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Avoidable thermodynamic inefficiencies and costs in an externally fired combined cycle power plant

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Abstract

The real thermodynamic inefficiencies in a thermal system are related to exergy destruction and exergy loss. An exergy analysis identifies the system components with the highest exergy destruction and the processes that cause them. However, only a part of the exergy destruction in a component can be avoided. A minimum exergy destruction rate for each system component is imposed by physical, technological, and economic constraints. The difference between the total and the unavoidable exergy destruction rate represents the avoidable exergy destruction rate, which provides a realistic measure of the potential for improving the thermodynamic efficiency of a component.

The calculation of avoidable cost rates associated with both exergy destruction and capital investment is described in the paper and is applied to the exergoeconomic evaluation of an externally fired combined cycle power plant. For each plant component, avoidable and unavoidable exergy destructions and investment costs are calculated. The assumptions required for these calculations are discussed. Modified exergoeconomic variables assist in identifying the real potential of improving single plant components. In addition, some aspects of the design and improvement of externally fired combined cycles are discussed. The results of this study show that the concepts of avoidable exergy destruction and avoidable investment cost are very useful in designing cost-effective energy conversion systems.

Keywords: Avoidable thermodynamic inefficiencies; Cost of thermal systems; Exergoeconomic evaluation; Externally fired combined cycle

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Nomenclature

c	cost per unit of exergy (€/GJ)
$\dot{C}$	cost rate associated with an exergy stream (€/h)
d	diameter (mm)
$\dot{E}$	exergy rate (MW)
f	exergoeconomic factor
h	specific enthalpy (kJ/kg)
$\dot{H}$	enthalpy rate (MW)
$\dot{m}$	mass flow rate (kg/s)
p	pressure (bar)
$\dot{Q}$	time rate of heat transfer (MW)
S	specific entropy (kJ/kg K)
$\dot{S}$	entropy rate (MW/K)
T	temperature (K)
w	specific work (kJ/kg)
$\dot{W}$	power (MW)
v	specific volume (m^3/kg)
x, y	exergy destruction ratio
$\dot{Z}$	cost rate associated with capital investment (€/h)

Greek symbols

Δ	difference
ε	exergetic efficiency
λ	excess air fraction
η	energetic efficiency

Subscripts

a	average
c	cold stream
c	compressor
D	exergy destruction
e	exit
F	fuel (cxergy)
h	hot stream
i	inlet
k	component
L	loss
m	mean
min	minimum
p	polytropic
P	product (exergy)
q	heat transfer
s	isentropic

sat	saturated
t	turbine
tot	overall system
0	ambient conditions

Superscripts

AV	avoidable
UN	unavoidable
*	modified

1. Introduction

The real thermodynamic inefficiencies in a thermal system are related to exergy destruction and exergy loss. The exergy destruction is caused by effects such as chemical reaction, heat transfer through a finite temperature difference, mixing of matter at different compositions or states, unrestrained expansion, and friction. At any given state of technological development, some exergy destruction within a system component will always be unavoidable due to physical and economic constraints.

The purpose of the paper is to present an approach for estimating the avoidable part of exergy destruction (and the costs associated with it) as well as the avoidable part of investment cost associated with a system component.

The concepts of avoidable exergy destruction and investment cost are combined with an exergoeconomic evaluation technique. Modified exergetic and exergoeconomic variables are defined using avoidable exergy destruction rates and costs. The general procedure of calculating the avoidable part of exergy destruction rate in a system component and the avoidable part of investment cost is described in Ref. [1]. Here, this approach is extended and applied to different types of system components. Conventional exergoeconomic techniques are discussed in detail in Refs. [2,3].

This new approach is applied to the exergoeconomic evaluation of a conceptual design of an advanced externally fired combined cycle (*EFCC*). In this design, air is the working fluid in a gas turbine system, the combustion chamber of which is replaced by two high-temperature heat exchangers. A coal-fired combustion chamber is placed downstream the gas turbine and operates at nearly atmospheric pressure. Thermal energy is transferred from the hot combustion products to the compressed air in the gas turbine.

For simplicity, only steady-state processes are considered and the operating and maintenance costs are assumed to be constant and independent of the selection of the design point for the component being considered. In the following discussion and calculations, operating and maintenance costs are neglected.

2. Methodology

The exergy destruction rate $\dot{E}_{D,k}$ due to irreversibilities within the kth system component is obtained in exergoeconomics from an exergy balance

$$\dot{E}_{D,k} = \dot{E}_{F,k} - \dot{E}_{P,k} - \dot{E}_{L,k} \tag{1}$$

where $\dot{E}_{F,k}$, $\dot{E}_{P,k}$, and $\dot{E}_{L,k}$ denote the fuel, product, and exergy loss of the kth component, respectively. In this study, the thermodynamic inefficiencies of a component consist exclusively of exergy destruction ($\dot{E}_{L,k} = 0$, [1]). The unavoidable exergy destruction $\dot{E}_{D,k}^{UN}$ cannot be reduced due to technological limitations (e.g. availability and costs of materials and manufacturing methods) regardless of the amount of investment.

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{AV} + \dot{E}_{D,k}^{UN} \tag{2}$$

In practical applications, the unavoidable exergy destruction per unit of product exergy $(\dot{E}_D/\dot{E}_P)_k^{UN}$ is determined by appropriately selecting the most important thermodynamic parameters of the kth component to obtain an extremely low minimum exergy destruction rate that just could be realized in the foreseeable future. Such a design will necessarily have a very large investment cost. For a design A of the same type of component with a value of the exergetic product $\dot{E}_{P,k,A}$, the ratio $(\dot{E}_D/\dot{E}_P)_k^{UN}$ can be used to calculate the unavoidable exergy destruction $\dot{E}_{D,k,A}^{UN}$

$$\dot{E}_{D,k,A}^{UN} = \dot{E}_{P,k,A}\left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN} \tag{3}$$

The avoidable part of the exergy destruction rate is obtained from Eq. (2). It is apparent that this procedure is associated with some more or less arbitrary decisions.

The exergetic efficiency ε_k of a component is defined as the ratio between product $\dot{E}_{P,k}$ and fuel $\dot{E}_{F,k}$. An in-depth discussion of the definition of exergetic efficiencies for different types of plant components can be found in [3,4]. A modified exergetic efficiency ε_k^* that considers the avoidable exergy destruction within the kth component may be defined as [1]:

$$\varepsilon_k^* = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k} - E_{D,k}^{UN}} = 1 - \frac{\dot{E}_{D,k}^{AV}}{\dot{E}_{F,k} - E_{D,k}^{UN}} \tag{4}$$

This modified exergetic efficiency enables the comparison of exergetic efficiencies of dissimilar components. The avoidable exergy destruction rate in a system component can be related to the exergy rate of the fuel supplied to the overall system to provide a measure of the potential of improving the efficiency of the overall system by improving the performance of the kth component

$$y_{D,k}^{AV} = \frac{\dot{E}_{D,k}^{AV}}{\dot{E}_{F,tot}} \tag{5}$$

The ratio $x_{D,k}^{AV}$ is a thermodynamic measure for the potential of improvement when the kth component is considered in isolation. It shows the percentage of the incurred exergy destruction in a component that could be avoided if an extremely efficient component would be used.

$$x_{D,k}^{AV} = \frac{\dot{E}_{D,k}^{AV}}{\dot{E}_{D,k}} \tag{6}$$

In an exergoeconomic analysis, a cost value is assigned to each exergy stream. The cost rate $\dot{C}_{D,k,A}$ associated with the exergy destruction consists of an unavoidable part $\dot{C}_{D,k,A}^{UN}$ and an avoidable part

$\dot{C}^{AV}_{D,k,A}$ [1].

$$\dot{C}_{D,k,A} = c_{F,k}\dot{E}_{D,k,A} = \underbrace{c_{F,k}\dot{E}^{UN}_{D,k,A}}_{\dot{C}^{UN}_{D,k,A}} + \underbrace{c_{F,k}\dot{E}^{AV}_{D,k,A}}_{\dot{C}^{AV}_{D,k,A}} \tag{7}$$

where $c_{F,k}$ is the average cost per exergy unit of fuel [2]. The cost rate $\dot{C}^{AV}_{D,k,A}$ is the cost of the fuel used to cover the avoidable exergy destruction in the component when the product $\dot{E}_{p,k}$ is fixed.

The unavoidable investment cost per unit of product exergy $(\dot{Z}/\dot{E}_P)^{UN}_k$ is obtained by considering an extremely inefficient version of the kth component. In practical applications the term $(\dot{Z}/\dot{E}_P)^{UN}_k$ is determined by arbitrarily selecting a set of thermodynamic parameters for this component that lead to a very inefficient solution and by estimating the investment cost for this solution.

At a given design point A, the cost rate associated with the unavoidable investment cost $\dot{Z}^{UN}_{k,A}$ is

$$\dot{Z}^{UN}_{k,A} = \dot{E}_{P,k,A}\left(\frac{\dot{Z}}{\dot{E}_P}\right)^{UN}_k \tag{8}$$

and the avoidable cost is obtained from

$$\dot{Z}^{AV}_{k,A} = \dot{Z}_{k,A} - \dot{Z}^{UN}_{k,A} \tag{9}$$

Exergy destruction and capital costs are the real cost sources of a plant component. In a conventional exergoeconomic evaluation, the exergoeconomic factor expresses the contribution of capital cost to the sum of capital cost and cost of exergy destruction:

$$f_k \equiv \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{D,k}} = \frac{\dot{Z}_k}{\dot{Z}_k + c_{F,k}\dot{E}_{D,k}} \tag{10}$$

In [1], a modified exergoeconomic factor f^*_k is suggested based exclusively on avoidable costs:

$$f^*_k \equiv \frac{\dot{Z}^{AV}_k}{\dot{Z}^{AV}_k + \dot{C}^{AV}_{D,k}} = \frac{\dot{Z}^{AV}_k}{\dot{Z}^{AV}_k + c_{F,k}\dot{E}^{AV}_{D,k}} \tag{11}$$

In Section 4, the calculation of the unavoidable exergy destruction in components commonly used in thermal systems, such as compressors, turbines, heat exchangers, and combustion chambers is discussed.

3. The EFCC power plant

3.1. Process description

The conceptual design of an 126 MW EFCC power plant shown in Fig. 1 is used to demonstrate the application of the concept of avoidable thermodynamic inefficiencies and costs. It is neither a thermodynamic nor a cost optimal design.

Compressed air is heated in a metallic heat exchanger $C2$ up to a temperature of 1073 K. In an additional ceramic heat exchanger $C3$ the air temperature is increased to the gas turbine inlet

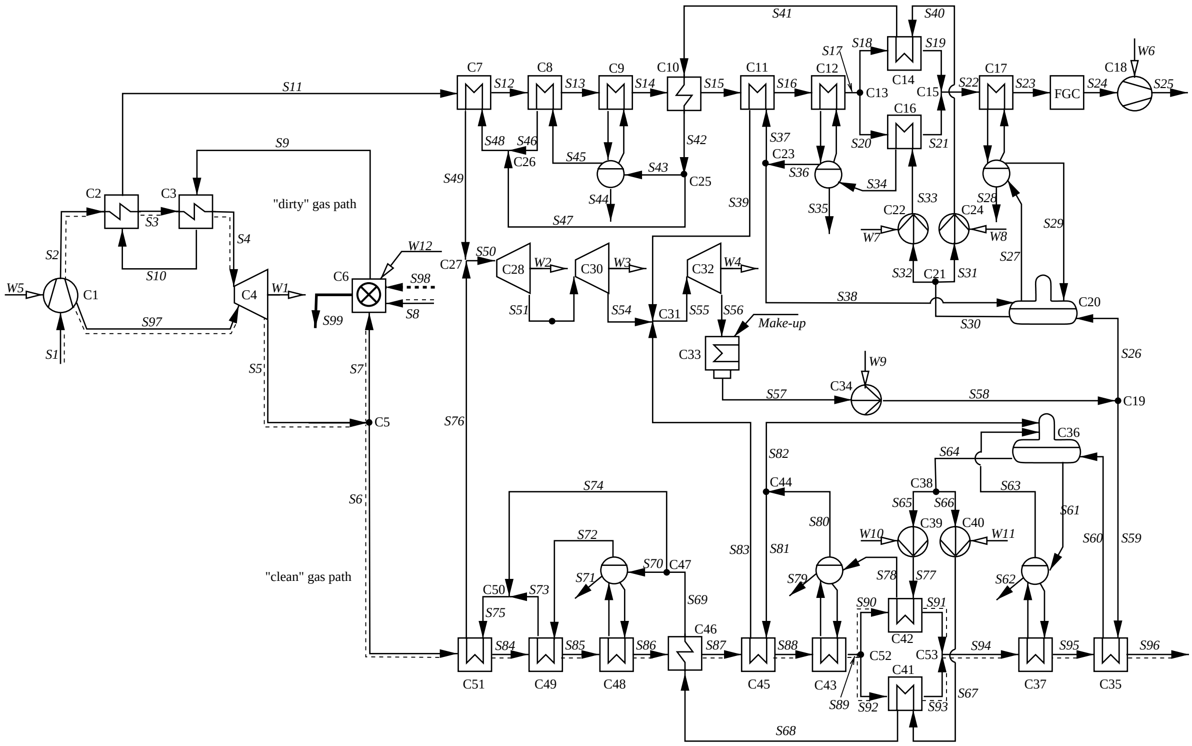

Fig. 1. Flow diagram and design-point performance of an externally fired combined cycle power plant.

temperature (1623 K). This hot and compressed air stream is expanded to 1.043 bar in a gas turbine expander $C4$ where 20% of the air supplied to the gas turbine compressor $C1$ is used for cooling the blades and the rotor (stream $S97$). At the turbine outlet the hot air stream ($T_5 = 829$ K) is split in two parts. A partial stream is used as preheated combustion air (secondary air) in a coal-fired combustion chamber $C6$ (excess air fraction $\lambda = 0.31$, calculated exhaust gas temperature $T_9 = 2273$ K) whereas the rest is directly supplied to a 'clean' HRSG.[1] Additional primary air (stream S8) is supplied to the combustion chamber in an equal amount of the coal flow rate. A part of the thermal energy of the combustion products is transferred in two high-temperature heat exchangers. For safety reasons, the combustion chamber operates below ambient pressure so that a stack fan is required in the combustion gas path. The mass flow rate of secondary air supplied to the combustion chamber is given by the desired excess air ratio.

The bottoming cycle consists of two non-reheat heat-recovery steam generators (HRSG). In both HRSGs, steam is generated at three pressure levels (83, 5, and ≈ 3 bar). In the 'dirty' HRSG larger values for the minimum temperature differences were used since the presence of fly ash and acid gases (SO_2, SO_3) requires the use of more expensive materials. Larger temperature differences result in smaller

[1] The term 'clean' denotes the gas path of the pure air stream. Gas cleaning is required to remove particulates and harmful combustion products such as SO_2 and NO_x from the exhaust gas in the combustion gas path.

heat transfer areas. For simplicity, the same overall heat transfer coefficient was applied to the same type of component in both HRSGs. However, in practical applications fouling effects due to deposits and erosion are more significant in the 'dirty' HRSG so that lower values for the heat transfer coefficient U should be used in this subsystem. High-pressure liquid water is sprayed into the partly superheated steam flows $S73$ and $S46$ to control the temperature at the outlets of superheaters $C7$ and $C51$ to 803 K. To avoid low-temperature corrosion on the gas side in the low-pressure evaporator $C17$, the pressure in the steam drum was adjusted properly.

Material considerations limit the air temperature that can be reached in the high-temperature heat exchangers so that supplementary firing with natural gas may be required to reach the desired high temperature at the gas turbine inlet. However, in recent research projects (e.g. [5]) ceramic heat exchangers are developed to achieve high temperatures of the working fluid without any supplementary firing. In this study, only coal is used. A comprehensive discussion of EFCC power plants can be found in Refs. [6,7].

Designing the high-temperature heat exchangers at acceptable cost is one of the main issues to be solved before EFCC power plants may become a cost-effective alternative to existing power plant designs such as natural-gas-fired combined cycles, supercritical steam power plants, or integrated gasification combined cycles. In Ref. [6], it is suggested to split the ceramic heat exchanger into a counter-flow and a parallel-flow section to maintain the temperature of the heat transfer surface below 1673–1723 K. A detailed discussion of the design of the high-temperature heat exchangers can be found in Refs. [5,7].

It is assumed that the power plant under investigation operates at base load. For simplicity, only steady-state operation is considered.

3.2. Simulation and modeling

The GateCycle™ energy balance software [8] was used to build a thermal model of the EFCC power plant. Typical engineering design parameters, such as pinch point temperature differences and economizer approach temperatures, pressure losses, and overall heat transfer coefficients were used. The designs of the gas turbine system, the high-temperature heat exchangers, and the external combustion chamber are based on data published in [7]. Plant performance was evaluated at ISO ambient conditions i.e. 288.15 K, 1.013 bar, and 60% relative humidity. Coal data were taken from Ref. [7].

Table 1 summarizes the values of selected design parameters. In general, larger temperature differences were selected for the heat exchangers operating in the 'dirty' HRSG and at higher temperature levels. The heat loss to the surroundings was assumed to be 1% of the actual heat transfered to the cold fluid in the HRSGs. In the combustion chamber 1% of the fuel energy is lost through radiation.

In both heat recovery steam generators, the gas streams flow in parallel through the intermediate-pressure economizer ($C16$ and $C42$) and the first high-pressure economizer ($C14$ and $C41$). The required heat transfer is calculated for each heat exchanger and the gas stream is split so that the gas outlet temperature is the same for both partial streams. Consequently, no exergy destruction due to temperature differences occurs in the subsequent mixer ($C15$ and $C53$).

The GateCycle code provides the mass flow rate, temperature, pressure, and chemical composition of all streams but does not calculate exergy values. Hence the THESIS software package [9] was used for

Table 1
Values of selected key design parameters used for the following calculations: design-point (DP), unavoidable thermodynamic inefficiencies (UTI), unavoidable investment costs (UIC)

Component	Parameter	Unit	DP	UTI	UIC
Turbomachines					
Compressor C1	$\eta_{p,C1}$	(%)	89.5	94	85
Expander *C*4	$\eta_{s,C4}$	(%)	90	92	70
HP steam turbine *C*28	$\eta_{s,C28}$	(%)	88	95	82
IP steam turbine *C*30	$\eta_{s,C30}$	(%)	88	95	82
LP steam turbine *C*32	$\eta_{s,C32}$	(%)	88	91	75
Condensate pump *C*34	$\eta_{s,C34}$	(%)	82	84	65
Feedwater pumps	η_{s}	(%)	85	88	75
Stack fan *C*18	$\eta_{s,C18}$	(%)	65	75	50
Combustion chamber					
Excess air	λ	(–)	0.31	0.15	0.31
Exit temperature	T_7	(K)	829	1000	829
Pulverizer spec. work	w_{12}	(kJ/kg)	72	25	72
High temperature heat exchangers					
Metallic	$\Delta T_{\min,C2}$	(K)	260	50	425
Ceramic	$\Delta T_{\min,C3}$	(K)	425	50	650
'*Clean*' *HRSG*					
Evaporators	$\Delta T_{\min,C48}$	(K)	18	5	35
	$\Delta T_{\min,C43}$	(K)	12	5	35
	$\Delta T_{\min,C37}$	(K)	10	5	35
Superheaters	$\Delta T_{\min,C51}$	(K)	26	2	150
	$\Delta T_{\min,C49}$	(K)	56	2	150
	$\Delta T_{\min,C45}$	(K)	25	2	150
Economizers	$\Delta T_{\min,C46}$	(K)	53	2	100
	$\Delta T_{\min,C41}$	[K]	7	2	100
	$\Delta T_{\min,C42}$	[K]	11	2	100
	$\Delta T_{\min,C35}$	[K]	10	2	100

this purpose. Both programs use JANAF data for the properties of ideal gases and IAPWS-IF97 for the properties of water and steam. All exergy values are calculated using $T_0 = 288.15$ K as the temperature of the environment, $p_0 = 1.013$ bar as the pressure of the environment, and Ahrendts' model (model I in [3]) for calculating standard molar chemical exergy values.

4. Calculation of unavoidable exergy destruction and cost

For calculating the term $(\dot{E}_{\mathrm{D}}/\dot{E}_{\mathrm{P}})_k^{\mathrm{UN}}$, each plant component is considered in isolation. Although stream data from the actual design are used as far as possible, temperatures, pressures, mass flow rates, and key design parameters are adjusted to achieve a high thermodynamic performance. The overall heat transfer coefficients were kept constant throughout these calculations and pressure losses were neglected. All plant components are adiabatic.

4.1. Unavoidable exergy destruction

4.1.1. Turbomachines

Friction causes exergy destruction in a turbine, compressor, pump, or fan. The lower the temperature, the more significant the effect of friction. The key design parameter for the turbomachines is the polytropic or isentropic efficiency. For a compressor, the isentropic efficiency is always lower than the polytropic efficiency whereas for an expander the reverse is true. The isentropic efficiency depends on the pressure ratio whereas the polytropic efficiency enables the comparison of machines with different pressure ratios. For calculating the unavoidable exergy destruction per unit of product exergy $(\dot{E}_{\mathrm{D}}/\dot{E}_{\mathrm{P}})^{\mathrm{UN}}$, the largest technically achievable values of the pressure ratio and the polytropic or isentropic efficiency should be selected. A similar conclusion can be drawn for a turbine. The values given in the middle column of Table 1 were used to calculate the unavoidable exergy destruction. Both the pressure ratio and the inlet temperature were taken from the actual design.

4.1.2. Combustion chamber

Chemical reaction is the most significant source of exergy destruction in a combustion chamber. Exergy destruction in the combustion chamber is mainly affected by the excess air fraction and the temperature of the air at the inlet. The thermodynamic inefficiencies of combustion can be reduced by preheating the combustion air and reducing the air–fuel ratio. For the calculation of the unavoidable exergy destruction in a combustion chamber, a high combustion air temperature at the inlet and the lowest technically meaningful value of the air–fuel ratio should be selected. Table 1 shows the values of these parameters used in this study. The use of a more efficient pulverizer indicated by the specific pulverizer work w_{12} was also considered.

4.1.3. Heat exchangers

The exergy destruction $\dot{E}_{\mathrm{D},q}$ due to heat transfer from a hot stream (index h) to a cold stream (index c) in an adiabatic heat exchanger is given by

$$\dot{E}_{\mathrm{D},q} = T_0 \dot{Q} \frac{T_{\mathrm{ha}} - T_{\mathrm{ca}}}{T_{\mathrm{ha}} T_{\mathrm{ca}}} \tag{12}$$

where T_{ha} and T_{ca} represent the thermodynamic average temperatures of the hot and cold streams, respectively [2].

Eq. (12) shows that the difference in the average thermodynamic temperatures $(T_{\mathrm{ha}} - T_{\mathrm{ca}})$ is a measure of the exergy destruction. Mismatched heat capacity rates of the two streams, i.e. $(\dot{m}c_{\mathrm{p}})_{\mathrm{h}}/(\dot{m}c_p)_{\mathrm{c}} \neq 1$, and a finite minimum temperature difference $\Delta T_{\min}$ are the causes of the thermodynamic inefficiencies associated with heat transfer. Matching streams of significantly different heat capacity rates $\dot{m}c_p$ in a heat exchanger should be avoided. Furthermore, the lower the temperature levels T_{ha} and T_{ca}, the greater the exergy destruction at a given temperature difference $(T_{\mathrm{ha}} - T_{\mathrm{ca}})$.

Parallel temperature profiles and a small minimum temperature difference are chosen to calculate the unavoidable exergy destruction in heat exchangers without phase change (Table 1). In general, the stream with the larger heat capacity rate $\dot{m}c_p$ is split and the temperature level of the cold stream is increased until a maximum allowable outlet temperature $T_{\mathrm{out,max}}$ is reached. This temperature is restricted by the actual inlet temperature of the hot stream, the minimum temperature difference $\Delta T_{\min}$,

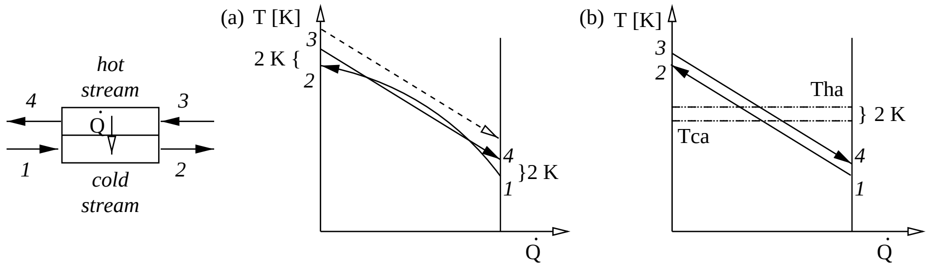

Fig. 2. (a) Crossing temperature profiles in a heat exchanger, (b) approximation used to calculate the unavoidable exergy destruction.

material limitations (e.g. in the high-temperature heat exchangers and in the superheaters) and the operation of the power plant (e.g. required minimum exit subcooling in economizers): Here, the maximal temperature was assumed to be 1675 K for the ceramic heat exchanger $C3$, 1075 K for superheaters, and ($T_{sat}(p)-3$ K) for economizers. The term $T_{sat}(p)$ is the saturation temperature of the cold fluid at given pressure in the economizer.

For calculating the unavoidable exergy destruction in a heat exchanger, a small value of the minimum temperature difference should be selected and the heat should be transferred at a high temperature level. Pressure losses should be neglected in these calculations.

A parallel temperature profile may be obtained by adjusting the mass flow rate of the stream with the larger heat capacity rate $\dot{m}c_p$ so that the temperature differences at the inlet and at the outlet of the heat exchanger are the same. However, caution is required since crossing temperature profiles might occur, if the specific heat capacity c_p of a stream changes significantly with temperature.

Fig. 2(a) shows the temperature profiles in an economizer when a small temperature difference is assumed and the specific heat capacity of the water changes. The hot stream has a higher temperature than the cold stream at the inlet and the outlet of the heat exchanger. Within this component, however, the design temperature of the cold stream exceeds the temperature of the hot stream, leading to an infeasible design. In such a case, the temperature profile of one stream has to be adjusted (e.g. the hot stream = dashed line in Fig. 2(a)). This requires a detailed calculation of the temperature profiles in each heat exchanger. For simplicity, the following approximation is applied in this study to calculate the unavoidable exergy destruction in such a component operating above ambient temperature.

The average thermodynamic temperature $T_{a,c}$ is calculated for the cold stream

$$T_{c,a} = \frac{\dot{H}_2 - \dot{H}_1}{\dot{S}_2 - \dot{S}_1}, \qquad p_2 = p_1 \tag{13}$$

For a given value of the minimum temperature difference ΔT_{min} the average thermodynamic temperature $T_{h,a}$ of the hot stream ($p_4 = p_3$) is

$$T_{h,a} = T_{c,a} + \Delta T_{min} \tag{14}$$

The exergy rates $\dot{E}_{q,h}$ and $\dot{E}_{q,c}$ associated with the heat transfer $\dot{Q} = \dot{Q}_{12} = |\dot{Q}_{34}|$ is

$$\dot{E}_{q,c} = \left(1 - \frac{T_0}{T_{c,a}}\right)\dot{Q}; \qquad \dot{E}_{q,h} = \left(1 - \frac{T_0}{T_{h,a}}\right)\dot{Q} \tag{15}$$

Thus, the unavoidable exergy destruction rate $\dot{E}_{D,k}^{UN}$ is obtained from[2]

$$\dot{E}_{D,k}^{UN} = \dot{E}_{q,h} - \dot{E}_{q,c} \tag{16}$$

The term $(\dot{E}_D/\dot{E}_P)^{UN}$ obtained in this way is lower than the value, which would be obtained if the real temperature profiles would be considered.

In an evaporator operating below the critical pressure of the cold fluid, the exergy destruction is mainly due to the large temperature differences between the hot and the cold streams. By decreasing the minimum temperature difference at constant operation pressure only small improvements of the thermodynamic performance can be achieved. Some exergy destruction is associated with the mixing of the subcooled feedwater with the water in the steam drum. For calculating the term $(\dot{E}_D/\dot{E}_P)^{UN}$, the feedwater is assumed to be supplied to the drum at saturation conditions.

4.1.4. Mixers and deaerators

Equal temperatures and pressures are assumed for the streams being mixed. Thus, the unavoidable exergy destruction rate for the water and steam streams in the plant shown in Fig. 1 is zero.

4.2. Unavoidable investment cost

The unavoidable investment costs per unit of product exergy $(\dot{Z}/\dot{E}_P)_k^{UN}$ are obtained by considering an extremely inefficient version of the kth component. Values for the key design parameters of the component are selected to lead to a very inefficient solution for which the investment costs are estimated. Hence, low values for the isentropic or polytropic efficiency and the pressure ratio should be chosen for turbomachines. A small heat transfer area in a heat exchanger can be achieved by selecting a large minimum temperature difference. Less expensive materials and no cooling of the walls may be sufficient for a combustion chamber when the chemical reaction takes place at a lower temperature. Thus, a large air–fuel ratio and ambient temperature of the air at the inlet to the combustion chamber should be used for calculating the unavoidable investment cost.

Table 1 summarizes the estimated values of the parameters used to calculate the term $(\dot{Z}/\dot{E}_P)^{UN}$. For the mixers, deaerators, and combustion chamber it is assumed that the unavoidable investment costs are equal to the values at the design-point.

5. Exergy analysis

The exergetic efficiency of the total plant ($\varepsilon_{tot} = 45.78\%$, Table 2) indicates that the conceptual design of such an externally fired combined cycle is thermodynamically more efficient than that of a supercritical coal-fired steam power plant ($\varepsilon_{tot} \approx 42\%$). However, some potential still exists for improving the overall efficiency and reducing the cost of electricity. In this section the real thermodynamic inefficiencies (exergy destruction and exergy loss) will be analyzed.

Exergy losses are mainly associated with the exhaust gases *S*25 and *S*96 as well as the exergy transfer to the environment in the condenser and the gas-cleaning unit *FGC*. All exergy losses together account

[2] If the heat exchanger operates below ambient temperatures, the fuel is represented by $|E_{q,c}|$ and the product by $|E_{q,h}|$.

Table 2
Results of the exergy analysis including avoidable and unavoidable exergy destruction rates for selected plant components

No	$\dot{E}_{P,k}$ (MW)	$\left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN}$ (–)	$\dot{E}_{D,k}^{UN}$ (MW)	$\dot{E}_{D,k}^{AV}$ (MW)	$y_{D,k}$ (%)	$y_{D,k}^{AV}$ (%)	$x_{D,k}^{AV}$ (%)	ε_k (%)	ε_k^* (%)
$C3$	103.17	0.0125	1.29	9.82	4.03	3.56	88.39	90.28	91.31
$C2$	59.98	0.0151	0.91	9.68	3.84	3.51	91.41	85.00	86.10
$C4$	178.75	0.0288	5.15	7.80	4.70	2.83	60.23	93.24	95.82
$C6$	205.37	0.3239	66.51	3.62	25.43	1.31	5.16	74.54	98.27
$C1$	92.03	0.0306	2.82	3.23	2.19	1.17	53.39	93.83	96.61
$C8$	6.84	0.0024	0.02	1.48	0.54	0.54	98.67	82.05	82.21
$C20$	1.68	0.0000	0.00	1.20	0.43	0.44	100.00	58.38	58.38
$C10$	4.08	0.0049	0.02	0.85	0.32	0.31	97.70	82.41	82.74
$C49$	4.85	0.0064	0.03	0.75	0.28	0.27	96.15	86.10	86.58
$C32$	25.67	0.1035	2.66	0.72	1.22	0.26	21.36	88.38	97.28
$C46$	3.00	0.0050	0.01	0.70	0.26	0.25	98.59	80.82	81.15
$C35$	1.41	0.0226	0.03	0.61	0.23	0.22	95.31	68.77	69.86
$C9$	16.26	0.1773	2.88	0.60	1.26	0.22	17.24	82.36	96.44
$C48$	11.52	0.1360	1.57	0.39	0.71	0.14	20.00	85.51	96.75
$C7$	2.23	0.0008	0.00	0.27	0.10	0.10	96.43	89.00	89.07
$C30$	13.48	0.0727	0.98	0.22	0.43	0.08	18.33	91.85	98.42
$C12$	3.50	0.2243	0.78	0.19	0.35	0.07	19.59	78.23	94.87
$C43$	4.11	0.2331	0.96	0.15	0.40	0.05	13.51	78.79	96.51
$C18$	1.58	0.3312	0.52	0.13	0.24	0.05	20.00	70.90	92.67
$C26$	0.54	0.0000	0.00	0.13	0.05	0.05	100.00	81.25	81.25
$C51$	1.58	0.0012	0.00	0.13	0.05	0.05	100.00	92.18	92.29
$C28$	10.10	0.0498	0.50	0.11	0.22	0.04	17.74	94.26	98.90
$C50$	0.38	0.0000	0.00	0.09	0.03	0.03	100.00	81.26	81.26
$C27$	54.63	0.0000	0.00	0.00	0.00	0.00	–	100.00	100.00
Tot	126.27	–	–	–	47.50	–	–	45.78	–

The components are ranked in descending order of the value $y_{D,k}^{AV}$. Overall system: $\dot{E}_{F,tot} = 275.81$ MW, $\dot{E}_{D,tot} = 131.02$ MW.

for 6.72% or 18.525 MW of the fuel exergy supplied to the overall plant. More than 47% of the fuel exergy is destroyed within the plant components (Table 2).

Table 2 shows for each plant component and the total plant the exergy of product $\dot{E}_{P,k}$, the exergy destruction ratio $y_{D,k}$, and the exergetic efficiency ε_k. Calculated values for the avoidable and unavoidable parts of the exergy destruction ($\dot{E}_{D,k}^{AV}$ and $\dot{E}_{D,k}^{UN}$), the term $(\dot{E}_D/\dot{E}_P)_k^{UN}$, the variables $y_{D,k}^{AV}$ and $x_{D,k}^{AV}$ as well as the calculated modified exergetic efficiency ε_k^* are also given.

The exergy destruction ratio $y_{D,k}$ shows that the main contributors to the exergy destruction in the total plant are in descending order of importance: the combustion chamber $C6$, the gas turbine expander $C4$, the ceramic heat exchanger $C3$, the metallic heat exchanger $C2$, the gas turbine compressor $C1$, the high-pressure evaporator $C9$, the low-pressure steam turbine $C32$, the high-pressure evaporator $C48$, and the superheater $C8$. However, the avoidable exergy destruction rate $\dot{E}_{D,k}^{AV}$ and the ratio $x_{D,k}^{AV}$ indicates that in some of these components only a small part of $\dot{E}_{D,k}$ can be reduced by using the best available technology. For example, in the combustion chamber only 5% of the thermodynamic inefficiencies can be avoided by further preheating the reactants or decreasing the excess air fraction. Improvement efforts should be centered on the efficiencies that can be avoided [1]. In the following, the plant components with the highest $\dot{E}_{D,k}^{AV}$ values will be discussed in descending order of this value.

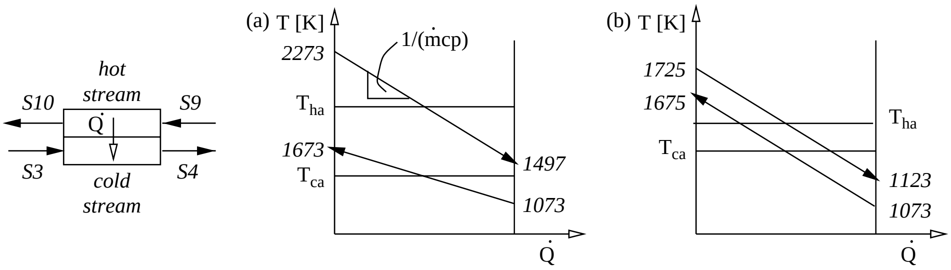

Fig. 3. Temperature profiles of hot and the cold stream in the ceramic heat exchanger $C3$. (a) Actual profiles; (b) profiles used to calculate the unavoidable exergy destruction.

A significant part of the exergy destruction in the high-temperature heat exchangers $C3$ and $C2$ can be avoided ($x^{AV}_{D,C3} \approx 88\%$, $x^{AV}_{D,C2} \approx 91\%$). Exergy destruction associated with heat transfer and pressure losses as well as the exergy loss associated with heat transfer to the surroundings are the main causes of the inefficiencies in both components. Fig. 3 illustrates the temperature profiles of the hot and the cold stream in the actual ceramic heat exchanger $C3$ and the profiles used for calculating the term $(\dot{E}_D/\dot{E}_P)^{UN}_{C3}$ assuming counter-current flow. Exergy destruction associated with heat transfer is lower at higher temperature levels and for smaller temperature differences (Eq. (12)). For the calculation of the unavoidable exergy destruction in the ceramic heat exchanger, a maximum cold side outlet temperature of 1675 K and a minimum temperature difference of 50 K are assumed. To achieve parallel temperature profiles, the mass flow rate of the cold stream is adjusted accordingly (Fig. 3). Similar conclusions can be drawn from the evaluation of the metallic heat exchanger $C2$.

Almost 60% of the exergy destruction $\dot{E}_{D,C4}$ in the gas turbine expander $C4$ can be avoided by selecting a component with the highest available isentropic efficiency (here: $\eta_{s,t} = 0.92$) and by avoiding the use of cooling air ($\dot{m}_{97} = 0$). In this calculation, both the actual turbine inlet temperature and the actual pressure ratio were used.

Still a significant amount of the exergy destruction in the overall plant can be avoided by improving the thermodynamic performance of the combustion chamber ($\dot{E}^{AV}_{D,C6} = 3.62$ MW) through a higher temperature of the air and a lower excess-air fraction. In the actual plant, the temperature of the combustion air depends on the temperature T_4 at the gas turbine inlet and the pressure ratio p_4/p_5 in the expander $C4$. Increasing this temperature and decreasing the pressure ratio would result in the desired increase of the combustion air temperature. A higher mass flow rate $\dot{m}_7$ increases the excess air fraction in the combustion chamber.

In the high-pressure superheater $C8$ the avoidable exergy destruction is mainly due to mismatched temperature profiles. Splitting the hot stream (S12), which has the larger heat capacity rate $\dot{m}c_p$, decreasing the minimum temperature difference, for example, to 2 K, and avoiding any heat transfer to the environment could reduce the actual exergy destruction up to 98%.

Exergy destruction in a deaerator is mainly caused by differences in temperature and pressure of the water streams being mixed.[3] These thermodynamic inefficiencies can be completely avoided when the feedwater and the steam have the same temperature and pressure and the plant component is considered in

[3] In a real power plant, the feedwater contains soluble gases that are removed in the deaerator so that small differences in the chemical composition of the streams also occur.

Table 3
Breakdown of the total purchased-equipment costs

Gas turbine ($C2$ and $C3$) (%)	21.5
High-temperature heat exchangers (%)	17.7
Combustion chamber (%)	14.2
Steam turbines and generator (%)	13.3
HRSG (%)	10.7
Pumps and stack fan $C2$ (%)	1.0
Others (e.g. gas cleaning) (%)	21.6

isolation. In this power plant design, not enough energy in the evaporator $C17$ is available to provide saturated steam at the proper temperature level for deaerating the feedwater $S26$ in deaerator $C20$. Consequently, also intermediate-pressure steam $S38$ has to be used for this purpose. The pressure level and the amount of steam generated in the evaporator $C17$ depend on the minimum acceptable exhaust gas temperature T_{23} which must not be lower than the dew point of the acid gases in stream $S23$. Although, there is only a small potential to reduce the exergy destruction in deaerator $C20$ through an increased steam production in evaporator $C17$, all or a part of the water $S26$ might be deaerated in one deaerator (e.g. $C36$). This component could be operated at a higher pressure level and would be supplied with steam from the 'clean' HRSG. In the clean HRSG more flexibility exists since an additional economizer $C35$ can use the low temperature energy and reduce the exergy loss associated with the exhaust gas $S96$.

The $x_{D,k}^{AV}$-value is equal to 100% for all mixers and deaerators since only differences in temperature and pressure occur between the streams being mixed. In general, such differences can be avoided (see for example mixer $C27$ in Table 2). However, the purpose of the mixers $C26$ and $C50$ is to control the temperature of superheated steam. This cannot be achieved by mixing streams of equal temperatures and pressures.

6. Economic analysis

Engineering economics based on EPRI Technical Assessment Guide [10] were used to perform the economic analysis and to calculate the levelized cost of electricity in constant Euro (reference year: 2003). Purchased-equipment costs were estimated using data published in the literature (e.g. [7]) and assumptions made by the authors. Table 3 shows the breakdown of the total-purchased equipment costs.

Table 4 summarizes important economic assumptions made in this study. The levelized cost of electricity for this design is 38.7 €/MWh. Carrying charges (depreciation, dividends, interest, taxes and

Table 4
Selected parameter and assumptions for the economic analysis

Plant economic life (years)	20
Levelization period (years)	10
Average general inflation rate (%)	2
Average real escalation rate for coal (%)	0.5
Average real cost of money (%)	6.8
Plant service date	Jan. 2006
Average capacity factor (%)	85
Unit cost of coal (€/GJ-LHV)	1.8

Table 5
Results of the exergoeconomic analysis for selected plant components

No	$c_{\mathrm{F,k}}$ (€/GJ)	$c_{\mathrm{P,k}}$ (€/GJ)	$(\dot{Z}/\dot{E}_p)_k^{\mathrm{UN}}$ (€/MWh)	$\dot{Z}_k^{\mathrm{UN}}$ (€/h)	$\dot{Z}_k^{\mathrm{AV}}$ (€/h)	$\dot{C}_{D,k}^{\mathrm{AV}}$ (€/h)	$\dot{C}_{\mathrm{D},k}^{\mathrm{UN}}$ (€/h)	$\mathrm{f_k}$ (%)	f_k^* (%)	$\dot{Z}_k + \dot{C}_{\mathrm{D},k}$ (€/h)	$(\dot{Z}_k + \dot{C}_{\mathrm{D},k})^{\mathrm{AV}}$ (€/h)	$\frac{(\dot{Z}_k+\dot{C}_{\mathrm{D},k})^{\mathrm{AV}}}{(\dot{Z}_k+\dot{C}_{\mathrm{D},k})}$ (%)
*C*4	5.97	7.22	1.0615	190	338	168	111	65.5	66.9	806	506	62.7
*C*3	3.36	4.75	2.9592	305	76	119	16	73.9	38.9	515	195	37.7
*C*1	7.22	8.14	0.4684	43	103	84	73	48.2	55.1	304	187	61.7
*C*2	3.36	4.77	2.2813	137	38	117	11	57.8	24.7	304	156	51.3
*C*32	7.42	10.51	3.5087	90	105	19	71	68.4	84.5	285	124	43.5
*C*30	7.00	10.19	6.0705	82	43	5	25	80.5	88.7	155	48	31.2
*C*28	7.00	10.14	6.4082	65	34	3	13	86.4	92.3	114	37	32.1
*C*14	3.36	14.08	4.2877	4	29	1	0	97.0	96.8	33	29	88.7
*C*6	1.78	2.99	2.1717	446	0	23	427	49.8	0.0	896	23	2.6
*C*46	5.97	8.81	3.0968	9	6	15	0	50.2	28.9	31	21	68.7
*C*8	3.36	5.05	3.2367	22	1	18	0	56.4	6.5	42	19	46.2
*C*41	5.97	19.41	4.4038	2	18	1	0	93.7	93.4	21	19	90.7
*C*48	5.97	7.89	2.5691	30	8	8	34	47.3	49.4	80	16	20.6
Tot	1.76	10.75	–	–	–	–	–	79.1	–	3969	–	–

The components are ranked in descending order of the value $(\dot{Z}_k + \dot{C}_{\mathrm{D},k})^{\mathrm{AV}}$. Overall system: $\dot{C}_{\mathrm{F,tot}} = 1745$ €/h, $\dot{Z}_{\mathrm{tot}} = 3139$ €/h, $\dot{C}_{\mathrm{P,tot}} = 4505$ €/h, $\dot{C}_{L,\mathrm{tot}} = 379$ €/h.

insurance) contribute 64% of this value. The rest are fuel-related costs. Operating and maintenance costs were neglected in this study. The specific investment cost calculated at the beginning of plant operation (i.e. including cost escalation and allowance for funds used during construction time) are 1431 €/kW$_e$.

7. Exergoeconomic analysis and evaluation

Table 5 shows the values of the exergoeconomic variables for selected plant component. Plant components with a large value of the sum $(\dot{Z}+\dot{C}_D)^{AV}$ should be improved first. Hence, the components are ranked in descending order of this value. Based on the assumptions in this study, the ratio $(\dot{Z}+\dot{C}_D)^{AV}/(\dot{Z}+\dot{C}_D)$ indicates the percentage of the costs that could theoretically be avoided in today's technological and economic environment for each component.

Consideration of the avoidable costs emphasizes the need to improve the cost effectiveness of the gas turbine components ($C4$,$C1$), the high-temperature heat exchangers ($C3$,$C2$), and the steam turbines ($C32$,$C30$,$C28$). Only a small part (14–23%) of the costs $(\dot{Z}+\dot{C}_D)$ are avoidable in the design of the HP and IP evaporators (e.g. $C48$). The combustion chamber $C6$ has the highest cost of exergy destruction ($\dot{C}_{D,k}=\dot{C}^{AV}_{D,k}+\dot{C}^{UN}_{D,k}$) and the second highest investment cost ($\dot{Z}_k=\dot{Z}^{AV}_k+\dot{Z}^{UN}_k$). However, the potential of improving this component is very low.

The exergoeconomic factor f_k is used to identify the major cost source (capital investment or cost of exergy destruction, Eq. 10) associated with a system component. In a conventional exergoeconomic evaluation, the value of the exergoeconomic factor is compared with a target value of the same type of component. If the f_k value is high, it should be investigated whether it is cost effective to reduce the capital investment at the expense of component efficiency. If the f_k value is low, the component efficiency should be improved by increasing the capital investment. The modified exergoeconomic f^*_k considers avoidable investment cost and cost of exergy destruction.

The f^*_k values shown in Table 5 indicate that the cost effectiveness of the EFCC power plant might be improved by reducing the capital investment for the gas turbine expander $C4$, the steam turbines ($C28$, $C30$, and $C32$), and the high-pressure economizers ($C14$ and $C41$). The high-temperature heat exchangers ($C2$ and $C3$), the high-pressure economizer $C46$, and the superheater $C8$ have relatively low

Table 6
Exergy rate $\dot{E}_j$, cost per exergy unit c_j, and cost rate $\dot{C}_j$ for electric power W_j

No.	$\dot{E}_j$ (MW)	c_j (€/GJ)	$\dot{C}_j$ (€/h)
W1	178.746	7.22	4646
W2	10.102	10.14	369
W3	13.483	10.19	495
W4	25.671	10.51	971
W5	98.082	7.22	2550
W6	2.228	7.90	63
W7	0.014	7.90	0
W8	0.324	7.90	9
W9	0.077	7.90	2
W10	0.017	7.90	0
W11	0.234	7.90	7
W12	0.758	7.90	22

Table 7
Mass flow rate $\dot{m}_j$, temperature T_j, pressure p_j, physical exergy rate $\dot{E}_j^{\mathrm{PH}}$, chemical exergy rate $\dot{E}_j^{\mathrm{CH}}$, total exergy rate $\dot{E}_j^{\mathrm{TOT}}$, cost per exergy unit c_j, and cost rate $\dot{C}_j$ for the jth stream.

No.	$\dot{m}_j$ (kg/s)	T_j (K)	p_j (bar)	$\dot{E}_j^{\mathrm{PH}}$ (MW)	$\dot{E}_j^{\mathrm{CH}}$ (MW)	$\dot{E}_j^{\mathrm{TOT}}$ (MW)	c_j (€/GJ)	$\dot{C}_j$(€/h)
1	250.00	288.15	1.013	0.000	0.389	0.389	0.00	0
2	200.00	667.32	15.246	73.626	0.311	73.937	8.10	2157
3	200.00	1073.00	14.941	133.608	0.311	133.920	6.61	3187
4	200.00	1623.00	14.642	236.780	0.311	237.091	5.80	4952
5	250.00	829.16	1.043	63.491	0.389	63.880	5.97	1372
6	135.36	829.16	1.043	34.376	0.211	34.587	5.97	743
7	114.64	829.16	1.043	29.115	0.178	29.294	5.97	629
8	5.00	288.15	1.013	0.000	0.008	0.008	0.00	0
9	129.23	2273.00	0.991	228.898	5.773	234.670	3.36	2842
10	129.23	1497.65	0.961	114.618	5.773	120.390	3.36	1458
11	129.23	928.44	0.942	44.046	5.773	49.819	3.36	603
12	129.23	904.95	0.941	41.544	5.773	47.317	3.36	573
23	129.23	418.13	0.926	2.689	5.773	8.462	3.36	102
25	129.23	349.65	1.013	1.328	5.773	7.101	5.47	140
26	22.90	309.64	3.834	0.080	0.057	0.137	10.80	5
38	3.95	425.00	5.000	3.099	0.010	3.109	6.45	72
39	1.26	500.82	5.000	1.069	0.003	1.072	6.43	25
54	36.07	454.97	5.000	29.144	0.090	29.234	7.00	737
55	43.41	464.13	5.000	35.391	0.108	35.499	7.42	949
83	6.07	512.49	5.000	5.204	0.015	5.219	9.95	187
96	135.36	359.75	1.015	1.069	0.211	1.280	5.97	28
97	50.00	667.32	15.246	18.406	0.078	18.484	8.10	539
98	10.52	288.15	1.013	0.000	275.413	275.413	1.76	1745

f_k^* values. Most of the avoidable costs associated with these components are due to the cost of exergy destruction. An increase of the exergetic efficiencies of these components might reduce the overall product cost $c_{P,\mathrm{tot}}$ even if the cost associated with capital investment will increase.

The parameters discussed in Section 4 should be modified to achieve the objective of the design changes (increasing ε_k or decreasing $\dot{Z}_k$) for each plant component. However, changes suggested by the evaluation of a component should only be considered if they do not contradict changes suggested by components with a significantly higher value of the sum $(\dot{Z}_k + \dot{C}_{\mathrm{D},k})^{\mathrm{AV}}$. Due to the interactions among the plant components, several iterations will be required to achieve a cost optimal design. Tables 6 and 7 show important variables for selected material streams to enable the reader to verify values shown in the previous tables.

8. Conclusions

Some potential still exists for improving the thermodynamic performance of the system. In the water steam/cycle, thermodynamic inefficiencies can be reduced, for example, by decreasing the minimum temperature differences, increasing the temperatures of high-pressure and intermediate-pressure steam, adjusting the pressure levels, or adding reheat sections. However, in this subsection only 36% of the net electric power is generated. Hence, the effect of design improvements in the bottoming cycle on the overall exergetic efficiency is less than in the gas turbine section including the combustion chamber.

So far, the avoidable exergy destruction rate is calculated for a plant component in isolation. Interactions among the plant components may prevent that the expected reduction in the thermodynamic inefficiencies is achieved. A part of the exergy destruction within a system component is in general caused by the inefficiencies of the remaining system components (*exogenous* exergy destruction). If no irreversibilities and exergy losses occur in all the remaining system components, the exergy destruction is due exclusively to the component being considered (*endogenous* exergy destruction). A change in the exergy destruction in one subsystem affects in general the exergy destruction in other subsystems too. Thus, the change in the total exergy input to a system is usually different from the change in the exergy destruction in one system component. More attention should be paid to interactions among the plant components and their effects on the thermodynamic inefficiencies and costs.

Through the distinction between avoidable and unavoidable thermodynamic inefficiencies and costs, recommendations with respect to changes expected to improve the cost effectiveness of the overall plant can be made with increased certainty.

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