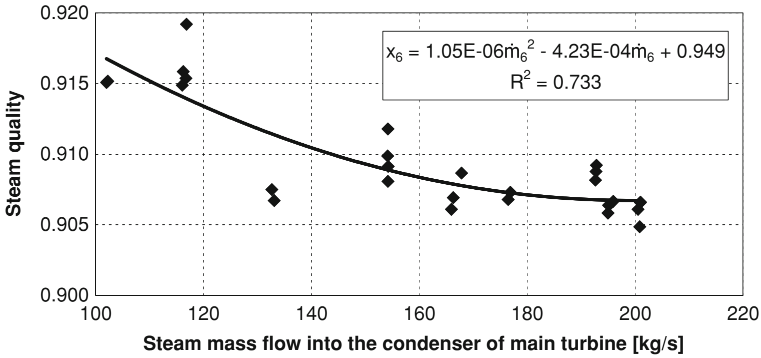


Fig. 3.18 Steam quality in the condenser of the main turbine in relation to the steam mass flow in the turbine

the pump, i.e. 3.7°C, a constant efficiency of the pump equal to 77% along with the constant 99.5% mechanical efficiency of the turbine.

3.4.6 KQ1, KQ2 Condensers in Main and Auxiliary Turbines

It is necessary to determine the characteristic of the temperature of cooling water fed into it in the function of the ambient temperature (Fig. 3.17). In accordance with the exploitation instructions of the turbine it was assumed that the temperature of the cooling water for the operation of the power unit with the rated capacity may not drop below 15°C.

For both condensers one can assume constant values of the heat transfer coefficient and heating surface (kF_n) determined for the rated power unit capacity $N_{el}^{ST} = 370$ MW and consider them as constant over the entire range of the variability of the water and steam mass streams. A summary of these parameters is given in Table 3.6.

It is also necessary to apply the characteristic of steam quality in the condenser of the main turbine in relation to the mass stream of steam into the turbine (Fig. 3.18).

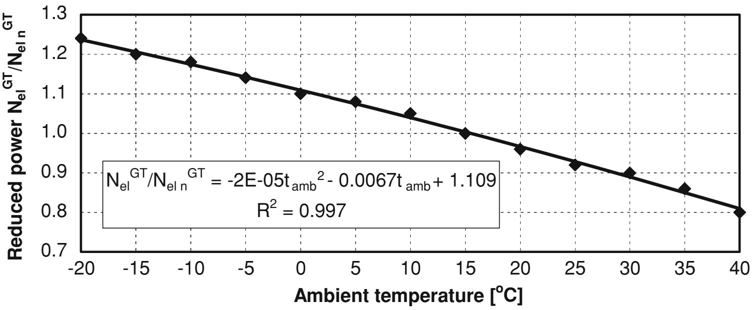


Fig. 3.19 Characteristic of reduced electrical capacity of the gas turbogenerator in the function of ambient temperature

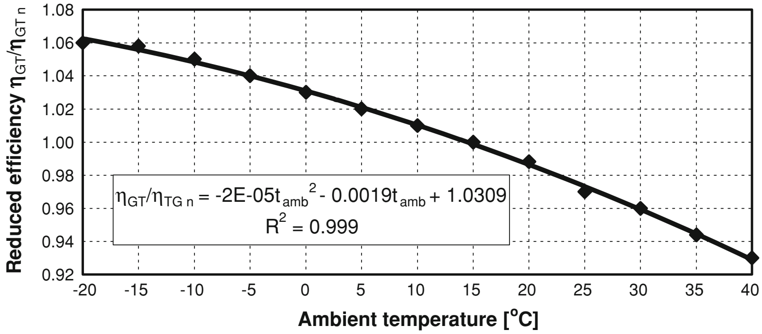


Fig. 3.20 Characteristic of reduced efficiency of the gas turbogenerator in the function of ambient temperature

In order to calculate the specific enthalpy of the steam before the condenser one can apply the relation: $h_{31} = 2260$ (kJ/kg) + specific enthalpy of condensate behind the condenser.

3.4.7 Gas Turbogenerator

Gas turbine is characterized by a variable power, efficiency, temperature and stream of exhaust gases from the turbine (and therefore, their heat capacity stream) as a result of variable air density depending on the ambient temperature. These changes are accounted for in the model by the application of the reduced characteristics of its operation and scheduled chart of ambient temperatures.

The presented characteristics are prepared for the GTM7A gas turbogenerator (Figs. 3.19, 3.20, 3.21).

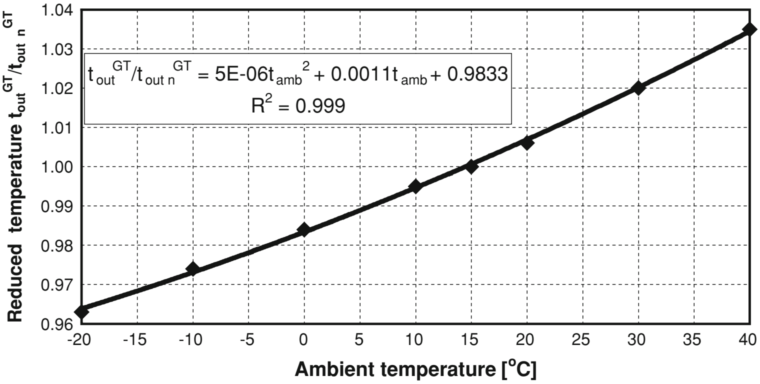


Fig. 3.21 Characteristic of reduced temperature of the exhaust gases from the gas turbine in the function of ambient temperature

The characteristics of other gas turbogenerators are similar; therefore, no significant error can result from their application with regard to every turbogenerator. In addition, the presented model offers a selection of an optimum power output of the gas turbine, thus accounting for the variability of its value vary within a very wide range of the output value, from zero to almost infinite value. Therefore, it is a priori impossible to determine its characteristics for an unfamiliar turbine output. It is only possible to account for it during the calculations performed with the purpose of verification of the results. In addition, the optimum power of the gas turbine for various price relations between energy carriers will be different.

The characteristic of the heat capacity stream $\dot{C}$ of the exhaust gases from the gas turbine result from its energy balance (3.76) and (3.77) and the characteristics in Figs. 3.19, 3.20 and 3.21.

Figure 3.22 presents the ambient temperature regression equation and curve for the largest climatic zone in Poland, i.e. zone III. It is based on the scheduled chart of temperature [2] in this zone.

3.5 Equations of Mass and Energy Balance

Energy balance forms the basic tool necessary for solving thermodynamic issues. Hence, mass and energy balances of the particular facilities have been prepared for the examined power unit. System boundaries of the particular groups are marked with broken lines and each has a specific number attributed to it (Figs. 3.19, 3.20, 3.21a, 3.23, 3.24, 3.25).

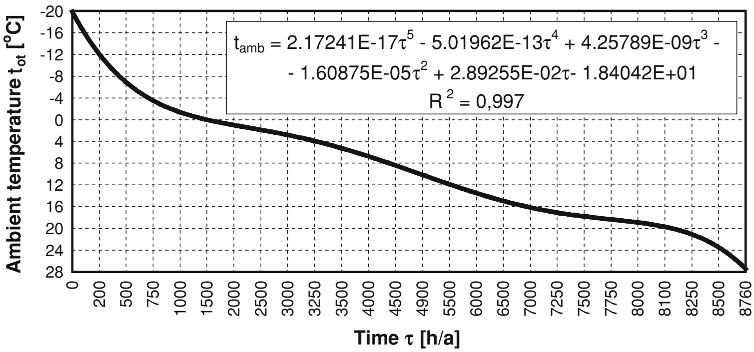
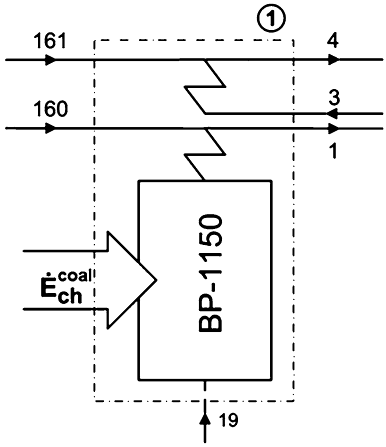


Fig. 3.22 Regression curve and equation for ambient temperature in climatic zone III

Fig. 3.23 Diagram of BP-1150 boiler



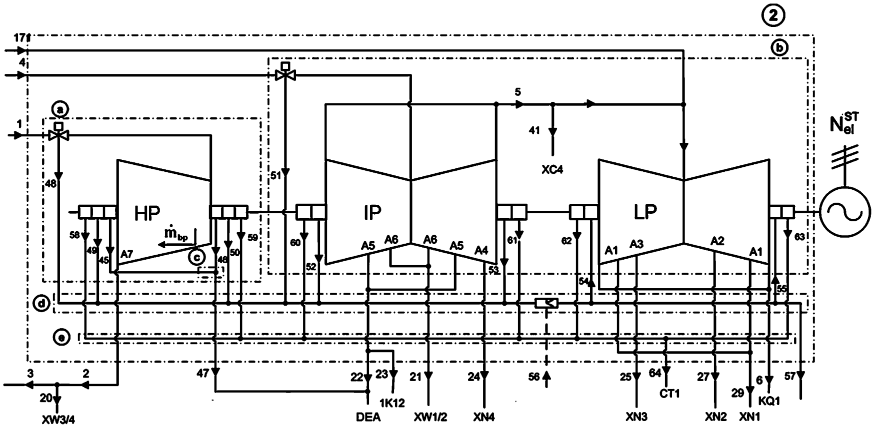


Fig. 3.24 Diagram of 18K370 steam turbine

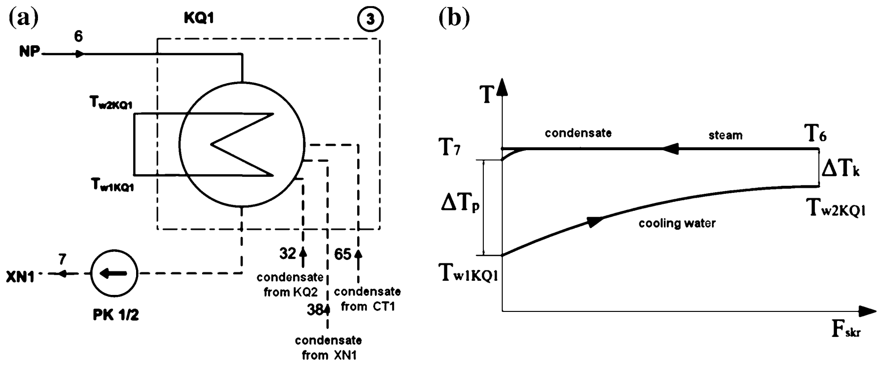


Fig. 3.25 **a** Schematic diagram of KQ1 condenser. **b** Distribution of temperatures in the condenser

For each device it is possible to determine the general form of the balance of substance and energy equations:

- for energy balance:

$$\sum_{i=1}^{n} \dot{m}_i h_i + \dot{E} = 0 \tag{3.1}$$

where: $\dot{E} = N_i$ or $\dot{E}_{amb}$

- for substance balance

$$\sum_{i=1}^{n} \dot{m}_i = 0 \tag{3.2}$$

The balance equations presented below are based on the following assumptions:

- steady and continuous operation of the power unit,
- value of the pressure of fresh steam varies in accordance with the characteristics of the regulation (Fig. 3.7),
- constant temperatures of fresh and superheated steam on the boiler's exhaust,
- chemical energy comes only from the combustion of coal,
- steam is not fed into air preheaters,
- steam is not fed into the collectors between power units,
- accounting for heat losses in fresh steam and superheated steam pipelines between the boiler and the turbine; the remaining losses are disregarded,
- water is not fed into primary and interstage superheaters,
- accounting for pressure losses in pipelines feeding steam into LP and HP regenerative feed water preheaters, heat exchangers and the reheater; the remaining losses are disregarded,
- assumptions of no subcooling of condensate in the condensers together with a constant temperature of subcooling behind regenerative feed water preheaters and heat exchangers is made,

- the effect of pressure resulting from the operation of the feed pump on water enthalpy is accounted for; the influence of the remaining pumps and valves is disregarded,
- losses resulting from steam leakages from the diaphragm of the main turbine and steam from under balance piston are accounted for,
- the in-parallel HP regenerative feed water preheaters are considered as in-series operating dual heat exchangers (XW1/2 and XW3/4).

3.5.1 Steam Boiler

3.5.1.1 System Boundary No. 1: BP-1150 Steam Boiler

- Energy balance equation:

$$\eta_b \eta_p \dot{E}_{ch}^{coal} + \dot{m}_3(h_3 - h_4) + \dot{m}_{19}(h_{19} - h_1) = 0 \tag{3.3}$$

$$h_{160} = h_1 \tag{3.4}$$

$$h_{161} = h_4 \tag{3.5}$$

- Mass balance equations:

$$\dot{m}_{19} + \dot{m}_{160} - \dot{m}_1 = 0 \tag{3.6}$$

$$\dot{m}_3 + \dot{m}_{161} - \dot{m}_4 = 0 \tag{3.7}$$

The balance equations of the steam boiler assume zero value injection of water in the primary heater and interstage reheater. This simplification results from the fact that water injection into the primary heater does not affect the energy balance of the system and its value is not measured for purposes of exploitation. The value of the water injection into interstage reheater has assumed zero value throughout the entire cycle of the operation of the power unit.

3.5.2 Steam Turbogenerator

3.5.2.1 System Boundary No. 2: 18K370 Steam Turbine

- Energy balance equations:

$$N_{i\,HP}^{ST} = (\dot{m}_1 - \dot{m}_{48} - \dot{m}_{bp} - \dot{m}_{46} - \dot{m}_{50} - \dot{m}_{59})\,(h_1 - h_2) \tag{3.8}$$

$$N_{i\,IP}^{ST} = \dot{m}_4 h_4 - \dot{m}_{51} h_4 - \dot{m}_{21} h_{21} - \dot{m}_{22} h_{22} - \dot{m}_{23} h_{23} - \dot{m}_{24} h_{24} - \dot{m}_5 h_5 \tag{3.9}$$

$$N_{\mathrm{i\,LP}}^{\mathrm{ST}} = (\dot{m}_{171} + \dot{m}_5 - \dot{m}_{41})h_5 - \dot{m}_{29}h_{29} - \dot{m}_{27}h_{27} - \dot{m}_{25}h_{25} - \dot{m}_6 h_6 \tag{3.10}$$

$$N_{\mathrm{el}}^{\mathrm{ST}} = (N_{\mathrm{i\,HP}}^{\mathrm{ST}} + N_{\mathrm{i\,IP}}^{\mathrm{ST}} + N_{\mathrm{i\,LP}}^{\mathrm{ST}})\eta_{\mathrm{em}} \tag{3.11}$$

- Mass balance equations:

system boundary a

$$\dot{m}_1 - \dot{m}_{48} - \dot{m}_2 - \dot{m}_{45} - \dot{m}_{46} - \dot{m}_{49} - \dot{m}_{50} - \dot{m}_{58} - \dot{m}_{59} = 0 \tag{3.12}$$

system boundary b

$$\begin{aligned} &\dot{m}_4 + \dot{m}_{171} - \dot{m}_{41} - \dot{m}_{51} - \dot{m}_{21} - \dot{m}_{22} - \dot{m}_{23} - \dot{m}_{24} - \dot{m}_{52} - \dot{m}_{53} \\ &- \dot{m}_{60} - \dot{m}_{61} - \dot{m}_{25} - \dot{m}_{27} - \dot{m}_{29} - \dot{m}_{62} - \dot{m}_{63} - \dot{m}_{54} - \dot{m}_{55} - \dot{m}_6 = 0 \end{aligned} \tag{3.13}$$

system boundary c

$$\dot{m}_{45} + \dot{m}_{46} - \dot{m}_{47} = 0 \tag{3.14}$$

system boundary d

$$\dot{m}_{48} + \dot{m}_{49} + \dot{m}_{50} + \dot{m}_{51} + \dot{m}_{52} + \dot{m}_{53} + \dot{m}_{56} - \dot{m}_{54} - \dot{m}_{55} - \dot{m}_{57} = 0 \tag{3.15}$$

system boundary e

$$\dot{m}_{58} + \dot{m}_{59} + \dot{m}_{60} + \dot{m}_{61} + \dot{m}_{62} + \dot{m}_{63} - \dot{m}_{64} = 0 \tag{3.16}$$

The energy balance for the HP section of the steam turbine accounts for the mass stream of steam from under balance piston $\dot{m}_{\mathrm{bp}}$ (symbolically marked in the HP section of the turbine). The mass balances thereof account for the values of the steam leakages from the diaphragms and turbine valves and water flow into the sealing steam collector $\dot{m}_{45} - \dot{m}_{63}$.

3.5.3 KQ1 Condenser

3.5.3.1 System Boundary No. 3: KQ1 Condenser

- Energy balance equation:

$$\dot{m}_6 h_6 + \dot{m}_{32}h_{32} + \dot{m}_{38}h_{38} + \dot{m}_{65}h_{65} - \dot{m}_7 h_7 - \dot{m}_{\mathrm{KQ1}}c_{\mathrm{w}}(T_{\mathrm{w2KQ1}} - T_{\mathrm{w1KQ1}}) = 0 \tag{3.17}$$

- Mass balance equation:

$$\dot{m}_6 + \dot{m}_{32} + \dot{m}_{38} + \dot{m}_{65} - \dot{m}_7 = 0 \tag{3.18}$$

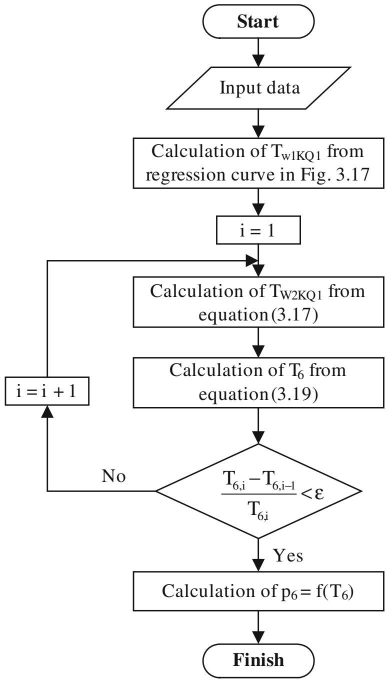


Fig. 3.26 Algorithm for the calculation of pressure p_6 in the condenser

The solution of a system of equations consisting of energy and mass balance that constitute the characteristics of the operations of a power unit (Sect. 3.4) forms its mathematical model and starts with the calculation of the saturation pressure p_6 (and hence, the saturation temperature T_6) in the condenser. This is so since this pressure determines the conditions of the operation of the turbine system. The calculations of the pressure require iterations. The input data include: mass flows and specific enthalpies of the medium in the particular points of the condenser and the ambient temperature T_{amb}. The schematic diagram for the calculation algorithm of pressure p_6 is presented in Fig. 3.26. Under the assumption of temperatures T_6 and T_7 ($T_6 \cong T_7$), after determination of the temperature T_{w1KQ1} from the regression equation in Fig. 3.17 (Sect. 3.4.6) for a given temperature T_{amb} and on the basis of relation (3.17) it is possible to obtain T_{w2KQ1}. Subsequently, temperature T_6 is derived from the equation:

$$T_6 = \frac{-T_{w2KQ1} + T_{w1KQ1} \exp \dfrac{-(\mathrm{kF})_{\mathrm{cond}}}{\dot{m}_{KQ1}\ \mathrm{c_w}}}{-1 + \exp \dfrac{-(\mathrm{kF})_{\mathrm{cond}}}{\dot{m}_{KQ1}\ \mathrm{c_w}}} \tag{3.19}$$

along with the pressure p_6 corresponding to it.

Equation 3.19 is derived as a result of the combination of relations in (3.20) and (3.21):

$$\dot{Q}_{\mathrm{cond}} = \dot{\mathrm{m}}_{\mathrm{KQ1}}\, \mathrm{c_w}(T_{\mathrm{w2KQ1}} - T_{\mathrm{w1KQ1}}) \tag{3.20}$$

$$\dot{Q}_{\mathrm{cond}} = (\mathrm{kF}\ \Delta T_{\log})_{\mathrm{cond}} \tag{3.21}$$

while $(\triangle T_{\log})_{\mathrm{cond}}$ (on condition of disregarding the effect of subcooling, i.e. when $T_6 = T_7$) is equal to:

$$\left(\Delta T_{\log}\right)_{\mathrm{cond}} = \frac{\Delta T_{\mathrm{p}} - \Delta T_{\mathrm{k}}}{\ln \frac{\Delta T_{\mathrm{p}}}{\Delta T_{\mathrm{k}}}} = \frac{(T_6 - T_{\mathrm{w1KQ1}}) - (T_6 - T_{\mathrm{w2KQ1}})}{\ln \dfrac{T_6 - T_{\mathrm{w1KQ1}}}{T_6 - T_{\mathrm{w2KQ1}}}}. \tag{3.22}$$

The calculations of the saturation temperature T_6 and the corresponding saturation pressure p_6 are concluded after the required precision of the calculations is obtained.

3.5.4 Low Pressure Regenerative Feed Water Preheaters: XN1, XN2, XN3, XN4, CT1

3.5.4.1 System Boundary No. 4: Manifold Before CT1

- Energy balance equations

$$\dot{m}_7 h_7 - \dot{m}_8 h_8 - \dot{m}_{56} h_{56} - \dot{m}_{170} h_{170} = 0 \tag{3.23}$$

$$h_7 = h_8 = h_{56} = h_{170} \tag{3.24}$$

- Mass balance equation:

$$\dot{m}_7 - \dot{m}_8 - \dot{m}_{56} - \dot{m}_{170} = 0 \tag{3.25}$$

3.5.4.2 System Boundary No. 5: Steam Subcooler Supplied from External Diaphragms CT1

- Energy balance equation:

$$\dot{m}_8 h_8 + \dot{m}_{64} h_{64} - \dot{m}_9 h_9 - \dot{m}_{65} h_{65} = 0 \tag{3.26}$$

- Mass balance equations:

$$\dot{m}_8 - \dot{m}_9 = 0 \tag{3.27}$$

$$\dot{m}_{64} - \dot{m}_{65} = 0 \tag{3.28}$$

3.5.4.3 System Boundary No. 6: XN1 Heater

- Energy balance equation:

$$\dot{m}_9 h_9 + \dot{m}_{30} h_{30} - \dot{m}_{10} h_{10} - \dot{m}_{38} h_{38} = 0 \tag{3.29}$$

- Mass balance equations:

$$\dot{m}_9 - \dot{m}_{10} = 0 \tag{3.30}$$

$$\dot{m}_{30} - \dot{m}_{38} = 0 \tag{3.31}$$

3.5.4.4 System Boundary No. 7: XN2 Heater

- Energy balance equation:

$$\dot{m}_{10} h_{10} + \dot{m}_{28} h_{28} + \dot{m}_{36} h_{36} - \dot{m}_{11} h_{11} - \dot{m}_{37} h_{37} = 0 \tag{3.32}$$

- Mass balance equations:

$$\dot{m}_{10} - \dot{m}_{11} = 0 \tag{3.33}$$

$$\dot{m}_{28} + \dot{m}_{36} - \dot{m}_{37} = 0 \tag{3.34}$$

3.5.4.5 System Boundary No. 8: Manifold Before XN3

- Energy balance equation:

$$\dot{m}_{11} h_{11} + \dot{m}_{37} h_{37} + \dot{m}_{43} h_{43} - \dot{m}_{12} h_{12} = 0 \tag{3.35}$$

- Mass balance equation:

$$\dot{m}_{11} + \dot{m}_{37} + \dot{m}_{43} - \dot{m}_{12} = 0 \tag{3.36}$$

3.5.4.6 System Boundary No. 9: XN3 Heater

- Energy balance equation:

$$\dot{m}_{12} h_{12} + \dot{m}_{26} h_{26} - \dot{m}_{13} h_{13} - \dot{m}_{36} h_{36} = 0 \tag{3.37}$$

- Mass balance equations:

$$\dot{m}_{12} - \dot{m}_{13} = 0 \tag{3.38}$$

$$\dot{m}_{26} - \dot{m}_{36} = 0 \tag{3.39}$$

3.5.4.7 System Boundary No. 10: XN4 Heater

- Energy balance equation:

$$\dot{m}_{13}h_{13} + \dot{m}_{24}h_{24} - \dot{m}_{14}h_{14} - \dot{m}_{35}h_{35} = 0 \tag{3.40}$$

- Mass balance equations

$$\dot{m}_{13} - \dot{m}_{14} = 0 \tag{3.41}$$

$$\dot{m}_{24} - \dot{m}_{35} = 0 \tag{3.42}$$

3.5.4.8 System Boundary No. 11: Manifold After XN4 Heater

- Energy balance equation:

$$\dot{m}_{14}h_{14} + \dot{m}_{35}h_{35} + \dot{m}_{44}h_{44} + \dot{m}_{169}h_{169} - \dot{m}_{15}h_{15} = 0 \tag{3.43}$$

- Mass balance equation:

$$\dot{m}_{14} + \dot{m}_{35} + \dot{m}_{44} + \dot{m}_{169} - \dot{m}_{15} = 0 \tag{3.44}$$

3.5.4.9 System Boundary No. 12: Manifold Before XN1 Heater

- Energy balance equation:

$$\dot{m}_{29}h_{29} + \dot{m}_{57}h_{57} - \dot{m}_{30}h_{30} = 0 \tag{3.45}$$

- Mass balance equation:

$$\dot{m}_{29} + \dot{m}_{57} - \dot{m}_{30} = 0 \tag{3.46}$$

3.5.4.10 System Boundary No. 13: Manifold Before XN2 Heater

- Energy balance equations:

$$\dot{m}_{27}h_{27} - \dot{m}_{28}h_{28} - \dot{m}_{40}h_{40} = 0 \tag{3.47}$$

$$h_{27} = h_{28} = h_{40} \tag{3.48}$$

- Mass balance equation:

$$\dot{m}_{27} - \dot{m}_{28} - \dot{m}_{40} = 0 \tag{3.49}$$

3.5.4.11 System Boundary No. 14: Manifold before XN3 Heater

- Energy balance equations:

$$\dot{m}_{25}h_{25} - \dot{m}_{26}h_{26} - \dot{m}_{39}h_{39} = 0 \tag{3.50}$$

$$h_{25} = h_{26} = h_{39} \tag{3.51}$$

- Mass balance equation:

$$\dot{m}_{25} - \dot{m}_{26} - \dot{m}_{39} = 0 \tag{3.52}$$

The temperature of water at the exhaust of heat exchangers is calculated using Peclet's formula and energy balance of the heaters (analogical as for the case of the condensers; compare Eq. 3.19):

$$T_{k2} = T_s - (T_s - T_{p2})e^{\frac{-\mathrm{kF_n}}{\dot{m}_w \mathrm{c_w}}} \tag{3.53}$$

where:

T_{k2} temperature of water at the exhaust from regenerative feed water heater,
T_s saturation temperature of condensate at the exhaust from the heater,
T_{p2} temperature of water supplied to regenerative feed water heater,
$\mathrm{kF_n}$ product of heat transfer coefficient and heating surface of the heater calculated for rated operating conditions (Table 3.5), which is considered as a constant over the entire range of operating conditions of heater,
$\mathrm{c_w}$ specific heat of the water passing through the heater,
$\dot{m}_w$ water mass flow through the heater.

The calculations were started by determination of temperature behind the condensate cooler CT1, and the temperatures behind the remaining LP heaters were subsequently calculated. Water enthalpies corresponding to the temperature

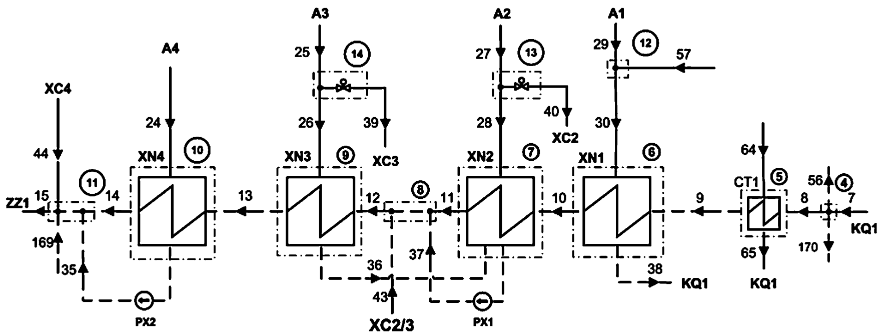


Fig. 3.27 Diagram of LP regeneration

determined were calculated for the steam quality $x = 0$, which disregards the influence of the condensate pumps (Fig. 3.27).

3.5.5 Deaerator, Main Feed Water Pump, KQ2 Condenser

3.5.5.1 System Boundary No. 15: ZZ1 Feed Water Tank

- Energy balance equation:

$$\dot{m}_{15}h_{15} + \dot{m}_{22}h_{22} + \dot{m}_{34}h_{34} + \dot{m}_{47}h_{47} - \dot{m}_{16}h_{16} = 0 \tag{3.54}$$

- Mass balance equation:

$$\dot{m}_{15} + \dot{m}_{22} + \dot{m}_{\dot{m}34} + \dot{m}_{47} - \dot{m}_{16} = 0 \tag{3.55}$$

3.5.5.2 System Boundary No. 16: Feed Water Pump PZ1

- Energy balance equation:

$$(\dot{m}_{23}h_{23} - \dot{m}_{31}h_{31})\eta_{\mathrm{m1K12}} - (\dot{m}_{17}h_{17} - \dot{m}_{116}h_{116})/\eta_{\mathrm{mPZ1}} = 0 \tag{3.56}$$

$$h_{16} = h_{116} = h_{168} \tag{3.57}$$

- Mass balance equations:

$$\dot{m}_{116} - \dot{m}_{17} = 0 \tag{3.58}$$

$$\dot{m}_{23} - \dot{m}_{31} = 0 \tag{3.59}$$

$$\dot{m}_{168} + \dot{m}_{116} - \dot{m}_{16} = 0 \tag{3.60}$$

3.5.5.3 System Boundary No. 17: KQ2 Condenser

- Energy balance equation:

$$\dot{m}_{31}h_{31} - \dot{m}_{32}h_{32} - \dot{m}_{KQ2}\mathrm{c_w}\left(T_{w2KQ2} - \mathrm{T_{w1KQ2}}\right) = 0 \tag{3.61}$$

- Mass balance equation:

$$\dot{m}_{31} - \dot{m}_{32} = 0 \tag{3.62}$$

3.5.6 High Pressure Regenerative Feed Water Preheaters: XW1/2, XW3/4

3.5.6.1 System Boundary No. 18: XW1/2 Regenerative Feed Water Preheater

- Energy balance equation:

$$\dot{m}_{17}h_{17} + \dot{m}_{21}h_{21} + \dot{m}_{33}h_{33} - \dot{m}_{18}h_{18} - \dot{m}_{34}h_{34} = 0 \tag{3.63}$$

- Mass balance equations:

$$\dot{m}_{17} - \dot{m}_{18} = 0 \tag{3.64}$$

$$\dot{m}_{21} + \dot{m}_{33} - \dot{m}_{34} = 0 \tag{3.65}$$

3.5.6.2 System Boundary No. 19: XW3/4 Regenerative Feed Water Preheater

- Energy balance equation:

$$\dot{m}_{18}h_{18} + \dot{m}_{20}h_{20} - \dot{m}_{19}h_{19} - \dot{m}_{33}h_{33} = 0 \tag{3.66}$$

- Mass balance equations:

$$\dot{m}_{18} - \dot{m}_{19} = 0 \tag{3.67}$$

$$\dot{m}_{20} - \dot{m}_{33} = 0 \tag{3.68}$$

Fig. 3.28 Schematic diagram of deaerator, main feed pipe and KQ2 condenser

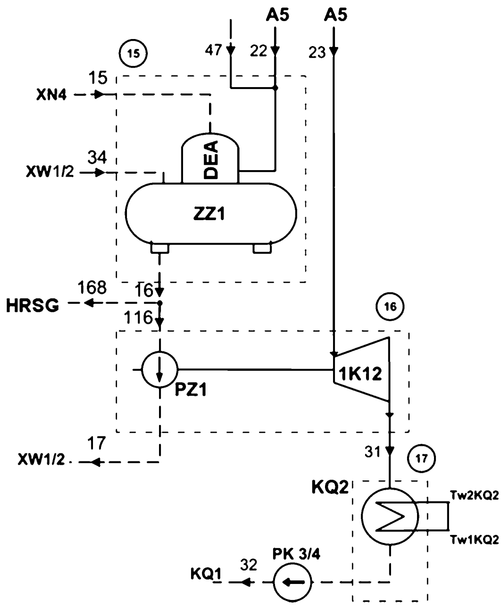
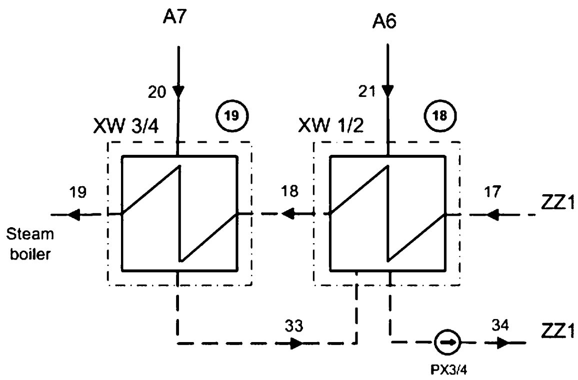


Fig. 3.29 Schematic diagram of HP regenerative feed water preheaters: XW1/2, XW3/4



The temperatures of the water at the exhaust of regenerative feed water preheaters were calculated on the basis of the relation (3.53). The values of enthalpies corresponding to them were determined from the regression equation for the feed water at the output of the pump (Fig. 3.16). In this process the pressure drops in the preheaters were disregarded (Figs. 3.28, 3.29, 3.30, 3.31).

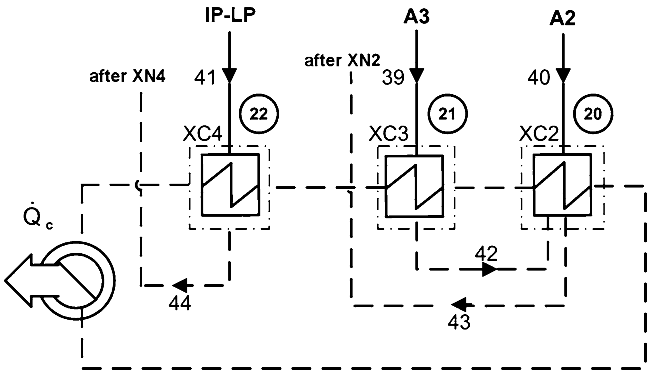


Fig. 3.30 Schematic diagram of the heater structure

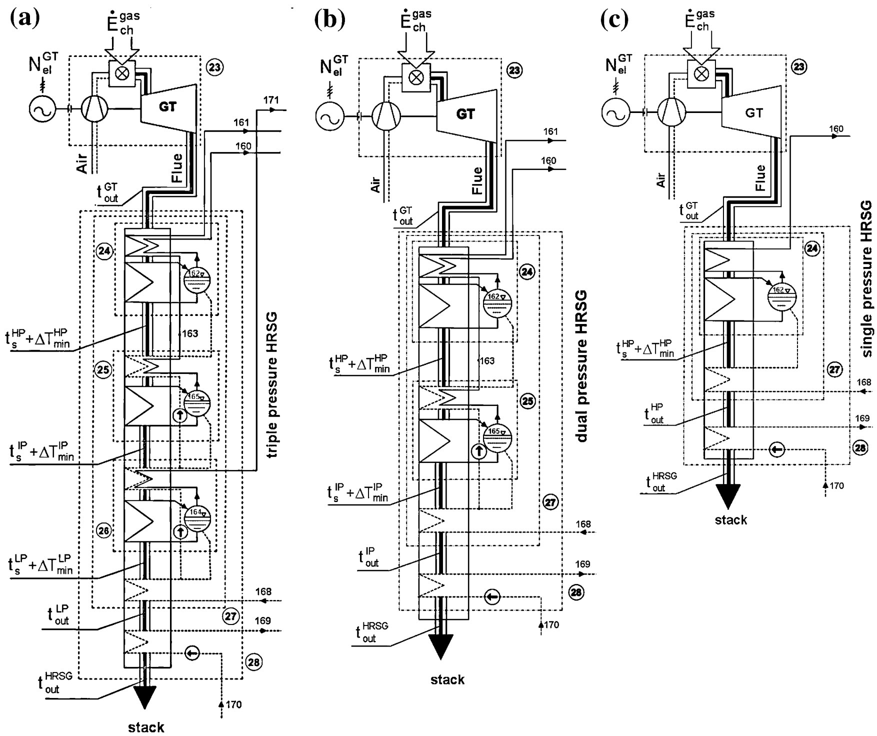


Fig. 3.31 Schematic diagram of gas turbine and heat recovery steam generators **a** triple pressure HRSG, **b** dual pressure HRSG, **c** single pressure HRSG

3.5.7 *XC2, XC3, XC4 Heaters*

3.5.7.1 System Boundary No. 20: XC2 Heater

- Energy balance equation:

$$\dot{Q}_{\mathrm{XC2}} = \dot{m}_{40}h_{40} + \dot{m}_{42}h_{42} - \dot{m}_{43}h_{43} \tag{3.69}$$

- Mass balance equation:

$$\dot{m}_{40} + \dot{m}_{42} - \dot{m}_{43} = 0 \tag{3.70}$$

3.5.7.2 System Boundary No. 21: XC3 Heater

- Energy balance equation:

$$\dot{Q}_{\mathrm{XC3}} = \dot{m}_{39}h_{39} - \dot{m}_{42}h_{42} \tag{3.71}$$

- Mass balance equation:

$$\dot{m}_{39} - \dot{m}_{42} = 0 \tag{3.72}$$

3.5.7.3 System Boundary No. 22: XC4 Heater

- Energy balance equation:

$$\dot{Q}_{\mathrm{XC4}} = \dot{m}_{41}h_{41} - \dot{m}_{44}h_{44} \tag{3.73}$$

- Mass balance equation:

$$\dot{m}_{41} - \dot{m}_{44} = 0 \tag{3.74}$$

while:

$$\dot{Q}_{\mathrm{XC2}} + \dot{Q}_{\mathrm{XC3}} + \dot{Q}_{\mathrm{XC4}} = \dot{Q}_{\mathrm{c}} = \dot{Q}_{\mathrm{u}} + \dot{Q}_{\mathrm{dhw}}. \tag{3.75}$$

A constant pressure drop equal to 10 kPa was assumed in the pipeline feeding the heaters. Moreover, an assumption of a constant value of condensate subcooling behind the heaters was assumed at 2°C.

3.5.8 Gas Turbine and Heat Recovery Steam Generator

3.5.8.1 System Boundary No. 23: Gas Turbogenerator

- Energy balance equations:

$$\dot{C}(t_{out}^{GT} - t_{amb}) = \dot{E}_{ch}^{gas} - N_{el}^{GT} \tag{3.76}$$

$$\dot{E}_{ch}^{gas} = \frac{N_{el}^{GT}}{\eta_{GT}} \tag{3.77}$$

3.5.8.2 System Boundary No. 24: High Pressure Section of HRSG

- Energy balance equations:

$$\dot{C}[t_{out}^{GT} - (t_s^{HP} + \Delta T_{min}^{HP})] = \dot{m}_{160}(h_1 - h'_{162}) + \dot{m}_{161}(h_4 - h_{163}) \tag{3.78}$$

$$h'_{162} = h'_{162}(\text{water};\ \mathrm{x} = 0;\ \mathrm{t} = t_s^{HP};\ p_{162} = p_1) \tag{3.79}$$

$$h_{163} = h_{163}(\text{steam};\ \mathrm{t} = t_s^{HP};\ p_{163} = p_4) \tag{3.80}$$

3.5.8.3 System Boundary No. 25: Intermediate Pressure Section of HRSG

- Energy balance equations:

$$\dot{C}[(t_s^{HP} + \Delta T_{min}^{HP})] - (t_s^{IP} + \Delta T_{min}^{IP}) = \dot{m}_{160}(h_{162} - h'_{165}) + \dot{m}_{161}(h_{163} - h_{165}) \tag{3.81}$$

$$h'_{165} = h'_{165}(\text{water}; x = 0;\ \mathrm{t} = t_s^{IP};\ p_{165} = p_{163} = p_4) \tag{3.82}$$

3.5.8.4 System Boundary No. 26: Lowd Pressure Section of HRSG

- Energy balance equations:

$$\dot{C}[(t_s^{IP} + \Delta T_{min}^{IP}) - (t_s^{LP} + \Delta T_{min}^{LP})] = (\dot{m}_{160} + \dot{m}_{161})(h'_{165} - h'_{164}) + \dot{m}_{171}(h_5 - h'_{164}) \tag{3.83}$$

$$h'_{164} = h'_{164}(\text{water};\ \mathrm{x} = 0;\ \mathrm{t} = t_s^{LP}; p_{164} = p_5) \tag{3.84}$$

$$h_{171} = h_5 \tag{3.85}$$

3.5.8.5 System Boundary No. 27: HRSG without of Low Pressure Regenerative Feed Water Preheater

- Energy balance equations:

$$\dot{C}(t_{\mathrm{out}}^{\mathrm{GT}} - t_{\mathrm{out}}^{\mathrm{LP}}) = \dot{m}_{160}(h_1 - h_{168}) + \dot{m}_{161}(h_4 - h_{168}) + \dot{m}_{171}(h_5 - h_{168}) \quad (3.86)$$

$$h_{168} = h_{16} \quad (3.87)$$

- Mass balance equation:

$$\dot{m}_{168} = \dot{m}_{160} + \dot{m}_{161} + \dot{m}_{171} \quad (3.88)$$

3.5.8.6 System Boundary No. 28: HRSG

- Energy balance equations:

$$\begin{aligned}\dot{C}(t_{\mathrm{out}}^{\mathrm{GT}} - t_{\mathrm{out}}^{\mathrm{HRSG}}) &= \dot{m}_{160}(h_1 - h_{168}) + \dot{m}_{161}(h_4 - h_{168}) \\ &\quad + \dot{m}_{171}(h_5 - h_{168}) + \dot{m}_{170}(h_{169} - h_{170})\end{aligned} \quad (3.89)$$

$$h_{170} = h_7 \quad (3.90)$$

$$h_{169} = h_{15} \quad (3.91)$$

- Mass balance equation:

$$\dot{m}_{169} = \dot{m}_{170} \quad (3.92)$$

In Eqs. 3.78, 3.86, 3.88 and 3.89 for the case of a single-pressure boiler $\dot{m}_{161} = \dot{m}_{171} = 0$ and $t_{out}^{LP} \equiv t_{out}^{HP}$, while in a dual-pressure boiler $\dot{m}_{171} = 0$ and $t_{\mathrm{out}}^{\mathrm{LP}} \equiv t_{\mathrm{out}}^{\mathrm{IP}}$.

In Eqs. 3.86, 3.89 for the case of a triple-pressure boiler $h_{168} = h_{169}$ ($p_{164} < p_{168}$), in Eq. 3.60 $\dot{m}_{168} = 0$ and in Eqs. 3.43, 3.92 $\dot{m}_{169} = \dot{m}_{170} - (\dot{m}_{160} + \dot{m}_{161} + \dot{m}_{171})$.

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Chapter 4
Algorithm for the Calculation of an Optimum Structure of Heat Exchangers for the Modernization of a 370 MW Power Unit to Combined Heat and Power Cycle

The opportunity decreases the consumption of the chemical energy of fuels and the following reduction in the emission of hazardous products of their combustion plays a very significant role. Hence, an important advantage from the ecological point of view is offered by the modernization of existing condensing power stations to operate with combined heat and power, thus enabling heat generation $\dot{Q}_c = \dot{Q}_u + \dot{Q}_{dhw}$ besides the production of electricity. Such heat can be used for the purposes of district heating and air conditioning $\dot{Q}_u$ and utilized for the purposes of supply of domestic hot water $\dot{Q}_{dhw}$ (Fig. 3.4).

Moreover, the modernization of a power plant is associated with the improvement of its energy efficiency. It was indicated that such modernization can also improve the economic effectiveness of the power station. The final effect will mainly depend on the price relations between energy carriers, ratios of heat to power prices and fuel prices as well as the tariffs imposed on the pollution of the environment. At this point it is important to directly state the thesis that the familiar technology and technical facilities along with existing networks that guarantee the rational use of primary fuels should decide on the threshold (determined for the given boundary value of economic efficiency in terms of NPV) price relations between energy carriers and tariffs on the pollution of the environment in a way that ensures that the whole undertaking is feasible from the economic perspective. In the era of the greenhouse effect such considerations play an increasingly important role.

Chapter 2 was devoted to the comparison of the possible methods of modernization of coal-fired power stations to combined heat and power from the thermal and economic perspective. For the current energy carrier prices the most justified option involves steam bleed from turbine extractions from the low-pressure section of the turbine for feeding heaters. Hence, a more detailed technical and economic analysis of this solution is a further necessity.

For the purpose of analysis a mathematical model of the power unit with the rated capacity of 370 MW has been developed. Such units constitute basic power

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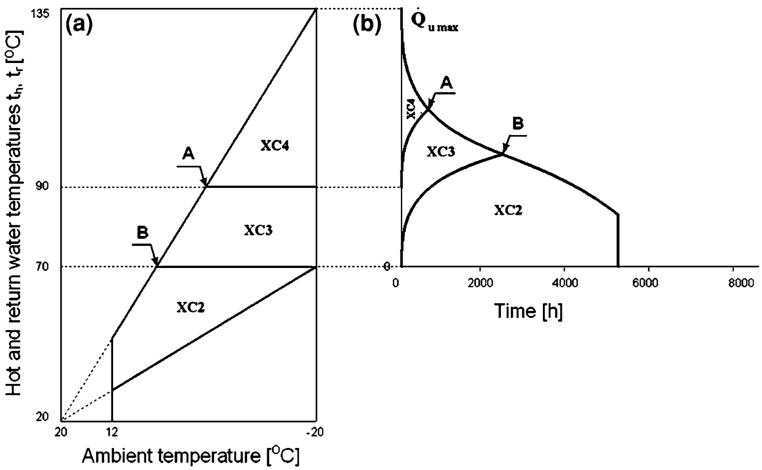

Fig. 4.1 Qualitative regulation of thermal power output $\dot{Q}_u$ from the power station for purposes of heating, air conditioning and ventilation of residential areas for the alternative with three heaters XC2, XC3 and XC4: **a** linear regulation chart, **b** annual scheduled chart of demand for thermal power (t_h, t_r temperatures of network hot water and return water)

unit models operating in Polish power stations. This model is based on equations of energy and mass balance, relations characterizing the operation of the facilities and equations of the state of the circulating medium.

The model of the actual thermal technology unit was supplemented by three heaters, XC2, XC3, XC4 connected in a series with steam bleed from two steam extractions A2, A3 in the LP reheat stage and from IP–LP crossoverpipe. The maximum temperature of circulating water in XC2 heater is $t_h = 70°\text{C}$, $t_h = 90°\text{C}$ in XC3 heater and $t_h = 135°\text{C}$ in XC4 (Fig. 4.1).

The extraction of steam for purposes of heating constitutes the cause for the divergence of the turbine from its nominal operating conditions at which it operates with its highest efficiency. Consequently, steam pressures are altered in the particular stages of the turbine. Moreover, variable demand for network heat imposes the necessity of variable mass stream of steam for the supply of heat exchangers and, hence, make these divergences variable in time. The mathematical model of the power unit has to, consequently, account for these changes and in order to reflect them Stodola–Flügel turbine passage equation is applied [1].

The fundamental question to be set during the modernization unit to combined heat and power involves the selection of an appropriate structure of heaters to guarantee an optimum investment strategy. Another question regards the optimum alternative for the supply of district heating $\dot{Q}_c = \dot{Q}_u + \dot{Q}_{dhw}$, i.e. in accordance with the annual schedule of heat demand (Fig. 4.1): would an optimum alternative involve the application of all three heaters, i.e. XC2, XC3, XC4, or may be only two of them, i.e. XC3 and XC4 or XC2 and XC4, or just one: XC4? What would be an optimum structure of heaters needed for the supply of a large greenhouse farm used for growing flowers and vegetables? Would an optimum structure require the use of two heaters i.e. XC2 and XC3 or XC3 and XC4, or just one: XC4?

The stream of mass of the steam to feed the particular heat exchangers vary depending on ambient temperature in accordance with the qualitative regulation chart (Fig. 4.1a).

For instance, mass stream of steam from IP–LP crossoverpipe to feed XC4 heater has the highest value in peak demand period for network heat and assumed zero value at point A to the right from this point (Fig. 4.1b). Concurrently, at this point and to the left from it, the flow from extraction A3, the mass stream of steam assumes a constant maximum value, while at point B it assumes zero value while mass stream of steam from extraction A2 feeding XC2 heat exchanger assumes its maximum value at this point.

In the alternative with two heaters the mass streams of steam from extractions A2 and A3 assume zero values for the respective systems with XC3 and XC4 heaters and XC2, XC4 over the entire area of the time chart.

If the structure of the heaters were to change from XC2, XC3 and XC4 into XC2 and XC4 due to the fact that the mass stream of steam from extraction A3 assumes zero value over the entire range of heat demand periods, the value of steam extraction from IP–LP crossoverpipe changes accordingly. In this alternative, the variable steam from extraction IP–LP crossoverpipe is slightly smaller than the total variable mass stream from A3 and IP–LP crossoverpipe extractions in the alternative with three heaters (specific enthalpy of heating steam from extraction IP–LP crossoverpipe is higher than that of extraction A3). Concurrently, the changes in the value of the steam bleed from extraction A2 are the same in both alternatives.

In the system with XC3 and XC4 heaters the total heat production in XC2 has to be taken over by XC3 heater. Hence, the variable mass stream of steam from A3 extraction is virtually formed by the total of the variable streams from two extractions (A3 and A2) from the alternative with three heaters, while the values of the mass stream of steam from IP–LP crossoverpipe extraction are the same over the entire area of the annual chart summarizing the demand for heat as in the alternative with XC2, XC3 and XC4.

The greater the number of in-series installed heaters the smaller the exergy losses in the system and, hence, the higher the production of electricity under the assumption of a constant heat demand. However, the investment cost will increase as well. Thus, there is an optimum solution in terms of technology economy, which will be sought further on.

The economic criterion should govern the process of selection of an optimum structure of heaters. This criterion can be expressed as the maximization of the profit resulting from the modernization of the power unit to combined heat and power. The economic criterion is superior to the technology design aspect. This is so since in the market economy, the economic efficiency decides on the application of specific technology solutions when undertaking an investment.

The economic criterion for the selection of an optimum structure of heat exchangers is expressed by the relation (Eq. 2.1).

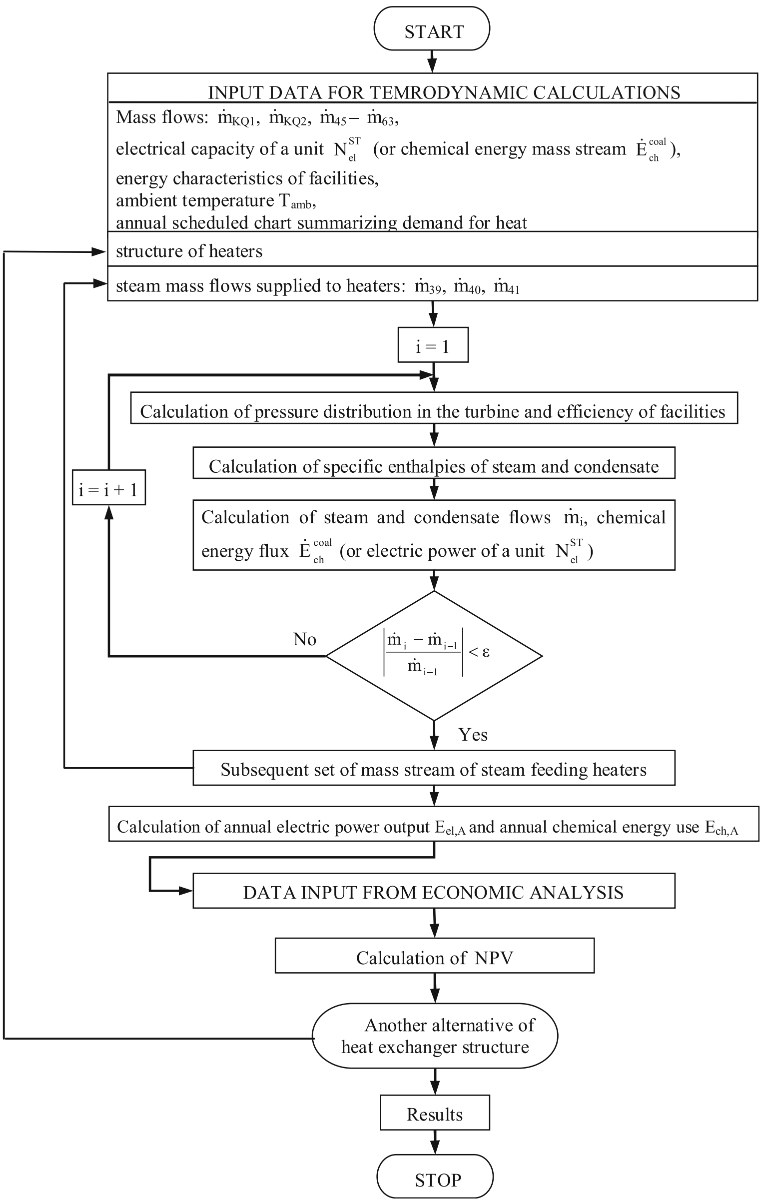

Fig. 4.2 Algorithm for the calculation of an optimum structure of heat exchangers

Figure 4.2 presents an overview of the algorithm applied for the selection of an optimum structure of heat exchangers during the modernization of a power unit to combined heat and power.

First, thermodynamic calculations are conducted. The results gained form the input into economic calculations.

The input data include: electric power output: for the condition $N_{\mathrm{el}}^{\mathrm{ST}} = \text{const.}$ or chemical energy mass stream of the fuel (for the condition $\dot{E}_{\mathrm{ch}}^{\mathrm{coal}} = \text{const.}$), current structure of heaters, energetic characteristics of facilities, heat power of the unit, mass stream of cooling water in the condensers: $\dot{m}_{\mathrm{KQ1}}$ and $\dot{m}_{\mathrm{KQ2}}$, values of thermal parameters of fresh and superheated steam, thermal parameters of the condensate, ambient temperature T_{amb}, complete set of steam leakages from diaphragms and valves $\dot{m}_{45} - \dot{m}_{63}$ and mass stream of bleed steam from extractions $\dot{m}_{39}$, $\dot{m}_{40}, \dot{m}_{41}$ that feed the particular heater in their assumed structure.

The values of these mass streams are derived from the annual scheduled chart summarizing the demand for district or technology heat and curves that form the boundaries of the capacities of particular heaters resulting from qualitative regulation (Fig. 4.1). The calculations of the electric power of the units (or chemical energy streams) for a given structure of heat exchangers were conducted for several hundred flux values that cover with a relatively small time step equal to $\Delta\tau = 24$ h the entire area of the annual chart summarizing the demand for heat. The calculation of the power for each of these sets is a necessity since it is aimed at determining the annual output of electric energy of a power unit. This value forms an input into the calculations of the economic efficiency during the exploitation of a power unit.

The calculations of the electric power (and chemical energy stream) of a power unit with regard to each set of input data requires iterations. This comes as a consequence of the divergence of the turbine from its nominal operating conditions (denoted by n index in Eq. 4.1); the current flow through the particular stages of the turbine are not marked with any indices; the pressures in the extractions before and behind the particular stages are marked as p_{p} and p_{k}, respectively; the current temperatures before a stage as T_{p}. It is notable that not accounting for simplex regarding temperature $T_{\mathrm{p}}/T_{\mathrm{n,p}}$ in Eq. 4.1 as a result of the fact that its value is close to one does not considerably influence the value of pressure p_{p} during steam extraction to feed heaters. For the same reasons, i.e. due to the pressure variations in the turbine extractions, the values of the heating steam streams $\dot{m}_{39}, \dot{m}_{40}, \dot{m}_{41}$ undergo iterative correlation applying the condition $\dot{Q}_{\mathrm{XC2}} = \text{const}$, $\dot{Q}_{\mathrm{XC3}} = \text{const}$, $\dot{Q}_{\mathrm{XC4}} = \text{const}$. The first iteration step involves determination of mass stream of steam through the particular stages of the turbine for the adopted input data. Subsequently, the calculated mass stream of steam is applied for the new distribution of pressures in the turbine with the aid of the Stodola–Flügel turbine passage equation. This makes it possible to illustrate pressure variations in the extraction resulting from the changes in the mass stream of steam through them [1]:

$$p_{\mathrm{p}} = \sqrt{\left(\frac{\dot{m}}{\dot{m}}\right)^2 \frac{T_p}{T_{n,p}} \left(p_{n,p}^2 - p_{n,k}^2\right) + p_k^2} \tag{4.1}$$

Calculations start with the final group of the LP section stages. For this purpose, first, the saturation pressure p_6 in the condenser is calculated by means of iterations with the aid of its energy balance and relation $\dot{Q}_{\text{cond}} = (\text{kF}\Delta T_{\log})_{\text{cond}}$. Following that, saturation pressures are calculated for IP sections, subsequently leading to calculation of p_4. Pressure p_2 is derived by adding the pressure drop in the interstage reheater p_{2-4} to pressure p_4. Pressure p_1 is derived from the relation of its given value behind the boiler to the mass stream of fresh steam from the boiler. This pressure cannot be derived from the turbine passage equation (during the operation with the rated capacity the opening of the regulation valve is in the range 80–100% depending on the quality of fuel, pressure in the condenser, steam admission into interstage collector among others). The nominal values in the Stodola–Flügel turbine passage equation for the specific stages include: values of steam flow, temperatures and pressures for the operation of the power unit with nominal capacity without assumption of combined heat and power, i.e. no steam extraction to feed heaters. On their basis the nominal values of specific enthalpy of the steam and condensate are calculated. New values of specific enthalpy of steam and condensate are derived for the new values of pressures and specific efficiency of the turbine η_{iHP}, η_{iIP}, η_{iLP}, which in turn, are determined on the basis of the characteristics of efficiency and steam bleed into HP, IP and LP sections of the turbine on the basis of iterations. These values are applied in the successive step, i.e. calculation of the new values of steam and condensate flow (under the assumption of constant temperatures T_1 and T_4 in the calculations of specific enthalpy). Subsequently, these data are used to calculate a new distribution of pressures in the turbine, etc. until the required precision of calculations is obtained. In the second and successive iteration steps the temperatures of the condensate behind the LP and HP regenerative feed water preheaters and the temperatures of the saturation in the condensers are derived with the aid of a relation stating the heat flow that is exchanged in a heater: $\dot{Q} = \text{kF}\Delta T_{\log}$, where $\Delta T_{\log}$ denotes algorithmic mean of the difference of the medium that is fed into a heater.

The electric power output from the unit derived in the final step of iteration as well the chemical energy flux combusted in the boiler form the values that had to be determined.

Using the calculated electric power and chemical energy mass streams for each set of the input data it is necessary to derive the annual electric output from a unit and the annual use of chemical energy. The calculations of the annual values have to be repeated for all alternatives of the heater structure after modernization in order to compare their economic efficiency and select an alternative in which the value of NPV is the highest (Eq. 2.1).

At this point the process of thermodynamic calculations is complete and it is possible to proceed to economic calculations for which the input data include, besides the values of the investment, annual generation of electric energy and heat, annual use of chemical energy of the coal: prices of energy carriers and heat, ecological fees, interest rate, distance from the power plant to the end heat consumers (which will determine the economically justified distance of heat transmission pipeline), etc. (Eq. 2.1).

For an optimum structure, i.e. the alternative for which the highest value of NPV is gained, it is necessary to repeat the calculations under the assumption of increased fuel use in the boiler thus restoring the initial electric power of the power plant. It is predictable that the economic efficiency of the undertaking in this case will be even greater.

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Chapter 5
Testing Calculations of the Mathematical Model of a Power Unit

5.1 Methods for Solving the System of Equations

The development of a mathematical model (mathematical modeling) is not limited to the formulation of the relations between the particular variables and their transformation into the simplest form. Neither it is sufficient to state these relations in the form of analytical equations. In order to obtain a comprehensive mathematical model it is also necessary to present an appropriate method for solving such equations. This is the numerical algorithm which decides on the quality and precision of solving the investigated problem.

The range of solutions to systems of equations, both linear and nonlinear is very extensive and the number of practically applied algorithms exceeds a few dozen. Initially the idea of solving the system of algebraic linear equations consisted in the application of Gaussian elimination method together with numerical procedures for deriving thermal and calorific parameters of water and steam. However, a decision was made to use *Engineering Equation Solver* program (*EES*). It is a window-based application operating under MS Windows, which includes a solver for solving algebraic nonlinear equations and containing an extensive databases of thermodynamic parameters encountered in widely understood thermal technology, including water and steam systems.

The solving of the system of equations is based on one of the version of the Newton's algorithm, which displays the following characteristics:

- terms of the Jacobean are derived on the basis of numerical calculations,
- system of equations is divided into blocks in order to enhance the coherence, the division occurs by probing Jacobean by the Tarjan method.

The software was manufactured by US-based F-Chart Software (www.fchart.com).

R. Bartnik and Z. Buryn, *Conversion of Coal-Fired Power Plants to Cogeneration and Combined-Cycle*, DOI: 10.1007/978-0-85729-856-0_5,

Table 5.1 Summary of basic quantities in the steam and water balance

L.p.	Symbols in use in Fig. 3.4	Parameter	Unit
1	$\dot{m}_1$	Stream of mass of fresh steam	kg/s
2	T_1	Fresh steam temperature	°C
3	P_1	Fresh steam pressure	kPa
4	T_4	Superheated steam temperature	°C
5	T_{w1KQ1}	Temperature of cooling water before KQ1	°C
6	$N_{\mathrm{el}}^{\mathrm{ST}}$	Generator's continuous capacity	MW
7	$\dot{m}_2$	Steam stream of mass behind HP section of turbine	kg/s
8	T_2	Steam temperature behind HP section of turbine	°C
9	P_2	Steam pressure behind HP section of turbine	kPa
10	$\dot{m}_3$	Steam stream of mass into reheat stage	kg/s
11	$\dot{m}_4$	Superheated steam stream of mass into IP section of turbine	kg/s
12	p_4	Superheated steam pressure	°C
13	$\dot{m}_5$	Steam stream of mass into LP section of turbine	kg/s
14	T_5	Steam temperature in IP–LP crossoverpipe	°C
15	p_5	Steam pressure in IP–LP crossoverpipe	kPa
16	$\dot{m}_6$	Steam stream of mass into condenser KQ1	kg/s
17	T_6	Steam temperature in condenser KQ1	°C
18	p_6	Steam pressure in condenser KQ1	kPa
19	$\dot{m}_7$	Condensate stream behind pumps PK1/2	kg/s
20	T_7	Condensate temperature behind pumps PK ½	°C
21	$\dot{m}_{10}$	Condensate stream behind heater XN1	kg/s
22	T_{10}	Condensate temperature behind heater XN1	°C
23	$\dot{m}_{11}$	Condensate stream behind heater XN2	kg/s
24	T_{11}	Condensate temperature behind heater XN2	°C
25	$\dot{m}_{12}$	Condensate stream before heater XN3	kg/s
26	T_{12}	Condensate temperature before heater XN3	°C
27	$\dot{m}_{13}$	Condensate stream behind heater XN3	kg/s
28	T_{13}	Condensate temperature behind heater XN3	°C
29	$\dot{m}_{14}$	Condensate stream behind heater XN4	kg/s
30	T_{14}	Condensate temperature behind heater XN4	°C
31	$\dot{m}_{15}$	Condensate stream into feedwater tank ZZ1	kg/s
32	T_{15}	Condensate temperature into feedwater tank ZZ1	°C
33	$\dot{m}_{16}$	Feedwater flow into feed pump PZ1	kg/s
34	T_{16}	Condensate temperature in feedwater tank ZZ1	°C
35	$\dot{m}_{17}$	Feedwater flow before reheat stage XW	kg/s
36	T_{17}	Temperature of feedwater before reheat stage XW	°C
37	$\dot{m}_{18}$	Feedwater flow behind heater XW1/2	kg/s
38	T_{18}	Temperature of feedwater behind heater XW1/2	°C
39	$\dot{m}_{19}$	Feedwater flow into boiler before reheat stage XW	kg/s
40	T_{19}	Temperature of feedwater in boiler behind reheat stage XW	°C
41	$\dot{m}_{20}$	Steam stream of mass from extraction A7 to XW3/4 heater	kg/s
42	T_{20}	Steam temperature in extraction A7	°C
43	p_{20}	Steam pressure in extraction A7	kPa
44	$\dot{m}_{21}$	Steam stream of mass in extraction A6	kg/s

(continued)

Table 5.1 (continued)

L.p.	Symbols in use in Fig. 3.4	Parameter	Unit
45	T_{21}	Steam temperature from extraction A6 to XW1/2 heater	°C
46	p_{21}	Steam pressure in extraction A6	kPa
47	$\dot{m}_{22}+\dot{m}_{23}+\dot{m}_{47}$	Steam stream of mass from extraction A5 to TP and ZZ1	kg/s
48	T_{22}	Steam temperature in extraction A5	°C
49	p_{22}	Steam pressure in extraction A5	kPa
50	$\dot{m}_{24}$	Steam stream of mass from extraction A4 to XN4 heater	kg/s
51	T_{24}	Steam temperature in extraction A4	°C
52	p_{24}	Steam pressure in extraction A4	kPa
53	$\dot{m}_{25}$	Steam stream of mass from extraction point A3 to feed XN3 heater	kg/s
54	T_{25}	Steam temperature in extraction A3	°C
55	p_{25}	Steam pressure in extraction A3	kPa
56	$\dot{m}_{27}$	Steam stream of mass at extraction point A2 to XN2 heater	kg/s
57	T_{27}	Steam temperature in extraction A2	°C
58	p_{27}	Steam pressure in extraction A2	kPa
59	$\dot{m}_{29}+\dot{m}_{57}$	Steam stream of mass from extraction point A1 to feed XN1 heater	kg/s
60	p_{29}	Steam pressure in extraction A1	kPa
61	T_{w2KQ1}	Temperature of cool water behind KQ1	°C

5.2 Results of Testing Calculations

Table 5.1 summarizes the basic parameters of the flows, temperatures and pressures in the water-steam circulation and their symbols (Fig. 3.4). The calculations were performed in accordance with the algorithm given in Chap. 4 under the assumption of a system operating without combined heat and power. The comparison between these data (measured and calculated) made it possible to assess the correctness of the mathematical model presented in Chap. 3.

The testing of the mathematical model of the power unit involved the comparison of the variables resulting from the calculations using EES and parameters gained during the measurements on the live system. The input values included streams of mass, temperatures and pressures of fresh steam, temperature of superheated steam and temperature of cooling water before KQ1 condenser, whose values in the program were equal to the measured values. The ambient temperature in the program has been selected in a way that ensures an equivalence of the measured value with the calculated temperature of the cool water T_{w1KQ1}.

Tables 5.2 and 5.3 contain a comparison between values of measurements and the results of calculations for the steam stream of mass in the range of 171.6–305.6 kg/s, which corresponds to electric power capacity of 218–370 MW. For each measurement the relative error was derived from the relation:

Table 5.2 Comparison between measured values and results of calculations for various values of fresh mass stream of steam, measurements numbers 1, 2, 3

No.	Parameter from Fig. 3.4	Measurement 1			Measurement 2			Measurement 3		
		Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)
1	$\dot{m}_1$	–	305.6	–	–	277.4	–	–	255.3	–
2	T_1	–	537.1	–	–	538	–	–	534.9	–
3	p_1	–	18,216	–	–	18,156	–	–	18,310	–
4	T_4	–	535.4	–	–	532.6	–	–	516.3	–
5	T_{w1KQ1}	–	30.5	–	–	27.6	–	–	30.5	–
6	N_{el}^{ST}	369.8	369.7	0.0	340.3	338.9	0.4	307.4	309.5	−0.7
7	$\dot{m}_2$	302.8	304.2	−0.5	274.8	272.9	0.7	252.9	254.3	−0.6
8	T_2	325.6	328.2	−0.8	319.4	319.3	0.0	309.5	310.4	−0.3
9	p_2	4,208	4,277	−1.6	3,842	3,859	−0.4	3,542	3,552	−0.3
10	$\dot{m}_3$	271.0	274.8	−1.4	247.1	247.4	−0.1	228.0	231.3	−1.4
11	$\dot{m}_4$	271.0	274.5	−1.3	247.5	247.1	0.2	228.4	231.1	−1.2
12	p_4	3,940	4,001	−1.5	3,599	3,610	−0.3	3,319	3,330	−0.3
13	$\dot{m}_5$	223.7	228.1	−1.9	204.3	206.7	−1.2	188.4	193.3	−2.5
14	T_5	264.3	264.0	0.1	262.6	263.3	−0.3	250.8	251.2	−0.2
15	p_5	548	558	−1.8	501	506	−1.0	461	466	−1.1
16	$\dot{m}_6$	193	195.1	−1.0	175.7	176.5	−0.5	162.9	166.3	−2.0
17	T_6	47.9	46.7	2.6	42.2	42.6	−0.9	44.1	44.3	−0.5
18	p_6	10.3	10.43	−1.0	8.3	8.44	−1.7	9.2	9.3	−1.1
19	$\dot{m}_7$	218	223.2	−2.2	197.4	201.6	−2.1	182.8	189.6	−3.6
20	T_7	47.9	47.0	1.9	42.2	43.1	−2.1	44.1	45.3	−2.6
21	$\dot{m}_{10}$	218	223.2	−2.2	197.3	201.6	−2.1	182.7	189.6	−3.6
22	T_{10}	67.9	68.9	−1.5	66.3	66.6	−0.5	64.9	66.1	−1.8
23	$\dot{m}_{11}$	218.2	223.2	−2.2	197.3	201.6	−2.1	182.7	189.6	−3.6
24	T_{11}	90.5	91.0	−0.5	88.6	88.8	−0.2	86.7	87.4	−0.8
25	$\dot{m}_{12}$	242	248.3	−2.6	218.6	224.2	−2.5	201.9	210.4	−4.0

(continued)

Table 5.2 (continued)

No.	Parameter from Fig. 3.4	Measurement 1			Measurement 2			Measurement 3		
		Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)
26	T_{12}	90.8	91.0	−0.2	88.8	88.8	0.0	86.9	87.4	−0.6
27	$\dot{m}_{13}$	241.8	248.3	−2.6	218.6	224.2	−2.5	201.9	210.4	−4.0
28	T_{13}	126.3	129.2	−2.2	124	126.8	−2.2	121.7	126.8	−4.0
29	$\dot{m}_{14}$	242	248.3	−2.6	218.6	224.2	−2.5	201.9	210.4	−4.0
30	T_{14}	149.0	153.5	−2.9	146	150.4	−2.9	143.2	147.8	−3.1
31	$\dot{m}_{15}$	251.8	259.4	−2.9	227.3	233.8	−2.8	209.8	219.2	−4.3
32	T_{15}	149.1	153.5	−2.9	146.1	150.4	−2.9	143.2	147.8	−3.1
33	$\dot{m}_{16}$	304.2	313.9	−3.1	276.2	281.7	−2.0	254.2	262.1	−3.0
34	T_{16}	180.4	181.2	−0.4	176.5	177.3	−0.5	173.1	177.5	−2.5
35	$\dot{m}_{17}$	304.2	313.6	−3.0	276.2	281.7	−2.0	254.2	262.1	−3.0
36	T_{17}	184.1	184.7	−0.3	180.2	180.8	−0.3	176.8	180.9	−2.3
37	$\dot{m}_{18}$	304.2	313.6	−3.0	276.2	281.5	−1.9	254.2	261.7	−2.9
38	T_{18}	209.7	210.5	−0.4	205.8	206.3	−0.2	202.4	203.3	−0.4
39	$\dot{m}_{19}$	304.3	313.6	−3.0	276.2	281.5	−1.9	254.2	261.7	−2.9
40	T_{19}	249.2	249.8	−0.2	244.5	244.8	−0.1	240.4	241.8	−0.6
41	$\dot{m}_{20}$	28.3	29.0	−2.4	24.7	25.1	−1.6	22.8	22.2	2.7
42	T_{20}	326	328.2	−0.7	319.4	318.5	0.3	309.5	310.4	−0.3
43	p_{20}	4,218	4,277	−1.4	3,842	3,857	−0.4	3,542	3,552	−0.3
44	$\dot{m}_{21}$	12.0	12.5	−4.0	10.9	11.2	−2.7	10.1	10.3	−1.9
45	T_{21}	435	436.4	−0.3	432.4	435.5	−0.7	417.5	418.8	−0.3
46	p_{21}	2,050	2,086	−1.7	1,866	1,877	−0.6	1,719	1,724	−0.3
47	$\dot{m}_{22} + \dot{m}_{23} + \dot{m}_{47}$	27.1	26.2	3.4	24.4	22.6	8.0	23.1	21.5	7.4
48	T_{22}	343	343.0	0.1	341.1	344	−0.8	327.7	329.0	−0.4
49	p_{22}	1,049	1,067	−1.7	956	968	−1.2	880	887	−0.8
50	$\dot{m}_{24}$	10.0	11.1	−9.9	8.7	9.6	−9.4	7.9	8.8	−10.2

(continued)

Table 5.2 (continued)

No.	Parameter from Fig. 3.4	Measurement 1			Measurement 2			Measurement 3		
		Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)
51	T_{24}	264	262.3	0.8	262.6	263.6	−0.4	250.8	250.7	0.0
52	p_{24}	548	558	−1.8	500	507	−1.4	461	466	−1.1
53	$\dot{m}_{25}$	15.6	17.1	−8.8	13.9	15.5	−10.3	12.7	14.3	−11.2
54	T_{25}	207	208.6	−1.0	205.2	205.5	−0.1	194.5	194.0	0.3
55	p_{25}	312	316	−1.3	285	289	−1.4	263	267	−1.5
56	$\dot{m}_{27}$	8.0	7.9	1.3	7.3	7.2	1.4	6.5	6.5	0.0
57	T_{27}	99.2	94.9	4.5	98.2	92.1	6.6	91.8	90.5	1.4
58	p_{27}	89.1	90.3	−1.3	81.5	81.9	−0.5	75.2	78.6	−4.3
59	$\dot{m}_{29} + \dot{m}_{57}$	7.8	8.3	−6.0	8.1	8.6	−5.8	6.7	6.6	1.5
60	p_{29}	35.6	35.7	−0.3	32.5	31	4.8	29.9	35.4	−15.5
61	T_{w2KQ1}	40.0	40.0	0.0	36.2	36.3	−0.3	38.6	38.7	−0.3

Table 5.3 Comparison between measured values and results of calculations for various values of fresh mass stream of steam, measurements numbers 4, 5, 6

No.	Parameter from Fig. 3.4	Measurement 4			Measurement 5			Measurement 6		
		Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)
1	$\dot{m}_1$	–	234.6	–	–	201.4	–	–	171.6	–
2	T_1	–	538.0	–	–	535.9	–	–	538.4	–
3	p_1	–	17,883	–	–	17,036	–	–	16,552	–
4	T_4	–	524.9	–	–	509.8	–	–	516.3	–
5	T_{w1KQ1}	–	29.6	–	–	26.5	–	–	28.2	–
6	N_{el}^{ST}	292.4	289.7	0.9	246.0	249.3	−1.3	214.9	217.9	−1.4
7	$\dot{m}_2$	232.4	234.0	−0.7	199.5	200.1	−0.3	170.0	177.0	−4.0
8	T_2	305.4	309.9	−1.5	301.4	300.6	0.3	291.1	293.3	−0.8
9	p_2	3,265	3,305	−1.2	2,816	2,813	0.1	2,410	2,447	−1.5
10	$\dot{m}_3$	210.2	213.3	−1.5	181.5	183.4	−1.0	155.4	158.5	−2.0
11	$\dot{m}_4$	210.6	213.0	−1.1	181.8	183.0	−0.7	155.7	158.3	−1.6
12	p_4	3,062	3,093	−1.0	2,643	2,640	0.1	2,264	2,299	−1.5
13	$\dot{m}_5$	174.9	179.0	−2.3	151.6	154.5	−1.9	130.8	139.5	−6.2
14	T_5	257.7	258.4	−0.3	247.3	248.0	−0.3	252.8	253.5	−0.3
15	p_5	428	435	−1.6	371	374	−0.8	320	327	−2.1
16	$\dot{m}_6$	151.5	154.2	−1.8	131.1	133.1	−1.5	114.3	116.7	−2.1
17	T_6	42.2	42.4	−0.5	37.5	38.0	−1.3	37.7	38.2	−1.3
18	p_6	8.31	8.37	−0.7	6.44	6.63	−2.9	6.5	6.7	−3.0
19	$\dot{m}_7$	169.4	175.2	−3.3	147.1	151.3	−2.8	127.2	132.1	−3.7
20	T_7	42.2	43.5	−3.0	37.5	39.4	−4.8	37.7	40.2	−6.2
21	$\dot{m}_{10}$	169.4	175.2	−3.3	147.0	151.3	−2.8	127.1	132.1	−3.8
22	T_{10}	63.6	64.5	−1.4	60.7	61.2	−0.8	58.6	59.1	−0.8
23	$\dot{m}_{11}$	169.4	175.2	−3.3	147	151.3	−2.8	127.1	132.1	−3.8
24	T_{11}	85.0	85.7	−0.8	81.4	82.3	−1.1	77.8	79.0	−1.5
25	$\dot{m}_{12}$	186.8	194.2	−3.8	161.7	167.3	−3.3	139.1	145.6	−4.5

(continued)

Table 5.3 (continued)

No.	Parameter from Fig. 3.4	Measurement 4			Measurement 5			Measurement 6		
		Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)
26	T_{12}	85.1	85.7	−0.7	81.5	82.3	−1.0	77.8	79.0	−1.5
27	$\dot{m}_{13}$	186.6	194.2	−3.9	161.7	167.3	−3.3	139.1	145.6	−4.5
28	T_{13}	119.6	123.1	−2.8	115.3	119.1	−3.2	110.8	115.5	−4.1
29	$\dot{m}_{14}$	186.8	194.2	−3.8	161.7	167.3	−3.3	139.1	145.6	−4.5
30	T_{14}	140.6	145.6	−3.4	135.7	140.4	−3.3	130.6	136.0	−4.0
31	$\dot{m}_{15}$	193.8	203.9	−5.0	167.6	176.1	−4.8	143.9	150.8	−4.6
32	T_{15}	140.6	145.6	−3.4	135.7	140.4	−3.3	130.6	136.0	−4.0
33	$\dot{m}_{16}$	233.5	241.6	−3.4	200.5	206.7	−3.0	170.8	177.4	−3.7
34	T_{16}	170.0	171.0	−0.6	164.2	165.0	−0.5	158.4	159.5	−0.7
35	$\dot{m}_{17}$	233.5	241.6	−3.4	200.5	206.7	−3.0	170.8	177.4	−3.7
36	T_{17}	173.7	174.4	−0.4	167.9	168.3	−0.2	162.1	163.2	−0.7
37	$\dot{m}_{18}$	233.5	241.6	−3.4	200.5	206.7	−3.0	170.8	177.4	−3.7
38	T_{18}	199.0	199.9	−0.5	192.6	193.4	−0.4	185.8	187.8	−1.1
39	$\dot{m}_{19}$	233.5	241.6	−3.4	200.5	206.7	−3.0	170.8	177.4	−3.7
40	T_{19}	236.2	237.2	−0.4	228.6	229.4	−0.3	220.5	222.1	−0.7
41	$\dot{m}_{20}$	19.7	20.4	−3.4	15.9	16.5	−3.6	12.9	13.4	−3.7
42	T_{20}	305.4	309.5	−1.3	301.4	300.6	0.3	291.1	293.3	−0.8
43	p_{20}	3,265	3,302	−1.1	2,816	2,813	0.1	2,409	2,447	−1.6
44	$\dot{m}_{21}$	9.1	9.5	−4.2	7.7	8.0	−3.8	6.1	6.8	−10.3
45	T_{21}	425.5	426.2	−0.2	412.0	411.9	0.0	418.2	417.2	0.2
46	p_{21}	1,588	1,602	−0.9	1,371	1,369	0.1	1,176	1,191	−1.3
47	$\dot{m}_{22} + \dot{m}_{23} + \dot{m}_{47}$	20.6	19.2	7.3	17.6	15.5	13.5	14.5	13.3	9.0
48	T_{22}	335.3	336.0	−0.2	323.4	323.0	0.1	329.4	329.0	0.1
49	p_{22}	815	827	−1.5	705	707	−0.3	607	614	−1.1
50	$\dot{m}_{24}$	7.0	7.9	−11.4	5.9	6.4	−7.8	4.8	5.3	−9.4

(continued)

Table 5.3 (continued)

No.	Parameter from Fig. 3.4	Measurement 4			Measurement 5			Measurement 6		
		Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)	Calculated value	Measured value	ε (%)
51	T_{24}	257.7	257.4	0.1	247.3	247.0	0.1	252.8	252.5	0.1
52	p_{24}	428	435	−1.6	371	375	−1.1	320	327	−2.1
53	$\dot{m}_{25}$	11.5	13.0	−11.5	9.8	11.0	−10.9	8.1	9.3	−12.9
54	T_{25}	200.9	202.3	−0.7	191.6	194.5	−1.5	196.9	200.1	−1.6
55	p_{25}	244	250	−2.4	212	217	−2.3	183	190	−3.7
56	$\dot{m}_{27}$	5.8	6.0	−3.3	4.9	5.1	−3.9	3.9	4.2	−7.1
57	T_{27}	95.0	88.3	7.6	87.4	84.5	3.4	92.2	80.8	14.1
58	p_{27}	69.9	74.0	−5.5	60.6	64.9	−6.6	52.7	59.4	−11.3
59	$\dot{m}_{29} + \dot{m}_{57}$	6.3	6.2	1.6	6.0	5.7	5.3	4.6	4.3	5.8
60	p_{29}	27.8	28.1	−1.1	23.8	24.2	−1.7	21.0	22.3	−5.8
61	T_{w2KQ1}	38.1	38.7	−1.6	33.0	33.3	−0.9	33.8	34.0	−0.6

$$\varepsilon = \frac{x_c - x_m}{x_m} \tag{5.1}$$

where:

x_c calculated value

x_m value obtained from measurement

From the results in the tables it stems that the results of calculations gained with the developed model are compatible with the results of measurements for condensing operation of the power plant. For the majority of parameters the value of the relative error does not exceed 5% and for the electric power, which is the most relevant parameter is lower than 1.5%. The largest differences are found in the values of mass stream of steam from extractions A1–A5. This could be due to the small values of these quantities in comparison to the mass flow of condensate encountered in the overall balance.

Chapter 6
Thermodynamic Analysis of a Combined Heat and Power Unit Using Extractions A2, A3 and Crossoverpipe IP–LP for Heater Supply

Thermodynamic calculations of the operating parameters of a power unit takes into consideration thermal power output in accordance with the chart summarizing the demand for district heat in the total of $\dot{Q}_{\mathrm{cmax}} = 220$ MW (Fig. 6.2), while its output for the purposes of preparing network hot water was assumed at $\dot{Q}_{\mathrm{dhw}} = 15$ MW. The maximum temperature of hot water in the network was assumed at $t_{\mathrm{h\ max}} = 135$°C, while the maximum temperature of return water at $t_{\mathrm{r\ max}} = 70$°C.

The extraction of considerable amounts of steam to feed heat exchangers results in the decrease of the pressures in the extractions. These pressures can be further reduced following the decrease of electrical power output from a unit, which cannot be completely ruled out. The reduction of pressures and, hence, the resulting saturation temperatures lead to the decrease of steam temperature below 70°C in A1 extraction, i.e. below the minimum temperature of network hot water at the output of a section of dedicated to its generation. For this reason A1 extraction is not suitable for application for heat generation. There are additional obstacles associated with steam extraction and its regulation as it does not have a regulation valve that is capable of maintaining a constant pressure level. As a consequence, the use of A1 extraction would require installing a new, completely redesigned LP turbine section. Moreover, saturation temperature in A3 extraction drops below 120°C following the increase of steam extraction in accordance with annual chart of the demand for heat. It is not possible to gain the maximum temperature of water in district heating system of $t_{\mathrm{h\ max}} = 135$°C.

For these reasons the optimum heater structure will have to be sought in a heater structure supplied with steam from extractions A2 and A3 and from crossoverpipejoining IP and LP turbine sections. This chapter is devoted to presentation of a number of alternatives involving thermodynamic calculations of this type of heater supply (Fig. 3.4).

The decrease of the annual electric energy capacity of the power plant $\Delta E_{\mathrm{el,A}}^{\mathrm{El}}$ has a considerable effect on the specific cost of heat production Q_{A} (per unit of energy). However, one can note that as a result of using latest generation turbine

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blading this decrease may not be the case. For the condition $\dot{E}_{\mathrm{ch}} = \mathrm{const}$ this decrease is expressed with the Eq. 2.17 (Chap. 2):

$$\Delta E_{\mathrm{el,A}}^{\mathrm{El}} = \int_0^{\tau A} \varepsilon \dot{Q}_{\mathrm{c}}^{\mathrm{EL}} \mathrm{d}\tau = \varepsilon Q_{\mathrm{A}} \tag{6.1}$$

where:

ε mean weighted annual decrease of electricity production in a power plant resulting from heat generation Q_{A}.

In order to ensure that the modernization of the power plant to combined heat and power is justified from the economic perspective, the following relation has to be fulfilled (for the condition $\dot{E}_{\mathrm{ch}} = \mathrm{const}$) (Eq. 2.19, Chap. 2):

$$\varepsilon \leq \frac{e_h}{e_{\mathrm{el}}} - \frac{(z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_P}{e_{\mathrm{el}} Q_A} \tag{6.2}$$

[if in relation (6.2) there is the equality, the price of heat e_{h} also expresses the specific cost k_{h} of its generation]
where:

e_{el} specific sales price of electric energy. One can note that the electricity produced in a combined process "red energy" has a higher price than "black energy," which adds another rationale for the modernization of a power plant to combined heat and power,

K_{p} cost of electricity needed to power network hot water pumps and auxiliary pumps,

$(z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}}$ depreciation and annual cost of maintenance and overhaul of the facilities following capital expenditureJ_{mod} for adapting a power plant to combined heat and power

The selection of an optimum structure of heaters therefore involves the optimization of the value of the weighted coefficient ε.

The value of ε is relative to the time variable values of coefficients ε_i for the specific extractions. Values of ε_i vary depending on the changing steam mass flows extracted from them in accordance with the qualitative regulation of thermal power output (Chap. 4, Fig. 4.1) and are relative only to the fluctuations of the pressure in extractions. In accordance with Stodola–Flügel turbine passage equation the reduction of steam feed into the following stages of the turbine leads to the increase of coefficients ε_i and consequently, to increase of ε [5]. Consequently, for the case when a power plant consists of several units, which is a rule, a more beneficial alternative involves feeding heaters not from a single turbine but from two of them despite the fact that it would be sufficient to extract the required thermal power from only one of them. This could apply extraction A2 and A3 and the crossoverpipe joining IP and LP turbine sections. It is also necessary to ensure a reserve for the case of emergency shut-down of one of the power units. Such a reserve can take the form of an inexpensive water boiler or a turbogenerator adapted to combined heat and power that is capable of meeting total required heat demand.

Steam bleeding from two turbogenerators could also be imposed by the lack of technical capability of extracting the needed volumes of steam from a single turbine.

For the above reasons the calculations are additionally conducted for the steam extraction from a single and two power units.

Moreover, an assumption was made of a constant chemical fuel mass stream (Sect. 6.2) that is equal to the stream prior to the modernization of the unit to combined heat and power. This stream is $\dot{E}_{ch} = 934.8$ MW (corresponding to maximum continuous capacity $N_{el}^{ST} = 380$ MW). In addition, constant electrical power output (Sect. 6.3) is assumed at a level that is equal to the one prior to the modernization to the combined heat and power, that is, $N_{el}^{ST} = 380$ MW. This operation is made possible by the parameters of the steam boiler, as it has certain efficiency surplus as well as due to additional capacity of the flow system in the turbine.

The operation of the power unit with a constant electrical power output, regardless of the varied thermal power supplied for the purposes of district heating and network hot water in accordance with the annual diagram of its demand guarantees a higher economic efficiency of the combined heat and power (Chap. 2). As a consequence, the profit from the undertaking increases. This increment of the profit is equal to δZ_A the product of the decrease of the specific cost of heat production Δk_h during the combined heat and power and its annual production Q_A (Chap. 2):

$$\delta Z_A = \Delta Z_A^{com} - \Delta Z_A = Q_A \Delta k_h = \Delta E_{el,A}^{El,gross} e_{el}(1 - \varepsilon_{el}) - \Delta E_{ch,A} e_{coal} - \Delta K_{env}^{com} \tag{6.3}$$

where:

e_{el}, e_{coal} specific price of electric energy and coal,

$\Delta E_{el,A}^{El,gross}$ decrease of annual electrical energy output from a power plant as a result of its modernization to combined heat and power without power compensation,

ε_{el} relative coefficient of power station internal load,

ΔK_{env}^{com} increase of charge on pollution of natural environment

The value of δZ_A involves the decrease in the receipts from sales of electricity $\Delta E_{el,A}^{El,gross} e_{el}(1 - \varepsilon_{el})$ when a unit operates without power compensation (Eq. 6.8), minus additional cost of coal used if power compensation occurs $\Delta P_A(\mathrm{NCV}) e_{coal}$ (Eq. 6.5) and increase of the charges on polluting the environment ΔK_{env}^{com} resulting from additional mass stream of coal combusted in the boiler for the purposes of power compensation. When comparing with the profit prior to the modernization, the increase of gross profit ΔZ_A^{com} in Eq. 6.3 after restoring the initial electrical capacity of the power station is equal to:

$$\Delta Z_{\mathrm{A}}^{\mathrm{com}} = Q_{\mathrm{A}} e_{\mathrm{h}} - \Delta K_{\mathrm{A}}^{\mathrm{El,com}} \tag{6.4}$$

where:
e_{h} specific price of heat.

It is gained as a result of combustion of additional volumes of coal ΔP_{A} with the net calorific value NCV (Chap. 2):

$$\Delta E_{\mathrm{ch,A}} = \Delta P_{\mathrm{A}}(\mathrm{NCV}) = \int_0^{\tau_A} \Delta \dot{\mathrm{P}}(\mathrm{NCV}) \mathrm{d}\tau. \tag{6.5}$$

In order to ensure that the combined heat and power is justified from the economic point of view, the following relation has to be fulfilled:

$$\Delta Z_{\mathrm{A}}^{\mathrm{com}} \geq 0. \tag{6.6}$$

The increase of the annual operating cost of the power station $\Delta Z_{\mathrm{A}}^{\mathrm{El,com}} \geq 0$ in Eq. 6.4 is associated with capital expenditure on the modernization of the power plant to the combined heat and power $J_{\mathrm{mod}} = J_{\mathrm{blid}} + J_{\mathrm{switchgear}} + J_{\mathrm{aut}} + J^{\mathrm{mod}} + J_{\mathrm{pip}}^{\mathrm{dist}} + J_{\mathrm{water}}^{\mathrm{inst}} + J_{\mathrm{stat}}^{\mathrm{heat}}$ (investment in buildings, switchgear, systems of automatic control, modernization of turbines for heating steam extraction for combined heat and power, heat distribution line with network pumps, installation maintaining adequate static pressure of water in the heat transmission pipeline and heat exchanger station and an additional cost K_{p} of the power needed to supply pumps necessary to put in motion the pumps in the main heat line and the accessory ones, additional cost of coal $\Delta P_{\mathrm{A}}(\mathrm{NCV}) e_{\mathrm{coal}}$ and increase in environmental charges $\Delta K_{\mathrm{env}}^{\mathrm{com}}$):

$$\Delta K_{\mathrm{A}}^{\mathrm{El,com}} = (z\rho + \delta_{\mathrm{serv}}) J_{\mathrm{mod}} + K_{\mathrm{P}} + \Delta P_{\mathrm{A}}(\mathrm{NCV}) e_{\mathrm{coal}} + \Delta K_{\mathrm{env}}^{\mathrm{com}} \tag{6.7}$$

where:
$z\rho J_{\mathrm{mod}}$ depreciation [1–4],
$\delta_{\mathrm{serv}} J_{\mathrm{mod}}$ cost of maintenance of overhaul of facilities [1–4]

The increase of the gross profit ΔZ_{A} in Eq. 6.3 resulting from the modernization of the power plant to the combined heat and power without restoring is its electrical capacity at the time prior to its modernization is equal to (Chap. 2):

$$\Delta Z_{\mathrm{A}} = Q_{\mathrm{A}} e_{\mathrm{h}} - \Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}} e_{\mathrm{el}} (1 - \varepsilon_{\mathrm{el}}) - \Delta K_{\mathrm{A}}^{\mathrm{El}} \geq 0, \tag{6.8}$$

while the annual increase in the overall cost $\Delta K_{\mathrm{A}}^{\mathrm{El}}$ is equal to:

$$\Delta K_{\mathrm{A}}^{\mathrm{El}} = (z\rho + \delta_{\mathrm{serv}}) J_{\mathrm{mod}} + K_{\mathrm{P}}. \tag{6.9}$$

6.1 Alternatives of Steam Supply to Heaters

The analysis involved the following alternatives of the supply of heatersduring combined heat and power with the reheat cycle involving network hot water heating to the following temperatures: 70°C in XC2 heater using steam extracted from A2, 90°C in XC3 heater using steam extracted from A3 and 135°C in XC4 heater using steam from the crossoverpipe between IP–LP turbine sections as well alternatives with network hot water heating in XC2 and XC3 heaters to 80 and 120°C instead of 70 and 90°C, respectively, in accordance with the annual chart of heat demand. For the same use of the chemical energy the operation of the unit with higher water temperatures results in a higher electrical power output from the power plant. This comes as a consequence of the close-to-zero extraction of heating steam from the crossoverpipe between IP–LP turbine sections.

Figure 6.1 presents the diagram of steam supply to heaters from two per units (Sects. 6.1.2, alternative f and 6.1.3, alternative d).

6.1.1 Heating Steam Extraction from a Single Power Unit

Here are alternatives for supply of heat exchangers with steam extracted from a single power:

1. steam is extracted from extractions A2, A3 and crossoverpipe between IP–LP turbine section and fed to XC2, XC3, XC4 heaters;
    - Table 6.1, item 1.; Table 6.2, item 1., Fig. 6.3.—constant stream of chemical energy of fuel,
    - Table 6.3, item 1.; Table 6.4, item 1., Fig. 6.13.—constant electric power output
2. steam is extracted from extractions A2 and crossoverpipe between IP–LP turbine section and is fed to XC2 and XC4 heaters; XC3 is excluded;
    - Table 6.1, item 2.; Table 6.2, item 1.—constant stream of chemical energy of fuel
    - Table 6.3, item 2.; Table 6.4, item 1.—constant electric power output
3. steam is extracted from extraction A3 and crossoverpipe between IP–LP turbine section and is fed to XC3 and XC4 heaters; XC2 is excluded;
    - Table 6.1, item 3.; Table 6.2, item 2.—constant stream of chemical energy of fuel
    - Table 6.3, item 3.; Table 6.4, item 2.—constant electric power output

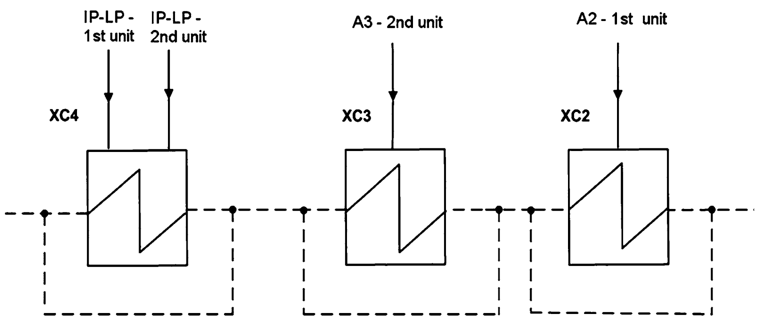

Fig. 6.1 Diagram of steam supply to heaters from two per units (Sects. 6.1.2, alternative f and 6.1.3, alternative d)

4. steam is extracted from crossoverpipe between IP–LP turbine sections and is fed to XC4 heater; XC2 and XC3 heaters are excluded;
   - Table 6.1, item 4.; Table 6.2, item 3.—constant stream of chemical energy of fuel
   - Table 6.3, item 4.; Table 6.4, item 3.—constant electric power output

6.1.2 Heating Steam Extraction from Two Power Units

Here are the alternatives of heaters which supply from two power units:

1. steam is extracted from turbine extractions A2 and A3 in the first unit to supply XC2 and XC3 heaters, steam to feed XC4 heater is extracted from crossover-pipe between IP–LP turbine sections in the second unit;
   - Table 6.1, item 6.; Table 6.2, item 1.—constant flux of chemical energy of fuel
   - Table 6.3, item 6.; Table 6.4, item 1.—constant electric power output
2. steam is extracted from turbine extraction A2 in the first unit to feed XC2 heater, steam to feed XC3 and XC4 heaters is extracted from A3 extraction and crossoverpipe between IP–LP turbine sections in the second unit;
   - Table 6.1, item 7.; Table 6.2, item 1.—constant stream of chemical energy of fuel
   - Table 6.3, item 7.; Table 6.4, item 1.—constant electric power output
3. steam is extracted from turbine extraction A2 and crossoverpipe between IP–LP turbine sections in the first unit to feed XC2 and XC4 heaters, steam to feed XC3 heater is extracted from A3 extraction in the second unit;
   - Table 6.1, item 8.; Table 6.2, item 1.—constant stream of chemical energy of fuel
   - Table 6.3, item 8.; Table 6.4, item 1.—constant electric power output

4. steam is extracted from turbine extraction A2 in the first power unit to feed XC2 heater, steam to supply XC4 heater is extracted from crossoverpipe between IP–LP turbine sections in the second unit; XC3 is excluded;
   - Table 6.1, item 9.; Table 6.2, item 1.—constant stream of chemical energy of fuel
   - Table 6.3, item 9.; Table 6.4, item 1.—constant electric power output
5. steam is extracted from turbine extraction A3 in the first power unit to feed XC3 heater, steam to supply XC4 heater is extracted from crossoverpipe between IP–LP turbine sections in the second unit; XC2 is excluded;
   - Table 6.1, item 10.; Table 6.2, item 2.—constant stream of chemical energy of fuel
   - Table 6.3, item 10.; Table 6.4, item 2.—constant electric power output
6. steam is extracted from turbine extraction A2 and crossoverpipe between IP–LP turbine sections in the first power unit to feed XC2 and XC4 heaters, while the steam to supply XC3 and XC4 heaters is extracted from A3 extraction and crossoverpipe between IP–LP turbine sections in the second unit (in this alternative XC4 heater has a parallel supply with an equal rate from two units (Fig. 6.1));
   - Table 6.1, item 11.; Table 6.2, item 1., Figs. 6.8 and 6.9—constant stream of chemical energy of fuel
   - Table 6.3, item 11.; Table 6.4, item 1., Figs. 6.16 and 6.17—constant electric power output.

6.1.3 Raised Temperatures of Network Hot Water Heating in XC2 and XC3 Heaters for Steam Extraction from a Single and Two Power Units

This section undertakes the analysis of heater supply for the case of increasing the temperature of network hot water behind the heaters. An assumption was made that steam from extraction A2 will enable production of hot water with the temperature of 80°C, while the water at the exhaust of A3 extraction will reach 120°C as a result of larger surface of the heat exchange. The heating steam delivered from XC2, XC3 and XC4 for these temperatures is marked with broken lines in Fig. 6.2. The line that separates heat from XC3 to XC4 heaters is virtually absent as it overlaps with the Y-axis within the range of the operating conditions of XC4 heater, i.e. 173–220 MW (this section is enlarged). The amount of generated heat Q_{XC4} is inconsiderable since in these circumstances it is equal to 2 per mille of the total output Q_{w} in the heating season (Tables 6.1 and 6.3, item 5.).

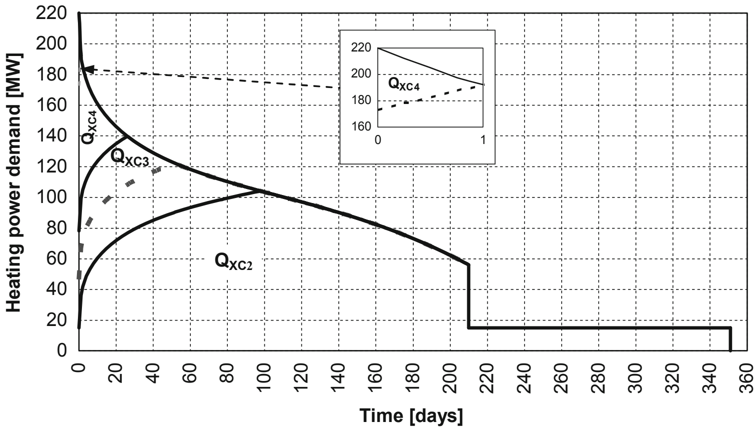

Fig. 6.2 Annual scheduled demand for heating power (*broken lines* indicate the shares of XC2, XC3 and XC4 heaters in satisfying the demand for heating power after increase of the temperatures of hot water at the output of XC2 and XC3 heaters to 80 and 120°C, respectively) calculated with the time step of $\Delta\tau = 24$ h

As a result of comparing the share of the particular heaters in satisfying heating demand one can note a considerable increase in the heat production in XC2 heater and decrease in XC3 and XC4 heaters, which will result in decrease in the demand for chemical energy of the fuel. If the temperature of network hot water were to be reduced from $t_{h\ max} = 135$ to 120°C for instance, it would be possible to abandon XC4 heater totally and the thermodynamic efficiency of the combined process would increase beside the savings in terms of the use of primary fuels.

The analysis invovled the following alternatives of supplying heat exchangers:

1. Combined heat and power with steam extraction from a single power unit:

(a) steam is extracted from extractions A2, A3 and crossoverpipe between IP–LP turbine section and fed to XC2, XC3 and XC4 heaters;

- Table 6.1, item 5.; Table 6.2, item 1., Fig. 6.10—constant stream of chemical energy of fuel
- Table 6.3, item 5.; Table 6.4, item 1., Fig. 6.18—constant electric power output

2. Combined heat and power with heating steam extraction from two power units:

(a) steam is extracted from turbine extractions A2 and A3 in the first power unit to feed XC2 and XC3 heaters; XC4 heater is supplied from crossoverpipe between IP–LP turbine section in the second unit

  - Table 6.1, item 12.; Table 6.2, item 1.—constant stream of chemical energy of fuel
  - Table 6.3, item 12.; Table 6.4, item 1.—constant electric power output

(b) steam is extracted from extraction A2 in the first power unit to feed XC2 heater; XC3 and XC4 heaters are fed from A3 turbine extraction and crossoverpipe between IP–LP turbine section in the second unit

- Table 6.1, item 13.; Table 6.2, item 1.—constant stream of chemical energy of fuel
- Table 6.3, item 13.; Table 6.4, item 1.—constant electric power output

(c) steam is extracted from extractions A2 and crossoverpipe between IP–LP turbine section in the first power unit to feed XC2 and XC4 heaters; XC3 heater is supplied from turbine extraction A3 in the second unit;

- Table 6.1, item 14.; Table 6.2, item 1.—constant stream of chemical energy of fuel
- Table 6.3, item 14.; Table 6.4, item 1.—constant electric power output

(d) steam is extracted from extractions A2 and crossoverpipe between IP–LP turbine section in the first power unit to feed XC2 and XC4 heaters; XC3 and XC4 heaters are supplied from turbine extraction A3 and crossoverpipe between IP–LP turbine section in the second unit (in this alternative XC4 heater is supplied from two units in a parallel system (Fig. 6.1));

- Table 6.1, item 15.; Table 6.2, item 1., Figs. 6.11 and 6.12—constant stream of chemical energy of fuel
- Table 6.3, item 15.; Table 6.4, item 1., Figs. 6.19 and 6.20—constant electric power output.

6.2 Operation with a Constant Stream of Chemical Energy of Fuel Combustion in Boiler

This section is devoted to the presentation of selected results of calculations regarding the operation of a power unit (for the case of two units applied in combined heat and power) for the alternatives of the combined heat and power presented in Sect. 6.1. An additional assumption is made that the unit operates without electrical capacity compensation. As a result, such decrease of the electrical power output from a unit is variable in time depending on the changing demand for district heating and hot water.

The calculations assume a constant value of chemical mass stream of fuel combustion in the boiler (for the case of two unit operation and two boiler system involvement) that is equal to the electrical power output from the unit prior to its modernization to the combined heat and power (Tables 6.1 and 6.2).

The calculations were also conducted for the alternative of supply of heaters from one to two power units.

To sum up: the presented results of thermodynamic calculation illustrate a combined heat and power for a constant mass stream of chemical energy of boiler

Table 6.1 Electrical energy output $E^{El}_{el,w}$ for analyzed alternatives of heater supply in the peak (winter) season; $\dot{E}_{ch} = 934.8$ MW = const, $\dot{Q}_{c\ max} = 220$ MW, $\dot{Q}_{dhw} = 15$ MW, $\tau_w = 211$ days

Item	Alternative	$E_{ch,w}$ (MWh)	Q_w (MWh)	Q_{XC2} (MWh)	Q_{XC3} (MWh)	Q_{XC4} (MWh)	$E^{El}_{el,w}$ (MWh)
Condensing unit operation							
		4,733,802	0	0	0	0	1,937,307
Combined heat and power, single unit—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C							
1	XC2/3/4	4,733,802	531,581	423,308	84,784	23,489	1,857,708
2	XC2/4	4,733,802	531,581	423,308	0	108,273	1,855,412
3	XC3/4	4,733,802	531,581	0	508,092	23,489	1,824,500
4	XC4	4,733,802	531,581	0	0	531,581	1,814,606
Combined heat and power, two units—A2 ≤ 80°C, A3 ≤ 120°C, IP–LP ≤ 135°C							
5	XC2/3/4	4,733,802	531,581	479,114	51,328	1,135	1,862,700
Combined heat and power, two units—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C							
6	XC2/3 (1st unit)	4,733,802	508,092	423,308	84,784	0	1,863,046
	XC4 (2nd unit)	4,733,802	23,489	0	0	23,489	1,931,748
	Total	9,467,604	531,581	423,308	84,784	23,489	3,794,794
7	XC2 (1st unit)	4,733,802	423,308	423,308	0	0	1,880,596
	XC3/4 (2nd unit)	4,733,802	108,273	0	84,784	23,489	1,913,367
	Total	9,467,604	531,581	423,308	84,784	23,489	3,793,963
8	XC2/4 (1st unit)	4,733,802	446,797	423,308	0	23,489	1,875,049
	XC3 (2nd unit)	4,733,802	84,784	0	84,784	0	1,918,780
	Total	9,467,604	531,581	423,308	84,784	23,489	3,793,829
9	XC2 (1st unit)	4,733,802	423,308	423,308	0	0	1,880,594
	XC4 (2nd unit)	4,733,802	108,273	0	0	108,273	1,911,877
	Total	9,467,604	531,581	423,308	0	108,273	3,792,471
10	XC3 (1st unit)	4,733,802	508,092	0	508,092	0	1,829,700
	XC4 (2nd unit)	4,733,802	23,489	0	0	23,489	1,931,748
	Total	9,467,604	531,581	0	508,092	23,489	3,761,448

(continued)

Table 6.1 (continued)

Item	Alternative	$E_{ch,w}$ (MWh)	Q_w (MWh)	Q_{XC2} (MWh)	Q_{XC3} (MWh)	Q_{XC4} (MWh)	$E^{El}_{el,w}$ (MWh)
11	XC2/4 (1st unit)	4,733,802	435,052	423,308	0	11,744	1,877,788
	XC3/4 (2nd unit)	4,733,802	96,528	0	84,784	11,744	1,916,028
	Total	9,467,604	531,580	423,308	84,784	23,488	3,793,816
Combined heat and power, two units—A2 ≤ 80°C, A3 ≤ 120°C, IP–LP ≤ 135°C							
12	XC2/3 (1st unit)	4,733,802	530,445	479,117	51,328	0	1,862,953
	XC4 (2nd unit)	4,733,802	1,135	0	0	1,135	1,937,039
	Total	9,467,604	531,580	479,117	51,328	1,135	3,799,992
13	XC2 (1st unit)	4,733,802	479,117	479,117	0	0	1,873,398
	XC3/4 (2nd unit)	4,733,802	52,463	0	51,328	1,135	1,925,911
	Total	9,467,604	531,580	479,117	51,328	1,135	3,799,309
14	XC2/4 (1st unit)	4,733,802	480,252	479,117	0	1,135	1,873,130
	XC3 (2nd unit)	4,733,802	51,328	0	51,328	0	1,926,138
	Total	9,467,604	531,580	479,117	51,328	1,135	3,799,268
15	XC2/4 (1st unit)	4,733,802	479,685	479,117	0	568	1,873,263
	XC3/4 (2nd unit)	4,733,802	51,896	0	51,328	568	1,926,011
	Total	9,467,604	531,580	479,117	51,328	1,135	3,799,274

Table 6.2 Electrical energy output $E^{\text{El}}_{\text{el,w}}$ for analyzed alternatives of heater supply in the off-peak (summer) season; $\dot{E}_{\text{ch}} = 934.8$ MW = const, $\dot{Q}_{\text{dhw}} = 15$ MW, $\tau_{\text{s}} = 140$ days

Item	Alternative	$E_{\text{ch,s}}$ (MWh)	Q_{s} (MWh)	Q_{XC2} (MWh)	Q_{XC3} (MWh)	Q_{XC4} (MWh)	$E^{\text{El}}_{\text{el,s}}$ (MWh)
Condensing unit operation							
		3,140,900	0	0	0	0	1,276,799
Combined heat and power							
1	XC2	3,140,900	50,400	50,400	0	0	1,270,097
2	XC3	3,140,900	50,400	0	50,400	0	1,265,971
3	XC4	3,140,900	50,400	0	0	50,400	1,264,995

fuel combustion (for the case of operation with two units and two boilers) that is equal to this stream prior to modernization of the units to combined heat and power for time variable electrical power output from the unit as a result of operation without electrical capacity compensation.

6.2.1 Results of Calculations

Table 6.1 contains a summary of the results of gross electrical energy output $E^{\text{El}}_{\text{el,w}}$, heat Q_{w} and consumption of chemical energy of the coal $E_{\text{ch,w}}$ in the peak season(winter), while Table 6.2 offers the summary of $E^{\text{El}}_{\text{el,s}}$, Q_s and $E_{\text{ch,s}}$ in the off-peak season(summer). The annual values of these variables are equal to the totals of the peak and off-peak values, i.e.:

$$E_{\text{ch,A}} = E_{\text{ch,w}} + E_{\text{ch,s}}, \tag{6.10}$$

$$E^{\text{El}}_{\text{el,A}} = E^{\text{El}}_{\text{el,w}} + E^{\text{El}}_{\text{el,s}}, \tag{6.11}$$

$$Q_{\text{A}} = Q_{\text{w}} + Q_{\text{s}}. \tag{6.12}$$

Heat generation in winter Q_{w} is composed of total heat for the purposes of district heating and district hot water, while in the summer it corresponds the total of heat for the purposes of network hot water only.

The maximum thermal power output from the power plant was assumed in accordance with the annual chart summarizing the demand for district heating at $\dot{Q}_{\text{c max}} = 220$ MW, the values of maximum temperatures of network hot water and return water at $t_{\text{h max}} = 135°\text{C}$ and $t_{\text{r max}} = 70°\text{C}$, respectively. In the alternative with the mass stream of chemical energy of boiler fuel combustion of $\dot{E}_{\text{ch}} = 934.8$ MW the value of the heating power demand for preparation of district hot water was assumed at $\dot{Q}_{\text{dhw}} = 15$ MW as a constant both in the peak and off-peak season. Additionally, an assumption was made that the peak season for the flux of $\dot{E}_{\text{ch}} = 934.8$ MW lasts $\tau_{\text{w}} = 211$ days, while the off-peak season $\tau_{\text{s}} = 140$ days.

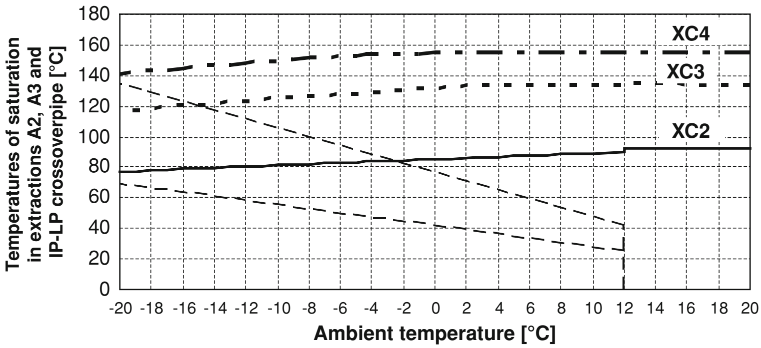

Fig. 6.3 Temperatures of saturation of bleed steam in the function of ambient temperature; A2 extraction is used to feed XC2 heater, A3 extraction is used to feed to XC3, XC4 is supplied from IP–LP crossoverpipe (broken lines mark the temperatures of network hot water and return water)

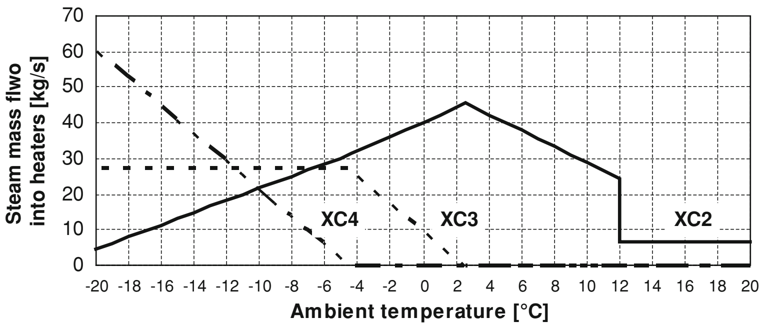

Fig. 6.4 Heating steam mass stream into XC2, XC3 and XC4 heaters in the function of ambient temperature (Sect. 6.1.1, alternative a)

6.2.1.1 Heating Steam Extraction from a Single Unit

Figure 6.3 presents the curve of temperature of steam saturation in the function of ambient temperature for steam feed to three heaters and chemical energy stream $\dot{E}_{ch} = 934.8$ MW (Sect. 6.1.1, alternative a).

The demand for thermal poweris relative to the ambient temperaturein accordance with the annual qualitative regulation chart. Following the fluctuations of temperature the steam stream of mass into the heaters also changes with time. Figures 6.4 and 6.5 present steam bleed from turbine extractions to heaters in the function of the ambient temperature for two alternatives of supply for a constant chemical energy stream of $\dot{E}_{ch} = 934.8$ MW. The reason for the step-wise increase of the steam feed into XC2 heater (Fig. 6.4) and XC4 heater (Fig. 6.5) for

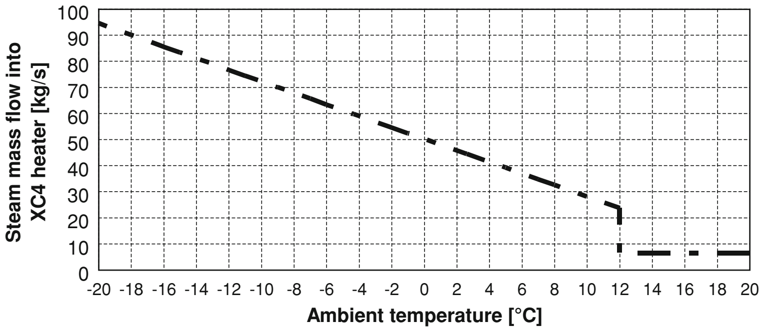

Fig. 6.5 Heating stream mass stream into XC4 heater in the function of ambient temperature (Sect. 6.1.1, alternative d)

the ambient temperature 12°C is associated with an increase in the demand for the steam resulting from the beginning of the heating season.

Following an increase of the thermal power as a result of decreasing ambient temperatures the effect of the application of the enthalpy of the steam is reduced and electrical power output decreases. Concurrently, the production of electricity and heat generation increases. Figure 6.6 presents the dependence of electrical power output and Fig. 6.7 presents the relation of the unit capacity and ambient temperature for two alternatives of heater supply (XC2/3/4 and XC4). For the purposes of comparison the electrical power output and efficiency of the power unit is presented for the case of operation in a conventional cycle (i.e. without steam extraction to feed heaters).

In Fig. 6.6 one can note the slight increase of the electrical power output during the condensing operation. This results from an increase in the efficiency of the power unit (Fig. 6.7) as a result of decrease in the pressure in the condenser, which in turn comes as a consequence of lower cooling water temperature (Fig. 3.17). As one can note the decrease of electricity production is greater for the alternative in which the steam is extracted only from the IP–LP crossoverpipe. The greatest difference between the decrease of electricity production resulting from steam extraction for the purposes of district heating is noted for the ambient temperature of 2.6°C and it is equal to around 11 MW. For this temperature the maximum amount of steam is extracted from A2 extraction to feed XC2 heater. Along with the decrease of temperatures steam stream of mass into XC2 falls and the feed rate into XC3 increases and subsequently into XC4. This results in a decrease of electricity production to only a value of around 2 MW for the ambient temperature of −20°C. The maximum decrease of electricity production is equal to maximum 46–48 MW depending on the alternative and the annual decrease is equal to 10–16 MW.

The decrease of electricity production is additionally followed by increase in power unit capacity defined as:

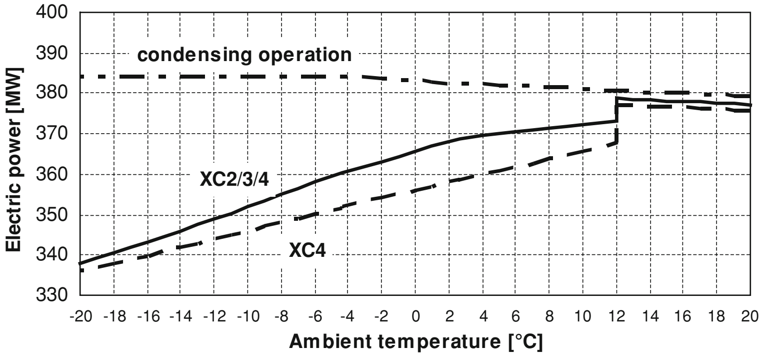

Fig. 6.6 Electrical power output in the function of ambient temperatures for various alternatives of heater supply (Sect. 6.1.1, alternatives a and d)

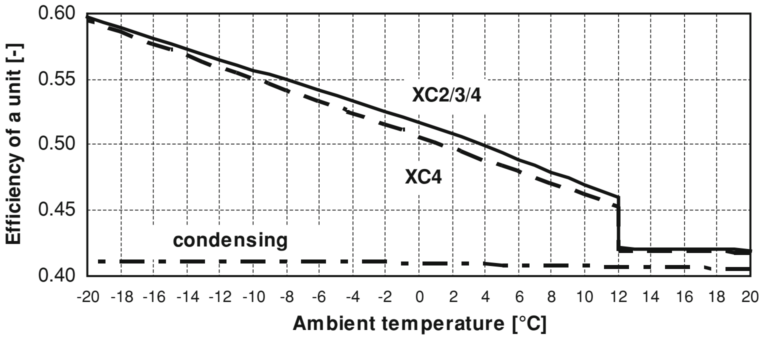

Fig. 6.7 Efficiency of a unit in the function of ambient temperature for various alternatives of heater supply (Sect. 6.1.1, alternatives a and d)

$$\eta_{\mathrm{Eec}} = \frac{N_{\mathrm{el}}^{\mathrm{ST}} + \dot{Q}_C}{\dot{E}_{\mathrm{ch}}} \tag{6.13}$$

where:

$\eta_{\mathrm{E\ ec}}$ efficiency of the unit adapted to combined heat and power
$N_{\mathrm{el}}^{\mathrm{ST}}$ electrical power output,
$\dot{Q}_C$ heat power of the unit adapted to cogeneration,
$\dot{E}_{\mathrm{ch}}$ stream of chemical energy of the fuel,

which for the ambient temperature of -20°C is equal to around 60–19% higher than the efficiency of the power plant prior to its modernization. The average annual increase in the efficiency is equal to 5.3–6.2%, depending on the alternative of heaters supply.

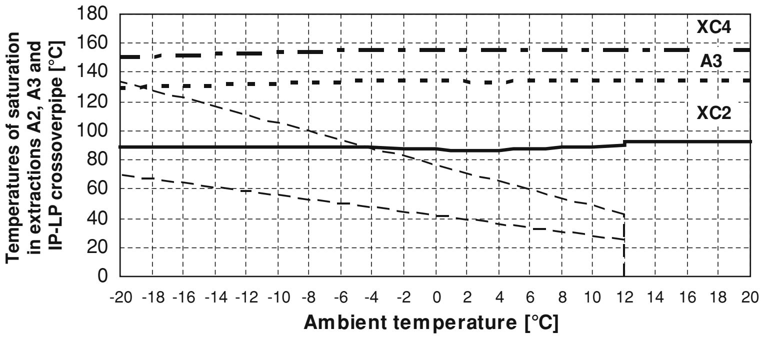

Fig. 6.8 Temperatures of bleed steam saturation in the function of ambient temperatures; XC2 and XC4 heaters are fed with steam from A2 extraction and IP–LP crossoverpipe in the first unit; XC4 heater is fed in a parallel system from IP–LP crossoverpipe in the second unit (*broken lines* mark the temperature of network hot water and return water)

6.2.1.2 Heating Steam Extraction from Two Power Units

Figures 6.8 and 6.9 present the curves of temperatures of bleed steam saturation in the function of ambient temperatures for the alternative of heater steam extraction from two power unit for the chemical energy mass stream of $\dot{E}_{ch} = 934.8$ MW (Sect. 6.1.2, alternative f).

6.2.1.3 Raised Temperatures of Network Hot Water Preheating in XC2 and XC3 Heaters for the Supply of Heaters from a Single and Two Units

Figure 6.10 presents the curves of temperatures of bleed steam saturation in the function of ambient temperaturesfor the alternative of heater steam supply from a single power unit for the chemical energy flux of $\dot{E}_{ch} = 934.8$ MW (Sect. 6.1.3, alternative 1a).

Figures 6.11 and 6.12 present the curves of temperatures of bleed steam saturation in the function of ambient temperatures for the alternative of heater steam supply from two power units for the chemical energy flux of $\dot{E}_{ch} = 934.8$ MW (Sect. 6.1.3, alternative 2d).

6.3 Operation with a Constant Electric Capacity of a Unit

Figures 6.13, 6.14, 6.15, 6.16, 6.17, 6.18, 6.19 and 6.20 present the results of thermodynamic calculations regarding the efficiency of a unit, chemical energy stream of the fuel and temperature of steam saturation in extractions A2, A3 and

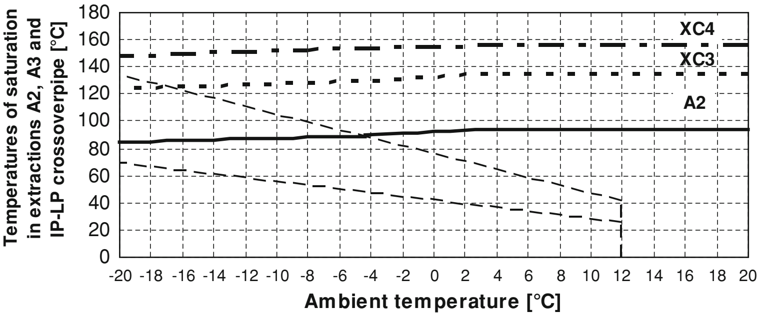

Fig. 6.9 Temperatures of steam saturation in the function of ambient temperatures; XC3 and XC4 heaters are fed with steam from A3 extraction and IP–LP crossoverpipe in the second unit; XC4 heater is fed in a parallel system from IP–LP crossoverpipe in the first unit (*broken lines* mark the temperature of network hot water and return water)

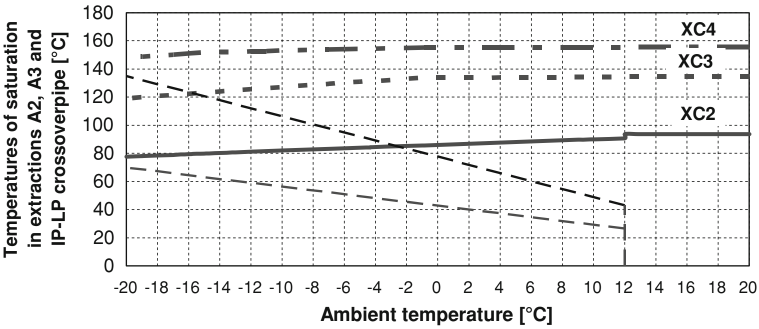

Fig. 6.10 Temperatures of steam saturation in the function of ambient temperature; XC2 heater is fed with steam from A2, XC3 from extraction A3 and XC 4 heater is fed from IP–LP crossoverpipe unit (*broken lines* mark the temperature of network hot water and return water)

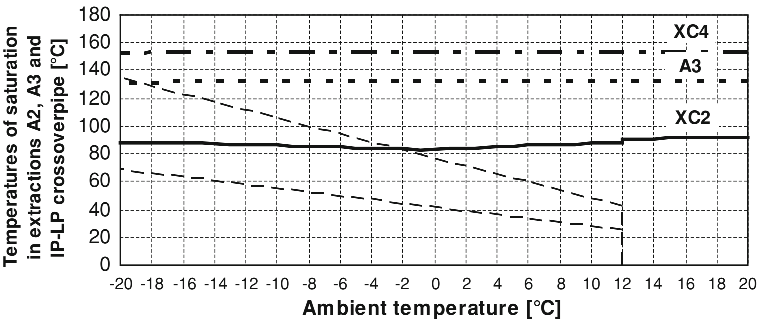

Fig. 6.11 Temperatures of steam saturation in the function of ambient temperatures; XC2 and XC4 heaters are fed with steam from A2 extraction and IP–LP crossoverpipe in the second unit; XC 4 heater is fed in a parallel system from IP–LP crossoverpipe in the first unit (*broken lines* mark the temperature of network hot water and return water)

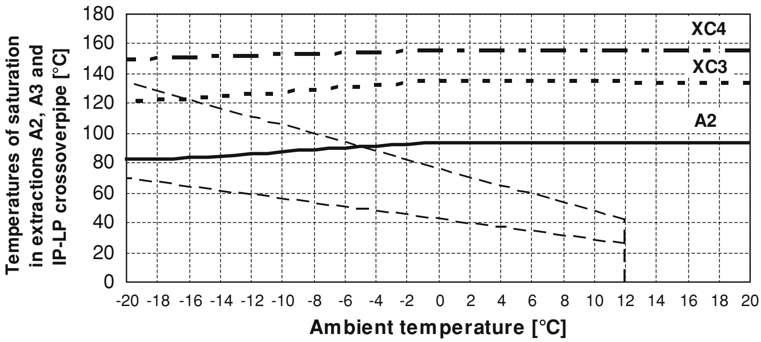

Fig. 6.12 Temperatures of steam saturation in the function of ambient temperature; XC3 and XC4 heaters are fed with steam from A3 extraction and IP–LP crossoverpipe in the second unit; XC 4 heater is fed in a parallel system from IP–LP crossoverpipe in the first unit (*broken lines* mark the temperature of network hot water and return water)

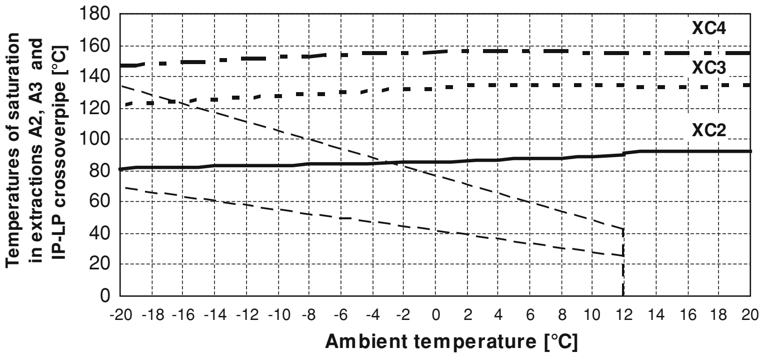

Fig. 6.13 Temperatures of steam saturation in the function of ambient temperatures; XC2 heater is supplied with steam from A2 extraction, XC3 from extraction A3 and XC4 from IP–LP crossoverpipe (*broken lines* mark the temperature of network hot water and return water)

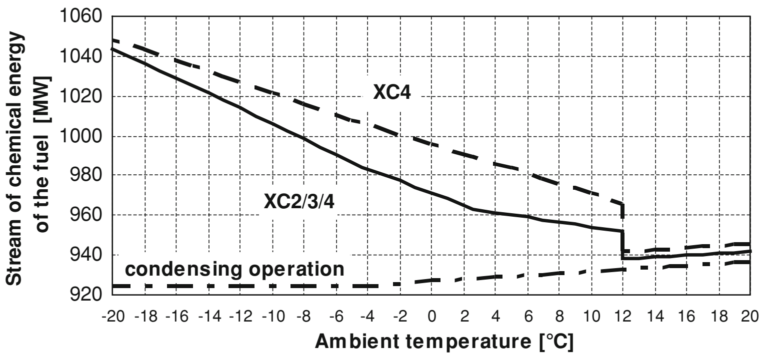

Fig. 6.14 Chemical energy flux in the function of ambient temperature for various alternatives of heater supply (Sect. 6.1.1, alternatives a and d)

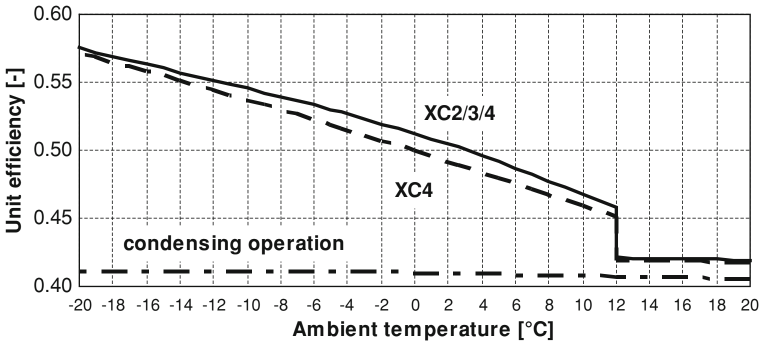

Fig. 6.15 Efficiency of a power unit in the function of ambient temperature for various alternatives of heater supply (Sect. 6.1.1, alternative a and d)

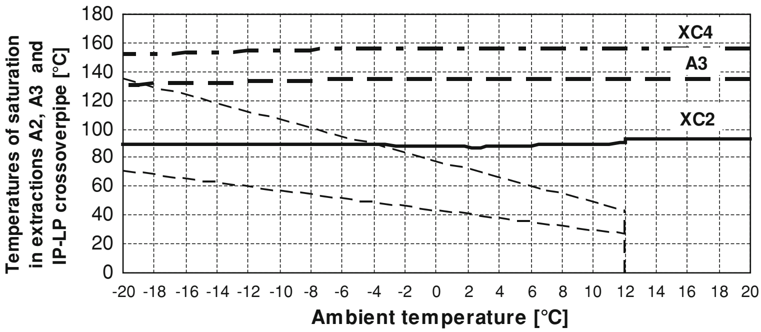

Fig. 6.16 Temperatures of extracted steam saturation in the function of ambient temperatures; XC2 and XC4 heaters are fed with steam from A2 extraction and IP–LP crossoverpipe in the first unit; XC4 heater is fed in a parallel system from IP–LP crossoverpipe in the second unit (*broken lines* mark the temperature of network hot water and return water)

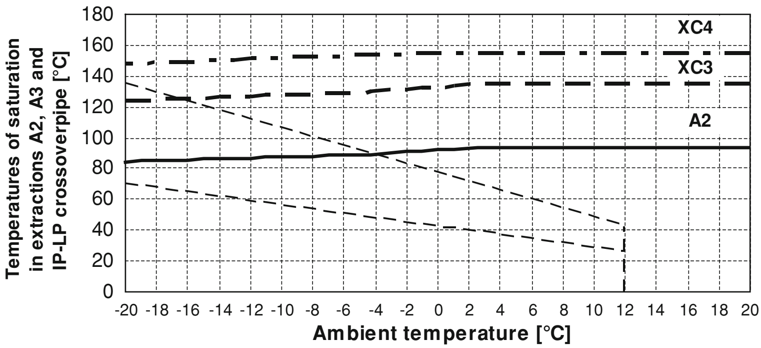

Fig. 6.17 Temperatures of extracted steam saturation in the function of ambient temperatures; XC3 and XC4 heaters are fed with steam from A3 extraction and IP–LP crossoverpipe in the second unit; XC4 heater is fed in a parallel system from IP–LP crossoverpipe in the first unit (*broken lines* mark the temperature of network hot water and return water)

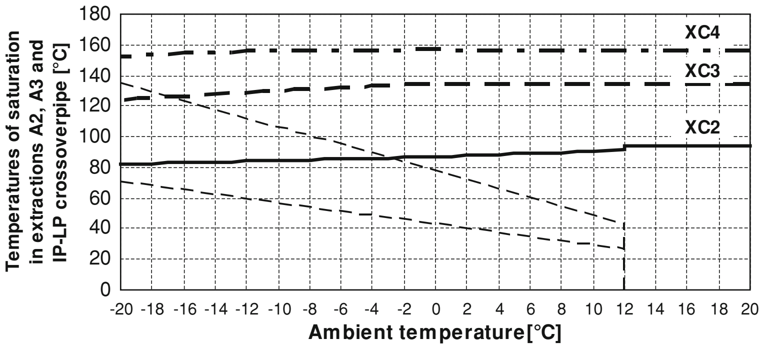

Fig. 6.18 Temperatures of saturation of extracted steam in the function of ambient temperature; XC2 heater is fed with steam from A2 extraction, XC3 from extraction A3 and XC4 from IP–LP crossoverpipe (*broken lines* mark the temperature of network hot water and return water)

IP–LP crossoverpipe for the alternatives of heater supply in Sect. 6.1 during a single and two unit operation of a power plant with a constant electric power output. The application of the combined heat and power under the assumption of maintaining a constant electric capacity imposes a need to increase steam production in a boiler. As a result, the pressure drops in the extractions (according to Eq. 4.1) as well as the decrease of saturation temperatures as a result of bleeding considerable amounts of steam from them are smaller than for the case of operation without compensation of electric capacity. As a result, it is possible to ensure higher temperatures of network hot water at the output of the heaters.

6.3.1 Results of Calculations

Calculations were conducted for value of electric capacity maximal of 380 MW. The maximum of thermal power output was taken in accordance with the annual chart of the demand for district heating, i.e. $\dot{Q}_{c\,max} = 220$ MW, values of maximum temperatures of network hot water and return water, i.e. $t_{h\ max} = 135°C$, $t_{r\ max} = 70°C$. A constant value of the demand for thermal power for the purposes of network hot water was assumed at $\dot{Q}_{dhw} = 15$ MW both in the peak and off-peak seasons. In addition, it was assumed that the peak season lasts for $\tau_w = 211$ days, and the off-peak season lasts for $\tau_s = 140$ days.

Tables 6.3 and 6.4 summarize the results of calculations for the peak and off-peak seasons, stream of chemical energy of coal $E_{ch,w}$, $E_{ch,s}$, gross electrical energy output $E_{el,w}$ $E_{el,s}$ and heat generation Q_w, Q_s during the combined heat and power in the power plant with constant electric capacity of $N_{el}^{ST} = 380$ MW for all analyzed alternatives of heater supply.

The annual output and production of these variables are equal to the sums of the values for the peak and off-peak seasons (Eqs. 6.10–6.12, Sect. 6.2.1).

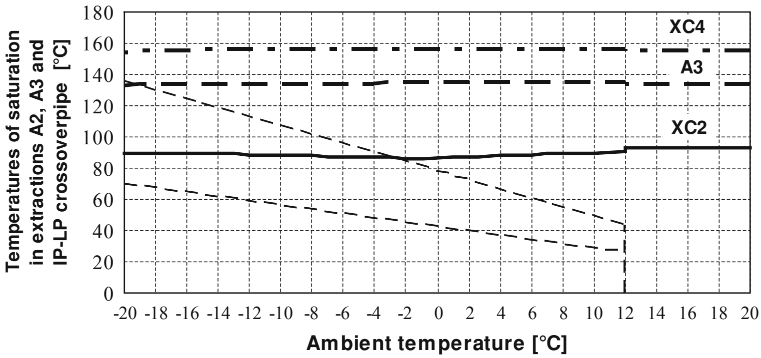

Fig. 6.19 Temperatures of steam saturation in the function of ambient temperatures; XC2 and XC4 heaters are fed with steam from A2 extraction and IP–LP crossoverpipe in the second unit; XC4 heater is fed in a parallel system from IP–LP crossoverpipe in the first unit (*broken lines* mark the temperature of network hot water and return water)

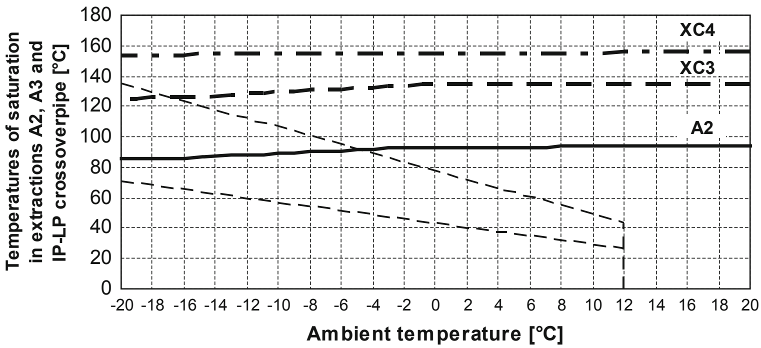

Fig. 6.20 Temperatures of steam saturation in the function of ambient temperatures; XC3 and XC4 heaters are fed with steam from A3 extraction and IP–LP crossoverpipe in the second unit; XC4 heater is fed in a parallel system from IP–LP crossoverpipe in the first unit (*broken lines* mark the temperature of network hot water and return water)

6.3.1.1 Heating Steam Extraction from a Single Unit

Figure 6.13 presents steam saturation temperatures in extractions A2, A3 and crossoverpipe joining IP and LP turbine sections for steam feed to heaters in a power plant adapted to combined heat and power (Sect. 6.1.1, alternative a) for electric capacity of $N_{el}^{ST} = 380$ MW.

Figure 6.14 presents the use of chemical energy of the fuel in the function of ambient temperature for steam supply to three heaters (Sect. 6.1.1, alternative a) and a single one (Sect. 6.1.1 alternative a). Figure 6.15 presents the relation between the efficiency of a power unit and ambient temperature for the same conditions.

Table 6.3 Use of chemical energy of coal $E_{ch,w}$ for the analyzed alternatives of heaters supply in the peak season (winter); $N_{el}^{ST} = 380$ MW = const, $\dot{Q}_{c\ max} = 220$ MW, $\dot{Q}_{dhw} = 15$ MW, $\tau_w = 211$ days

Item	Alternative	$E_{ch,w}$ (MWh)	Q_w (MWh)	Q_{XC2} (MWh)	Q_{XC3} (MWh)	Q_{XC4} (MWh)	$E_{el,w}^{El}$ (MWh)
Condensing operation of a unit							
		4,700,729	0	0	0	0	1,924,320
Single unit operation—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C							
1	XC2/3/4	4,902,448	531,581	423,308	84,784	23,489	1,924,320
2	XC2/4	4,908,304	531,581	423,308	0	108,273	1,924,320
3	XC3/4	4,988,169	531,581	0	508,092	23,489	1,924,320
4	XC4	5,013,378	531,581	0	0	531,581	1,924,320
Single unit operation—A2 ≤ 80°C, A3 ≤ 120°C, IP–LP ≤ 135°C							
5	XC2/3/4	4,889,726	531,581	479,114	51,328	1,135	1,924,320
Two unit operation—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C							
6	XC2/3 (1st nit)	4,888,796	508,092	423,308	84,784	0	1,924,320
	XC4 (2nd unit)	4,714,817	23,489	0	0	23,489	1,924,320
	Total	9,603,613	531,581	423,308	84,784	23,489	3,848,640
7	XC2 (1st unit)	4,844,442	423,308	423,308	0	0	1,924,320
	XC3/4 (2nd nit)	4,761,457	108,273	0	84,784	23,489	1,924,320
	Total	9,605,899	531,581	423,308	84,784	23,489	3,848,640
8	XC2/4 (1st unit)	4,858,445	446,797	423,308	0	23,489	1,924,320
	XC3 (2nd unit)	4,747,684	84,784	0	84,784	0	1,924,320
	Total	9,606,129	531,581	423,308	84,784	23,489	3,848,640
9	XC2 (1st unit)	4,844,440	423,308	423,308	0	0	1,924,320
	XC4 (2nd unit)	4,765,254	108,273	0	0	108,273	1,924,320
	Total	9,609,694	531,581	423,308	0	108,273	3,848,640
10	XC3 (1st unit)	4,974,710	508,092	0	508,092	0	1,924,320
	XC4 (2nd unit)	4,714,817	23,489	0	0	23,489	1,924,320
	Total	9,689,527	531,581	0	508,092	23,489	3,848,640

(continued)

Table 6.3 (continued)

Item	Alternative	$E_{ch,w}$ (MWh)	Q_w (MWh)	Q_{XC2} (MWh)	Q_{XC3} (MWh)	Q_{XC4} (MWh)	$E^{El}_{el,w}$ (MWh)
11	XC2/4 (1st unit)	4,851,520	435,052	423,308	0	11,744	1,924,320
	XC3/4 (2nd unit)	4,754,651	96,528	0	84,784	11,744	1,924,320
	Total	9,606,171	531,580	423,308	84,784	23,488	3,848,640
Two unit operation—A2 $\leq$ 80°C, A3 $\leq$ 120°C, IP–LP $\leq$ 135°C							
12	XC2/3 (1st unit)	4,889,090	530,445	479,117	51,328	0	1,924,320
	XC4 (2nd unit)	4,701,406	1,135	0	0	1,135	1,924,320
	Total	9,590,496	531,580	479,117	51,328	1,135	3,848,640
13	XC2 (1st unit)	4,862,564	479,117	479,117	0	0	1,924,320
	XC3/4 (2nd unit)	4,729,614	52,463	0	51,328	1,135	1,924 320
	Total	9,592,178	531,580	479,117	51,328	1,135	3,848 640
14	XC2/4 (1st unit)	4,863,241	480,252	479,117	0	1,135	1,924 320
	XC3 (2nd unit)	4,729,002	51,328	0	51,328	0	1,924 320
	Total	9,592,243	531,580	479,117	51,328	1,135	3,848 640
15	XC2/4 (1st unit)	4,862,904	480,252	479,117	0	568	1,924 320
	XC3/4 (2nd unit)	4,729,319	51,896	0	51,328	568	1,924 320
	Total	9,592,223	531,580	479,117	51,328	1,135	3,848 640

Table 6.4 Use of chemical energy of coal $E_{ch,s}$ for the analyzed alternatives of heaters supply in the off-peak season (summer); $N_{el}^{ST} = 380$ MW, $\dot{Q}_{dhw} = 15$ MW, $\tau_s = 140$ days

Item	Alternative	$E_{ch,s}$ (MWh)	Q_s (MWh)	Q_{XC2} (MWh)	Q_{XC3} (MWh)	Q_{XC4} (MWh)	$E_{el,s}^{El}$ (MWh)
Condensing operation of a unit							
		3,140,964	0	0	0	0	1,276,800
Combined heat and power							
1	XC2	3,158,177	50,400	50,400	0	0	1,276,800
2	XC3	3,168,774	50,400	0	50,400	0	1,276,800
3	XC4	3,171,282	50,400	0	0	50,400	1,276,800

As one can note in Fig. 6.14, the operation of a unit with electrical power compensation results in the increase of chemical energy use for the case of all alternatives of heater supply. Concurrently, following an increase of steam feed into heaters the total efficiency of the power plant increases as well (Fig. 6.15), which for maximum steam extractionfor ambient temperatureof −20°C is equal to over 57%, which is 16% higher than the efficiency of the condensing power plant. The mean annual increase of efficiency is equal to 5.3–6.0% depending on the alternative.

6.3.1.2 Heating Steam Extraction from Two Power Units

Figures 6.16 and 6.17 present steam saturation temperatures in extractions A2, A3 and crossoverpipe joining IP and LP turbine sections for the feeding of heaters from two power units in a power plant adapted to combined heat and power (Sect. 6.1.2, alternative f) for electric capacity of $N_{el}^{ST} = 380$ MW.

6.3.1.3 Raised Temperatures of Network Hot Water at the Exhaust of XC2 and XC3 Heaters for Supply of Heaters from a Single and Two Power Units

Figure 6.18 presents steam saturation temperatures in extractions A2 and A3 and crossoverpipe joining IP and LP turbine sections of a power plant adapted to combined heat and power while the temperatures of network hot water are raised at the output of XC2 and XC3 heaters for the case of steam bleed from single unit (Sect. 6.1.3, alternative 1a) for electric capacity of $N_{el}^{ST} = 380$ MW.

Figures 6.19 and 6.20 present steam saturation temperatures in extractions A2 and A3 and crossoverpipe joining IP and LP turbine sections of a power plant adapted to combined heat and power while the temperatures of network hot water are raised at the output of XC2 and XC3 heaters for the case of steam bleed from a single unit (Sect. 6.1.3, alternative 2d) for electric capacity of $N_{el}^{ST} = 380$ MW.

As it has already been noted at the beginning of Sect. 6.3, the drop of the temperature of steam saturation in turbine extractions during a combined heat and power with a constant electrical capacity is smaller than for the case of its operation without power compensation(Figs. 6.3, 6.9, 6.10, 6.11, 6.12).

6.4 Summary and Conclusions

1. On the basis of the analysis of thermodynamic calculations of presented alternatives of steam supply to heaters XC2, XC3 and XC4 from extractions A2 and A3 and crossoverpipe joining IP–LP turbine sections one can conclude that the maximum required temperature of network hot water $t_{\mathrm{h\ max}} = 135°\mathrm{C}$ is obtained in each one of them. It is also possible to have an output of hot water temperature with the temperature of 70°C.
2. For the case of steam feeding to three heaters XC2, XC3 and XC4 it is not possible to conclude which of the alternatives offers more benefits from thermodynamic perspective. Is it more beneficial to have steam extracted from a single or two power units? Thermodynamic equivalence is the case for both power plant operating with a constant electrical capacity from the units [equal to their power from prior to modernization to combined heat and power (Tables 6.3 and 6.4)] and higher and variable in time chemical energy stream of the fuel as a result of power compensation as well as for the case of a constant chemical energy stream of the fuel (Tables 6.1 and 6.2). The various most beneficial alternatives of operation whether involving single or two power units may come as a consequence of imprecision of the mathematical model of a power unit, which does not completely account for all the technical details in a power unit. This matter is also relevant for any other process, not only technical one regarding the justification for minuteness of a model at the expense of investing time and money. One can note that the calculations conducted for the purpose of testing the model used (Chap. 5) proved the conformity of the results of calculations (for the completely condensing unit operation) with the results of measurements at the level of 1.5%, which is an excellent result. Such precision level was also gained for the case of electric capacity, which is the most important input parameter to economic analysis. In addition, the correctness and adequacy of the results are relative both to the precision of measurements used for preparation of energy characteristics of facilities used in the model and the precision of calculations. The latter, in turn, is associated with the calculation error that is allowed for. For the case of calculations here the calculation process stops when the maximum relative difference in the following iterations is lower than 10^{-5}.
3. For the operation of a power unit with a constant electrical power output for the supply of two heaters (XC3 and XC4) or even one (XC4) the stream of chemical energy of fuel is considerably higher in comparison to the supply of all three heaters. Concurrently, for the case of operation with a constant

chemical energy of boiler fuel combustion, electric power output of the plant is considerably smaller in comparison to the output when all three heaters are supplied. Therefore, alternatives with a single XC4 heater and two heaters are unattractive from the thermodynamic point of view. Does this also pertain to the economics? It is clear that for the latter case the cost of modernization of the power plant will be lowest. In conclusion, this requires further analysis. It is so that the justification of a business undertaking is decided on or after an economic analysis.

4. It is very important to note here that the modernization in the form that is undertaken has to secure continuous supply of heat to end users. For this reason among the alternatives of heater supply presented in this book the most beneficial is the one in which heater XC4 is supplied in a parallel system from two power units. In this system steam extracted from one of the power units is supplied to XC2 and XC4 heaters, and the other unit supplies XC3 and XC4 heaters (Sects. 6.1.2, alternative f and 6.1.3, alternative d, Fig. 6.1). In the event of an emergency or service shut-down of one of the units, which cannot be completely ruled out, it is possible that the other one takes over the supply task. This alternative is, additionally, beneficial from the thermodynamic point of view.
5. The temperatures of network hot water in XC2 and XC3 heater that are raised from 70 to 80°C and from 90 to 120°C, respectively (which is enabled by steam from A2 and A3 extractions) result in a greater electrical power output for a constant steam of chemical energy of the fuel. This results from the almost zero heating steam extraction from IP–LP crossoverpipe and decreased extraction from A3. Thus, the thermodynamic efficiency of the combined heat and power increases. If the temperature of network hot water was to be reduced from $t_{\mathrm{h\ max}} = 135$ and 120°C for instance, it would be possible to abandon XC4 heater altogether and the thermodynamic efficiency of the combined process would increase beside the savings in terms of the use of primary fuels. In addition, the economic efficiency of the power plant would increase as well.
6. In the design of the structure of heaters it is also necessary to account for the possibility of heat supply to consumer in the case when the units adapted to combined heat and power have to operate under a considerably lower electrical output than the rated capacity or if it became completely out-of-service for some time. This is possible during regulation running of a power plant when the power output can fall to 180 MW. Such a reserve can be secured from heaters supplied from the steam collector. This part, in turn, is supplied from the exhausts of the HP turbine sections in the working units or from an emergency boiler.

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Chapter 7
Economic Efficiency of a Power Unit Adapted to Cogeneration

Chapter 6 presented the results of thermodynamic analysis of the operation of a power unit with the rated capacity of 370 MW adapted to combined heat and power with steam extractions from one and two power units (Figs. 3.4, 6.1). This chapter will focus on the results of the economic efficiency of such operation, values of specific cost of heat for all alternatives presented in Sect. 6.1, discounted net profit ΔNPV^{mod} gained as a result of combined heat and power in a power plant, internal rate of return IRR of capital investment J_{mod} needed for the modernization of the power plant to combined heat and power and dynamic pay-back period $DPBP^{mod}$ [1–4]. In addition, the presentation includes maximum justified distance of heat transmission pipeline L_{pip}^{max} from the power plant to consumers as well as maximum specific price $e_{CO_2}^{max}$ on the purchase of CO_2 allowances resulting from the emission of additional CO_2 volume associated with compensation of decrease of electric-energy production.

7.1 Specific Cost of Heat Production

Specific cost of heat generation k_h in a power plant without compensation of decrease of electric-energy production for a constant stream of chemical energy of boiler fuel combustion corresponding to electrical capacity of a power plant prior to its modernization to the combined heat and power is obtained from the equation for the increase of gross profit ΔZ_A resulting from the combined heat and power output (Eq. 6.8):

$$\Delta Z_A = Q_A e_h - \Delta E_{el,A}^{El,gross} e_{el}(1 - \varepsilon_{el}) - \Delta K_A^{El}. \tag{7.1}$$

From the equation in (7.1) it stems that the revenues from the sales of heat $Q_A e_h$ has to at least compensate for the decrease of the revenues from the sales of

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electric energy $\Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}} e_{\mathrm{el}}(1-\varepsilon_{\mathrm{el}})$ and increase of annual operating expenses of the power plant $\Delta K_{\mathrm{A}}^{\mathrm{El}}$.

If $\Delta Z_{\mathrm{A}} = 0$, then the specific price of heat e_{h} (per heat unit) is equal to the cost of its generation, i.e. $e_{\mathrm{h}} \equiv k_{\mathrm{h}}$:

$$k_{\mathrm{h}} = \frac{\Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}} e_{\mathrm{el}}(1-\varepsilon_{\mathrm{el}}) + \Delta K_{\mathrm{A}}^{\mathrm{El}}}{Q_{\mathrm{A}}} \tag{7.2}$$

while the annual increase in the operating cost of the power plant $\Delta K_{\mathrm{A}}^{\mathrm{El}}$ in the combined heat and power is expressed by the equation:

$$\Delta K_{\mathrm{A}}^{\mathrm{El}} = (z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_{\mathrm{P}} \tag{7.3}$$

where:

e_{el}	specific price of electric energy (per energy unit),
$\Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}}$	decrease of the annual electrical energy output from a power plant resulting from combined heat and power without compensation of decrease of electric power unit of the heat produced in cogeneration,
J_{mod}	capital expenditure for the modernization of the power plant to the combined heat and power,
K_{p}	cost of electric energy needed to power network hot water pumps and auxiliary pumps in the heat transmission pipeline,
Q_{A}	annual heat production in a power plant,
$z\rho J_{\mathrm{mod}}$	depreciation,
$\delta_{\mathrm{serv}} J_{\mathrm{mod}}$	cost of maintenance and overhaul of the facilities,
$\varepsilon_{\mathrm{el}}$	relative coefficient of power station internal load.

The annual increase of the cost $\Delta K_{\mathrm{A}}^{\mathrm{El}}$ is associated with the total “turnkey” investment J_{mod} (Eq. 7.8) for the modernization of the power plant to combined heat and power, cost K_{p} of electric energy needed to drive the pumps in the network hot water distribution system and accessory ones that maintain the adequate static water pressure level to enable its evaporation.

The specific cost of heat production $k_{\mathrm{h}}^{\mathrm{com}}$ in the power plant during its operation with compensation of decrease of electric-energy production, i.e. for electric capacity that is equal to the one prior to the modernization to combined heat and power as a result of increasing chemical energy of boiler fuel combustion is obtained from the relation (Eq. 6.4):

$$\Delta Z_{\mathrm{A}}^{\mathrm{com}} = Q_{\mathrm{A}} e_{\mathrm{h}} - \Delta K_{\mathrm{A}}^{\mathrm{El,com}}, \tag{7.4}$$

On this basis and for the condition that $\Delta Z_{\mathrm{A}}^{\mathrm{com}} = 0$:

$$k_{\mathrm{h}}^{\mathrm{com}} = \frac{\Delta K_{\mathrm{A}}^{\mathrm{El,com}}}{Q_{\mathrm{A}}}. \tag{7.5}$$

Concurrently, the annual increase in the operating cost of a power plant is expressed by the equation:

$$\Delta K_{\mathrm{A}}^{\mathrm{El,com}} = (z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_{\mathrm{P}} + \Delta P_{\mathrm{A}}(\mathrm{NCV})e_{\mathrm{coal}} + \Delta K_{\mathrm{env}}^{\mathrm{com}} \tag{7.6}$$

where:

e_{coal} specific price of coal (per energy unit).

In comparison to the cost $\Delta K_{\mathrm{A}}^{\mathrm{El}}$ (Eq. 7.3) the annual operating cost of a power plant $\Delta K_{\mathrm{A}}^{\mathrm{El,com}}$ increases as a result of additional cost of hard coal $\Delta P_{\mathrm{A}}(\mathrm{NCV})e_{\mathrm{coal}}$ needed for compensation of decrease of electric-energy production and expenses associated with environmental charges $\Delta K_{\mathrm{env}}^{\mathrm{com}}$.

The extra chemical energy steam of boiler fuel combustion $\Delta P_{\mathrm{A}}(\mathrm{NCV})$ per annum is expressed by the equation:

$$\Delta P_{\mathrm{A}}(\mathrm{NCV}) = \int_{0}^{\tau_{\mathrm{A}}} \Delta\dot{P}(\mathrm{NCV})\mathrm{d}\tau \tag{7.7}$$

The cost $\Delta K_{\mathrm{A}}^{\mathrm{El}}$, $\Delta K_{\mathrm{A}}^{\mathrm{El,com}}$ in the numerator of Eqs. 7.2 and 7.5 decreases as a result of extra revenues gained from the sales of certificates of origin of electric energy from the combined generation of electric energy and heat in a high-efficiency process. These revenues lead to reduction in the cost of heat generation. However, in the circumstances of a compensation of decrease of electric-energy production in a power plant the obligation of purchasing additional CO_2 allowances as a result of combustion of increased volume of coal ΔP_{A} leads to increase of the cost $\Delta K_{\mathrm{A}}^{\mathrm{El,com}}$ and can result in higher profitability of the operation without compensation.

Therefore, there is a boundary price $e_{\mathrm{CO_2}}^{\max}$ on the purchase of additional CO_2 allowances, which can be determined from the condition $k_{\mathrm{h}}^{\mathrm{com}} = k_{\mathrm{h}}$.

The total investment J_{mod} consists of the cost of investment specification, construction work, cost of machinery, engineering supervision, acceptance, start-up, reserve and other unpredictable expenses. Specifically speaking, investment J_{mod} can be divided into expenditure associated with modernization of turbine to heating steam extraction $J_{\mathrm{turb}}^{\mathrm{mod}}$, heat distribution pipeline $J_{\mathrm{pip}}^{\mathrm{dist}}$, installation that makes up hot water in the network along with circulation pumps $J_{\mathrm{water}}^{\mathrm{inst}}$, heat-exchanger stations $J_{\mathrm{stat}}^{\mathrm{heat}}$ and buildings needed for these facilities J_{bild}, electric switchgear $J_{\mathrm{switchgear}}$ and systems of automatic control J_{aut}:

$$J_{\mathrm{mod}} = J_{\mathrm{bild}} + J_{\mathrm{switchgear}} + J_{\mathrm{aut}} + J_{\mathrm{turb}}^{\mathrm{mod}} + J_{\mathrm{pip}}^{\mathrm{dist}} + J_{\mathrm{water}}^{\mathrm{inst}} + J_{\mathrm{stat}}^{\mathrm{heat}}. \tag{7.8}$$

The investment in the heat transmission pipeline is the function of its total length L_{pip}:

$$J_{\text{pip}}^{\text{dist}} = j_{\text{D}} L_{\text{pip}} \tag{7.9}$$

where:

j_{D} specific (per unit of length) investment in heat distribution pipeline; this investment is the function of the diameters of forward pipeline and return one.

The investment in heaters XC2, XC3 and XC4, Fig. 3.4 (together forming the capital expenditure on $J_{\text{stat}}^{\text{heat}}$), are relative to their maximum thermal power resulting from qualitative regulation of the thermal power generated in the power plant in accordance with required maximum temperatures of network hot water (Chaps. 4 and 6):

$$J_{\text{stat}}^{\text{heat}} = j_{\dot{\text{Q}}} \left(\dot{Q}_{\text{XC2}}^{\max} + \dot{Q}_{\text{XC3}}^{\max} + \dot{Q}_{\text{XC4}}^{\max} \right) \tag{7.10}$$

where:

$j_{\dot{\text{Q}}}$ specific investment in heaters (per unit of power).

Depending on the alternative of combined heat and power described in Chap. 6 the capital expenditure $J_{\text{stat}}^{\text{heat}}$ involves the investment in three heaters XC2, XC3, XC4 or only two, XC2 and XC4 (in this case $\dot{Q}_{\text{XC3}}^{\max} = 0$) or XC3 and XC4 (in this case $\dot{Q}_{\text{XC2}}^{\max} = 0$), or only in one XC4 (in this case $\dot{Q}_{\text{XC2}}^{\max} = \dot{Q}_{\text{XC3}}^{\max} = 0$).

In the examined case Eq. 7.8 accounting for Eqs. 7.9 and 7.10 takes the form:

$$\begin{aligned} J_{\text{mod}} &= J_{\text{bild}} + J_{\text{switchgear}} + J_{\text{aut}} + j_{\text{D}} L_{\text{pip}} \\ &\quad + j_{\dot{\text{Q}}} \left(\dot{Q}_{\text{XC2}}^{\max} + \dot{Q}_{\text{XC3}}^{\max} + \dot{Q}_{\text{XC4}}^{\max} \right) \text{ thousand PLN} \end{aligned} \tag{7.11}$$

Using relation in (7.11) it is possible to determine the maximum length of heat-distribution pipeline $L_{\text{pip}}^{\max}$ for which is profitable to supply a remote town with heat. This length $L_{\text{pip}}^{\max}$ is calculated from Eqs. 7.2 and 7.5 by substituting heat sales price e_{h}^{EC} in the place of k_{h} and $k_{\text{h}}^{\text{com}}$, which corresponds to heat price from the heating plant situated in the town.

7.1.1 Results of Calculations

The following input data were taken for the analysis:

- specific cost of heat transmission pipeline j_{D} = 6,000 PLN/m (for the diameters of forward and return pipelines of D = 900 mm),
- length of the pipeline L_{pip} = 12 km,
- specific cost of heat exchangers $j_{\dot{\text{Q}}}$= 31 PLN/kW,

- investment in buildings, switchgear and systems of automatic control $J_{bild} + J_{switchgear} + J_{aut} = 60$ million PLN,
- investment in modernization of turbine for extraction of heating steam $J_{turb}^{mod} = 2$ million PLN/turbine,
- relative coefficient of power station's internal load $\varepsilon_{el} = 8\%$,
- specific price of coal $e_{coal} = 11.5$ PLN/GJ,
- specific price of electric energy $e_{el} = 170$ PLN/MWh,
- specific price of electric energy needed to drive the pumps in the network hot water supply system and accessory ones in the heat-distribution pipeline $e_{el} = 140$ PLN/MWh; an assumption was made that the power used by the pumps is equal to 4 MW,
- specific cost for polluting environment (per unit of chemical energy of coal combustion)—0.3 PLN/GJ,
- specific (per energy unit) revenues gained from the sales of certificates of origin of electric energy from high-efficiency combined power and heat generation—$e_{cert} = 17.7$ PLN/MWh,
- annual depreciation rate, cost of overhaul and maintenance $z\rho + \delta_{serv} = 16\%$,
- specific sales price of heat from the communal heating station needed for calculation of boundary value of L_{pip}^{max}, $e_{h}^{EC} = 33$ PLN/GJ,
- specific purchase price of additional CO_2 allowances—$e_{CO_2} = 54.3$ PLN/Mg_{CO_2},
- CO_2 emission per unit of chemical energy of coal—94.13 kg_{CO_2}/GJ.

The calculated values of k_h, k_h^{com}, L_{pip}^{max}, $e_{CO_2}^{max}$ are summarized in Tables 7.1, 7.2.

The lowest cost k_h, k_h^{com} is obtained for the operation with three heaters. Due to the necessity of maintaining continuous supply of heat to consumers the optimal solution involves combined heat and power generation in a parallel system with steam extraction from two power units to feed XC4 heater (Fig. 6.1). The rationale for that is presented in Sect. 6.4. Basically, in the case of an emergency shut-down of power unit it is possible to supply sufficient volume of steam from the other one. The lowest cost is obtained for the case of electric power compensation in a power plant: k_h^{com}= 12.51 PLN/GJ, item 11, Table 7.1.

The maximum length of the heat distribution pipeline $L_{pip}^{max}$is equal to around 60 km. This value is derived on condition of the operation of a unit with its rated capacity.

The boundary purchase price of additional CO_2 allowances that was derived from calculations at $e_{CO_2}^{max} = 88.6$ PLN/Mg_{CO_2} (item 11, Table 7.1) exceeds the current price of $e_{CO_2} = 54.3$ PLN/Mg_{CO_2} by over 60%. Hence, combined heat and power in a power plant is more justified with electric power compensation.

Table 7.1 Specific cost of heat production for the analyzed alternatives of heater supply; $N_{el}^{ST} = 380$ MW = const, $\dot{Q}_{c\,max} = 220$ MW, $\dot{Q}_{dhw} = 15$ MW, $\tau_w = 211$ days

Item	Alternative	Capital expenditure	Not accounting for revenues from sales of certificates of origin and purchase of CO_2 allowances		Accounting for revenues from sales of certificates of origin and purchase of CO_2 allowances		Accounting for revenues from sales of certificates of origin and purchase of CO_2 allowances		Boundary purchase price of CO_2 allowances
		J_{mod} [mln PLN]	k_h^{com} (PLN/GJ)	L_{pip}^{max} (km)	k_h^{com} (PLN/GJ)	L_{pip}^{max} (km)	k_h^{com} (PLN/GJ)	L_{pip}^{max} (km)	$e_{CO_2}^{max}$ (PLN/Mg)
Single unit operation—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C									
1.	XC2/3/4	143.6	14.12	57.5	10.49	66.3	12.27	62.0	88.4
2.	XC2/4	143.6	14.23	57.3	10.63	65.9	12.46	61.5	88.3
3.	XC3/4	142.7	15.87	53.3	12.88	60.5	15.44	54.3	87.2
4.	XC4	140.8	16.26	52.4	13.45	59.2	16.24	52.4	105.6
Single unit operation—A2 ≤ 80°C, A3 ≤ 120°C, IP–LP ≤ 135°C									
5.	XC2/3/4	143.2	13.85	58.2	10.14	67.1	11.82	63.1	88.3
Two unit operation—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C									
6.	XC2/3 (1st unit) XC4 (2nd unit)	145.6	14.27	57.2	10.64	65.9	12.43	61.6	88.7
7.	XC2 (1st unit) XC3/4 (2nd unit)	145.6	14.31	57.1	10.70	65.8	12.50	61.4	88.6
8.	XC2/4 (1st unit) XC3 (2nd unit)	145.6	14.32	57.1	10.71	65.8	12.51	61.4	88.6
9.	XC2 (1st unit) XC4 (2nd unit)	145.6	14.38	56.9	10.79	65.6	12.63	61.1	88.3
10.	XC3 (1st unit) XC4 (2nd unit)	144.7	16.02	52.9	13.02	60.2	15.59	54.0	87.6

(Continued)

Table 7.1 (continued)

Item	Alternative	Capital expenditure	Not accounting for revenues from sales of certificates of origin and purchase of CO_2 allowances		Accounting for revenues from sales of certificates of origin and purchase of CO_2 allowances		Accounting for revenues from sales of certificates of origin and purchase of CO_2 allowances		Boundary purchase price of CO_2 allowances
		J_{mod} [mln PLN]	k_h^{com} (PLN/GJ)	L_{pip}^{max} (km)	k_h^{com} (PLN/GJ)	L_{pip}^{max} (km)	k_h^{com} (PLN/GJ)	L_{pip}^{max} (km)	$e_{CO_2}^{max}$ (PLN/Mg)
11.	XC2/4 (1st unit) XC3/4 (2nd unit)	145.6	14.31	57.1	10.71	65.8	12.51	61.4	88.6
Two unit operation—A2 $\leq$ 80°C, A3 $\leq$ 120°C, IP–LP $\leq$ 135°C									
12.	XC2/3 (1st unit) XC4 (2nd unit)	145.2	13.99	57.9	10.28	66.8	11.96	62.8	88.7
13.	XC2 (1st unit) XC3/4 (2nd unit)	145.2	14.02	57.8	10.32	66.7	12.01	62.6	88.6
14.	XC2/4 (1st unit) XC3 (2nd unit)	145.2	14.02	57.8	10.32	66.7	12.02	62.6	88.6
15.	XC2/4 (1st unit) XC3/4 (2nd unit)	145.2	14.02	57.8	10.32	66.7	12.01	62.6	88.6

Table 7.2 Specific cost of heat production for the analyzed alternatives of heater supply; $\dot{E}_{ch} = 934.8$ MW = const, $\dot{Q}_{c\,max} = 220$ MW, $\dot{Q}_{dhw} = 15$ MW, $\tau_w = 211$ days

Item	Alternative	Capital expenditure	Not accounting for revenues from sales of certificates of origin		Accounting for revenues from sales of certificates of origin	
		J_{mod} (mln PLN)	k_h (PLN/GJ)	L_{pip}^{max} (km)	k_h (PLN/GJ)	L_{pip}^{max} (km)
Single unit operation—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C						
1.	XC2/3/4	143.6	17.03	50.5	13.39	59.3
2.	XC2/4	143.6	17.22	50.1	13.61	58.8
3.	XC3/4	142.7	20.01	43.3	17.00	50.6
4.	XC4	140.8	21.72	39.2	18.88	46.1
Single unit operation—A2 ≤ 80°C, A3 ≤ 120°C, IP–LP ≤ 135°C						
5.	XC2/3/4	143.2	16.59	51.6	12.87	60.6
Two unit operation—A2 ≤ 70°C, A3 ≤ 90°C, IP–LP ≤ 135°C						
6.	XC2/3 (1st unit) XC4 (2nd unit)	145.6	17.19	50.1	13.56	58.9
7.	XC2 (1st unit) XC3/4 (2nd unit)	145.6	17.26	50.0	13.64	58.7
8.	XC2/4 (1st unit) XC3 (2nd unit)	145.6	17.27	50.0	13.65	58.7
9.	XC2 (1st unit) XC4 (2nd unit)	145.6	17.38	49.7	13.78	58.4
10.	XC3 (1st unit) XC4 (2nd unit)	144.7	20.17	42.9	17.17	50.2
11.	XC2/4 (1st unit) XC3/4 (2nd unit)	145.6	17.27	50.0	13.65	58.7
Two unit operation—A2 ≤ 80°C, A3 ≤ 120°C, IP–LP ≤ 135°C						
12.	XC2/3 (1st unit) XC4 (2nd unit)	145.2	16.73	51.2	13.02	60.2
13.	XC2 (1st unit) XC3/4 (2nd unit)	145.2	16.79	51.1	13.08	60.0
14.	XC2/4 (1st unit) XC3 (2nd unit)	145.2	16.79	51.1	13.09	60.0
15.	XC2/4 (1st unit) XC3/4 (2nd unit)	145.2	16.79	51.1	13.09	60.0

7.2 Discounted Parameters of Assessing Economic Efficiency of Combined Heat and Power in a Power Plant

The objective in any business undertaking is profit gaining. Profitability is the most important criterion of assessing an investment. This means that prior to taking a decision regarding involvement of capital an investor needs to be secure that the interest rate J_{mod} from an investment will be sufficiently high.

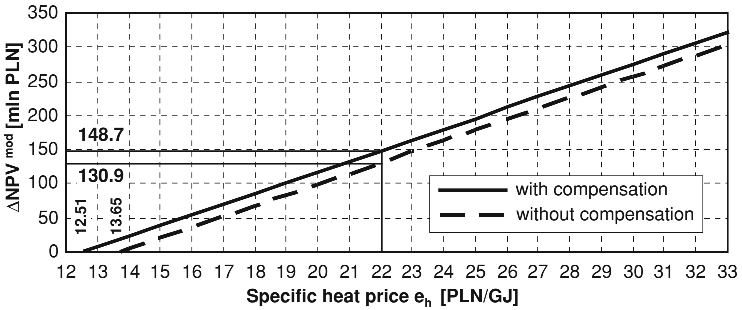

Fig. 7.1 Discounted net profit ΔNPV^{mod} in the function of specific heat price (for heater supply alternative as in item 11, Tables 7.1, 7.2)

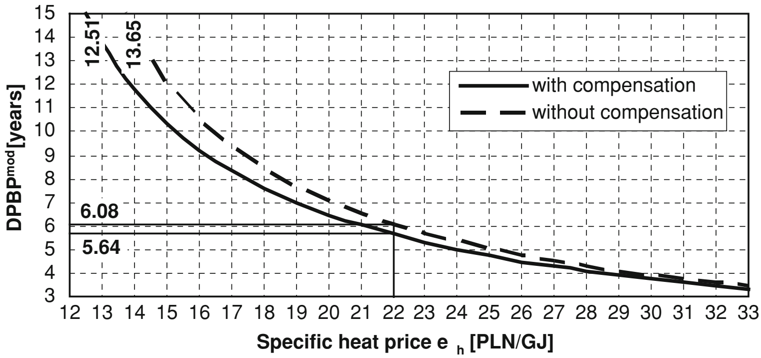

Fig. 7.2 Dynamic payback period $DPBP^{mod}$ of capital expenditure in the function of specific heat price (for heater supply alternative as in item 11, Tables 7.1, 7.2)

The reply to the question regarding the economic efficiency of an investment is offered by the values of calculated parameters of economic efficiency ΔNPV^{mod}, $DPBP^{mod}$, IRR (Figs. 7.1, 7.2, 7.3). The discounting parameters, i.e. ones that account for a changing time value of money are considered as the most effective criteria of evaluating an investment. In addition, it is necessary to conduct the analysis of sensitivity of these parameters to assess their changing value in the function of parameters they are relative to Figs. 7.4, 7.5, 7.6, 7.7, 7.8. The analysis of sensitivity offers a wider perspective regarding the profitability of an investment and helps assess its security. In addition, in the circumstances of competition it enables a business to conduct a price policy.

The total increase of discounted net profit (Eq. 2.1; Chap. 2) gained from the combined heat and power in a power plant not accounting for Independent Power Producer involvement is expressed by the equation [1–4]:

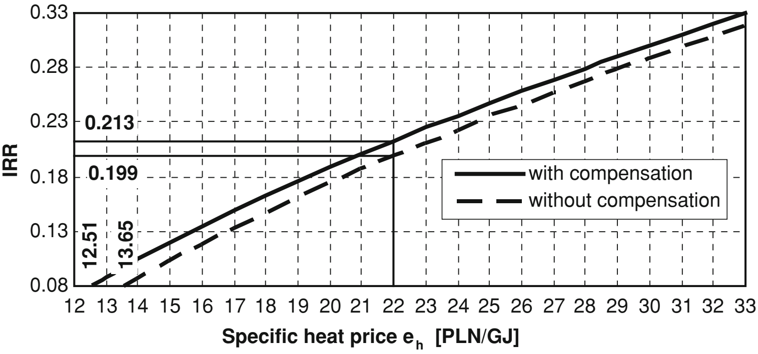

Fig. 7.3 Internal return rate IRR in the function of specific heat price (for heater supply alternative as in item 11, Tables 7.1, 7.2)

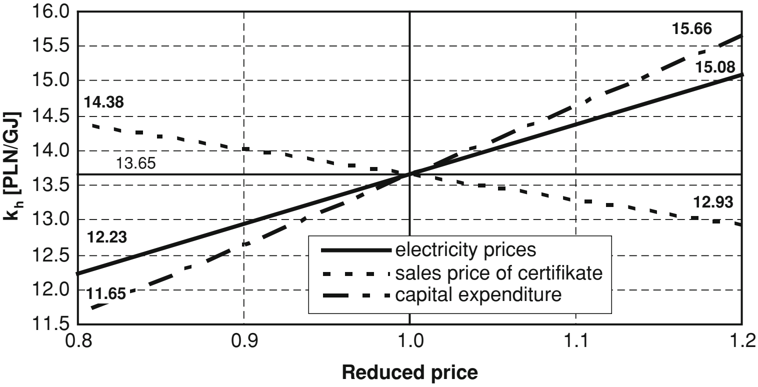

Fig. 7.4 Influence of electric-energy prices, sales price of certificates of electric energy origin from high-efficiency cogeneration and capital expenditure on the value of specific cost of heat production k_h (alternative of heater supply described in item 11, Table 7.2)

$$\Delta \mathrm{NPV}^{\mathrm{mod}} = \frac{\Delta Z_A (1 - p)}{\rho} \tag{7.12}$$

where:

p rate of profits tax ΔZ_A (an assumption was made that $p = 19\%$; this is the current tax rate in Poland),

ρ annual rate of progressive depreciation,

while the increase of gross profit (Eqs. 7.1 and 7.4) can be comfortably presented by the relation:

$$\Delta Z_A = Q_A (e_h - k_h). \tag{7.13}$$

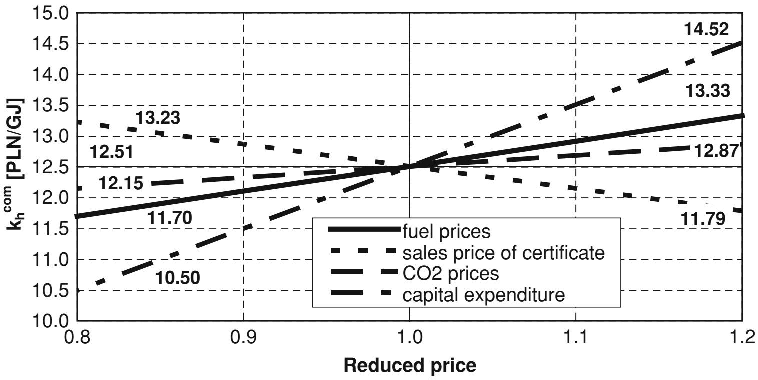

Fig. 7.5 Influence of fuel price, sales price of certificates of electric energy origin from high-efficiency cogeneration and purchase prices of CO2 allowances and capital expenditure on the value of the specific cost of heat production $k_{\mathrm{h}}^{\mathrm{com}}$ (alternative of heater supply described in item 11, Table 7.1)

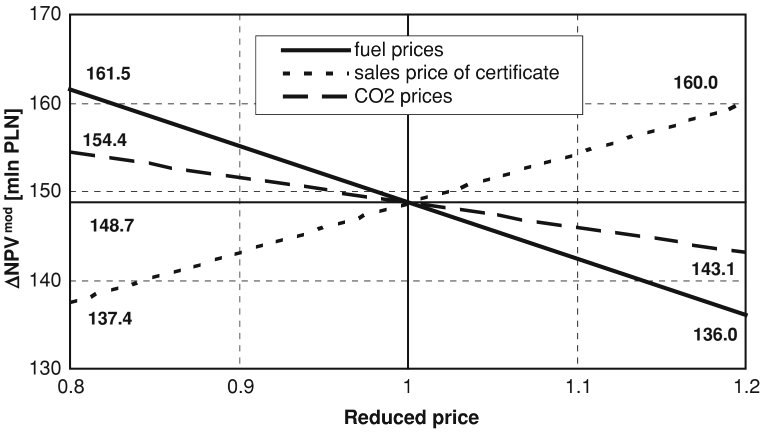

Fig. 7.6 Influence of fuel price, revenues from sales of certificates of electric-energy origin from high-efficiency cogeneration and purchase prices of additional CO_2 allowances on the value of $\Delta\mathrm{NPV}^{\mathrm{mod}}$ for the combined heat and power with compensation of decrease of electric-energy production (alternative of heater supply described in item 11, Table 7.1)

The rate of progressive depreciation ρ is expressed by the equation [1–4]:

$$\rho = \frac{r(1+r)^N}{(1+r)^N - 1}, \tag{7.14}$$

where:

N calculated exploitation period of a power plant, in years (in the calculations $N = 15$ years),

r interest rate (in the calculations $r = 8\%$).

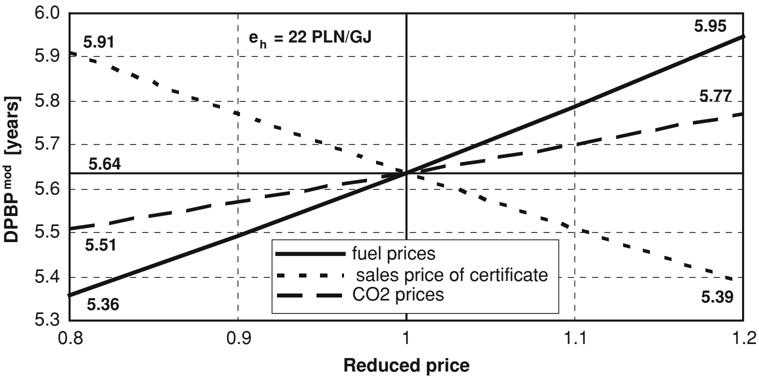

Fig. 7.7 Influence of fuel price, sales price of certificates of electric energy origin from high-efficiency cogeneration and purchase prices of additional CO_2 allowances on the value of $DPBP^{mod}$ for the combined heat and power with compensation of decrease of electric-energy production (alternative of heater supply described in item 11, Table 7.1)

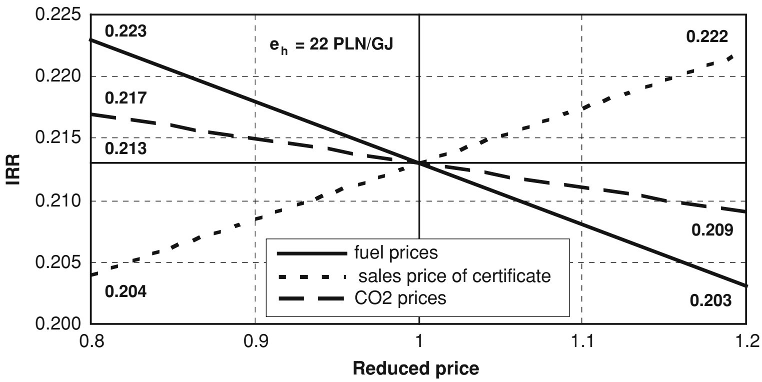

Fig. 7.8 Influence of fuel price, sales price of certificates of electric energy origin from high-efficiency cogeneration and purchase prices of additional CO_2 allowances on the value of IRR for the combined heat and power with compensation of decrease of electric-energy production (alternative of heater supply described in item 11, Table 7.1)

The rate of progressive depreciation guarantees a reward for an investment $J_{mod} = J_{cre} + J_{own}$ of capital involved in business undertaking that comes both from a loan as all as from investor's own funds. This means that own funds J_{own} and ones from a loan J_{cre} are considered in an equivalent manner.

The dynamic pay-back period that is associated with capital expenditure J_{mod} incurred in connection with the modernization of a power plant to combined heat and power together with the interest that invested capital would otherwise have brought is expressed with the equation [1–4]:

$$\mathrm{DPBP}^{\mathrm{mod}} = \frac{\ln \frac{\Delta Z_{\mathrm{A}}(1-p)+z\rho J_{\mathrm{mod}}}{\Delta Z_{\mathrm{A}}(1-p)+z\rho J_{\mathrm{mod}}-zrJ_{\mathrm{mod}}}}{\ln(1+r)} \tag{7.15}$$

where:

z discounting rate of a capital expenditure J_{mod} at the instant when the modernization of a power plant to combined is complete, $z > 1$; this rate accounts for the undesired effect of capital freezing during the construction period, since this money does not offer any profit in that period [1–4]:

$$z = \frac{\left[(1+r)^b - 1\right](1+r)}{br}, \tag{7.16}$$

where b denotes the construction period associated with the modernization of a power plant to combined heat and power, in years (the calculations assumed $b = 1$ year; then $z = 1 + r$).

The IRR rate that is brought by the capital investment J_{mod} is determined by the subsequent approximation from the equation [1–4]:

$$\mathrm{CF_A} = [z\rho]_{\mathrm{IRR}} J_{\mathrm{mod}} = [\Delta Z_{\mathrm{A}}]_{\mathrm{r}} + [z\rho]_{\mathrm{r}} J_{\mathrm{mod}}, \tag{7.17}$$

where as the notation $[\Delta Z_{\mathrm{A}}]_{\mathrm{r}}$ means that the value of ΔZ_{A} (Eqs. 7.1 and 7.4) were obtained for the discount rate of r.

Annual cash flow $\mathrm{CF_A}$ is determined by the relations:

- for combined heat and power without compensation of decrease of electric-energy production

$$\mathrm{CF_A} = Q_{\mathrm{A}} e_{\mathrm{h}} - \Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}} e_{\mathrm{el}}(1 - \varepsilon_{\mathrm{el}}) - K_{\mathrm{P}} - \delta_{\mathrm{serv}} J_{\mathrm{mod}} \tag{7.18}$$

- for combined heat and power with compensation of decrease of electric-energy production

$$\mathrm{CF_A} = Q_{\mathrm{A}} e_{\mathrm{h}} - K_{\mathrm{P}} - \delta_{\mathrm{serv}} J_{\mathrm{mod}} - \Delta P_{\mathrm{A}}(\mathrm{NCV}) e_{\mathrm{fuel}} - \Delta K_{\mathrm{env}}^{\mathrm{com}}, \tag{7.19}$$

while the annual rate δ_{serv} in the cost of overhaul and maintenance $\delta_{\mathrm{serv}} J_{\mathrm{mod}}$ was assumed at the level of $\delta_{\mathrm{serv}} = 3\%$.

To the cash flow $\mathrm{CF_A}$ in Eqs. 7.18 and 7.19 we need to add the revenues gained from the sales of certificates of origin of electric energy from high-efficiency combined power and heat generation. In the case of operation with electric power compensation from $\mathrm{CF_A}$ (Eq. 7.19) it is necessary to deduce the cost of purchase of additional CO_2 allowances.

The calculation of IRR rate in Eq. 7.17, in Eqs. 7.14 and 7.16 representing variables z and ρ instead of discount rate r we have to substitute the desired value of IRR. For $b = 1$ (Eq. 7.16), which is practically the case, IRR is therefore determined from the relation:

$$\mathrm{CF_A} = J_{\mathrm{mod}} \frac{\mathrm{IRR}(1 + \mathrm{IRR})^{N+1}}{(1 + \mathrm{IRR})^{N} - 1} \tag{7.20}$$

From Eqs. 7.17–7.19 it stems that if $e_{\mathrm{h}} = k_{\mathrm{h}}$ (Eqs. 7.2 and 7.5) then IRR is equal to the discount rate adopted in the calculations of $r = 8\%$, DPBP$^{\mathrm{mod}}$ is equal to calculated period of the exploitation of a power plant $N = 15$ years and $\Delta\mathrm{NPV}^{\mathrm{mod}} = 0$.

7.2.1 Results of Calculations

Figures 7.1, 7.2, and 7.3 present the values of $\Delta\mathrm{NPV}^{\mathrm{mod}}$, DPBP$^{\mathrm{mod}}$, IRR in the function of heat prices e_{h} for the alternative of combined heat and power with heating steam extraction from two power units and a parallel system of steam supply to XC4 heater (Fig. 6.1).

7.3 Analysis of Sensitivity

As already mentioned in Sect. 7.2, after the analysis of economic efficiency of any undertaking, after calculation of discounted parameters $\Delta\mathrm{NPV}^{\mathrm{mod}}$, DPBP$^{\mathrm{mod}}$, IRR it is necessary to conduct an analysis of their sensitivity in the function of the parameters that affect them. Analysis of sensitivity offers a wider perspective with regard to the profitability of an investment and enables an investor to assess its security as well as offers grounds for conducting price policy in the circumstances of a competitive market. It presents the range of primary fuel prices which will secure the profitability of an undertaking and the scope for reducing the price of a product that will ensure that they will not go of business. This level is determined as zero profits level, i.e. corresponds to zero value of $\Delta\mathrm{NPV}^{\mathrm{mod}}$.

7.3.1 Results of Calculations

Figures 7.4, 7.5, 7.6, 7.7 and 7.8 present the fluctuations of the parameters $\Delta\mathrm{NPV}^{\mathrm{mod}}$, DPBP$^{\mathrm{mod}}$ and IRR and changes in the specific cost of heat generation k_{h}, $k_{\mathrm{h}}^{\mathrm{com}}$in the function of capital expenditure J_{mod} as well as in the function of coal price e_{coal}, electric energy price e_{el}, sales price of certificates e_{cert} of electric-energy origin from high-efficiency cogeneration and the price e_{CO_2}on the purchase of additional CO_2 allowances. The values of the above taken into consideration vary in the range of ±20% from their basic values. The input prices are the following: $J_{mod} = 145.6$ million PLN, $e_{coal} = 11.5$ PLN/GJ, $e_{el} = 170$ PLN/MW, $e_{cert} = 17.7$ PLN/MWh, $e_{CO_2} = 54.3\,\text{PLN/Mg}_{CO_2}$. The reduced prices corresponding to basic prices assume value of 1 on the X axis in Figs. 7.4, 7.5, 7.6, 7.7 and 7.8. In the analysis of the sensitivity of parameters ΔNPV^{mod}, $DPBP^{mod}$, IRR (Figs. 7.6, 7.7, 7.8) an additional assumption was made that the basic heat price is $e_h = 22$ PLN/GJ (which is 33% lower than heat price $e_h^{EC} = 33$ PLN/GJ of the currently delivered heat from the commercial heating plant situated in the town).

From Figs. 7.4, 7.5, 7.6, 7.7 and 7.8 it results that the values of k_h, k_h^{com}, ΔNPV^{mod}, $DPBP^{mod}$, IRR are most sensitive to variations of capital expenditure J_{mod}, coal price e_{coal}, electric-energy price e_{el}, sales price of certificates e_{cert} of electric-energy origin from high-efficiency cogeneration. Concurrently, they are less sensitive to the price e_{CO_2} on the purchase of additional CO_2 allowances. For the price of heat of $e_h = 22$ PLN/GJ a change in the prices even by ±20% still secures that very high-economic efficiency of the operation of the power plant is maintained (for heat prices in excess of 22 PLN/GJ the situation will be more beneficial). For instance, if the price of coal e_{coal} were to increase by 20% from 11.5 to 13.8 PLN/GJ, the specific cost k_h^{com} would increase from 12.51 to 13.33 PLN/GJ (Fig. 7.5) and the profit from the undertaking ΔNPV^{mod} would fall from 148.7 to the value 136.0 million PLN (Fig. 7.6), time $DPBP^{mod}$ increases from 5.64 to 5.95 years (Fig. 7.7), and interest rate IRR falls from 0.213 to reach the value of 0.203 (Fig. 7.8). If profit ΔNPV^{mod} was to gain zero, the time $DPBP^{mod}$ would last for 15 years, and IRR 8% (the specific cost of heat generation k_h^{com} has to increase to the value of its sales price $e_h = 22$ PLN/GJ), it would be necessary that the price of coal rises by 331.5% to reach $e_{coal} = 38.12$ PLN/GJ.

7.4 Summary and Conclusions

1. The economic efficiency of a combined heat and power cycle in an adapted power station is very high. For instance, for the heat price of only $e_h = 22$ PLN/GJ (as it was mentioned before, this price is 33% lower than the price of heat $e_h^{EC} = 33$ PLN/GJ currently delivered to consumers from the existing district heating station located in the town) the increase of the discounted net profit ΔNPV^{mod} for the case of a combined heat and power with compensation of electrical power output in the alternative with heating steam extraction from two power units for the parallel steam supply to XC4 heater (Fig. 6.1; item 11, Table 7.1) is equal to 148.7 million PLN, IRR rate that the investment yields is equal to 21.3%, and the payback period $DPBP^{mod}$ lasts for only 5.64 years (Figs. 7.1, 7.2, 7.3, 7.4, 7.5, 7.6, 7.7, 7.8). This result can be assessed as a good one from the economic perspective. The maximum length of the transportation of heat from the power station to consumers is equal to $L_{pip}^{max} = 61.4$ km, while the maximum expense associated with the purchase of additional CO_2

allowances due to the combustion of additional coal volumes is equal to $e_{CO_2}^{max} = 88.6\,\mathrm{PLN/Mg}_{CO_2}$.

2. The effect of raising the temperatures of network hot water heating in XC2 and XC3 heaters from 70 to 80°C and from 90 to 120°C, respectively (as it is made possible due to parameters of steam from extractions A2 and A3) is associated with an increase of thermodynamic (Chap. 6) and economic efficiency of the combined heat and power. This comes as a consequence of zero steam extraction from IP–LP crossover pipe and reduced steam bleed from extraction A3). If the maximum temperature of network hot water could be reduced from $t_{h\ max} = 135$°C to for instance 120°C it would be possible to abandon XC4 heater altogether. As a result, the thermodynamic efficiency of the combined heat and power would increase substantially accompanied by decrease of primary fuel use, thus further increasing the economic efficiency of the power station operating in a combined heat and power.

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Chapter 8
Technical and Economical Effectiveness of Modernization 370 MW Power Unit Repowered by Gas Turbine with their Modernization to Cogeneration

The objective in the modernization of existing power plants involving their modernization to combined heat and power and concurrent repowering by a gas turbine and heat recovery steam generator is the improvement of the energy efficiency of electrical energy production, limitation of pollutant emission into the environment and improvement of the economic efficiency of the operation.

8.1 Methodology of Economical Analysis of Repowering 370 MW Power Unit by Gas Turbine with their Modernization to Cogeneration

During the analysis of the economic efficiency of the modernization of a power unit and involving its repowering by a gas turbine (Fig. 3.4) it is essential to determine the power output of the gas turbine and an optimum structure of the heat recovery steam generator that repowers the system. However, other factors also have to be considered. It is very important to find a reply to a question; what price relations between energy carriers are justified, between the price of coal, gas and electrical energy, what tariffs should be imposed on the emissions of CO_2, CO, NO_x, SO_2 and dust into the atmosphere due to which the application of the relatively expensive though ecological hydrocarbon based fuel (i.e. natural gas) in the power industry will be economically effective beside being justified from the thermodynamic perspective. These price relations and emission charges have an important effect on the selection of optimum output of the gas turbine.

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8.1.1 Necessary and Sufficient Conditions of Cost-Effective Modernization

The *necessary condition* for the economic efficiency of a power unit by a gas turbine and a heat recovery steam generator and its concurrent modernization to the combined heat and power is that the increase of the annual gross profit $\Delta Z_A = (Z_A)^{mod} - (Z_A)^{ex}$ resulting from the exploitation of a repowered unit is at less not less than zero, that is $\Delta Z_A \geq 0$. The profit from the modernization in a generalized case (i.e. when the sale price of electrical energy changes from $(e_{el})^{ex}$ to $(e_{el})^{mod}$) can be expressed by the equation

$$(Z_A)^{mod} = Q_A e_h + \left[\left(E_{el,A}\right)^{ex} + E^{GT}_{el,A} \mp \Delta E^{ST}_{el,A}\right](e_{el})^{mod} - \left[\left(K^{El}_A\right)^{ex} + \Delta K^{El}_A\right] \quad (8.1)$$

while the profit prior to repowering is expressed as

$$(Z_A)^{ex} = \left(E_{el,A}\right)^{ex}(e_{el})^{ex} - \left(K^{El}_A\right)^{ex} \quad (8.2)$$

Thus, the prerequisite for the undertaking is expressed by the relation (compare Eqs. 2.10a, 2.22b):

$$\begin{aligned}\Delta Z_A &= (Z_A)^{mod} - (Z_A)^{ex} \\ &= Q_A e_h + (E^{GT}_{el,A} \mp \Delta E^{ST}_{el,A})(e_{el})^{mod} - \Delta K^{El}_A + (E_{el,A})^{ex}[(e_{el})^{mod} - (e_{el})^{ex}] \geq 0\end{aligned} \quad (8.3)$$

where:

e_h, $(e_{el})^{ex}$, $(e_{el})^{mod}$	specific price of heat (per energy unit), electrical energy prior to and following its repowering,
$(E_{el,A})^{ex}$	annual electrical energy output of a power unit prior to its repowering,
$E^{GT}_{el,A}$	annual net electrical energy production in the gas turbogenerator,
$\Delta E^{ST}_{el,A}$	annual decrease (increase) of net electrical energy production, in the steam turbine;
$\Delta E^{ST}_{el,A}$	the function of the power of the gas turbine and structure of heat recovery steam generator; in the circumstances of partial exclusion of low-pressure regeneration in the power unit and replacing it with with regeneration in the heat recovery steam generator and production of intermediate- and low-pressure steam in it [2], an increase in the production of electrical energy will follow in the steam turbine despite the extraction of heating steam into heaters XC2, XC3, XC4 during the combined heat and power,
ΔK^{El}_A	increase of the annual operating cost of the power unit after its repowering,

Q_A annual heat generation in the unit adapted to combined heat and power.

From the Eq. 8.3 it stems that the revenues from the sales of heat $Q_A e_h$ and additional electrical energy $(E^{GT}_{el,A} \mp \Delta E^{ST}_{el,A})(e_{el})^{mod}$ as well as the revenues $(E_{el,A})^{ex}[(e_{el})^{mod} - (e_{el})^{ex}]$ resulting from a change in the price from $(e_{el})^{ex}$ to $(e_{el})^{mod}$ has to exceed the annual operating cost of the power unit after its modernization ΔK^{El}_A.

From the relation (8.3) we obtain the necessary condition for the minimum price e_h of heat for the purposes of district heating Q_A that ensures that the modernization of the power unit combined with its repowering by a gas turbine in a parallel system is economically justified.

$$e_h \geq \frac{\Delta K^{El}_A - (E^{GT}_{el,A} \mp \Delta E^{ST}_{el,A})(e_{el})^{mod} - (E_{el,A})^{ex}[(e_{el})^{mod} - (e_{el})^{ex}]}{Q_A} \tag{8.4}$$

When in the relation (8.4) there is an equality sign ($\Delta Z_A = 0$), the price e_h expresses the specific cost (per energy unit) of heat generation $(k_h)^{GT}$

$$(k_h)^{GT} = \frac{(z\rho + \delta_{serv})(J^{CHP} + J^{GT}) + K_P + K^{GT}_{gas} + K^{GT}_{env} - (\Delta K)^{CHP+GT} - (\Delta K^{coal}_{env})^{CHP+GT}}{Q_A} - \frac{(E^{GT}_{el,A} \mp \Delta E^{ST}_{el,A})(e_{el})^{mod} + (E_{el,A})^{ex}[(e_{el})^{mod} - (e_{el})^{ex}]}{Q_A} \tag{8.5}$$

while the total of the costs

$$\Delta K^{El}_A = (z\rho + \delta_{serv})(J^{CHP} + J^{GT}) + K_P + K^{GT}_{gas} + K^{GT}_{env} - (\Delta K^{coal})^{CHP+GT} - (\Delta K^{coal}_{env})^{CHP+GT} \tag{8.6}$$

constitutes the increase of the annual operating costs of a power unit associated with its repowering , i.e. costs that are related to development of a new heating and gas system minus the positive economic effects in its coal-fired section where:

J^{CHP} turnkey investment for modernization of a system to a combined heat and power,

J^{GT} turnkey investment for repowering a unit with a gas turbine and heat recovery steam generator,

K_p power needed to supply pumps necessary to put in motion the pumps in the main heat distribution pipeline and the accessory ones and to maintain adequate static pressure of water in heat distribution pipeline,

K^{GT}_{gas} cost of natural gasused in the gas turbine,

K^{GT}_{env} environmental charges incurred from gas combustion,

$(\Delta K^{coal})^{CHP+GT}$ decrease of the cost of coal purchased for the repowered unit,

$(\Delta K_{\mathrm{env}}^{\mathrm{coal}})^{\mathrm{CHP+GT}}$ decrease of the cost of environmental charges incurred due to gas combustion due to smaller volumes of coal combustion in the repowered unit,

$z\rho + \delta_{\mathrm{serv}}$ annual rate associated with handling investment capital and remaining fixed costs relative to capital expenditure (maintenance, overhauls), [1, 2].

The decrease of the costs $(\Delta K^{\mathrm{coal}})^{\mathrm{CHP+GT}}$, $(\Delta K_{\mathrm{env}}^{\mathrm{coal}})^{\mathrm{CHP+GT}}$ are relative to the power of the gas turbogenerator and the structure of the heat recovery steam generator.

Sufficient condition of cost-effective modernization of power plant is that decrease of specific cost of energy production Δk_{el} would guarantee adequately high increase of Net Present Value $\Delta\mathrm{NPV}^{\mathrm{mod}}$, relatively short Discounted Pay-Back Period DPBP $^{\mathrm{mod}}$ associated with expenditure on investment on modernisation and relatively high Internal Rate of Return IRR of this investments [1, 2]. As a rule investors expects higher rate of return that investment on capital market, due to the higher risk associated with this sort of investment.

For the case of only combined heat and power without repowering a power unit by a gas turbine and heat recovery steam generator the specific cost of heat generation is expressed by the following equations (Sect. 2.2):

- for combined heat and power without compensation of its electric capacity this cost comes from the relation

$$\begin{aligned}(Z_{\mathrm{A}})^{\mathrm{mod}} &= Q_{\mathrm{A}} e_{\mathrm{h}} + [(E_{\mathrm{el,A}})^{\mathrm{ex}} - \Delta E_{\mathrm{el,A}}^{\mathrm{El}}](e_{\mathrm{el}})^{\mathrm{mod}} - [(K_{\mathrm{A}}^{\mathrm{El}})^{\mathrm{ex}} + \Delta K_{\mathrm{A}}^{\mathrm{El}}] \geq (Z_{\mathrm{A}})^{\mathrm{ex}} \\ &= (E_{\mathrm{el,A}})^{\mathrm{ex}} (e_{\mathrm{el}})^{\mathrm{ex}} - (K_{\mathrm{A}}^{\mathrm{El}})^{\mathrm{ex}}\end{aligned} \tag{8.7}$$

and is equal to

$$k_{\mathrm{h}} = \frac{(z\rho + \delta_{\mathrm{serv}}) J^{\mathrm{CHP}} + K_{\mathrm{P}} + \Delta E_{\mathrm{el,A}}^{\mathrm{El}} (e_{\mathrm{el}})^{\mathrm{mod}} - (E_{\mathrm{el,A}})^{\mathrm{ex}} [(e_{\mathrm{el}})^{\mathrm{mod}} - (e_{\mathrm{el}})^{\mathrm{ex}}]}{Q_{\mathrm{A}}} \tag{8.8}$$

where:

$\Delta E_{\mathrm{el,A}}^{\mathrm{El}}$ annual decrease of net production of electrical energy in a power unit as a result of its operation in a combined heat and power,

- for the combined heat and power with the compensation of electrical capacity this cost results from the relation

$$\begin{aligned}(Z_{\mathrm{A}})^{\mathrm{mod}} &= Q_{\mathrm{A}} e_{\mathrm{h}} + (E_{\mathrm{el,A}})^{\mathrm{ex}} (e_{\mathrm{el}})^{\mathrm{mod}} - [(K_{\mathrm{A}}^{\mathrm{El}})^{\mathrm{ex}} + \Delta K_{\mathrm{A}}^{\mathrm{El}}] \\ &\geq (E_{\mathrm{el,A}})^{\mathrm{ex}} (e_{\mathrm{el}})^{\mathrm{ex}} - (K_{\mathrm{A}}^{\mathrm{El}})^{\mathrm{ex}}\end{aligned} \tag{8.9}$$

and is expressed with the equation

$$(k_{\mathrm{h}})^{\mathrm{com}} = \frac{(z\rho + \delta_{\mathrm{serv}})J^{\mathrm{CHP}} + K_{\mathrm{P}} + (\Delta K^{\mathrm{coal}})^{\mathrm{com}} + (\Delta K^{\mathrm{coal}}_{\mathrm{env}})^{\mathrm{com}} - (E_{\mathrm{el,A}})^{\mathrm{ex}}[(e_{\mathrm{el}})^{\mathrm{mod}} - (e_{\mathrm{el}})^{\mathrm{ex}}]}{Q_{\mathrm{A}}} \tag{8.10}$$

where:

$(\Delta K^{\mathrm{coal}})^{\mathrm{com}}$ annual increase in the volume of purchased coal for the purposes of compensation (restoring) the original electrical energy output from a power plant $\Delta E^{\mathrm{El}}_{\mathrm{el,A}}$,

$(\Delta K^{\mathrm{coal}}_{\mathrm{env}})^{\mathrm{com}}$ annual increase of environmental charge associated with the additional combustion of chemical energy of coal for the purposes of power compensation.

In order to ensure that the modernization of a power unit accompanied by repowering it by a gas turbogenerator and a heat recovery steam generator is economically justified in comparison to its modernization to combined heat and power only, the following conditions have to be fulfilled

$$(k_{\mathrm{h}})^{\mathrm{GT}} - k_{\mathrm{h}} \leq 0 \tag{8.11}$$

$$(k_{\mathrm{h}})^{\mathrm{GT}} - (k_{\mathrm{h}})^{\mathrm{com}} \leq 0. \tag{8.12}$$

The condition (8.12) is stricter than the one in (8.11), since usually the relation $(k_{\mathrm{h}})^{\mathrm{com}} < k_{\mathrm{h}}$ is fulfilled, as the total of revenues associated with the additional chemical energy of coal combustion in a power unit for the purposes of compensation of electrical energy production is smaller than the losses in revenues from the sales of electrical energy (Eqs. 8.8, 8.10)

$$(\Delta K^{\mathrm{coal}})^{\mathrm{com}} + (\Delta K^{\mathrm{coal}}_{\mathrm{env}})^{\mathrm{com}} < \Delta E^{\mathrm{El}}_{\mathrm{el,\,A}}(e_{\mathrm{el}})^{\mathrm{mod}} \tag{8.13}$$

From the relations (8.11) and (8.12) it is possible to derive the boundary (minimum) price of electrical energy for which case the modernization of a power unit involving its repowering by a gas turbine in a parallel system will be more economically justified in comparison to its modernizationto combined heat and power only:

- for the case without the compensation of electric capacity of a power unit

$$e^{\mathrm{min}}_{\mathrm{el}} \geq \frac{(z\rho + \delta_{\mathrm{serv}})J^{\mathrm{GT}} + K^{\mathrm{GT}}_{\mathrm{gas}} + K^{\mathrm{GT}}_{\mathrm{env}} - (\Delta K^{\mathrm{coal}})^{\mathrm{CHP+GT}} - (\Delta K^{\mathrm{coal}}_{\mathrm{env}})^{\mathrm{CHP+GT}}}{E^{\mathrm{GT}}_{\mathrm{el,A}} \mp \Delta E^{\mathrm{ST}}_{\mathrm{el,A}} + \Delta E^{\mathrm{El}}_{\mathrm{el,A}}} \tag{8.14}$$

- for the case with compensation

$$e_{\rm el}^{\min} \geq \frac{(z\rho + \delta_{\rm serv})J^{\rm GT} + K_{\rm gas}^{\rm GT} + K_{\rm env}^{\rm GT} - (\Delta K^{\rm coal})^{\rm CHP+GT} - (\Delta K_{\rm env}^{\rm coal})^{\rm CHP+GT} - (\Delta K^{\rm coal})^{\rm com} - (\Delta K_{\rm env}^{\rm coal})^{\rm com}}{E_{\rm el,A}^{\rm GT} \mp \Delta E_{\rm el,A}^{\rm ST}} \tag{8.15}$$

The cost of natural gas used in gas turbine is defined by the equation:

$$K_{\rm gas}^{\rm GT} = E_{\rm ch,A}^{\rm gas} e_{\rm g} \tag{8.16}$$

where:

$E_{\rm ch,A}^{\rm gas}$ indicates annual use of the chemical energy of gas which is dependent on the capacity of gas turbine and heat recovery generator structure,

$e_{\rm g}$ specific price of gas (price per energy unit).

Decrease of cost of coal use (or its increase for the case of combined heat and power with compensation of electric power) in the existing steam boiler is equal to:

$$\Delta K^{\rm coal} = \Delta E_{\rm ch,A}^{\rm coal} e_{\rm coal} \tag{8.17}$$

where:

$\Delta E_{\rm ch,A}^{\rm coal}$ indicates annual decrease (annual increase for the case of operation with power compensation) in usage of coal chemical energy which is dependent on power of gas turbine and heat recovery steam generator structure,

$e_{\rm coal}$ specific price of coal (price per energy unit).

For the case of combined heat and power of a power unit involving its repowering by a gas turbogenerator and heat recovery steam generator instead of $\Delta E_{\rm ch,A}^{\rm coal}$ in Eq. 8.17 it is necessary to substitute the value $(\Delta E_{\rm ch,A}^{\rm coal})^{\rm CHP+GT}$ while in the system without its repowering but with power compensation $(\Delta E_{\rm ch,A}^{\rm coal})^{\rm com}$.

Environmental cost $K_{\rm env}^{\rm GT}$ for gas unit and reduction (increase for the case of operation with power compensation) of cost $\Delta K_{\rm env}^{\rm coal}$ fixed with lowered (increase for the case of operation with power compensation) amount of annually used coal in power plant are dependent on the charges imposed on the use of environment and are described by

$$K_{\rm env}^{\rm GT} = E_{\rm ch,A}^{\rm gas}\left(\rho_{\rm CO_2}^{\rm gas} p_{\rm CO_2} + \rho_{\rm CO}^{\rm gas} p_{\rm CO} + \rho_{\rm SO_2}^{\rm gas} p_{\rm SO_2} + \rho_{\rm NO_x}^{\rm gas} p_{\rm NO_x}\right), \tag{8.18}$$

$$\Delta K_{\rm env}^{\rm fuel} = \Delta E_{\rm ch,\,A}^{\rm coal}\left(\rho_{\rm CO_2}^{\rm coal} p_{\rm CO_2} + \rho_{\rm CO}^{\rm coal} p_{\rm CO} + \rho_{\rm SO_2}^{\rm coal} p_{\rm SO_2} + \rho_{\rm NO_x}^{\rm coal} p_{\rm NO_x} + \rho_{\rm dust}^{\rm coal} p_{\rm dust}\right), \tag{8.19}$$

where:

$p_{CO_2}, p_{CO}, p_{NO_x}, p_{SO_2}, p_{dust}$	charges on emission of CO_2, CO, NO_x, SO_2, dust, PLN/kg,
$\rho^{gas}_{CO_2}, \rho^{gas}_{CO}, \rho^{gas}_{NO_x}, \rho^{gas}_{SO_2}$	charges on emission of CO_2, CO, NO_x, SO_2 per unit of gas chemical energy, kg/GJ,
$\rho^{coal}_{CO_2}, \rho^{coal}_{CO}, \rho^{coal}_{NO_x}, \rho^{coal}_{SO_2}, \rho^{coal}_{dust}$	emission of CO_2, CO, NO_x, SO_2, dust per unit of coal chemical energy, kg/GJ.

For the case of combined heat and power of a power unit involving its repowering by a gas turbogenerator and a heat recovery steam generator instead of $\Delta E^{coal}_{ch,A}$ in Eq. 8.19 it is necessary to substitute the value $(\Delta E^{coal}_{ch,A})^{CHP+GT}$, while in the system without its repowering but with power compensation $(\Delta E^{coal}_{ch,A})^{com}$.

Total cost of charge imposed on the protection of natural environment in coal fired power plant is described by the equation

$$\Delta K^{coal}_{env} = \Delta K^{coal}_{env} + \Delta K^{non\text{-}fuel}_{env} \tag{8.20}$$

The cost of non-fuel factors $\Delta K^{non\text{-}fuel}_{env}$ includes cost of neutralization of ash and slag, waste disposal, water consumption and sewage production, purchase and transport of chemicals for water treatment (demineralization and decarbonization), lime meal and other chemicals for wet-flue gases desulfurization IOS and cost of carbamide for NO_x reduction system.

In quantitative notion the Eq. 8.20 in the function of rates per unit emissions $p_{CO_2}, p_{CO}, p_{NO_x}, p_{SO_2}, p_{dust}$ takes the form:

$$\begin{aligned}\Delta K^{coal}_{env} = \Delta E^{coal}_{ch,A}[0.44 + \rho^{coal}_{CO_2}(p_{CO_2} - 0.00025) + \rho^{coal}_{CO}(p_{CO} - 0.11) \\ + \rho^{coal}_{SO_2}(p_{SO_2} - 0.46) + \rho^{coal}_{NO_x}(p_{NO_x} - 0.46) + \rho^{coal}_{dust}(p_{dust} - 0.31)], \quad [PLN]\end{aligned} \tag{8.21}$$

where decrease of coal use $\Delta E^{coal}_{ch,A}$ (increase for the case of operation with power compensation) is expressed in GJ/a, and rate per units of $p_{CO_2}, p_{CO}, p_{NO_x}, p_{SO_2}, p_{dust}$, in PLN/kg. The values of emissions of CO_2, CO, NO_x, SO_2 and dust in Eq. 8.21 are equal to: $\rho^{coal}_{CO_2} = 96.35\ kg_{CO2}/GJ$, $\rho^{coal}_{CO} = 0.01\ kg_{CO}/GJ$, $\rho^{coal}_{NO_x} = 0.164\ kg_{NOx}/GJ$, $\rho^{coal}_{SO_2} = 0.056\ kg_{SO2}/GJ$ (effectiveness of wet-flue gases desulfurization system $\eta_{FGD} = 0.913$), $\rho^{coal}_{dust} = 0.007\ kg_{dust}/GJ$ (effectiveness of electrofilter $\eta_{ef} = 0.9988$), and cost of 0.44 PLN/GJ was calculated with above amounts of emission for the current rates for emission charges in force Poland now: p_{CO_2}= 0.00025 PLN/kg, $p_{CO} = 0.11$ PLN/kg, p_{NO_x}= 0.46 PLN/kg, p_{SO_2}= 0.46 PLN/kg, $p_{dust} = 0.31$ PLN/kg.

We have to keep in mind that current rates are very low and do not encourage power producers to use highly effective and ecological technologies powered by ecological fuels in domestic power plants. What needs to be remarked is that emission of CO_2, CO, NO_x, SO_2 while burning natural gas in gas turbines is much lower and equal to: $\rho_{CO_2}^{gas} = 55\ kg_{CO2}/GJ$, $\rho_{CO}^{gas} = 0.0\ kg_{CO}/GJ$, $\rho_{NO_x}^{gas} = 0.02\ kg_{NOx}/GJ$, $\rho_{SO_2}^{gas} = 0.0\ kg_{SO2}/GJ$.

The relations (8.4), (8.14) and (8.15), which additionally apply relations (8.16)–(8.21) make it possible to extend the discussion to cover the effect of various variables and parameters on the economic effectiveness of modernization. The signs in inequalities (8.4), (8.14) and (8.15) will be mainly relative to the ratios of gas to coal prices and the power of the gas turbogenerator and structure of the heat recovery steam generator . As we already know, the power of the gas turbogenerator and structure of the heat recovery steam generator determine the capital expenditure and J^{GT} and annual gas use $E_{ch,A}^{gas}$ in the turbine and annual decrease of coal use $(\Delta E_{ch,A}^{coal})^{CHP+GT}$ in the power unit (for the case when it is repowered but the compensation of electrical energy production is accompanied by an increase in the use of chemical energy of the fuel by $(\Delta E_{ch,A}^{coal})^{com}$). By applying the relations (8.4), (8.14) and (8.15) it is, importantly, possible to determine such specific values of p_{CO_2}, p_{CO}, p_{NO_x}, p_{SO_2}, p_{dust} in the function of energy carrier prices e_{el}, e_g, e_{coal} (as well as current prices), for which case the majority relations defined by these relations are fulfilled. Hence, it is possible to determine such minimum specific charges, for which the application of ecological but relatively expensive hydrocarbon based fuel (natural gas) will be economically justified besides being thermodynamically efficient.

By substitution of relations (8.16)–(8.21) into (8.4), (8.14) and (8.15) we obtain the final form of necessary conditions for the economic efficiency of profitability of modernization of the existing coal-fired power unit by its repowering by a gas turbine and a heat recovery team generator with its subsequent modernization to combined heat and power:

- the necessary condition for the minimum price e_h of heat for district heating Q_A, in order to ensure the economic effectiveness of modernization of a power unit accompanied by its repowering by a gas turbine in a parallel system

$$
\begin{aligned}
e_h \geq (k_h)^{GT} = & \frac{(z\rho + \delta_{serv})(J^{CHP} + J^{GT}) + K_P + E_{ch,A}^{gas} e_g - (\Delta E_{ch,A}^{coal})^{CHP+GT} e_{coal}}{Q_A} \\
& - \frac{(E_{el,A}^{GT,gross} \mp \Delta E_{el,A}^{ST,gross})(1 - \varepsilon_{el}^{mod})(e_{el})^{mod} + (E_{el,A})^{ex}[(e_{el})^{mod} - (e_{el})^{ex}]}{Q_A} \\
& + \frac{E_{ch,A}^{gas}\left(\rho_{CO_2}^{gas} p_{CO_2} + \rho_{CO}^{gas} p_{CO} + \rho_{SO_2}^{gas} p_{SO_2} + \rho_{NO_x}^{gas} p_{NO_x}\right)}{Q_A} \\
& - (\Delta E_{ch,A}^{coal})^{CHP+GT}\left[\frac{0.44 + \rho_{CO_2}^{coal}(p_{CO_2} - 0.00025) + \rho_{CO}^{coal}(p_{CO} - 0.11)}{Q_A}\right. \\
& \left. + \frac{\rho_{SO_2}^{coal}(p_{SO_2} - 0.46) + \rho_{NO_x}^{coal}(p_{NO_x} - 0.46) + \rho_{dust}^{coal}(p_{dust} - 0.31)}{Q_A}\right]
\end{aligned} \tag{8.22}
$$

where:

$\varepsilon_{\mathrm{el}}^{\mathrm{mod}}$ internal load of the power plant

- necessary condition for the minimum (boundary) price of electrical energy for which repowering of a coal-fired power unit with a gas turbine and heat recovery steam generator after its modernization to combined heat and power will be economically justified.
- for the case of combined heat and power of a power unit without compensation of its electric capacity that was lowered due to combined heat and power production in it

$$
\begin{aligned}
e_{\mathrm{el}}^{\min} \geq\ & \frac{(z\rho + \delta_{\mathrm{serv}})J^{\mathrm{GT}} + E_{\mathrm{ch,\,A}}^{\mathrm{gas}} e_{\mathrm{g}} - (\Delta E_{\mathrm{ch,\,A}}^{\mathrm{coal}})^{\mathrm{CHP+GT}} e_{\mathrm{coal}}}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}) + \Delta E_{\mathrm{el,\,A}}^{\mathrm{El,\,gross}}(1-\varepsilon_{\mathrm{el}}^{\mathrm{ex}})} \\
& + \frac{E_{\mathrm{ch,\,A}}^{\mathrm{gas}}\left(\rho_{\mathrm{CO_2}}^{\mathrm{gas}} p_{\mathrm{CO_2}} + \rho_{\mathrm{CO}}^{\mathrm{gas}} p_{\mathrm{CO}} + \rho_{\mathrm{SO_2}}^{\mathrm{gas}} p_{\mathrm{SO_2}} + \rho_{\mathrm{NO_x}}^{\mathrm{gas}} p_{\mathrm{NO_x}}\right)}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}) + \Delta E_{\mathrm{el,\,A}}^{\mathrm{El,\,gross}}(1-\varepsilon_{\mathrm{el}}^{\mathrm{ex}})} \\
& - (\Delta E_{\mathrm{ch,\,A}}^{\mathrm{coal}})^{\mathrm{CHP+GT}} \left[\frac{0.44 + \rho_{\mathrm{CO_2}}^{\mathrm{coal}}(p_{\mathrm{CO_2}} - 0.00025) + \rho_{CO}^{coal}(p_{CO} - 0.11)}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}) + \Delta E_{\mathrm{el,\,A}}^{\mathrm{El,\,gross}}(1-\varepsilon_{\mathrm{el}}^{\mathrm{ex}})} \right. \\
& \left. + \frac{\rho_{\mathrm{SO_2}}^{\mathrm{coal}}(p_{\mathrm{SO_2}} - 0.46) + \rho_{\mathrm{NO_x}}^{\mathrm{coal}}(p_{\mathrm{NO_x}} - 0.46) + \rho_{\mathrm{dust}}^{\mathrm{coal}}(p_{\mathrm{dust}} - 0.31)}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}) + \Delta E_{\mathrm{el,\,A}}^{\mathrm{El,\,gross}}(1-\varepsilon_{\mathrm{el}}^{\mathrm{ex}})} \right]
\end{aligned} \tag{8.23}
$$

where:

$\varepsilon_{\mathrm{el}}^{\mathrm{ex}}$ internal load of the power plant prior to its modernization,

- for the case with power compensation

$$
\begin{aligned}
e_{\mathrm{el}}^{\min} \geq\ & \frac{(z\rho + \delta_{\mathrm{serv}})J^{\mathrm{GT}} + E_{\mathrm{ch,\,A}}^{\mathrm{gas}} e_{\mathrm{g}} - [(\Delta E_{\mathrm{ch,\,A}}^{\mathrm{coal}})^{\mathrm{CHP+GT}} + (\Delta E_{\mathrm{ch,\,A}}^{\mathrm{coal}})^{\mathrm{com}}] e_{\mathrm{coal}}}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}})} \\
& + \frac{E_{\mathrm{ch,\,A}}^{\mathrm{gas}}\left(\rho_{CO_2}^{\mathrm{gas}} p_{\mathrm{CO_2}} + \rho_{\mathrm{CO}}^{\mathrm{gas}} p_{\mathrm{CO}} + \rho_{\mathrm{SO_2}}^{\mathrm{gas}} p_{\mathrm{SO_2}} + \rho_{\mathrm{NO_x}}^{\mathrm{gas}} p_{\mathrm{NO_x}}\right)}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}})} \\
& - \left[(\Delta E_{\mathrm{ch,\,A}}^{\mathrm{coal}})^{\mathrm{CHP+GT}} + (\Delta E_{\mathrm{ch,\,A}}^{\mathrm{coal}})^{\mathrm{com}}\right] \left[\frac{0.44 + \rho_{\mathrm{CO_2}}^{\mathrm{coal}}(p_{\mathrm{CO_2}} - 0.00025) + \rho_{\mathrm{CO}}^{\mathrm{coal}}(p_{\mathrm{CO}} - 0.11)}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}})} \right. \\
& \left. + \frac{\rho_{\mathrm{SO_2}}^{\mathrm{coal}}(p_{\mathrm{SO_2}} - 0.46) + \rho_{\mathrm{NO_x}}^{\mathrm{coal}}(p_{\mathrm{NO_x}} - 0.46) + \rho_{\mathrm{dust}}^{\mathrm{coal}}(p_{\mathrm{dust}} - 0.31)}{(E_{\mathrm{el,\,A}}^{\mathrm{GT,\,gross}} \mp \Delta E_{\mathrm{el,\,A}}^{\mathrm{ST,\,gross}})(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}})} \right].
\end{aligned} \tag{8.24}
$$

Equations 8.22–8.24 after the prior calculation of value of $E_{\mathrm{el,A}}^{\mathrm{GT,gross}}$, $\Delta E_{\mathrm{el,A}}^{\mathrm{ST,gross}}$, $\Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}}$, $E_{\mathrm{ch,A}}^{\mathrm{gas}}$, $(\Delta E_{\mathrm{ch,A}}^{\mathrm{coal}})^{\mathrm{CHP+GT}}$, $(\Delta E_{\mathrm{ch,A}}^{\mathrm{coal}})^{\mathrm{com}}$, Q_{A} (the above variables with the exception of $\Delta E_{\mathrm{el,A}}^{\mathrm{El,gross}}$, $(\Delta E_{\mathrm{ch,A}}^{\mathrm{coal}})^{\mathrm{com}}$ i Q_{A} are functions of the power of the gas turbogenerator and structure of heat recovery steam generator) enable the analysis of the economic effectiveness of modernization with the aid of the mathematical model of a power unit (Chap. 3) for a wide range of power output from the gas turbine and an extensive selection of possible structures to be applied in the system of heat recovery steam generator. Thus, it is possible to establish an optimum solution , i.e. such power of a gas turbine and structure of heat recovery steam generator for which the values $(k_\mathrm{h})^{\mathrm{GT}}$, $e_{\mathrm{el}}^{\min}$ will assume the lowest possible values. The economic efficiency of the operation of a modernized 370 MW unit will be the greatest in this case. Subsequently, it is possible to establish the specific values of $p_{\mathrm{CO_2}}$, p_{CO}, $p_{\mathrm{NO_x}}$, $p_{\mathrm{SO_2}}$, p_{dust} charges in the function of energy carrier prices e_{el}, e_{g}, e_{coal}, for which case the modernization will ensure its profitability.

However, one has to bear in mind that in order to ensure that the application of a gas turbine is economically justified, the boundary price $e_{\mathrm{el}}^{\min}$ has to be at least not lower than the current sales price $(e_{\mathrm{el}})^{\mathrm{ex}}$ of electrical energyfrom a power unit, $e_{\mathrm{el}}^{\min} \le (e_{\mathrm{el}})^{\mathrm{ex}}$.

8.2 Calculation Results of Technical and Economical Effectiveness

With the application of the model of the power unit from Chap. 3 and methodology from Sect. 8.1 an analysis was made of a system involving a dual-pressure heat recovery steam generator with an additional closed feedwater heater for low-pressure regeneration. This heater is situated in the rear of the HRSG in the part with low-temperature exhaust gases (Fig. 3.4). The calculations W501F (SGT6-5000F) gas turbogenerator was adopted with the rated electrical energy output of $N_{\mathrm{el,n}}^{\mathrm{GT}}$= 202 MW, rated gas temperature from turbine exhaust of $t_{\mathrm{out,n}}^{\mathrm{GT}}$= 578°C and rated efficiency of electrical energy production of $\eta_{\mathrm{GT,n}} = 38.1\%$. The temperature of the exhaust gases from the heat recovery steam generator was assumed to be constant at $t_{\mathrm{out}}^{\mathrm{HRSG}}$= 90°C. The following estimation of capital expenditure necessary for the modernization of the power unit was taken: expenditure associated with repowering a unit with a gas turbine and heat recovery steam generator $J^{\mathrm{GT}} = 375$ mln PLN, expenditure for the modernization of the power unit to combined heat and power $J^{\mathrm{CHP}} = 145.6$ mln PLN (Chap. 7). The expenditure J^{GT} account for the purchase price of the new condenser (KQ1) and low-pressure section of the steam turbine (LP) with an increased flow system though it and the new electrical generator driven by the turbine with the output of 420 MW as well as the construction cost to be covered for installing it. The purchase of a new generator (around € 6,000,000) is even 50% cheaper than the cost necessary for modernization of an old one. The rate $z\rho + \delta_{\mathrm{serv}}$ [1, 2] was assumed to be equal to 16%.

The production of heat in the power plant for the purposes of central heating and network hot water were adopted in accordance with the qualitative regulation and annual scheduled chart of demand for thermal power presented in

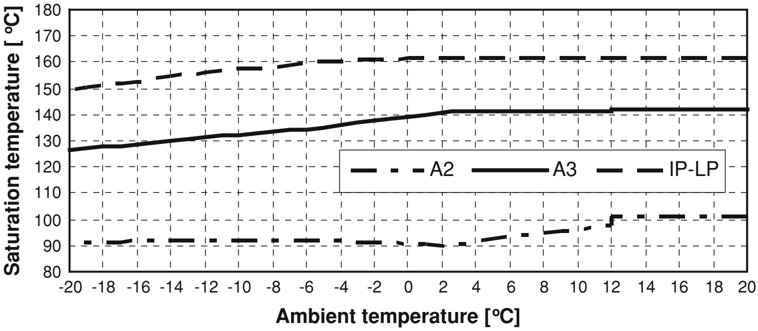

Fig. 8.1 Temperatures of steam saturation after modernization of the power unit in the function of ambient temperatures (*A2* temperature in extraction A2, *A3* temperature in extraction A3, *IP–LP* temperature in IP–LP crossoverpipe)

Figs. 4.1 and 6.2 in Chap. 4 and Sect. 6.1.3, respectively. In XC2 heater (Fig. 3.4) network hot water is heated to the temperature of 70°C, while in XC3 heater to 90°C, and in XC4 heater to 135°C. Figure 8.1 presents the distribution of steam saturation temperatures in A2, A3 extractions and in IP–LP crossoverpipe that supplies these heaters in the function of ambient temperature.

These distributions result from steam bleed for the purposes of district heating in accordance with the qualitative regulation chart of thermal power supplied from the power plant. They play a fundamental role since they determine the applicability of particular extractions for the purposes of central heating and network hot water. From Fig. 8.1 it stems that the temperatures of extracted steam saturation within the entire range of ambient temperatures guarantee that the network hot water is heated to the required temperatures in XC2, XC3 and XC4 heaters.

In thermal calculations an assumption was made that the fresh steam stream fed into the steam turbine assumes a constant value that is regardless of the ambient temperature, i.e. $\dot{m}_1 = 320$ kg/s. This is the total steam production in the boiler prior to the repowering. Following it this stream is given by the total of production of fresh steam in the existing coal-fired boiler and heat recovery steam generator (Fig. 3.4, $\dot{m}_{19} = 247$ kg/s, $\dot{m}_{160} = 73$ kg/s), which offers benefits from the thermal perspective. The greater the reduction of the workload on the coal-fired boiler (which is the source of greatest exergy losses in the system) that is taken over by the heat recovery steam generator, the higher the efficiency of electrical energy production in the repowered unit [2]. The only limitation is associated with the minimum required input into the coal-fired boiler. The mean annual efficiency of heat and power generation in the power unit is equal to $\eta_c = 55\%$ (Fig. 8.2), and its incremental and apparent efficiency (Sects. 2.3, 2.4) are equal to $\eta_\Delta = 44\%$, $\chi = 54\%$, respectively.

The ratio of annual use o chemical energy of the gas in a 202 MW turbine to the annual decrease of the use of the chemical energy of the coal in the steam boiler is equal to $E^{\text{gas}}_{\text{ch, A}}/\Delta E^{\text{coal}}_{\text{ch, A}} = 2.9$. This decrease (Fig. 8.3) results from the reduction in production of fresh steam in the boiler by $\dot{m}_{160}$= 73 kg/s.

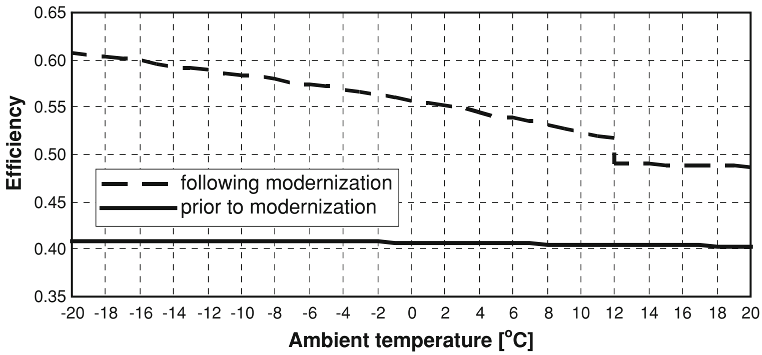

Fig. 8.2 Gross energy efficiency of the power unit in the function of ambient temperatures

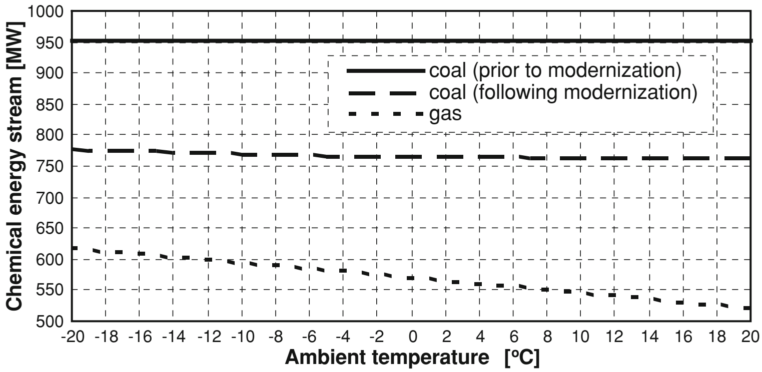

Fig. 8.3 Chemical energy streams of coal combustion in the boiler and steam of gas in the gas turbine in the function of ambient temperatures

As a consequence of repowering though despite the extraction of heating steam to feed XC2, XC3, XC4 heaters (Fig. 8.4), mean annual power output increases by $\Delta N^{ST} = 17$ MW (Fig. 8.5); hence the necessity purchase of a new electric 420 MW generator.

This is due to around 60% decrease of steam extraction into low-pressure regenerative heaters : XN1, XN2, XN3, XN4 (Figs. 8.6, 8.7) due to the partial replacement of it by regeneration in the heat recovery steam generator and smaller flow of the condensate from KQ1 condenser. This smaller stream comes as a consequence of above mentioned regeneration in the heat recovery steam generator as well as the combined heat and power of the power unit (Figs. 3.4, 8.8). Additionally, the production of intermediate-pressure steam in the heat recovery steam generator, equal to $\dot{m}_{161} = 8$ kg/s contributes to an increase of the power output of the steam turbogenerator.

Figure 8.9 presents the specific cost of heat production $(k_h)^{GT}$ in the repowered unit (Eq. 8.5, Sect. 8.1). When the sales price of electrical energy following the repowering does not change and is equal to the one prior to its repowering

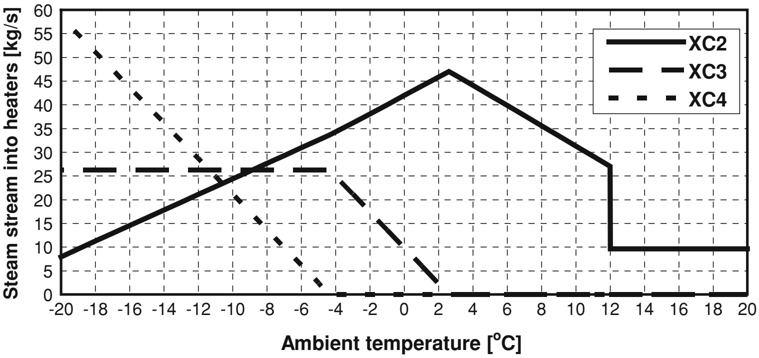

Fig. 8.4 Extraction steam streams into heaters in the function of ambient temperatures

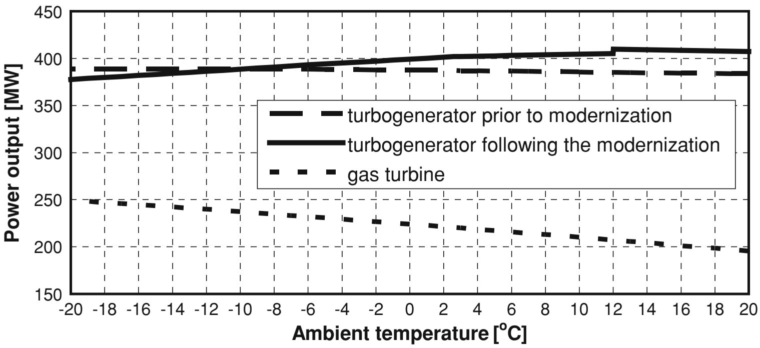

Fig. 8.5 Power of steam and turbine turbogenerator in the function of ambient temperatures

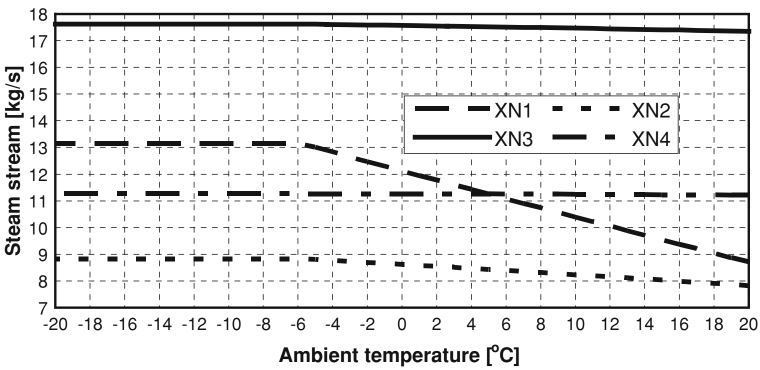

Fig. 8.6 Extraction steam feed into low-pressure regeneration heaters prior to modernization of the power unit in the function of ambient temperature

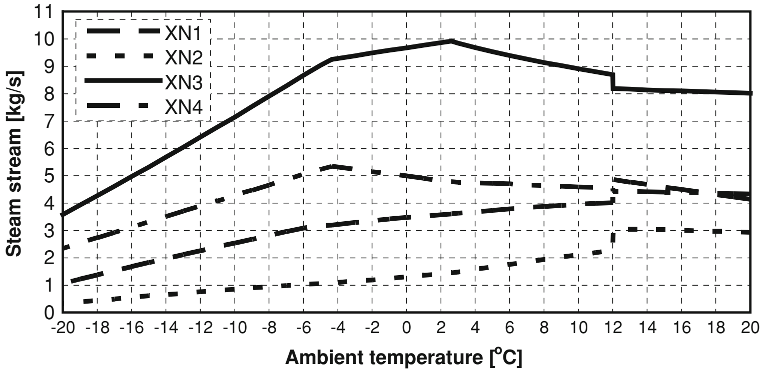

Fig. 8.7 Extraction steam feed into low-pressure regeneration heaters following modernization of the power unit in the function of ambient temperature

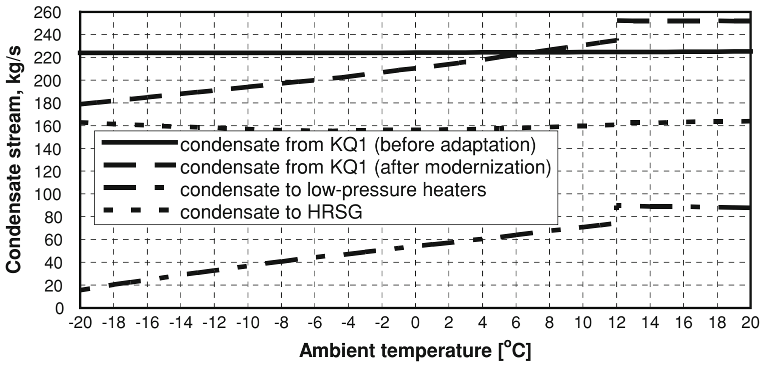

Fig. 8.8 Condensate from KQ1 condenser in the function of ambient temperatures (total stream of condensate from KQ1 after modernization is equal to the total condensate feed into HRSG and condensate into low-pressure regeneration heater)

$(e_{el})^{ex}$ = 170 PLN/MWh, the cost $(k_h)^{GT}$ for the current emission charges: p_{CO_2}= 0.00025 PLN/kg, p_{CO} = 0.11 PLN/kg, p_{NO_x}= 0.46 PLN/kg, p_{SO_2}= 0.46 PLN/kg, p_{dust} = 0.31 PLN/kg, and the current prices of imported gas e_g = 28 PLN/GJ and coal e_{coal} = 11.4 PLN/GJ is equal to as much as 70 PLN/GJ. In order to ensure that this cost were to be equal to the cost of heat generation k_h = 13.6 PLN/GJ in a combined heat and power system without the concurrent repowering by a gas turbine (Eq. 8.8, Sect. 8.1), the price of electrical energy would have to rise to reach the value $(e_{el})^{mod}$ = 196.6 PLN/MWh (Fig. 8.9). However, one can note that for this electrical energy price , a decrease of cost k_h will follow due to a positive sign of the term $(E_{el,A})^{ex}[(e_{el})^{mod} - (e_{el})^{ex}]$ in Eq. 8.8. For this reason, it is more beneficial to generate heat without repowering a unit by a gas turbine.

In order to ensure that the situation were to reverse (Eqs. 8.11, 8.13 and Sect. 8.1), the electrical energy price would have to increase even further. This price is

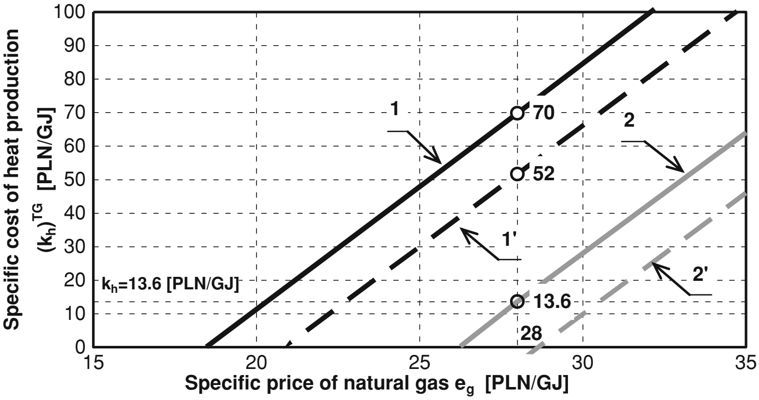

Fig. 8.9 Specific cost of heat generation $(1.1' - e_{el} = 170\ \mathrm{PLN/MWh};\ 2.2' - e_{el} = 196.6\ \mathrm{PLN/MWh};\ 1.2 - p_{NO_x} = p_{SO_2} = 0.46\ \mathrm{PLN/kg};\ 1'.2' - p_{NO_x} = p_{SO_2} = 46\ \mathrm{PLN/kg})$

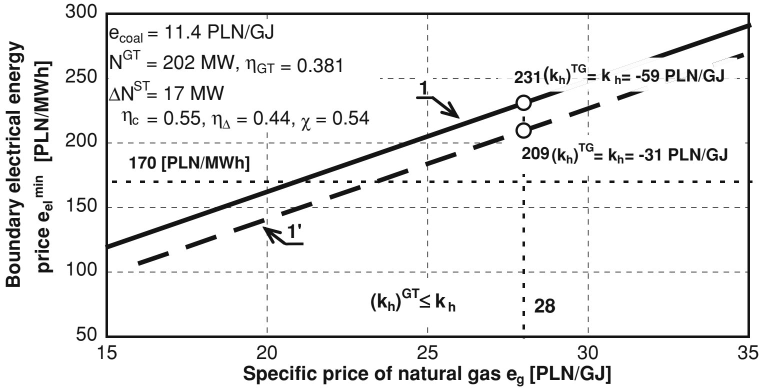

Fig. 8.10 Boundary electrical energy price $(1 - p_{NO_x} = p_{SO_2} = 0.46\ \mathrm{PLN/kg};\ 1' - p_{NO_x} = p_{SO_2} = 46\ \mathrm{PLN/kg})$

presented in Fig. 8.10 in the function of gas prices . In the examined case this price is equal to 231 PLN/MWh. The cost of heat generation in a repowered system and one without a gas turbine assumes a negative value of $(k_h)^{GT} = k_h = -59$ PLN/GJ. This is so because the increase in the revenue from electrical energy sales for this value exceeds the increase of annual operating cost of a repowered unit.

8.3 Summary and Conclusions

For the current price relations between energy carriers the modernization of a 370 MW coal-fired power unit involving its modernization to combined heat and power and repowering by a 200 MW gas turbogenerator is not economically justified. However, the modernization of this power unit only to combined heat and power yields considerable profits (Chap. 6). However, the repowering itself is very effective from the thermal dynamics perspective. This is not only due to modernization of a power unit to combined heat and power. It comes as a conse-quence of taking over by the heat recovery steam generator from the BP-1150 coal-fired boiler the partial production of fresh steam in the total of $\dot{m}_{160}$= 73 kg/s with the same parameters: temperature $t_{160} = t_1 = 540°C$ and pressure $p_{160} = p_1 = 18.3$ MPa (Fig. 3.4). This production is possible for the case of very low tempera-ture of exhaust gases from the gas turbine equal to $t^{GT}_{out,n}$= 578°C and small temperature increments, i.e. ones equal only to several degree Celsius [2]. The exhaust gases with the temperature of combustion of fine coal-air mixture of 1,300°C in the combustion chamber are applied for the production of steam (for the net calorific value of hard coal of 21.8 MJ/kg). This is accompanied by high temperature increments, as a result of which the efficiency of electrical energy generation in the power unit is low. For instance, the temperature difference between the combustion temperature and the temperature produced in the fresh steam boiler is equal to nearly 800°C (760 = 1,300 − 540), which leads to considerable exergy losses of the heat stream produced from coal combustion. Hence, the gross efficiency of electrical energy production in the power unit is equal to $\eta_{el} = 41\%$ (while net one to 37%). After the repowering the incremental efficiency of the steam tur-bogenerator, which is the derivate of the former, is equal to $\chi = 54\%$.

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2. Bartnik R (2009) Combined cycle power plants, thermal and economic effectiveness. WNT, Warszawa (in Polish)

Chapter 9
Technical and Economical Effectiveness of 370 MW Power Unit Repowered by Gas Turbine in Parallel System

A promising potential for the modernization of the power industry is associated with the so-called clean coal technologies based on gas turbines, since it is characterized by relatively high energy effectiveness and low emission of pollutants to the environment. Such technologies include, among the others, dual-fuel combined-cycle technologies in series or parallel systems. It is really important to note that the systems can be created on the basis of the presently existing coal structures by adding a gas turbine. In practice, the parallel system is more energetically and economically effective method of a power plant modernization [1]. This is why for such a system in this monograph presented are energetic and economic effectiveness of 370 MW power block repowered by 200 MW gas turbine with dual pressure heat recovery steam generator (Fig. 3.4).

9.1 Methodology of Economical Analysis of Modernization 370 MW Power Unit Repowered by Gas Turbine in Parallel System

Economic effectiveness of the modernization of an existing coal fired power plant by repowering it by gas turbine and heat recovery steam generator is equivalent with at least not increasing its specific cost of energy production k_{el}

$$(k_{el})^{mod} = \frac{(K_A)^{ex} + \Delta K_A}{(E_{el,A})^{ex} + \Delta E_{el,A}} \leq (k_{el})^{ex} = \frac{(K_A)^{ex}}{(E_{el,A})^{ex}} \tag{9.1}$$

where:

$(E_{el,A})^{ex}$, $\Delta E_{el,A}$ successive annual production of net electrical energy before modernization and increase of net production after modernization,

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$(K_A)^{ex}$ annual cost of operation of power plant before modernization,
ΔK_A increase of annual cost of operation of power plant after modernization.

9.1.1 Necessary Conditions of Cost-Effective Modernization

Necessary condition of cost-effective modernization can be described by the relation

$$\Delta k_{el} = (k_{el})^{mod} - (k_{el})^{ex} \le 0 \tag{9.2}$$

Another equivalent interpretation of relation (9.2) is the condition of increase of annual gross income attainable by operation of power plant after modernisation. This increase should be at least non-negative, which can be defined by the relation (compare Eqs. 2.10a, 2.22b, 8.3):

$$\begin{aligned}\Delta Z_A &= (Z_A)^{mod} - (Z_A)^{ex}\\ &= \left[(E_{el,A})^{ex} + \Delta E_{el,A}\right]\left[(e_{el})^{mod} - (k_{el})^{mod}\right] - (E_{el,A})^{ex}\left[(e_{el})^{ex} - (k_{el})^{ex}\right]\\ &= (E_{el,A})^{ex}\left[(e_{el})^{mod} - (e_{el})^{ex}\right] + \Delta E_{el,A}(e_{el})^{mod} - \Delta K_A \ge 0,\end{aligned} \tag{9.3}$$

where:

$(e_{el})^{ex}$, $(e_{el})^{mod}$ in succession selling price of electrical energy before and after modernization. Currently over-year average price $(e_{el})^{ex}$ is 170 PLN/MWh

If, the increase of profit ΔZ_A (9.3) is lower (higher when $(e_{el})^{mod} > (e_{el})^{ex}$ for value $[(E_{el,A})^{ex} + \Delta E_{el,A}][(e_{el})^{mod} - (e_{el})^{ex}]$ than increase of profit in case when price of electrical energy would not have changed, that is, when $(e_{el})^{ex} = (e_{el})^{mod}$. In that case increase of profit would be

$$\begin{aligned}\Delta Z_A &= (E_{el,A})^{ex}\left[(k_{el})^{ex} - (k_{el})^{mod}\right] + \Delta E_{el,A}\left[(e_{el})^{ex} - (k_{el})^{mod}\right]\\ &= \Delta E_{el,A}(e_{el})^{ex} - \Delta K_A.\end{aligned} \tag{9.4}$$

If reducing the price of electrical energy would even lead to the decrease of the specific cost of power production, $(e_{el})^{mod} - (e_{el})^{ex} = (k_{el})^{mod} - (k_{el})^{ex}$, the increase of profit would be low and equal

$$\Delta Z_A = \Delta E_{el,A}[(e_{el})^{ex} - (k_{el})^{ex}] \tag{9.5}$$

and Discounted Pay-Back Period of investment necessary for modernization DPBP $^{\mathrm{mod}}$ would be long, making the investment non-attractive from the economic point of view.

As Eq. 9.3 shows, increase of profit $\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}\left[(e_{\mathrm{el}})^{\mathrm{mod}} - (e_{\mathrm{el}})^{\mathrm{ex}}\right] + \Delta E_{\mathrm{el,A}}(e_{\mathrm{el}})^{\mathrm{mod}}$ has to completely cover the increase of annual cost of operation of power plant ΔK_{A}, that is costs attached with newly-built gas turbine, lowered by additional economic effects of coal powered turbine due to its modernization. Increase of those costs is described by equation

$$\begin{aligned}\Delta K_{\mathrm{A}} &= (K_{\mathrm{A}})^{\mathrm{mod}} - (K_{\mathrm{A}})^{\mathrm{ex}} \\ &= (z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_{\mathrm{gas}}^{\mathrm{GT}} + K_{\mathrm{env}}^{\mathrm{GT}} - \Delta K^{\mathrm{coal}} - \Delta K_{\mathrm{r,m,w}}^{\mathrm{coal}} - \Delta K_{\mathrm{env}}^{\mathrm{coal}},\end{aligned} \tag{9.6}$$

where:

J_{mod}	capital costs of modernization of power plant,
$K_{\mathrm{gas}}^{\mathrm{GT}}$	cost of natural gas burned in gas turbine,
$K_{\mathrm{env}}^{\mathrm{GT}}$	economic cost of natural environment usage in process of burning natural gas in gas turbine,
ΔK^{coal}	decrease of cost of coal purchase,
$\Delta K_{\mathrm{r,m,w}}^{\mathrm{coal}}$	in up-to-date coal-fired steam turbine decrease of costs of necessary maintenance and renovation, non-energetic resources and auxiliary materials, supplementary water; in calculation we can assume without making a major mistake that $\Delta K_{\mathrm{r,m,w}}^{\mathrm{coal}} = 0$,
$\Delta K_{\mathrm{env}}^{\mathrm{coal}}$	decrease of economic costs of natural environment exploitation due to decrease of annual coal usage in power plant,

Using relation (9.3) we can determine the relation stating the boundary (minimal) price of electrical energy, for what modernization of power plant would be profitable, it is for what increase of gross profit ΔZ_{A} from project would be at least non-negative.

$$e_{\mathrm{el}}^{\mathrm{min}} = \frac{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}(e_{\mathrm{el}})^{\mathrm{ex}} + \Delta K_{\mathrm{A}}}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}} + \Delta E_{\mathrm{el,A}}}. \tag{9.7}$$

One needs to say that the price $e_{\mathrm{el}}^{\mathrm{min}}$ determined from the Eq. 9.7 constitutes the price at which the total output of electrical energy $\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}} + \Delta E_{\mathrm{el,A}}$ produced in modernized power plant is sold. This means that the price that has to be valid so that the profit obtained from work of modernized power plant is not lowered in comparison to the profit obtained before modernization, i.e. $(Z_{\mathrm{A}})^{\mathrm{mod}} = (Z_{\mathrm{A}})^{\mathrm{ex}}$. Increase of the annual costs ΔK_{A} is covered not only by the revenues from $\Delta E_{\mathrm{el,A}}\, e_{\mathrm{el}}^{\mathrm{min}}$ (Eq. 9.4), but additionally from the revenues form $(E_{\mathrm{el,A}})^{\mathrm{ex}}[e_{\mathrm{el}}^{\mathrm{min}} - (e_{\mathrm{el}})^{\mathrm{ex}}]$ (Eq. 9.3).

The price of electrical energy sold from power plant after modernization $(e_{\mathrm{el}})^{\mathrm{mod}}$ of course needs to be not lower than $e_{\mathrm{el}}^{\mathrm{min}}$. Hence, the actual condition is that

$$(e_{\mathrm{el}})^{\mathrm{mod}} \geq e_{\mathrm{el}}^{\min}. \tag{9.8}$$

If the price of electrical energy sales after modernization has not changed, $(e_{\mathrm{el}})^{\mathrm{mod}} = (e_{\mathrm{el}})^{\mathrm{ex}}$ then from Eq. 9.4 we get the necessary condition

$$e_{\mathrm{el}}^{\min} = \frac{\Delta K_{\mathrm{A}}}{\Delta E_{\mathrm{el,A}}} = \frac{(z\rho + \delta_{\mathrm{serv}})J_{\mathrm{mod}} + K_{\mathrm{gas}}^{\mathrm{GT}} + K_{\mathrm{env}}^{\mathrm{GT}} - \Delta K^{\mathrm{coal}} - \Delta K_{\mathrm{env}}^{\mathrm{coal}}}{\Delta E_{\mathrm{el,A}}} \tag{9.9}$$

and we do not need to be familiar with the price $(e_{\mathrm{el}})^{\mathrm{ex}}$ and we do not need to know the production rate $\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}$ before modernization, which enables the analysis of economical effectiveness of power plant modernization by using only increase rates: increase of annual cost of operation of power plant ΔK_{A} and increase of net production after modernization $\Delta E_{\mathrm{el,A}}$. Therefore, incremental method offer only advantages and hence is comfortable, as it does not require the analysis of the current state of modernized power plant. Even if it were required to invest some money in the renovation, we could add that cost to J_{mod} (we need to remember that modernization of power plant make sense only when its technical condition allows long-term operation). The only necessary cost needed to calculate the threshold level for the effectiveness of modernization is associated with the annual cost of the operation of the new installed gas turbine (generally operation costs of new equipment) and reduction of operation costs of the existing coal-powered turbine, mainly due to decrease of coal purchase cost. However, we need to remember that the price $e_{\mathrm{el}}^{\min}$determined from Eq. 9.9 applies only to the sales of price of increased energy output $\Delta E_{\mathrm{el,A}}$, whereas price determined from Eq. 9.7 is a weighted mean of the price determined from Eq. 9.9 and price before modernization $(e_{\mathrm{el}})^{\mathrm{ex}}$, so it forms the minimal price of the sales of the total electrical energy production in the modernized power plant, i.e. the price for which profit from its operation would not decrease. When we enter symbol $e_{\mathrm{el}}^{\min}$ as $e_{\mathrm{el}}^{\min,\Delta E_{\mathrm{el,A}}}$ into Eq. 9.9 and substituting it into Eq. 9.7, we get the marginal price which actually is the weighted average of prices $(e_{\mathrm{el}})^{\mathrm{ex}}$ and $e_{\mathrm{el}}^{\min,\Delta E_{\mathrm{el,A}}}$

$$e_{\mathrm{el}}^{\min\,\mathrm{w.a.}} = \frac{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}} + \Delta E_{\mathrm{el,A}}}(e_{\mathrm{el}})^{\mathrm{ex}} + \frac{\Delta E_{\mathrm{el,A}}}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}} + \Delta E_{\mathrm{el,A}}} e_{\mathrm{el}}^{\min,\Delta E_{\mathrm{el,A}}}. \tag{9.10}$$

In the circumstances of high gas price (currently gas unit is three times more expensive than coal unit) condition in (9.9) is stricter that the condition in (9.7), i.e. the price $e_{\mathrm{el}}^{\min}$ calculated from (9.9) is higher than price calculated from (9.7) and, additionally, both prices are higher than $(e_{\mathrm{el}})^{\mathrm{ex}}$ (Fig. 9.1); in addition, the increase of annual costs ΔK_{A} is covered not only by income $\Delta E_{\mathrm{el,A}} e_{\mathrm{el}}^{\min}$, but additionally by the revenues from $\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}\left[e_{\mathrm{el}}^{\min} - (e_{\mathrm{el}})^{\mathrm{ex}}\right]$. When gas to coal price ratio is relatively low, and falls below about 2, then situation is reversed, the price is calculated from Eq. 9.7 is higher than the one from (9.9), because in these conditions they are lower than $(e_{\mathrm{el}})^{\mathrm{ex}}$ (Fig. 9.1).

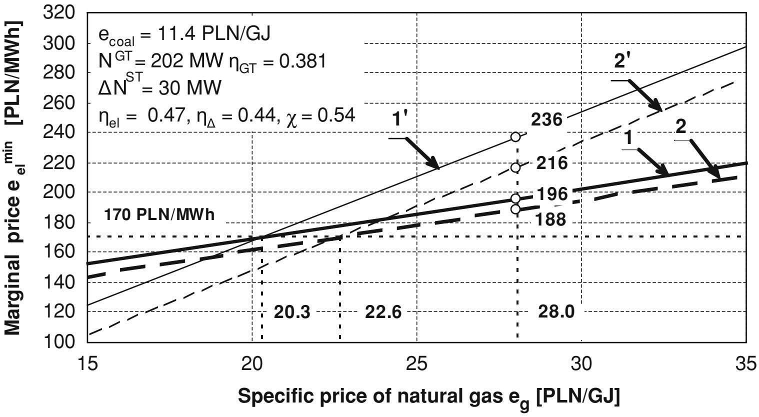

Fig. 9.1 Marginal price of electrical energy (*1*, *2* minimal weighted mean price of the sales of electrical energy produced in power plant; *1′*, *2′* minimal price on the sale of electrical energy produced in gas turbine and increase in steam turbine; *1*, *1′* $p_{NO_x} = p_{SO_2} = 0.46\text{PLN/kg}$; *2*, *2′* $p_{NO_x} = p_{SO_2} = 46\ \text{PLN/kg}$)

As a result of equalities (9.7) and (9.9) it is clear that the boundary price of electrical energy e_{el}^{min} is relative to investment costs J_{mod} (hence, relative to the capacity of gas turbine N_{el}^{GT} and structure of heat recovery), the cost of natural gas and decrease of the cost of coal purchase and environmental charges.

The increase of annual net production of electrical energy in modernized power plant is equal to:

$$\Delta E_{el,A} = \left(E_{el,A}^{GT,gross} + \Delta E_{el,A}^{ST,gross}\right)\left(1 - \varepsilon_{el}^{mod}\right) \tag{9.11}$$

where:

$E_{el,A}^{GT,gross}$, $\Delta E_{el,A}^{ST,gross}$ indicates annual gross production of electrical energy in gas turbine and annual gross increase of production in steam turbine, respectively,

ε_{el}^{mod} denotes the index of power station internal electrical energy use in the modernized power plant (in calculation it is assumed that $\varepsilon_{el}^{mod} = 4\%$).

The cost of natural gas used in gas turbine , decrease of cost of coal used in existing steam boiler, environmental cost K_{env}^{GT} for gas unit and reduction of cost ΔK_{env}^{coal} fixed with lowered amount of annually used coal in power plant dependent on the charges imposed on the use of environment, total cost of charge imposed on the protection of natural environment in coal fired power plant, the cost of non-fuel factors $\Delta K_{env}^{non\text{-}fuel}$ are defined by the Eqs. 8.16–8.21 in Sect. 8.1.1.

Substituting Eqs. 9.11, 8.16–8.21 into relation (9.7) we will obtain the final necessary condition for the economic effectiveness of conventional coal fired condensing power plant repowered by gas turbine in parallel system

$$
\begin{aligned}
e_{\mathrm{el}}^{\min} \geq\; & \frac{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}\left(e_{\mathrm{el}}\right)^{\mathrm{ex}}+(\mathrm{z}\rho+\delta_{\mathrm{serv}})J_{\mathrm{mod}}+E_{\mathrm{ch,A}}^{\mathrm{gas}}e_{\mathrm{g}}-\Delta E_{\mathrm{ch,A}}^{\mathrm{coal}}e_{\mathrm{coal}}}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}+\left(E_{\mathrm{el,A}}^{\mathrm{GT,gross}}+\Delta E_{\mathrm{el,A}}^{\mathrm{ST,gross}}\right)\left(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}\right)} \\
& + \frac{E_{\mathrm{ch,A}}^{\mathrm{gas}}\left(\rho_{\mathrm{CO_2}}^{\mathrm{gas}}p_{\mathrm{CO_2}}+\rho_{\mathrm{CO}}^{\mathrm{gas}}p_{\mathrm{CO}}+\rho_{\mathrm{SO_2}}^{\mathrm{gas}}p_{\mathrm{SO_2}}+\rho_{\mathrm{NO_x}}^{\mathrm{gas}}p_{\mathrm{NO_x}}\right)}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}+\left(E_{\mathrm{el,A}}^{\mathrm{GT,gross}}+\Delta E_{\mathrm{el,A}}^{\mathrm{ST,gross}}\right)\left(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}\right)} \\
& - \Delta E_{\mathrm{ch,A}}^{\mathrm{coal}}\left[\frac{0.44+\rho_{\mathrm{CO_2}}^{\mathrm{coal}}\left(p_{\mathrm{CO_2}}-0.00025\right)+\rho_{\mathrm{CO}}^{\mathrm{coal}}\left(p_{\mathrm{CO}}-0.11\right)}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}+\left(E_{\mathrm{el,A}}^{\mathrm{GT,gross}}+\Delta E_{\mathrm{el,A}}^{\mathrm{ST,gross}}\right)\left(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}\right)}\right. \\
& \left. + \frac{\rho_{\mathrm{SO_2}}^{\mathrm{coal}}\left(p_{\mathrm{SO_2}}-0.46\right)+\rho_{\mathrm{NO_x}}^{\mathrm{coal}}\left(p_{\mathrm{NO_x}}-0.46\right)+\rho_{\mathrm{dust}}^{\mathrm{coal}}\left(p_{\mathrm{dust}}-0.31\right)}{\left(E_{\mathrm{el,A}}\right)^{\mathrm{ex}}+\left(E_{\mathrm{el,A}}^{\mathrm{GT,gross}}+\Delta E_{\mathrm{el,A}}^{\mathrm{ST,gross}}\right)\left(1-\varepsilon_{\mathrm{el}}^{\mathrm{mod}}\right)}\right].
\end{aligned}
\tag{9.12}
$$

Relation (9.12) can enable one to open a discussion about the influence of various numbers and parameters (for example charges on the emission of CO_2, CO, NO_x, SO_2, dust) on the economic effectiveness of modernization. The sign in the minority relation in inequality (9.12) mainly depends on proportion of price of electrical energy to price of gas and coal and on the capacity of gas turbine and structure of heat recovery steam generator. Values of $E_{\mathrm{el,A}}^{\mathrm{GT,gross}}$, $\Delta E_{\mathrm{el,A}}^{\mathrm{ST,gross}}$, $\left(E_{\mathrm{el,A}}\right)^{ex}$, $E_{\mathrm{ch,A}}^{\mathrm{gas}}$, $\Delta E_{\mathrm{ch,A}}^{\mathrm{coal}}$, excluded from the function of power of gas turbine and structure of heat recovery steam generator, were calculated with aid of the mathematical block showed in Chaps. 2 and 3.

Using inequality (9.12) it is possible to determine the minimal value of specific price of $p_{\mathrm{CO_2}}$, p_{CO}, $p_{\mathrm{NO_x}}$, $p_{\mathrm{SO_2}}$, in function of energy carriers e_{el}, e_{g}, e_{coal} (therefore forming the current prices), when we will get majority relation described by this relation. It allows to establish the minimal specific price for which the use is ecological, but relatively expensive natural gas fuel in the power engineering sector will be highly effective not only in terms of thermodynamics but also from the economic point of view. It allows to find an optimal solution that is, one for which the economic effectiveness of repowering 370 MW conventional coal-fired condensing power plant by gas turbine in parallel system would be highest.

9.2 Calculation Results of Technical and Economical Effectiveness of Repowering 370 MW Power Unit by Gas Turbine in Parallel System

A unit with dual pressure heat recovery steam generator and gas turbine W501F (SGT6-5000F) were selected for analysis (Fig. 3.4) with the nominal capacity $N_{\mathrm{el,n}}^{\mathrm{GT}} = 202\ \mathrm{MW}$, nominal temperature of exhaust flue gases from turbine

$t_{\mathrm{out,n}}^{\mathrm{GT}} = 578\mathrm{C}$ and nominal efficiency of electrical energy generation $\eta_{\mathrm{GT,n}} = 38.1\%$. The investment needed for modernization of power plant was estimated at $J_{\mathrm{mod}} = 375$ mln PLN. For the purposes of estimation the price of necessary purchase of new 420 MW electric generator and the new condenser (KQ1) and low-pressure section of the steam turbine (LP) with an increased flow system though it was taken into account. Rate $z\rho + \delta_{\mathrm{serv}}$ was set at 16%. After repowering we can achieve an increase of power plant of up to 602 MW; therefore, it is higher than its initial power by over 60%. Moreover, gross effectiveness of electrical power production increases by 6% to the value $\eta_{\mathrm{el}} = 47\%$, while the increased efficiency and apparent efficiency of power plant (Chap. 2) are equal to $\eta_{\Delta} = 44\%$, $\chi = 54\%$, respectively.

The capacity of the steam turbine increases by $\Delta N^{\mathrm{ST}} = 30$ MW. This occurs as a result of about 50% decrease in the use of extracted steam into low pressure heat exchangers XN1, XN2, XN3, XN4 which were substituted by regeneration in heat recovery generator and also due to the production of intermediate pressure steam whose mass is equal to 8 kg/s.

The rate of annual use of natural gas chemical energy in 202 MW gas turbine to decrease of use of coal chemical energy in steam turbine ratio is equal to $E_{\mathrm{ch,A}}^{\mathrm{gas}} / \Delta E_{\mathrm{ch,A}}^{\mathrm{coal}} = 3$, rate of this decrease to usage of coal chemical energy before modernization is equal to $\Delta E_{\mathrm{ch,A}}^{\mathrm{coal}} / E_{\mathrm{ch,A}}^{\mathrm{coal}} = 0.2$. It comes as a result of the decrease of fresh steam production in coal boiler by 73 kg/s, which is “taken over” by heat recovery steam generator.

Figure 9.1 shows the results of the economic calculation. As it is shown for the current specific price of imported natural gas $e_{\mathrm{g}} = 28$ PLN/GJ and coal $e_{\mathrm{coal}} = 11.4$ PLN/GJ and actual charges on the emissions of CO_2, CO, NO_x, SO_2 and dust, the threshold price of electrical energy that would not generate losses after power plant modernization is equal to 196 PLN/MWh. It is higher than current average price by about 26 PLN/MWh.

9.3 Summary and Conclusions

With current price relation between different fuels, repowering conventional coal-fired condensing power plant by gas turbine in parallel system is not justified from economic point of view. Even very aggressive price politics with one hundred times increase of rates for emission of SO_2, NO_x in current situation when power plant have denitriding and desulfurization systems does not make it justify the undertaking of modernization with a view of its profitable operation, it only decreases the minimal price $e_{\mathrm{el}}^{\mathrm{min}}$of electrical energy sales from 196 to 188 PLN/MWh (Fig. 9.1). More increase of price rate for emission of CO_2 would only make things worse, despite the fact that natural gas offers almost two times lower emission rate of CO_2 than coal for one unit of energy, it needs three times more chemical energy $E_{\mathrm{ch,A}}^{\mathrm{gas}}$ to power the gas turbine. This leads to actual increase of environmental cost of CO_2. Cost-effectiveness of modernization with current rates for emission of CO_2, CO, NO_x, SO_2 and dust is only possible when after increase in the price of electrical energy by about 15% (=196/170; Fig. 9.1). For the facility in which expensive fuel is used it is also necessary to have a higher price of electrical energy produced from that expensive fuel. The reduction in the price of natural gas, and therefore decrease of gas price to coal price ratio can increase the effectiveness of modernization. The use of domestically extracted gas, which is cheaper for about 20% than imported, can assure the effectiveness of modernization in current price range of electrical energy and coal. Its needs to be stated that this type of modernization greatly increases the power output of a power plant. It would not be necessary to build highly necessary new power plants. What is more in this type of modernization is it is about four times cheaper per power unit than building a new power plant with super-critical parameters and the same effectiveness of producing electrical energy as in modernized power plant.

Reference

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Chapter 10
Summary and Final Conclusions

The conclusions resulting from the discussion along this book are summarized along the particular sections and chapters in the book. The presentation below focuses on the most important issues and general remarks arising out of the discussion.

- In the conditions of market economy the selection of a structure of heat exchangers should be based on economic criteria. Economic criterion is superior to thermodynamic design aspects. The thermal analysis (based on energy and entropy) makes it possible to seek for the opportunity of improving technology and engineering solutions to be applied in machines and facilities. In the market economy it is the economic criterion based on profit maximization that decides on justification of applying a specific technical solution and the analysis of economic viability decides on undertaking an investment. As it was indicated in this study, the economic efficiency of a power plant operating in a combined heat and power is very high.
- The presented methodology, algorithm and computer program applied for the selection of an optimum structure of heat exchangers that enable the conversion of a power plant to a combined heat and power are general in nature; thus, they can be applied with regard to power units with various capacities and various values of heat extracted from the power units. A change in the type of the power unit will only require modifications to some of the energy and mass-balance equations.

Calculations conducted for a number of alternatives with regard to a power plant for units with a rated capacity of $N_{\mathrm{el}}^{\mathrm{ST}} = 370\,\mathrm{MW}$ an optimum structure of heaters both in terms of economy and thermodynamic variables is the one based on three heaters XC2, XC3 and XC4 with steam bleed from A2, A3 extraction and IP–LP crossover pipe for the steam supply to XC4 heater in a parallel system (Fig. 6.1).

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