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Advanced exergetic analysis: Approaches for splitting the exergy destruction into endogenous and exogenous parts

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ABSTRACT

The irreversibilities (exergy destruction) within a component of an energy conversion system can be represented by two parts. The first part depends on the inefficiencies of the considered component while the second part depends on the system structure and the inefficiencies of the other components of the overall system. Thus, the exergy destruction occurring within a component can be split into two parts: (a) endogenous exergy destruction due exclusively to the performance of the component being considered and (b) exogenous exergy destruction caused also by the inefficiencies within the remaining components of the overall system. The paper discusses four different approaches developed by the authors for calculating the endogenous part of exergy destruction as well as the approach based on the structural theory. The advantages, disadvantages and restrictions for applications associated with each approach are presented. It is concluded that all approaches developed by the authors lead to comparable and acceptable results, whereas the structural theory approach should not be used for calculating the endogenous part of exergy destruction because it delivers unacceptable results. Splitting the exergy destruction into endogenous and exogenous parts improves our understanding of the interactions among system components and provides very useful information for improving an exergy conversion system, particularly when this concept is combined with the concept of avoidable and unavoidable exergy destruction.

1. Introduction

Exergy analysis is relevant in identifying and quantifying both the consumption of useful energy (exergy) used to drive a process as well as the irreversibilities (exergy destructions) and the losses of exergy. The latter are the true inefficiencies and, therefore, an exergy analysis can highlight the areas of improvement of a system. Exergy measures the material's true potential to cause a change. Throughout the years such analysis has been extensively discussed and applied to a wide variety of energy conversion systems, for example see Refs. [1–4].

All real processes are irreversible due to effects such as chemical reaction, heat transfer through a finite temperature difference, mixing of matter at different compositions or states, unrestrained expansion, and friction [1–4]. A conventional exergy analysis identifies the system components with the highest exergy destruction and the process that cause them. Efficiencies within a system's component can then be improved by reducing the exergy being destroyed within the component. However, given present technical limitations, part of the exergy destruction and losses may be unavoidable (described by splitting the exergy destruction into unavoidable and avoidable parts [6–9]), part may be due to the exergy destruction occurring within the other components of the energy conversion system being considered (exogenous exergy destruction [8–12]), and hence it may be worthwhile to improve the other components and not just the component with the highest exergy destruction.

It is therefore important to understand the genesis of the rate of exergy being destroyed in a component's process. Hence by splitting the exergy destruction within a component a better approach concerning the improvement of the energy conversion system can be attained.

A detailed exergy analysis, in which the exergy destruction is split into the previously mentioned parts, is called *advanced exergy analysis*. Such analysis facilitates the improvement of an exergy conversion system from the viewpoints of thermodynamics, economics and environmental impact.

The theory of splitting the exergy destruction allows for the further understanding of the exergy destruction values obtained from an exergy analysis and hence improves the accuracy of the analysis, thereby facilitating the improvement of energy conversion systems.

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Nomenclature

$\dot{E}$	exergy rate (W)
e	specific exergy (J/kg)
$\dot{m}$	mass flow rate (kg/s)
p	pressure (bar)
$\dot{Q}$	heat rate (W)
s_{gen}	specific entropy generation (J/kg K)
T	temperature (K)
$\dot{W}$	power (W)

Greek symbols

Δ	difference
ε	exergetic efficiency (dimensionless)
η	isentropic efficiency (dimensionless)
λ	air fuel ratio (dimensionless)

Abbreviations

AC	air compressor
CC	combustion chamber
CM	compressor
CD	condenser
EV	evaporator
GT	gas turbine (expander)
TV	throttling valve

Subscripts

CD	condensation
D	destruction
EV	evaporation
F	fuel
H	point of a hybrid cycle
k	*k*th component
L	losses
others	other components
P	product
R	point of a real cycle
RU	point of a cycle with unavoidable exergy destruction
T	point of a theoretical cycle
tot	overall system
0	thermodynamic environment

Superscripts

ch	chemical exergy
EN	endogenous
EX	exogenous
ID	ideal system
k	*k*th component
ph	physical exergy
R	real system

The main equations for the exergy analysis of the *k*th component and of the overall system are the same [1], but there is one difference associated with the treatment of the exergy losses: It is assumed that the system boundaries used for all exergy balances are at the temperature T_0 of the reference environment, and therefore, there are no exergy loses associated with the *k*th component [5]. Exergy losses appear only at the level of the overall system. Thus, the exergy balances are:

- for the *k*th component

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

- for the overall system

$$\dot{E}_{F,tot} = \dot{E}_{P,tot} + \sum_k \dot{E}_{D,k} + \dot{E}_{L,tot}. \tag{2}$$

The exergetic efficiency for the *k*th component is

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

with the exergy destruction given by

$$\dot{E}_{D,k} = T_0 \dot{m}_k s_{gen,k}. \tag{4}$$

The irreversibility within a component of an energy conversion system can be represented by two parts. The first part depends on the inefficiencies of the considered component (expressed by the specific entropy generation within the component $s_{gen,k}$) and the second part depends on the system structure and the inefficiencies of the other components of the overall system (expressed mainly by changes in the mass flow rate $\dot{m}_k$).

For considering the interactions among system components, the idea of introducing the *endogenous* and *exogenous* exergy destruction associated with the *k*th component was formulated in [4]

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX}, \tag{5}$$

where the *endogenous* exergy destruction associated with the *k*th component ($\dot{E}_{D,k}^{EN}$) is that part of the entire exergy destruction within the same component ($\dot{E}_{D,k}$) that would still appear when all other components operate in an ideal way and the *k*th component operates with its real exergetic efficiency. It is apparent that the endogenous exergy destruction is associated only with the inefficiencies within the *k*th component.

The *exogenous* exergy destruction ($\dot{E}_{D,k}^{EX}$) is the remaining part of the entire exergy destruction within the *k*th component, i.e. the exogenous exergy destruction is simultaneously due to the inefficiencies of the *k*th component and to the inefficiencies of the remaining components.

The determination of the endogenous and exogenous exergy destruction in the *k*th component indicates a way for optimizing the *k*th component and the overall system. Decreasing the value of the endogenous exergy destruction in the *k*th component (through improving the *k*th component itself) promotes, in general, also the decrease of the exogenous part of the exergy destruction in other components, i.e. other components will, in general, show a reduced exergy destruction "automatically".

2. Energy conversion systems

2.1. Theoretical system

A detailed analysis of a theoretical energy conversion system consisting of three components in series for illustrating the idea of splitting the exergy destruction in endogenous and exogenous parts (Fig. 1) have been presented in [4,9,10]. All assumptions for the analysis are given in Fig. 1. In addition, we assume that the rate at which the fuel exergy is converted to product exergy in a component is specific to the component itself hence the values of exergetic efficiencies (ε_A, ε_B and ε_C) of the components are independent from each other. If $\dot{E}_{P,tot} = const$, then the exergy destruction in component C is totally endogenous because the value $\dot{E}_{D,C}$ is a function of the irreversibilities in this component only. The exergy destruction in component B depends on the exergetic efficiencies of both components B and C; similarly the exergy destruction in component A depends on the exergetic efficiencies of components A, B and C. Therefore, there are *endogenous* and *exogenous* parts of the exergy destruction for components A and B. It should be emphasized that the assumptions associated with the theoretical system shown in Fig. 1 are not necessary for calculating the endogenous and exogenous parts of exergy destruction in a real system.

Theoretically, the endogenous exergy destruction of the kth component within a defined system can be found by setting the exergetic efficiency of all other components within the system to 1 and noting the exergy destruction of the said component operating at its specified exergetic efficiency ε_k. However, a problem arises when there is chemical reaction taking place within any of the components within the system or when there is heat transfer, because no ideal conditions ($\varepsilon_k = 1$) can be defined, in general, for a chemical reactor or a heat exchanger. Such is the case of the combustion chamber within a power system.

Hence we need to develop a different approach for splitting the exergy destruction into endogenous and exogenous parts for the kth component of a real energy conversion system.

2.2. Vapor-compression refrigeration machine

Here we consider a simple compression refrigeration machine (Fig. 2) consisting of the four components: compressor (*CM*), condenser (*CD*), throttling valve (*TV*) and evaporator (*EV*). The exergy destruction rate in the components of this simple refrigeration machine are calculated by $\dot{E}_{D,CM} = \dot{W}_{CM} - (\dot{E}_2 - \dot{E}_1)$, $\dot{E}_{D,CD} = (\dot{E}_2 - \dot{E}_3) - (\dot{E}_7 - \dot{E}_6)$, $\dot{E}_{D,EV} = (\dot{E}_4 - \dot{E}_1) - (\dot{E}_9 - \dot{E}_8)$ and $\dot{E}_{D,TV} = (\dot{E}_3^M - \dot{E}_4^M) - (\dot{E}_4^T - \dot{E}_3^T) = \dot{E}_3 - \dot{E}_4$. The thermal part of the physical exergy at points 3 and 4 is denoted by superscript T and the mechanical part denoted by superscript M. Notice that only the physical exergy associated with all material streams of the vapor-compression refrigeration machine using one-component working fluid is needed. The product of the overall refrigeration machine is the product of evaporator $\dot{E}_{P,tot} = \dot{E}_{P,EV} = \dot{E}_9 - \dot{E}_8$ which remains constant for the analysis.

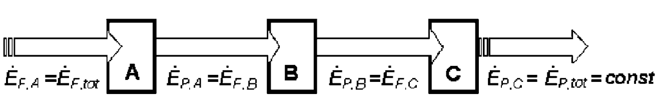

Fig. 1. Theoretical case of an energy-conversion system.

For demonstration purposes, the following operation conditions are assumed: The working fluid for the refrigeration machine is R22, $\dot{Q}_{cold} = 100$ kW; the condenser is cooled by air, $T_6 = 293$ and $T_7 = 303$ K, therefore T_{CD} assumed to be 313 K; the secondary working fluid is also air $T_8 = 268$ K; $T_9 = 258$ K, in this way $T_{EV} = 248$ K; the isentropic efficiency of the compressor is $\eta_{CM} = 0.8$. To simplify the calculations, the irreversibilities due to friction in the heat exchangers were not included in the analysis.

The thermodynamic data of the real cycle of the refrigeration machine and the results from the conventional exergetic analysis are given in Tables 1 and 2, respectively.

2.3. Gas-turbine power system

Fig. 3 shows a simple gas-turbine power system consisting of three components: air compressor (*AC*), combustion chamber (*CC*) and expander (*GT*). The product of the overall system is the power rate and it is kept constant in the analysis: $\dot{E}_{P,tot} = \dot{W}_{net} = const$. The following equations are used to calculate the exergy destruction of the components of this system: $\dot{E}_{D,AC} = \dot{W}_{AC} - (\dot{E}_2 - \dot{E}_1)$, $\dot{E}_{D,CC} = \dot{E}_3 - (\dot{E}_4 - \dot{E}_2)$ and $\dot{E}_{D,GT} = (\dot{E}_4 - \dot{E}_5) - \dot{W}_{GT}$.

For demonstration purposes, the following operation conditions are assumed: $\dot{W}_{net} = 30$ MW, $\eta_{AC} = 0.8$, $\eta_{GT} = 0.88$, $T_1 = 298$ K, $T_4 = 1230$ K and $p_4/p_1 = 10$.

The thermodynamic data of the real cycle of the gas-turbine power system and the results from the conventional exergetic analysis are given in Tables 3 and 4, respectively.

Table 1
Thermodynamic data for the real cycle of a simple refrigeration machine (for the exergetic values we assumed $T_0 = 293$ K and $p_0 = 1$ bar).

Stream	Working fluid	$\dot{m}$ (kg/s)	T (K)	p (bar)	e^{ph} (kJ/kg)
1R	R22	0.692	248	2.015	21.81
2R	R22	0.692	362	15.34	76.17
3R	R22	0.692	313	15.34	59.95
4R	R22	0.692	248	2.015	47.93
6	Air	14.39	293	1	0
7	Air	14.39	303	1	0.168
8	Air	9.942	268	1	1.138
9	Air	9.942	258	1	2.285

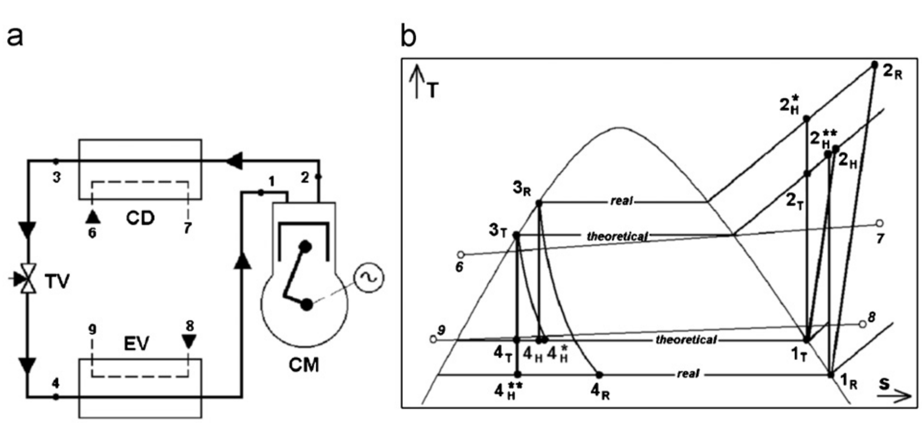

Fig. 2. Simple refrigeration machine: (a) schematic; (b) real, theoretical and hybrid cycles on a *T*–*s* diagram.

Table 2
Summary of the results from the conventional exergy analysis and some advanced exergy analyses of a simple compression refrigeration machine.

Component	Real cycle				Approach based on thermodynamic cycles		Engineering approach	
	$\dot{E}_{F,k}$ (kW)	$\dot{E}_{P,k}$ (kW)	$\dot{E}_{D,k}$ (kW)	ε_k (dimensionless)	$\dot{E}^{EN}_{D,k}$ (kW)	$\dot{E}^{EX}_{D,k}$ (kW)	$\dot{E}^{EN}_{D,k}$ (kW)	$\dot{E}^{EX}_{D,k}$ (kW)
CM	44.871	37.446	**7.425**	0.83	4.531 (61%)	2.894 (39%)	4.838 (65.2%)	2.587 (34.8%)
CD	11.069	2.416	**8.653**	0.22	6.350 (73.4%)	2.303 (26.6%)	6.769 (78.2%)	1.884 (21.8%)
TV	28.105	19.870	**8.235**	0.71	3.383 (41.1%)	4.852 (58.9%)	Not available	
EV	18.135	11.410	**6.725**	0.63	6.725 (100%)	0	6.725 (100%)	0
Overall system	44.871	11.410	**31.038**	0.25	20.989 (67.6%)	10.049 (32.4%)		

Bold values represent the total exergy destruction in the corresponding component.

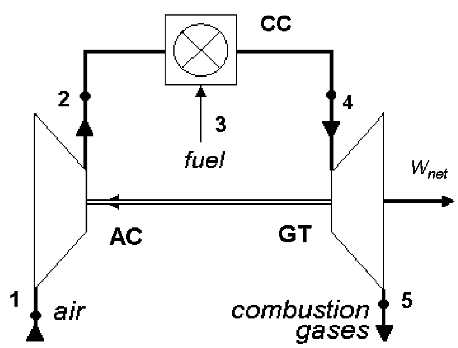

Fig. 3. Schematic of a simple gas-turbine power system.

Table 3
Thermodynamic data for the real cycle of a simple gas-turbine power system (for the exergetic values we assumed $T_0 = 298$ K and $p_0 = 1.013$ bar).

Stream	Working fluid	$\dot{m}$ (kg/s)	T (K)	p (bar)	e^{ph} (MJ/kg)	e^{ch} (MJ/kg)	e (MJ/kg)
1	Air[a]	140.70	298	1.013	0	0	0
2	Air[a]	140.70	635	10.13	0.314	0	0.315
3	CH4	2.16	298	12.00	0.382	51.382	51.763
4	Comb. gases[b]	142.86	1230	10.13	0.794	0.002	0.796
5	Comb. gases[b]	142.86	764	1.013	0.205	0.002	0.207

[a] Molar composition of air: Ar—0.92%, CO_2—0.03%, H_2O—1.89%, N_2—76.61% and O_2—20.55%.
[b] Molar composition of combustion gases: Ar—0.89%, CO_2—2.74%, H_2O—7.10%, N_2—74.59% and O_2—14.68%.

3. Approaches for splitting the exergy destruction

3.1. Approach based on thermodynamic cycles

This approach for splitting the exergy destruction into endogenous and exogenous parts is based on the analysis of a thermodynamic cycle. It has been described in detail in [8–11] where applications to different types of refrigeration machines are given.

The *real cycle* of the vapor-compression refrigeration machine is 1_R–2_R–3_R–4_R (Fig. 2b). All irreversibilities are included here.

In the *theoretical cycle* (1_T–2_T–3_T–4_T in Fig. 2b), the operating conditions for each component should correspond to either $\dot{E}_{D,tot} = 0$ (where it is possible) or to $\dot{E}_{D,tot} = \min$ (for example in a heat exchanger with different heat capacity rates of the working fluids: in this case $\Delta T_{min} = 0$). For the theoretical cycle, $T_{EV} = T_9 = 258$ K and $T_{CD} = 302$ K. In calculating the value of the endogenous exergy destruction in a component, we neglect the effect that the small exergy destruction within some theoretical heat exchangers has on these values.

The endogenous part of the exergy destruction in the kth component is calculated through an analysis of the hybrid cycle. The hybrid cycle represents the theoretical cycle with irreversibilities in the kth component only. The number of the hybrid cycles that should be created for the analysis is equal to the number of the components in the overall system: hybrid cycle for the compressor ($\dot{E}^{EN}_{D,CM}$) is 1_T–2_H–3_T–4_T; hybrid cycle for the condenser ($\dot{E}^{EN}_{D,CD}$) is 1_T–$2_H{}^*$–3_R–4_H; hybrid cycle for the throttling valve ($\dot{E}^{EN}_{D,TV}$) is 1_T–2_T–3_T–$4_H{}^*$; and hybrid cycle for the evaporator ($\dot{E}^{EN}_{D,EV}$) is 1_R–$2_H{}^{**}$–3_T–$4_H{}^{**}$. The results obtained for the endogenous part of the exergy destruction using the thermodynamic cycle approach are given in Table 2.

This approach cannot be applied to an energy conversion system when it is not possible to create such a theoretical cycle for the system being considered. For example, in a gas-turbine power system, it is not possible to define an ideal combustion process using the approach of thermodynamic cycles.

3.2. Engineering approach

An engineering approach for calculating the endogenous and exogenous parts of the exergy destruction is based on results obtained from the sensitivity exergetic analysis of the overall energy conversion system and on a further graphical representation of these results. This approach, which is described in detail in [12], is based on Eq. (2).

For an ideal system (superscript *ID*) producing a constant supply of product, the exergy balance can be written as

$$\dot{E}^{ID}_{F,tot} - \dot{E}^{ID}_{L,tot} = \dot{E}_{P,tot}. \quad (6)$$

If irreversibility is introduced in one component (kth component—superscript k) in the system, then Eq. (6) should be changed to

$$(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{k}_{F,tot}) - (\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{k}_{L,tot}) = \dot{E}_{P,tot} + \dot{E}^{EN}_{D,k} \quad (7)$$

because additional exergetic resources $\Delta\dot{E}^{k}_{F,tot}$ need to be supplied while the loss from the overall system increases by $\Delta\dot{E}^{k}_{L,tot}$. The value of the exergy destruction within the kth component $\dot{E}_{D,k}$ (which, under the conditions considered here, is equal to the exergy destruction within the overall system) is equivalent to the

Table 4
Summary of the results from the conventional exergy analysis and some advanced exergy approaches for calculating the values of endogenous exergy destruction for a simple gas-turbine power system.

Component	Real cycle				Approaches for calculating the value of $\dot{E}^{EN}_{D,k}$ (MW)			
	$\dot{E}_{F,k}$ (MW)	$\dot{E}_{P,k}$ (MW)	$\dot{E}_{D,k}$ (MW)	ε_k (dimensionless)	Engineering	Exergy balance	Equivalent component	Structural theory
AC	48.92	44.04	**4.88**	0.90	4.20 (86.1%)	4.30 (88.1%)	4.00 (82.0%)	4.88 (100%)
CC	112.13	69.73	**42.39**	0.62	27.85 (65.7%)	27.85 (65.7%)	27.85 (65.7%)	35.05 (82.7%)
GT	84.24	79.80	**4.44**	0.95	3.58 (80.6%)	3.62 (81.5%)	3.46 (84.7%)	4.40 (99.1%)
Overall system	112.13	30.0	**51.71**	0.27	35.63 (68.9%)	35.77 (69.2%)	35.31 (68.3%)	44.33 (85.7%)

Bold values represent the total exergy destruction in the corresponding component.

endogenous exergy destruction for the kth component, i.e. $\dot{E}_{D,tot} = \dot{E}_{D,k} = \dot{E}^{EN}_{D,k}$.

For the real energy conversion system (superscript R), Eq. (2) can be written as

$$(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{R}_{F,tot}) - (\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{R}_{L,tot}) = \dot{E}_{P,tot} + \dot{E}_{D,k} + \dot{E}_{D,others}, \tag{8}$$

where $\Delta\dot{E}^{R}_{F,tot}$ and $\Delta\dot{E}^{R}_{L,tot}$ represent the increases in the exergy of fuel and in the exergy loss, respectively, as a result of the exergy destructions in all components.

Let us assume that $\dot{E}_{D,others}$ in Eq. (8) tends to zero. In the same equation the expression $(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{R}_{F,tot}) - (\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{R}_{L,tot})$ approaches the expression $(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{k}_{F,tot}) - (\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{k}_{L,tot})$ in Eq. (7) and in parallel $\dot{E}_{D,k}$ approaches $\dot{E}^{EN}_{D,k}$. Hence, by plotting $[(\dot{E}^{ID}_{F,tot} + \Delta\dot{E}^{R}_{F,tot}) - (\dot{E}^{ID}_{L,tot} + \Delta\dot{E}^{R}_{L,tot}) - \dot{E}_{P,tot}]$ vs. $\dot{E}_{D,others}$ the value of $\dot{E}^{EN}_{D,k}$ can be obtained at the intercept where $\dot{E}_{D,others} = 0$ (Fig. 4).

Since the endogenous exergy destruction within a component is a function of the component's exergetic efficiency, the exergetic efficiency of the kth component must be kept constant ($\varepsilon_k = const$) while $\dot{E}_{D,others}$ is being varied.

The equation of this regression line is of the type $y = bx+c$, where the value of coefficient c is equal to the value of the endogenous part of the exergy destruction in the kth component ($c \equiv \dot{E}^{EN}_{D,k}$).

The main question is: Is this line a straight line or a curve? The theoretical energy conversion system (Fig. 1) is used to prove the linear dependence between $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ and $\dot{E}_{D,others}$.

The value of the total exergy destruction is

$$\begin{aligned}\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot} &= \dot{E}_{D,tot} = \dot{E}_{D,A} + \dot{E}_{D,B} + \dot{E}_{D,C} \\ &= \underbrace{\frac{\dot{E}_{P,tot}}{\varepsilon_C \varepsilon_B}\left(\frac{1}{\varepsilon_A} - 1\right) + \underbrace{\frac{\dot{E}_{P,tot}}{\varepsilon_C}\left(\frac{1}{\varepsilon_B} - 1\right) + \dot{E}_{P,tot}\left(\frac{1}{\varepsilon_C} - 1\right)}_{f_2(z)}}_{f_1(z)}.\end{aligned} \tag{9}$$

We define the variable $z = 1/\varepsilon_C\varepsilon_B$. If a linear dependence exists between two functions, $f_1(z)$ and $f_2(z)$, then any two selected values a_1 and a_2, where a_1 and a_2 are real numbers and

$$a_1 f_1(z) + a_2 f_2(z) = 0 \tag{10}$$

should not be both equal to zero [13].

Considering the theoretical process in Fig. 1 and Eq. (9), and given that the endogenous exergy destruction in component A is being investigated, then, $f_1(z)$ represents $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ and

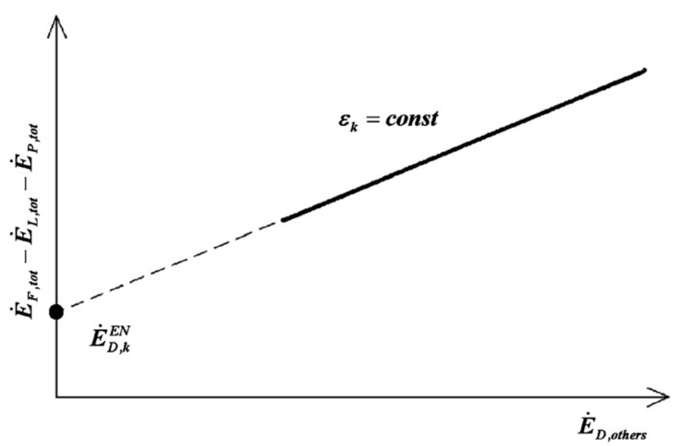

Fig. 4. Illustration for the engineering approach.

$f_2(z)$ represents $\dot{E}_{D,others}$. Eq. (9) in the form of Eq. (10) becomes

$$a_1 \frac{\dot{E}_{P,tot}}{\varepsilon_C \varepsilon_B}\left(\frac{1}{\varepsilon_A}\right) - a_1 \dot{E}_{P,tot} + a_2 \frac{\dot{E}_{P,tot}}{\varepsilon_C \varepsilon_B} - a_2 \dot{E}_{P,tot} = 0 \tag{11a}$$

or

$$a_1 \frac{\dot{E}_{P,tot}}{\varepsilon_A} z - a_1 \dot{E}_{P,tot} + a_2 \dot{E}_{P,tot} z - a_2 \dot{E}_{P,tot} = 0. \tag{11b}$$

Differentiating Eq. (11), we obtain

$$a_1 \frac{\dot{E}_{P,tot}}{\varepsilon_A} + a_2 \dot{E}_{P,tot} = 0 \tag{12a}$$

or

$$a_1 \frac{1}{\varepsilon_A} + a_2 = 0. \tag{12b}$$

Therefore, a_1 can be 1 and a_2 can be $-1/\varepsilon_A$. Hence, there exists a combination of non-zero values for a_1 and a_2. Therefore a linear dependence exists.

This approach is attractive because it does not need additional simulations of the energy conversion system (as it is necessary for other approaches). The following additional guidelines are suggested in [12] in plotting the graph $\dot{E}_{F,tot} - \dot{E}_{L,tot} - \dot{E}_{P,tot}$ vs. $\dot{E}_{D,others}$ for correctly determining the value of $\dot{E}^{EN}_{D,k}$:

- Before reducing the exergy destruction in the other components, set the pressure drops in these components to zero with the exception of the component under study.

- The exergy destruction in the other components must be reduced in such a way that at the point where $\dot{E}_{D,others} = 0$, the values of exergy destruction within each individual component $\dot{E}_{D,1}, \ldots, \dot{E}_{D,n-1}$ (where n is the number of components in the system) are all zero.
- When the system is large, it is better to concentrate on reducing the exergy destruction in the components with the highest exergy destruction rates.

The engineering approach can be applied to any energy conversion system and particularly to systems in which chemical exergy is converted to other forms. However, this approach cannot be used for calculating the endogenous exergy destruction in a component if the main condition required by this method $\varepsilon_k = const$, cannot be satisfied while $\dot{E}_{D,others}$ is being varied. An example of such a component is the throttling valve.

The following equations were obtained in this approach for calculating the value of $\dot{E}^{EN}_{D,k}$:

- For the simple refrigeration machine (Fig. 2a): compressor $y = 1.1111x+4.838$ (i.e., $\dot{E}^{EN}_{D,CM} = 4.838\,\text{kW}$), condenser $y = 0.0842x+6.769$ (i.e., $\dot{E}^{EN}_{D,CD} = 6.769\,\text{kW}$), evaporator $y = 6.725$ (i.e., $\dot{E}^{EN}_{D,CM} = 6.725\,\text{kW}$).
- For the simple gas-turbine power system (Fig. 3): air compressor $y = 0.99x+4.20$ (i.e., $\dot{E}^{EN}_{D,AC} = 4.20\,\text{MW}$), combustion chamber $y = 3.44x+27.85$ (i.e., $\dot{E}^{EN}_{D,CC} = 27.85\,\text{MW}$), gas turbine $y = 0.9965x+3.58$ (i.e., $\dot{E}^{EN}_{D,CM} = 3.58\,\text{MW}$).

3.3. Exergy balance method

The application of the exergy balance method is illustrated for the combustion chamber of the simple gas-turbine power system (Fig. 3). The adiabatic air compressor is ideal when $\varepsilon_{AC} = 1$ (i.e., $\eta_{AC} = 1$); the adiabatic expander is ideal when $\varepsilon_{EX} = 1$ (i.e., $\eta_{EX} = 1$); the ideal operating conditions for the combustion chamber (i.e., $\varepsilon_{EX} = 1$, or $\dot{E}_{D,CC} = 0$) are based on an exergy balance for the combustion chamber:

$$\dot{E}_4 = \dot{m}^*_4 \cdot e_4 = \dot{E}_2 + \dot{E}_3. \tag{13}$$

In this method an ideal combustion chamber is defined as one where the exergy balance (Eq. (13)) is fulfilled. In this case the ideal combustion chamber is defined as follows: The air fuel ratio $\lambda = \dot{m}_2/\dot{m}_3$ remains unchanged as for the real case, and the value of $\dot{E}_4$ is the exergy rate of the combustion gases required to satisfy Eq. (13). The temperature, the composition, and thus, the specific exergy e_4 of the combustion gases exiting the combustion chamber are kept the same as in the real case. At the "ideal" operation, the energy balance and the mass balance for the combustion chamber are not fulfilled when this method is applied. From Eq. (13) a fictitious mass flow rate $\dot{m}^*_4$ is determined that satisfies this equation. This mass flow rate replaces the mass flow rate of combustion gases in all downstream components, when their ideal operation is considered. The results are presented in Table 4.

3.4. Equivalent component method

In this method the ideal combustion chamber is approximated with an ideal heat exchanger. All state point pressure and temperature values are maintained and the working fluid used throughout the cycle is air. The results are also shown in Table 4. The main disadvantage of this method is that different working fluids are used when studying the real and the ideal operations.

3.5. Structural theory and malfunction/dysfunction analysis

Another approach for determining the effect of the inefficiencies within one component on the other components was proposed by Valero and co-workers and applied to the structural theory [14] and to the thermoecomonic diagnosis (for example, [15,16]). The analysis of *malfunction/dysfunction* is based on Eqs. (1)–(2). For the analysis we need to know the design conditions and the real operating conditions for an existing energy conversion system.

Any additional irreversibilities (irreversibilities which are not included in the design case of the exergy conversion system) within the kth component implies additional fuel being supplied to the overall system. According to this theory, the additional fuel being supplied to the overall system ($\Delta\dot{E}_{F,tot}$) is distributed to the system components as additional fuel for each component ($\Delta\dot{E}_{F,k}$). The additional fuel for each component is caused through a *malfunction* (produced by an increase of the specific consumption of the component itself) and *dysfunction* (introduced in the component being considered by malfunctions of other components).

It is necessary to note that the analysis of *malfunction/dysfunction* and the theory of splitting the exergy destruction into *endogenous/exogenous* parts are completely different approaches. The main differences between these approaches are the following:

Malfunction/dysfunction analysis means splitting only the additional part of the fuel in the kth component $\Delta\dot{E}_{F,k}$, which should be known. The main assumptions for the analysis of *malfunction/dysfunction* include: (a) $\dot{E}_{P,(k-1)} = \dot{E}_{F,k}$ (Fig. 5) which is possible only for a theoretical system, for example, the one shown in Fig. 1, and (b) each component is analyzed in isolation from the overall system, therefore the effect of the mass flow rate $\dot{m}_k$ (Eq. (4)) is neglected. In this way, this theory can be used only for the diagnosis of a real energy conversion system where the design conditions and the real operation conditions are not so far from each other.

On the other hand, the theory of splitting the exergy destruction into endogenous/exogenous parts is more general and can be applied to the diagnosis as well as to the design analysis and optimization of an energy conversion system. The definition of the endogenous part of the exergy destruction creates freedoms because it consists of two parts: necessary and sufficient. The necessary part is: The *endogenous* exergy destruction depends on the inefficiency of the kth component itself. The sufficient part is: when all other components operate in an ideal way.

The differences between malfunction/dysfunction analysis and splitting the exergy destruction into endogenous and exogenous parts become apparent when the algebraic formula which calculates the exergy destruction of the total system [14] is used

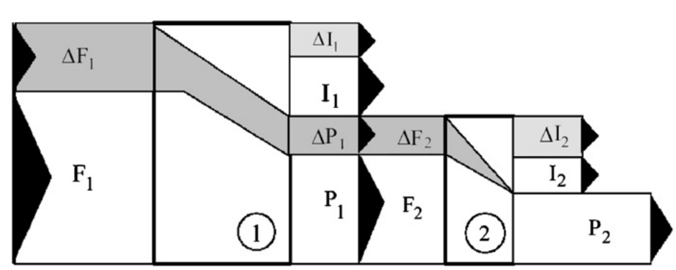

Fig. 5. Malfunction and fuel impact according to [15]. The variable ΔI_k in this figure represents the exergy destruction $E_{D,k}$. F_k and P_k correspond to $E_{F,k}$ and $E_{P,k}$.

for estimating the endogenous part of the exergy destruction within the kth component. The main assumption behind this method is that the product of the previous system becomes the fuel of the subsequent system. The latter allows one to create a fuel/product or $\langle FP \rangle$ matrix which can be used to mathematically define the exergy destruction in each component.

The total exergy destruction of any given system is defined as follows:

$$\dot{E}_{D,tot} = \sum_k \dot{E}_{D,k} = {}^t\mathbf{U}\mathbf{E}_{\mathbf{D,tot}} = {}^t\mathbf{U}(\mathbf{K}_D - \mathbf{U}_D)\mathbf{P}, \tag{14}$$

where ${}^t\mathbf{U}$ is the transpose unit vector, ${}^t\mathbf{U}\mathbf{E}_{\mathbf{D,tot}}$ is the matrix vector of the exergy destruction rate of the entire system, $\mathbf{K}^D$ is the inverse diagonal matrix containing the exergetic efficiency of each component, and $\mathbf{P}$ is the matrix vector of the product exergy from each component.

Applying Eq. (14) to the simple gas-turbine power system, the following equations for the exergy destruction in each component are obtained:

$$\dot{E}_{D,AC} \approx \frac{(\varepsilon_{AC} - 1)(\dot{E}_1 + d_2\dot{W}_{net})}{-1 + d_2}, \tag{15}$$

$$\dot{E}_{D,CC} \approx \frac{(1 - \varepsilon_{CC})(\dot{E}_1(-1 + d_1\varepsilon_{AC}\varepsilon_{GT}) + (-1 + d_1 d_2\varepsilon_{AC}\varepsilon_{CC})\dot{W}_{net})}{d_2(-1 + d_2)\varepsilon_{CC}\varepsilon_{GT}}, \tag{16}$$

$$\dot{E}_{D,GT} \approx \frac{(\varepsilon_{GT} - 1)(\dot{E}_1 + \dot{W}_{net})}{(-1 + d_2)\varepsilon_{GT}}, \tag{17}$$

where $d_1 = 1 - \dot{E}_5/\dot{E}_4$ is the ratio between the exergy used to operate the turbine and the exergy available for this purpose, and $d_2 = \dot{W}_{AC}/(\dot{W}_{AC} + \dot{E}_{P,tot})$ is the ratio between the power used to operate the compressor and the power generated by the turbine.

The endogenous exergy destruction in each component according to the method of structural theory was found by setting the exergetic efficiency of the other components in the system to 1. The results are shown in the last column of Table 4.

4. Results and discussion

The last four columns of Table 2 compare the results obtained by applying the approach based on thermodynamic cycles and the engineering approach to a simple compression refrigeration machine. From the advanced exergetic analysis (using the approach based of thermodynamic cycles) of the simple refrigeration machine as well as from work published elsewhere [17], we know that the exergy destruction within the evaporator of a vapor-compression refrigeration machine is only endogenous ($\dot{E}_{D,EV} = \dot{E}^{EN}_{D,EV}$). The same result we obtain using the engineering approach. The relative difference between the values of the endogenous exergy destruction is for the compressor 6.8%, and for the condenser 6.6%. Note that the values obtained by the engineering approach are higher. This difference is affected through the mass flow rate (Eq. (4)). In the thermodynamic approach, the real expansion process (throttling with $h_3 = h_4$) is replaced by a theoretical expander. In this way, we calculate the values $\dot{E}^{EN}_{D,CM}$, $\dot{E}^{EN}_{D,CD}$ and $\dot{E}^{EN}_{D,EV}$ by making the assumption that $s_3 = s_4$, therefore the expansion process does not affect the mass flow rate of the working fluid. In the engineering approach, the real expansion process in the throttling valve participates in the calculations and affects the mass flow rate of the working fluid within the refrigeration machine. A comparison of the values for the mass flow rate between the cycles 1_T–2_T–3_T–4_T and 1_T–2_T–3_T–4_H^* (Fig. 2b) shows that the value of $\dot{m}_{1_T-2_T-3_T-4_T}$ is by approximately 7% lower than the value of $\dot{m}_{1_T-2_T-3_T-4^*_H}$.

The results from the conventional and advanced exergetic analysis of the simple gas-turbine power system are given in Table 4. Initially we will discuss the results obtained from the authors' approaches (from "engineering" through "equivalent component"). The maximal difference among the values of $\dot{E}^{EN}_{D,AC}$ is 7.5%, and among the values of $\dot{E}^{EN}_{D,GT}$ is 4.6%. There is no deviation in the values of the endogenous exergy destruction for the combustion chamber because for this component of a simple open gas-turbine system an exact calculation of the endogenous exergy destruction is possible in all authors' approaches. We can conclude that the agreement of the results is in general satisfactory.

It is not possible to have experimental results from splitting the exergy destruction into endogenous and exogenous parts. Only a sensitivity analysis (calculations), or experimental data from a long time operation of a real energy conversion system can give us information on changes of the exergy destruction within the kth component due to irreversibilities in other components. For all components of the simple gas-turbine power system the value of $\dot{E}^{EN}_{D,k}$ is lower than the value of $\dot{E}_{D,k}$, therefore the value of $\dot{E}^{EX}_{D,k}$ is always positive. This means that decreasing the endogenous exergy destruction within a component leads to a decrease in the exogenous exergy destruction within other components. This fact is well established for simple gas-turbine systems.

Now we consider the data from the advanced exergy analysis using the symbolic algebraic approach (structural theory). For the air compressor and gas turbine we have $\dot{E}^{EN}_{D,k} \cong \dot{E}_{D,k}$. These results contradict the previous conclusion. The correct value for the endogenous exergy destruction within the combustion chamber can be obtained by applying the thermodynamic cycle approach and setting the exergetic efficiency of both the air compressor and the expander to 1. Then we obtain $\dot{E}^{EN}_{D,CC} = 28.54$ MW. This value, obtained by all the authors' approaches, deviates significantly from the value ($\dot{E}^{EN}_{D,CC} = 35.05$ MW) supplied by the symbolic algebraic approach. The following is a possible explanation for the unacceptably high values delivered by this approach. The mass flow rate of the working fluid at real operation conditions is higher than for any other operating conditions in which at least one component is ideal ($\varepsilon_k = const$). If a component is analyzed in isolation, then the value of $\dot{m}$ remains unchanged. In this way, all values of $\dot{E}^{EN}_{D,k}$ calculated by the symbolic algebraic approach are higher than the corresponding values calculated by the authors' approaches, where both variables of Eq. (4) ($\dot{m}$ and $s_{gen,k}$) are varied when calculating the value of $\dot{E}^{EN}_{D,k}$. Eqs. (15)–(17) are correct if the value of ε_k changes within a small range, but they cannot be used when only for one component $\varepsilon_k = const$ while for all other components $\varepsilon_k = 1$.

5. Conclusions

A general theory of splitting the exergy destruction into endogenous and exogenous parts as well as different approaches for realizing this splitting are presented in this paper. Knowledge of the endogenous and exogenous exergy destruction parts improves our understanding of the interactions among system components, and facilitates the optimization of the overall system.

The approach based on thermodynamic cycles and the engineering approach were applied to a vapor-compression refrigeration machine. The thermodynamic cycle approach is the most convenient one and provides the best results for systems for which a thermodynamic cycle can be defined. If this is not possible, two additional approaches developed by the authors and the engineering approach, which provide similar and acceptable results may be used. Application of these methods was demonstrated here with the aid of a simple gas-turbine system. The agreement of the results is in general satisfactory. In future applications the exergy balance method should be used in conjunction with more complex systems.

The application of a symbolic algebraic approach based on the structural theory to a simple gas-turbine system indicates that this approach is not appropriate to be used for calculating values of endogenous exergy destruction for system components, see for example Refs. [18,19].

More useful information is obtained when the concept of endogenous and exogenous exergy destruction is combined with the concept of avoidable and unavoidable exergy destruction [8,9,12,17–19].

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