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ADVANCED EXERGETIC ANALYSIS AS A TOOL FOR THE THERMODYNAMIC EVALUATION OF SUPERCRITICAL CO_2 POWER CYCLES

Tatiana Morosuk
Professor
Technische Universität Berlin
Berlin, Germany
morozyuk@iet.tu-berlin.de

George Tsatsaronis
Professor
Technische Universität Berlin
Berlin, Germany
tsatsaronis@iet.tu-berlin.de

Tatiana Morosuk is professor of Exergy-Based Methods for Refrigeration Systems at the Technische Universität Berlin, Germany. She studied refrigeration engineering in the Odessa State Academy of Refrigeration, Ukraine, and received her Diploma in 1990. She received her Ph.D. in 1994 and Professorship in 2001, all in Ukraine. Her field of interest is the application of modern exergy-based methods to the analysis and improvement of energy conversion systems. She is the author or co-author of 7 books and approximately 250 publications, and has 10 patents.

George Tsatsaronis is since 1994 the Bewag Professor of Energy Engineering and Environmental Protection at the Technische Universität Berlin, Germany. He received a Diploma in Mechanical Engineering (NTU Athens, Greece), and his MBA, PhD in Combustion, and Dr Habilitatus Degree in Thermoeconomics, all from the RWTH Aachen, Germany. In the time period between 1982 and 1994 he worked in the USA. His areas of interest include the design, development, analysis and optimization of energy conversion systems. He contributed significantly to the fundamentals and terminology of exergy-based methods. He co-authored the book Thermal Design and Optimisation (Wiley, 1996), has published more than 300 papers, received several international awards and recognitions, and has served as Chairman or Co-chairman of 18 international conferences.

ABSTRACT

The idea of developing supercritical CO_2 power cycles and applying them to industrial processes became increasingly popular in the last decade. Significant research has been done in this field, including the development of new thermodynamic cycles, investigation of characteristics of equipment, and parametric optimization of power systems. An exergetic analysis complements and enhances an energetic analysis in the thermodynamic and economic improvements (optimization) of supercritical CO_2 power cycles. An exergetic analysis is a powerful tool for developing, evaluating and improving energy conversion systems.

Conventional exergetic analyses have some limitations, which are significantly reduced by the so-called advanced analyses. In addition to conventional analyses, the latter evaluate, (a) the interactions among components of the overall system, and (b) the real potential for improving a system component. The main objective of an advanced exergetic analysis is to provide engineers with additional information that is useful for understanding and improving the design and operation of energy conversion systems, and that cannot be supplied by any other approach.

This paper deals with an application of an integrated conventional and advanced exergetic analysis to a supercritical CO_2 power cycle. The objective is to obtain detailed useful information about the optimization of the structure and the parameters of the system being considered.

NOMENCLATURE

E	exergy (J)
e	specific exergy (J/kg)
H	enthalpy (J)
h	specific enthalpy (J/kg)
j	j th stream of matter (-)
k	k th component (-)
m	mass (kg)
p	pressure (Pa)
Q	heat (J)
S	entropy (J/K)
s	specific entropy (J/kg·K)
T	temperature (K or °C)
w	specific work (J/kg)
y	exergy destruction ratio (-)

Greek symbols

ε	exergetic efficiency (%)
η	energetic efficiency (%)

Subscripts

$cycle$	(thermodynamic) cycle
D	(exergy) destruction
F	(exergy of) fuel
gen	(entropy) generation
L	(exergy) loss
P	(exergy of) product
tot	refers to the total system
0	reference state (environment)

Superscripts

·	time rate
AV	avoidable (exergy destruction)
EN	endogenous (exergy destruction)
EX	exogenous (exergy destruction)
opt	optimal
UN	unavoidable (exergy destruction)

Abbreviations

CM	compressor and motor
COL	cooler
EG	electric generator
EX	expander (turbine)
HE	heat exchanger
RHE	recuperative heat exchanger

INTRODUCTION

The potential for the application of supercritical CO_2 cycles is high for both power generation systems and refrigeration systems. The number of existing publications (particularly during the last decade) is very high. Here we will only mention the pioneering paper by Angelino (Angelino, 1968) for applications to power cycles and the papers by Lorentzen (for example, Lorentzen, 1994) and Kim et al. (Kim et al., 2004) that refer to CO_2 refrigeration cycles.

Note that most existing publications discuss the development and analysis of new thermodynamic cycles, some characteristics of equipment, the parametric optimization of these power systems, etc., and mainly refer to systems operating in an environment, the temperature of which allows condensation of the working fluid (CO_2). One of the few publications on refrigeration using CO_2, where hot climatic conditions were considered, is a recently published paper by Fazelpour and Morosuk (Fazelpour & Morosuk, 2014).

The present paper deals with supercritical CO_2 cycles used for power generation when the temperature of the environment is high enough, so that a simple CO_2 cycle must operate above the critical temperature of CO_2 (i.e., T_{CO2} >31.1 ºC). Then none of the known condensation cycles (Figure 1) can be applied.

The evaluation of the CO_2 cycles for power systems reported here has been conducted using results from both energy and exergy analyses.

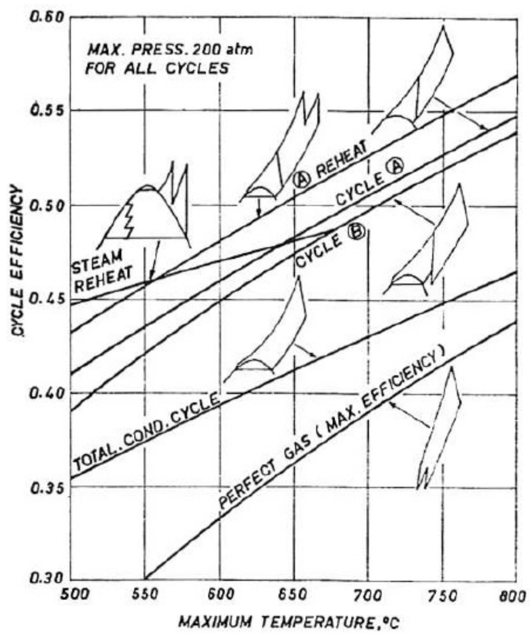

Figure 1. Efficiency of CO_2 condensation cycles compared with the performance of steam and perfect-gas cycles according to Angelino (1968)

EXERGY ANALYSIS

Conventional Exergy Analysis

The rate of physical exergy $\dot{E}_j^{PH}$ associated with the j th material stream is

$$\dot{E}_j^{PH} = \dot{m} \cdot e_j^{PH} = \dot{m}\left[\left(h_j - h_0\right) - T_0\left(s_j - s_0\right)\right] \quad (1)$$

Here $\dot{m}$ is the mass flow rate, while e, h, and s denote the specific exergy, enthalpy and entropy, respectively, of the j th material stream. The subscript 0 refers to the property values of the same mass flow rate at temperature T_0 and pressure p_0 of the reference state. For the systems discussed here only the physical exergy of the working fluid needs to be considered.

Central elements for an exergy-based analysis are the general concepts of *fuel* and *product* introduced 30 years ago (Tsatsaronis, 1984). The *exergy of product* is the desired result (expressed in exergy terms) achieved by the system (e.g., the *k*th component) being considered, and the *exergy of fuel* is the exergetic resources expended to generate the exergy of the product. Definitions of the exergy of fuel and the exergy of product for different system components as well as for overall systems are given in Tsatsaronis and Cziesla (2009) and Bejan et al. (1996).

In a conventional exergetic evaluation of the *k*-th component of a system, the following variables are used (see, for example, Bejan et al., 1996):

- The exergy destruction rate ($\dot{E}_{D,k}$) depends on the mass flow rate ($\dot{m}_k$) through the component and on the specific entropy generation ($s_{gen,k}$) within it:

$$\dot{E}_{D,k} = T_0 \dot{S}_{gen,k} = T_0\, \dot{m}_k\, s_{gen,k}. \tag{2}$$

- The exergetic efficiency is defined as the ratio between the exergy of product ($\dot{E}_{P,k}$) and the exergy of fuel ($\dot{E}_{F,k}$)

$$\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = 1 - \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}}. \tag{3}$$

- The exergy destruction ratio is defined by

$$y_k = \frac{\dot{E}_{D,k}}{\dot{E}_{F,tot}}. \tag{4}$$

The exergy balance for the *k*-th component can be written as

$$\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k}. \tag{5}$$

The advantages of exergy analyses have been discussed in many previous publications. However, a conventional exergetic analysis cannot accurately evaluate the mutual interdependencies among the system components. This becomes possible in an advanced exergetic analysis (for example, Tsatsaronis, 1999; Morosuk & Tsatsaronis, 2008; Tsatsaronis & Morosuk, 2012), in which the exergy destruction in each component is split into *endogenous* and *exogenous* parts as well as *avoidable* and *unavoidable* parts. Finally a combination of these two splitting approaches provides the designer and operator of an energy conversion system with unambiguous and valuable detailed information with respect to options for improving the overall efficiency.

Advanced Exergy Analysis

The total exergy destruction within the *k*-th component is split into *endogenous* and *exogenous* parts $\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX}$. Here $\dot{E}_{D,k}^{EN}$ is the *endogenous* part of exergy destruction, associated with the irreversibilities occurring within the *k*-th component when all other components operate in an ideal way and the component being considered operates with its current efficiency. $\dot{E}_{D,k}^{EX}$ is the *exogenous* part of exergy destruction in the *k*-th component and is caused within the *k*-th component also by the inefficiencies of the remaining components.

Only a part of the exergy destruction within a component can be avoided. The exergy destruction that cannot be further reduced due to technological limitations such as availability and cost of materials and manufacturing methods is the *unavoidable* ($\dot{E}_{D,k}^{UN}$) part of the exergy destruction. The remaining part represents the *avoidable* ($\dot{E}_{D,k}^{AV}$) part of the exergy destruction. Thus, splitting the exergy destruction within the *k*-th component into *unavoidable* and *avoidable* parts $\dot{E}_{D,k} = \dot{E}_{D,k}^{UN} + \dot{E}_{D,k}^{AV}$ provides a realistic measure of the potential for improving the thermodynamic efficiency of a component.

By combining the two concepts mentioned above, we obtain the unavoidable endogenous exergy destruction and subsequently the avoidable endogenous, the unavoidable exogenous and the avoidable exogenous parts of exergy destruction within the *k*-th component (Tsatsaronis and Morosuk, 2012).

The *endogenous unavoidable* ($\dot{E}_{D,k}^{EN,UN}$) part of the exergy destruction cannot be reduced because of technological limitations for the *k*-th component. The *exogenous unavoidable* ($\dot{E}_{D,k}^{EX,UN}$) part of the exergy destruction cannot be reduced because of technological limitations in the other components of the overall system for its given structure.

The *endogenous avoidable* ($\dot{E}_{D,k}^{EN,AV}$) part of the exergy destruction can be reduced by improving the efficiency of the *k*-th component. The *exogenous avoidable* ($\dot{E}_{D,k}^{EX,AV}$) part of the exergy destruction can be reduced by an improvement in the structure of the overall system, or by improving the efficiency of the remaining system components, and of course by improving the efficiency in the *k*-th component.

CASE STUDY

The supercritical CO_2 cycle, which was considered here as an example, is shown in Figure 2. From the thermodynamic point of view it represents a closed-cycle gas-turbine system with a recuperative heat exchanger (RHE). The following assumptions were used for the simulation:

- $\dot{W}_{net}$, the net power generated by the system, remains always equal to 10 MW.
- The isentropic efficiency of expander (EX) is equal to 0.90.
- The isentropic efficiency of the compressor (CM) is equal to 0.85.
- The temperature of the cooling medium at the inlet of the cooler (T_{env}) is equal to 40 ºC.
- The minimal temperature difference within the cooler ($T_4 = T_{env} + \Delta T$) is equal to 10 K.
- The minimal temperature difference within the regeneration heat exchanger ($T_3 = T_5 + \Delta T$) is equal to 20 K.
- The pressure drop within the heat exchanger (HE), the cooler (COL), and for each stream of the regeneration heat exchanger (RHE) is equal to 3%.

The effect of the ambient temperature on the system efficiency is shown in Figure 3. For the assumptions made here, and in the temperature range 10 °C to 40 °C the efficiency drops by 0.17 percentage points for every degree increase in the ambient temperature.

In order to find some optimal operation conditions, a sensitivity analysis has been conducted for pressure values at state 2 equal to 200, 250 and 300 bars. The results are shown in Figure 4. Optimal pressures at state 4 are also indicated in the same figure.

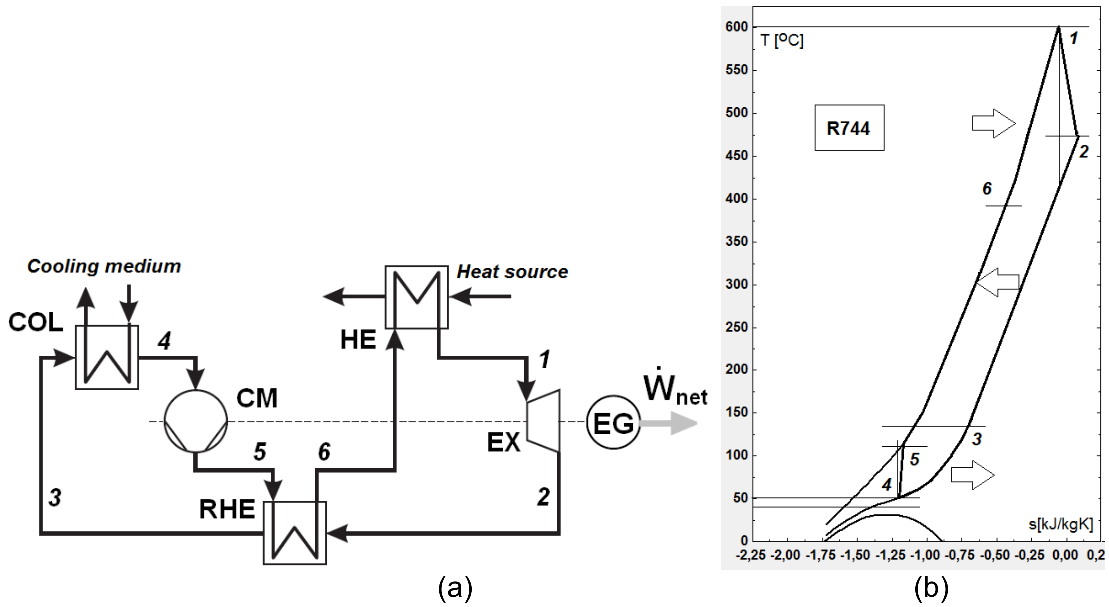

Figure 2. Schematic (a) and thermodynamic cycle (b) of the supercritical CO_2 power system studied here

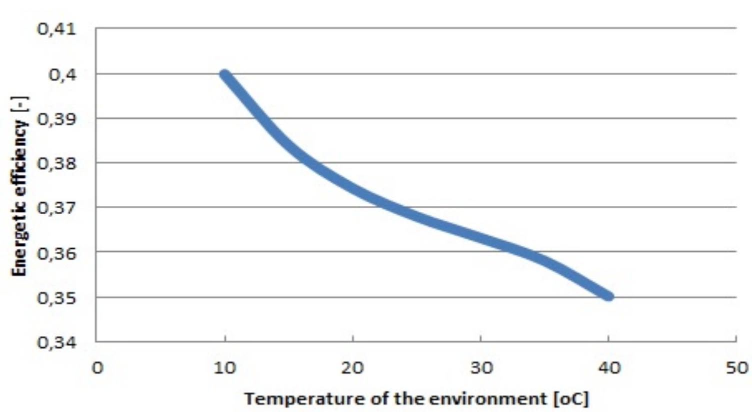

Figure 3. Effect of the ambient temperature on the efficiency of the supercritical CO2 cycle shown in Figure 2 when T_1 =600°C.

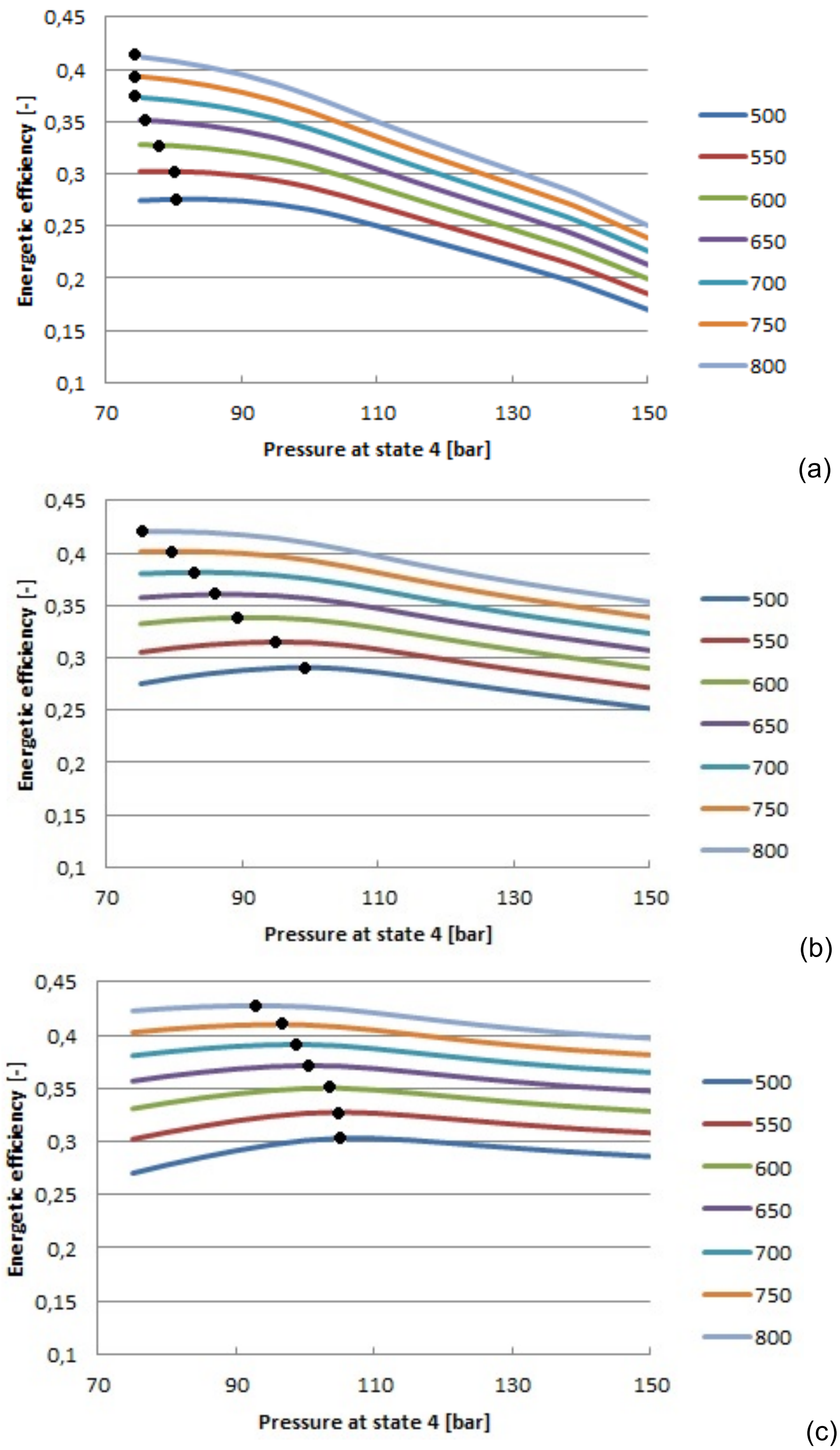

Figure 4. Sensitivity analysis for the supercritical CO_2 power system shown in Fig 2 at different temperatures of state 1 (from 500 °C to 800 ºC) and different pressures of state 1: (a) p_1 =200 bars, (b) p_1 =250 bars, and (c) p_1 =300 bars

From the thermodynamic point of view the differences between energetic efficiencies (at optimal pressure values at state 4) are not as high as one would expect; however, the operation conditions will significantly affect the economic characteristics of the supercritical CO_2 power systems and the cost of their product.

Taking into account "universal" operation conditions for different heat sources (nuclear power systems, solar power power systems, gas turbine power plants, etc.), and applications of the supercritical CO_2 power cycle, the maximal temperature at state 1 (T_1) was set equal to 600 ºC. Table 1 shows three important variables for the evaluation of the cycle: the efficiency (η), the specific work (w_{net}), and the mass flow rate of the working fluid in the cycle ($\dot{m}_{cycle}$). The differences in the energetic efficiency are rather small compared to the corresponding differences in the specific power of the cycle and the mass flow rate of the working fluid. Based on the results obtained from the sensitivity analysis, the cycle with p_1 =300 bars has been selected for further evaluation. The simulation data are given in Table 2 and illustrated in Figure 2b.

We obtained the following results: η=0.35%; $\dot{Q}_{COL}$= 18,565 kW; $\dot{Q}_{HE}$= 28,565 kW; $\dot{Q}_{RHE}$= 43,013 kW; $\dot{W}_{CM}$= 4,689 kW, and $\dot{W}_{EX}$= 14,689 kW.

Table 1. Selected characteristics of the supercritical CO2 cycle with T_1 =600ºC

p_1 [bar]	p_4^{opt} [bar]	η [-]	w_{net} [kJ/kg]	$\dot{m}_{cycle}$ [kJ/kg]
200	75	0.33	73.15	136.7
250	90	0.34	83.95	119.1
300	105	0.35	94.97	105.3

Table 2. Thermodynamic data for the supercritical CO2 cycle with T_1 =600ºC

Stream	T [°C]	p [bar]	h [kJ/kg]	e [kJ/kg]
1	600	300	585.5	590.2
2	476	111.4	446.0	444.2
3	132	108.2	37.6	263.4
4	50	105	-138.7	242.4
5	112	318.3	-94.2	281.5
6	386	309	314.3	432.4
0	40	1	11.9	0

For the exergetic analysis, we assumed for the reference state T_0 =40ºC and p_0 =1 bar. Since no chemical reaction takes place within the system, only physical exergies have been considered (Table 2). The exergetic analysis is conducted using the approach "exergy of fuel/exergy of product" (Bejan et al., 1996). The results are given in Table 3.

Based on these results we can conclude that the main source of thermodynamic inefficiencies (exergy destruction) is the RHE followed by the cooler.

To better understand the operation of the system from the thermodynamic viewpoint, we conducted an advanced exergetic analysis. The results are presented in Table 4. The methodology and applications of an advanced exergetic analysis can be found in several publications by Morosuk and Tsatsaronis (for example, Morosuk & Tsatsaronis, 2008 and 2012).

Table 3. Conventional exergetic analysis of the supercritical CO2 cycle with T_1 =600ºC

Component	$\dot{E}_{F,k}$ [kW]	$\dot{E}_{P,k}$ [kW]	$\dot{E}_{D,k}$ [kW]	ε_k [%]	y_k [%]
CM	4,688	4,114	574	87.7	3.4
COL	2,215	1,249	966	56.4	5.7
EX	15,377	14,689	688	95.5	4.1
HE	16,939	16,623	316	98.1	1.9
RHE	19,036	15,891	3,145	83.5	18.6
Overall system	16,939	10,000	5,689	59.0	33.6

Table 4. Advanced exergetic analysis of the supercritical CO2 cycle with T_1 =600ºC

Compo-nent	$\dot{E}_{D,k}^{EN}$ [kW]	$\dot{E}_{D,k}^{EX}$ [kW]	$\dot{E}_{D,k}^{UN}$ [kW]	$\dot{E}_{D,k}^{AV}$ [kW]	Splitting $\dot{E}_{D,k}^{real}$ [kW]			
					$\dot{E}_{D,k}^{UN}$		$\dot{E}_{D,k}^{AV}$	
					$\dot{E}_{D,k}^{UN,EN}$	$\dot{E}_{D,k}^{UN,EX}$	$\dot{E}_{D,k}^{AV,EN}$	$\dot{E}_{D,k}^{AV,EX}$
CM	324	250	313	261	157	156	167	94
COL	739	257	505	490	386	119	353	137
EX	578	110	330	358	274	56	304	54
HE	258	58	198	118	142	56	116	2
RHE	2,297	148	1,985	1160	1,913	72	1,084	76

Table 4 shows that for all components of the system the endogenous inefficiencies are always higher than the exogenous ones ($\dot{E}_{D,k}^{EN} > \dot{E}_{D,k}^{EX}$). This is particularly true when the *avoidable* endogenous inefficiencies are compared with the *avoidable* exogenous (last two columns of Table 4). This means that the interconnections among the components of the considered system are not very strong and each component can be optimized pretty much in isolation, to obtain an overall optimal design.

The values of unavoidable inefficiencies ($\dot{E}_{D,k}^{UN}$) are comparable with those of the avoidable inefficiencies ($\dot{E}_{D,k}^{AV}$). This indicates that the system has still some potential for improvement. By splitting the values given in the first four columns of Table 4 into $\dot{E}_{D,k}^{UN,EN}$, $\dot{E}_{D,k}^{UN,EX}$, $\dot{E}_{D,k}^{AV,EN}$ and $\dot{E}_{D,k}^{AV,EX}$ (see last four columns of Table 4), more detailed useful information is obtained, for example: $\dot{E}_{D,k}^{AV,EN} >> \dot{E}_{D,k}^{AV,EX}$.

The conventional exergetic analysis emphasizes more the relative importance of RHE compared to the remaining four components than the advanced analysis does. This conclusion results from a comparison of the values among the components given in the third column of Table 3 and in the second to last column of Table 4.

Currently some options are under investigations for improving the overall system. The first results will be reported during the presentation in the conference.

CONCLUSIONS

The application of an advanced exergetic analysis to a simple supercritical CO_2 cycle was demonstrated. This system can be improved by improving the components in isolation, because the *avoidable* inefficiencies caused by the components interconnections are relatively low.

The most important component from the thermodynamic viewpoint is the regenerative heat exchanger. System designers should focus on this component.

Future investigations will include the evaluation of more complex configurations and the application of advanced exergoeconomics to identify the configurations and the system parameters that will lead to lower cost of electricity.

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