EXERGY ANALYSIS OF A CO-GENERATION PLANT

by

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Limitations of traditional first-law analysis, based upon thermodynamic performance of process unit coupled with mass and energy balances, are not a serious limitation when dealing with familiar systems. However, when dealing with more uncongenial, complex ones, it provides incomplete insight for such evaluation. These limitations came from the fact that first-law analysis does not indicate the sources or magnitudes of entropy production, which is, by the second law, essential criterion for scaling losses. An evaluation of plant performance will usually require a comparison of the thermodynamic performance of process units with available data from existing plants. Therefore, exergy analysis is more than useful, providing information about magnitudes of losses and their distribution throughout the system as well. Such analysis is very thankful at the level of process units but applied on higher system levels e.g. the comparison of overall plant performance (total system) or the performance of subsystems, represents the valuable method for indicating where research resources can be directed to best advantage.

Key words: *co-generation plant, exergy, efficiency, losses*

Introduction

In the year 1938 Prof. Bošnjaković published a paper called "Kampf den Nichtumkehrbarkeiten" (Fight against irreversibilities) [1] in which he emphasizes the importance of irreversibility detection as well as the effort to reduce it in all real heat processes, since according to Gouy-Stodola theorem the loss of available work is directly proportional to the total entropy production in an observed process. The work loss on the other hand directly means destruction of primer (valuable) energy. Exactly this entropy production is in a direct connection with exergy, *i. e.* exergy destruction, so in the last decades many papers appear in which the exergy efficiencies are analyzed that are tied also to steam-turbine installations. A great contribute to the systematization and detail entropic analyzes of different, and among them steam-turbine installations, gives Bejan in his book "Entropy Generation Minimization" [2].

In one steam-turbine installation the irreversibility appears as a consequence of three focal points: the sole process of burning (furnace), the transfer of heat (exchanger) and of the friction (turbine, exchangers). This paper also displays with a simplified method the exergy efficiency of one selected steam-turbine co-generation installation, in total as well as of separate devices through which a selected steam-turbine installation is covered.

Method

Definition of exergy

Exergy is the maximum amount of work we can gain from the steady, open flow system that interacts with only one thermal reservoir – environment [3, 4].

Overall exergy of open thermodynamic systems consists of several exergies: kinetic exergy, potential exergy, physical or thermal exergy, and chemical exergy. First two exergies will be neglected because corresponding energy changes are neglected also [5]. Physical or thermal exergy is defined as maximum amount of work we can gain from the steady, open flow, system when it is brought to mechanical and thermal equilibrium with environment:

$$\dot{W}_{max} \quad E \quad q_m[(h \quad h_0) \quad T_{en}(s \quad s_0)] \tag{1}$$

Index 0 marks a condition of operating substance in mechanical and thermal equilibrium with environment. Chemical exergy can be defined as follows [6].

Chemical exergy is equal to the maximum amount of work obtainable when the substance under consideration is brought from the environmental state to the dead state by processes involving heat transfer and exchange of substances only with the environment.

That means that system in its "dead state" is in thermal, mechanical, and chemical equilibrium with environment *i. e.* it has no potential *at all*. In this paper chemical exergy analisys will be omitted so the term "dead state" refers only to thermal and mechanical equilibrium.

Exergy efficiencies

Because exergy, by its definition, represents maximal value for given initial state, it can be used as a criterion in comparing other irreversible processes.

Considering the fact that reversible processes are the processes without losses (can go in both ways) it is obvious that in the best case, exergy (reversible process) can remain unchanged, but not increased.

This fact can be written as:

$$\Sigma E_{in} \quad \Sigma E_{out} \quad \Sigma \dot{W} \quad \Sigma E_{loss} \tag{2}$$

Based on this fact, exergy efficiency can be defined. Several authors mentioned in references, provided definitions for exergy efficiencies. Among others, it is necessary to extract two different kinds: so called simple or universal and rational or functional exergy efficiency.

Simple exergy efficiency is defined as sum of exergy of outgoing flows and work divided by sum of exergy of ingoing flows, what can be written using (2), [7-9]

$$\varepsilon_s \quad \frac{\Sigma E_{out} \quad \Sigma \dot{W}}{\Sigma E_{in}} \quad 1 \quad \frac{\Sigma E_{loss}}{\Sigma E_{in}} \tag{3}$$

The equation shown above suggests that simple exergy efficiency represents portion of overall ingoing exergy that is lost ($\Sigma E_{loss} / \Sigma E_{inn}$), *i. e.* it's measure for a system's losses. Despite simple efficiency, rational exergy efficiency is defined as [7-9]:

$$\varepsilon_r \quad \frac{\Sigma E_{product}}{\Sigma E_{source}} \tag{4}$$

in which:

$\Sigma E_{product}$ is amount of exergy which is considered to be the system's product, desired effect, and

ΣE_{source} is a part of ingoing exergy used to achieve desired effect, product of the system.

From this definition, it is obvious that, for defining rational exergy efficiency, purpose and better understanding of a process itself is needed. More detail explanation about these efficiencies can be found in appendices A and B. It is important to emphasize that if any number of systems are compared with exergy analysis, they all must refer to the same condition of environment.

Physical model of the refrence plant

This section will apply above explained methods to evaluate each part and overall exergy efficiency of one co-generation plant. In fig. 1 we can see scheme of a plant that will be analyzed.

In the scheme can be seen that fuel and air enter the system in steam generator (St) where heat from combustion is used to superheat steam (stream "5") from preheated feed water (stream "4"). From steam generator, flue gasses exit the system. Superheated steam from steam generator is further taken to the turbine (T), $P = 50$ MW where expands until the pressure of the condenser (C), $p_c = 0.07$ bar is reached. Using cooling water, steam from the turbine is condensed and with pumps taken to the steam generator. On its way to the steam generator, condensate is preheated with a steam taken from the turbine in three preheaters (PH1, PH2, and PH3) and one deareator (De). Steam is also taken (stream "e") for simultaneous heat production.
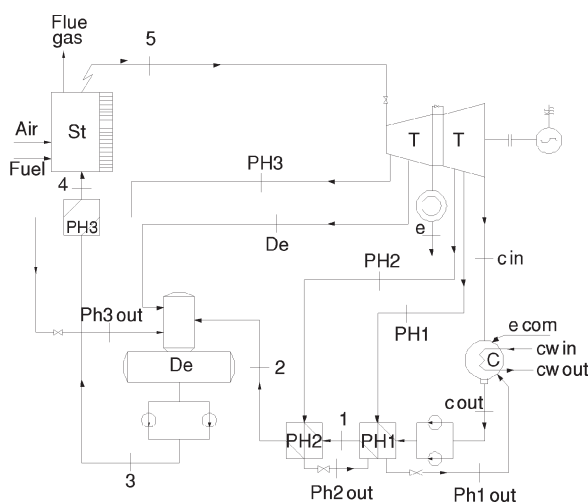

Figure 1. Thermodynamic scheme of a co-generation cycle

Mathematical model used in this chapter is reduced to form of energy and mass balance:

$$\sum_{j\ in(ij)} q_{mj} - \sum_{j\ out(ij)} q_{mj} = 0, \quad i \in I$$

$$\sum_{j\ in(ij)} q_{mj}h_j - \sum_{j\ out(ij)} q_{mj}h_j - W_i - Q_i = 0, \quad i \in I$$

$$Q_i = 0, \quad i \in \{T, C, PHi, De, Pm\}$$

All relevant stream's data can be seen in tab. 7, in appendix C. Data were calculated previously using this, mentioned model.

Calculation results

Steam generator

In steam generator, the feed water is brought to the saturation temperature, evaporated and superheated by fuel combustion, in our case, hard coal with specific lower heating value $\Delta h_L = 33.6$ MJ/kg. Flue gasses exit the system with 200 °C. Approximated calculation of combustion is given in appendix D.

Exergy balance equation in this case would be:

$$q_{m\,F}e_F \quad q_m e_4 \quad q_{m\,air}e_{air} \quad q_m e_5 \quad q_{m\,fg}e_{fg} \quad E_{loss} \tag{5}$$

Now, according to eq. (3), simple exergy efficiency can be written as:

$$\varepsilon_s \quad \frac{q_m e_5 \quad q_{m\,fg}e_{fg}}{q_{m\,F}e_F \quad q_m e_4 \quad q_{m\,air}e_{air}} \tag{6}$$

In steam generator, exergies of air and fuel are used to superheat the steam, so the rational exergy efficiency, according to eq. (4), is:

$$\varepsilon_r \quad \frac{q_m (e_5 \quad e_4)}{q_{m\,F}e_F \quad q_{m\,air}e_{air} \quad q_{m\,fg}e_{fg}} \tag{7}$$

Using the fact, that exergy of the fuel is few percentages higher than its lower heating value [10] ($\Delta h_L/e_F = 0.95$), and other known data from tab. 7, numerical values of these efficiencies are:

$$\varepsilon_s \quad 0.578 \quad 57.8\%$$

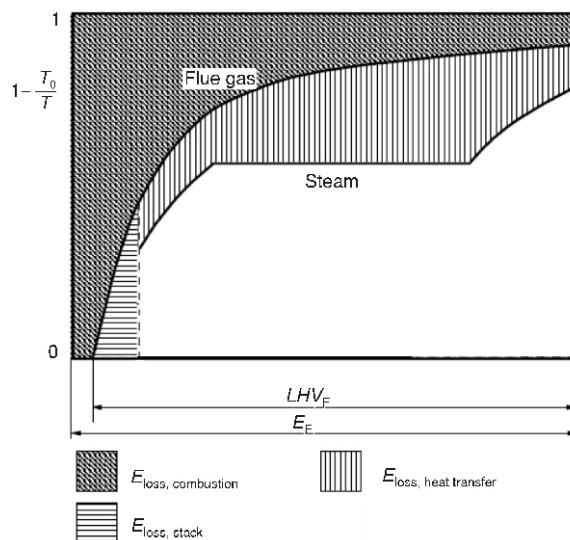$$\varepsilon_r \quad 0.487 \quad 48.7\%$$

Figure 2 shows exergy losses of steam generator in so called "value diagram" which is shown and explained in [7].

The process generates three kinds of losses. First loss can be seen as difference between exergy of the fuel and flue gasses marked as exergy loss due to combustion. Second loss exists due to heat transfer between flue gasses and steam. The last one is outgoing exergy of the flue gasses. All three can be clearly seen in fig. 2.

Turbine

For a turbine, simple exergy efficiency is:

Figure 2. Exergy losses in steam generator reduced to 1 kg of the fuel [7]

$$\varepsilon_s \quad \frac{\Sigma E_{out} \quad \Sigma \dot{W}}{\Sigma E_{in}} \quad \frac{E_{PH1} \quad E_{PH2} \quad E_{De} \quad E_e \quad E_{PH3} \quad E_{c\,in} \quad P}{E_5} \quad 0.896 \quad 89.6\% \quad (8)$$

The result will be the same if we calculate simple exergy efficiency by summing exergy losses for every turbine stage: $E_{loss} = E_{in} - E_{out} - P_i$. Previously calculated necessary data are:

$$\Sigma P_i \quad 5135 \text{ MW} \quad \Sigma \Delta E_i \quad 60.29 \text{ MW}$$

where P_i is the power obtained in each turbine stage and ΔE_i – the difference of ingoing and outgoing exergy and enthalpy of each stage.

In co-generation plants, desired effect is power and heat production and exergy used for the purpose is exergy decrease in each stage and exergy of the stream "e" itself. Considering this, rational exergy efficiency of a turbine is:

$$\varepsilon_r \quad \frac{\Sigma P_i \quad E_e}{\Sigma \Delta E_i \quad E_e} \quad 0.865 \quad 86.5\% \quad (9)$$

*Condenser**

As it can be seen in fig. 1, in condenser there are four ingoing streams: stream that exits the turbine (c in), cooling water inlet (cw in), steam from PH1 (PH1 out) and compensation for taking the steam for heat production (e com) and two outgoing streams: condensate (c out) and cooling water outlet (cw out). Table 1 shows calculated exergy of each stream.

It should be said that stream PH1 out was damped to the pressure of the condenser, so its exergy is decreased. Although the cooling water after condenser is thrown back to the environment, it must not be considered as loss, but as desired effect, because even though the steam had to be condensed, the process should be comprehended as the intention was to heat cooling water by condensing the steam. Then the efficiencies are:

Table 1. Exergy of each stream in condenser

Inlet				Outlet	
$E_{c\,in}$ [kW]	$E_{cw\,in}$ [kW]	$E_{e\,com}$ [kW]	$E_{PH1\,out}$ [kW]	$E_{c\,out}$ [kW]	$E_{cw\,out}$ [kW]
4881.1	1323.7	–0.663	47.38	161.64	5294.84

$$\varepsilon_s \quad \frac{E_{c\,out} \quad E_{cw\,out}}{E_{c\,in} \quad E_{cw\,in} \quad E_{PH1\,out} \quad E_{e\,com}} \quad 0.873 \quad 87.3\%$$

$$\varepsilon_r \quad \frac{\dot{E}_{cw\,out} \quad \dot{E}_{cw\,in}}{E_{c\,in} \quad E_{PH1\,out} \quad E_{e\,com} \quad E_{c\,out}} \quad 0.833 \quad 83.3\% \quad (10)$$

Preheater PH1

Ingoing flows in preheater PH1 are: PH1, PH2, and c out
Outgoing flows from preheater PH1 are: PH1 out and stream 1

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* In condenser and other heat echangers exergy losses due to pressure drop will be neglected!

Table 2. Exergy of each stream in preheater PH1

Inlet			Outlet	
E_{PH1} [kW]	$E_{c\,out}$ [kW]	$E_{PH2\,out}$ [kW]	E_1 [kW]	$E_{PH1\,out}$ [kW]
506.55	161.64	106.59	681.79	59.65

As earlier, exergy of the stream PH2 out was decreased due to damping to the pressure of the preheater PH1 (0.19 bar). Table 2 shows exergy of each stream in PH1, and beside tab. 3 are its numerical values of corresponding exergy efficiencies ε_s and ε_r, respectively.

$$\varepsilon_s = \frac{E_1 - E_{PH1\,out}}{E_{PH1} - E_{c\,out} - E_{PH2\,out}} = 0.957 = 95.7\%$$

$$\varepsilon_r = \frac{\dot{E}_1 - \dot{E}_{c\,out}}{E_{PH1} - E_{PH2\,out} - E_{PH1\,out}} = 0.940 = 94.0\%$$

(11)

Preheater PH2

Ingoing flows: PH2 and stream 1
Outgoing flows: PH2 out and stream 2

Table 3 shows exergy of each stream in PH1, and below tab. 4 are its numerical values of corresponding exergy efficiencies ε_s and ε_r, respectively.

Table 3. Exergy of each stream in preheater PH2

Inlet		Outlet	
E_{PH2} [kW]	E_1 [kW]	$E_{PH2\,out}$ [kW]	E_2 [kW]
1597.438	681.79	132.415	1750.27

$$\varepsilon_s = \frac{E_2 - E_{PH2\,out}}{E_{PH2} - E_1} = 0.826 = 82.6\%$$

$$\varepsilon_r = \frac{E_2 - E_1}{E_{PH2} - E_{PH2\,out}} = 0.729 = 72.9\%$$

(12)

Table 4. Exergy of each stream in deareator

Inlet			Outlet
E_{De} [kW]	E_2 [kW]	$E_{PH3\,out}$ [kW]	E_3 [kW]
2141.76	1750.27	2758.38	5624.51

Deareator

Ingoing flows are: stream De, stream 2, and outlet from preheater PH3 (PH3 out)
Outgoing flow is: stream 3

Decreased exergy of the stream PH3 out comes from damping to the pressure of the deareator.

$$\varepsilon_s = \frac{E_3}{E_{De} - E_2 - E_{PH3\,out}} = 0.846 = 84.6\%$$

$$\varepsilon_r = \frac{q_{m2}(e_3 - e_2)}{q_{mDe}(e_{De} - e_3) - q_{mPH3}(e_{PH3\,out} - e_3)} = 0.71 = 71.0\%$$

(13)

Preheater PH3

Ingoing flows: stream PH3 and stream 3
Outgoing flows: stream 4 and outlet from preheater PH3 ("PH3 out")

$$\varepsilon_s = \frac{E_4 - E_{PH3\,out}}{E_{PH3} - E_3} = 0.838 \approx 83.8\%$$

$$\varepsilon_r = \frac{E_4 - E_3}{E_{PH3} - E_{PH3\,out}} = 0.717 \approx 71.7\% \tag{14}$$

Table 5. Exergy of each stream in preheater PH3

Inlet		Outlet	
E_{PH3} [kW]	E_3 [kW]	$E_{PH3\,out}$ [kW]	E_4 [kW]
13917.9	5624.51	2758.38	13624.8

With analysis of preheater PH3, every major component of respective co-generation plant has been analyzed. Table 6 shows summarized calculated data for every component.

Table 6. Exergy losses of co-generation plant

Unit	ε_s [%]	ε_r [%]	Loss [MW]	Portion [%]	Portion of ingoing energy
Steam generator	57.8	48.7	79.57	84.793%	45.11%
Turbine	89.6	86.5	8.86	9.442%	5.02%
Condenser	87.3	83.3	0.79	0.842%	0.45%
Preheaters					
PH1	95.7	94	0.034	0.036%	0.02%
PH2	82.6	72.9	0.396	0.422%	0.22%
Deareator	84.6	71	1.03	1.098%	0.58%
PH3	83.3	71.7	3.16	3.367%	1.79%
Σ			93.84	100.00%	53.20%

In first two columns are numerical values of corresponding simple and rational exergy efficiencies. Next column shows amount of losses in particular apparatuses. While fourth column contains percentages of mentioned losses in respect to overall losses in the system, last column shows ratio of particular loss and ingoing exergy in observed facility. Diagram on fig. 3 graphically represents results from tab. 6, fourth column.

To calculate overall efficiencies of entire co-generation plant same approach as before will be used. Equation of exergy flow for the system that includes entire co-generation plant is:

$$E_F + E_{air} + E_{cw\,in} + E_{e\,com} = P + E_e + E_{fg} + E_{cw\,out} + E_{loss} \tag{15}$$

According to definition – eq. (3), follows a simple exergy efficiency of the plant:

$$\varepsilon_s = \frac{P + E_e + E_{fg} + E_{cw\,out}}{E_F + E_{air} + E_{cw\,in} + E_{e\,com}} = 0.468 \approx 46.8\% \tag{16}$$
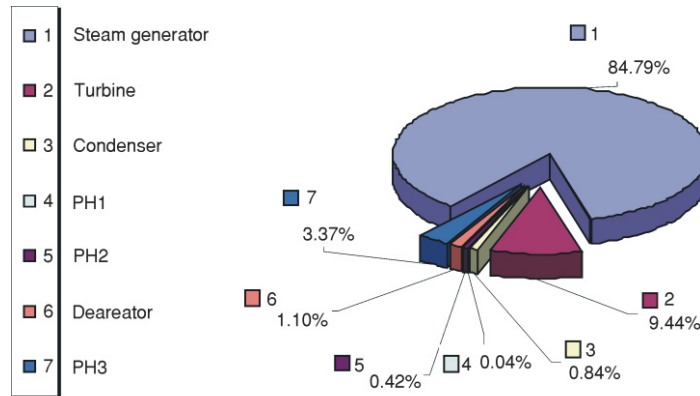
**Figure 3. Exergy losses in a co-generation plant
(color image see on our web site)**

In co-generation system, our intent, desired effect is heat and power production. To achieve that we use difference between ingoing exergy and exergy of all other outgoing streams except those, which are earlier defined as desired effect. Now, rational exergy efficiency is:

$$\varepsilon_r = \frac{P + E_e}{E_F + E_{air} + E_{cw\,in} + E_{e\,com} + E_{fg} + E_{cw\,out}} = 0.379 = 37.9\% \tag{17}$$

Diagram on fig. 4 shows portions of gained power on turbine shaft, co-generation heat, outgoing exergy and overall exergy losses in co-generation plant's ingoing exergy.
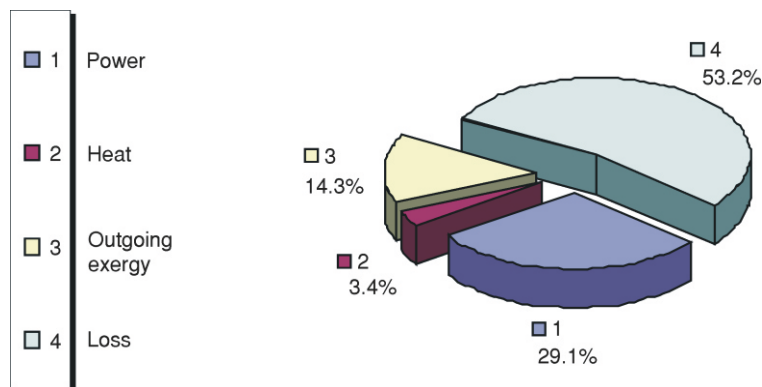

**Figure 4. Distribution of ingoing exergy
(color image see on our web site)**

Conclusions

Applied exergy analysis of this simple co-generation facility through both exergy efficiencies has shown that most exergy losses are generated in steam generator. Those losses are caused by irreversibility of combustion process and heat transfer at significant temperature dif-

ference between flue gasses and evaporating water. By applying co-generation process exergy efficiency of entire facility is increased in each segment as well as in steam generator. This efficiency increment is achieved by increasing the average temperature of heat abduction.

The co-generation process has a direct influence on fuel consumption what can be shown analytically for the respective plant.

Fuel consumption comparison of observed co-generation plant with a process, in which same quantity of heat and mechanical energy would be separately produced, using the same analysis, would show that co-generation plant saves up to 20,813 tons per year of hard coal with lower heating value of 33,600 kJ/kg. The given fact points out the cost effectiveness of such facility. Taken into consideration the harmfulness of combustion emissions and the fact that fossil fuels are running out, this data once more confirm that co-generation process represents an example of smart resource management and one of the intelligent solutions in the sense of sustainable development.

Nomenclature

(Nomenclature is defined according to 11)

E	– exergy, [W]		cw out	– cooling water outlet
e	– specific exergy, [Jkg^{-1}]		De	– deareator
$\dot{H}$	– entalpy flow, [W]		e	– stream e
h	– specific enthalpy, [JKg^{-1}]		en	– environment
Δh_L	– specific lower heating value, [kJkg^{-1}]		F	– fuel
P	– turbine power, [W]		fg	– flue gas
p	– pressure, [Pa]		in	– inlet
q_m	– mass flow rate, [kgs^{-1}]		loss	– loss
s	– specific entropy, [Jkg^{-1}K^{-1}]		max	– maximal
T	– temperature, [K]		out	– outlet
$\dot{W}$	– work flow, [W]		PH1	– preaheater PH1 outlet
			PH2	– preheater PH2
Greek letters			PH3	– preheater PH3
			PH1 out	– preheater PH1 outlet
ε_s	– simple exergy efficiency, [–]		PH2 out	– preheater PH2 outlet
ε_r	– rational exergy efficiency, [–]		PH3 out	– preheater PH3 outlet
			st	– steam generator
Subscripts			T	– turbine
			0	– environmental state
air	– air		1	– stream 1
c	– condenser		2	– stream 2
c in	– condenser inlet		3	– stream 3
c out	– condenser outlet		4	– stream 4
cw in	– cooling water inlet		5	– stream 5

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Appendix A

For better understanding rational exergy efficiency, following example will be intro-
duced. The system represents heat transfer in a heat exchanger with two streams where primary
flow is being heated with secondary flow.

It has been said that $\Sigma \dot{E}_{product}$ is an amount of exergy, which is considered as desired
effect of the process. In this case the intention was to heat a primary flow *i. e.* increase amount of
its exergy. So the term $\Sigma \dot{E}_{product}$ equals:

$$\Sigma \dot{E}_{product} \quad \dot{E}_{p\,out} \quad \dot{E}_{p\,in} \quad \Delta \dot{E}_{p} \tag{18}$$

Increasing primary's exergy has been done by decreasing secondary flow's exergy. So
the term $\Sigma \dot{E}_{source}$ equals:

$$\Sigma \dot{E}_{source} \quad \dot{E}_{s\,in} \quad \dot{E}_{s\,out} \quad \Delta \dot{E}_{s} \tag{19}$$

Simple exergy efficiency offers a clear inside information for variety of systems. This
comes from the fact that simple exergy efficiency says how much exergy, out of ingoing, re-
mained within outgoing flows, which can be, and often are, used further in a process. The main
disadvantage of simple efficiency is that it could be insensitive to changes in the system. That is
the reason for bringing rational exergy efficiency up. This efficiency must be considered as ad-
ditional information about the process because comprehending a process is crucial for its
definition.

Following example will illustrate above statement. Now, two heat exchangers each
with two streams will be taken with exception that each stream an explicit value of exergy will
be given (fig. 5).

Which of these two processes is better? In order to answer this question the simple
exergy efficiencies will be calculated.

$$(a) \ \varepsilon_s \quad \frac{\Sigma \dot{E}_{out}}{\Sigma \dot{E}_{in}} \quad \frac{20 \quad 10}{30 \quad 5} \quad \frac{30}{35}$$

$$(b) \ \varepsilon_s \quad \frac{\Sigma \dot{E}_{out}}{\Sigma \dot{E}_{in}} \quad \frac{18 \quad 12}{32 \quad 3} \quad \frac{30}{35}$$

It is obvious that simple exergy efficiency is not a tool, which can answer this question. Another, additional information is needed to compare these two processes and that is provided by rational exergy efficiency.

Figure 5. Two heat exchangers with equal simple efficiencies

(a) $\varepsilon_r = \dfrac{\Sigma \dot{E}_{product}}{\Sigma \dot{E}_{source}} = \dfrac{\Delta \dot{E}_p}{\Delta \dot{E}_s} = \dfrac{10-5}{30-20} = 0.5$

(b) $\varepsilon_r = \dfrac{\Sigma \dot{E}_{product}}{\Sigma \dot{E}_{source}} = \dfrac{\Delta \dot{E}_p}{\Delta \dot{E}_s} = \dfrac{12-3}{32-18} = 0.643$

It is now clear that the second process is more efficient than the first because it converted bigger part of secondary stream's exergy into exergy increase of primary stream.

It is important to emphasize that these two processes could have had equal rational exergy efficiencies, as well, but then the process with higher simple efficiency would be better because a bigger part of its ingoing exergy can be further used.

Appendix B

As a motivation for defining exergy efficiency, following example will be shown. First, it is useful to state how specific exergy is shown in *h-s* diagram (fig. 6).

Specific exergy, in *h-s* diagram, equals difference between enthalpy of the given stream and enthalpy of the stream in point, which lies on intersection of the stream's isentropic line and the line of environment [1].

Figure 6 shows isentropic (1-2$_{is}$) and real (1-2) expansion in one turbine stage. First law efficiency is defined as ratio between gained specific technical work and isentropic work that can be gained:

$$\eta_t = \frac{w_t}{w_{t\,is}} = \frac{h_1 - h_2}{h_1 - h_{2\,is}} \qquad (20)$$

On the other hand, simple exergy efficiency is defined, for this example as:

$$\varepsilon_s = \frac{e_2 + w_T}{e_1} \qquad (21)$$

This efficiency, as well as the first law efficiency, equals one when point 2 becomes 2$_{is}$ but if point 2$_{is}$ lies above the line of environment, simple exergy efficiency will always be higher than the first law efficiency and lower when 2$_{is}$ is under the line of environment. So what is the main difference between these two definitions? In the essence, first law efficiency is comparing two processes: isentropic process (1-2$_{is}$) and a real one (1-2). These two expansions are differing from each other only in their ending point. Outgoing flow of real expansion has higher exergy (end enthalpy)
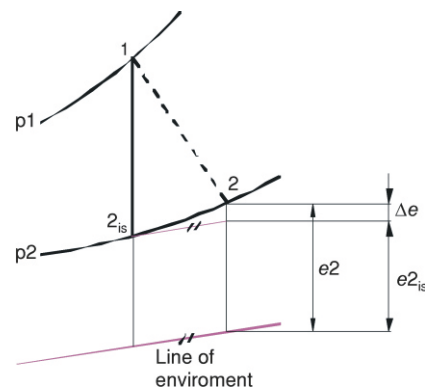

Figure 6. Turbine expansion in *h-s* diagram

than outgoing flow of an isentropic expansion, so it is still possible to gain more work from flow at point 2 than at point 2_{is} (Δe). For comparison to be complete, condition of an outgoing flow must be taken under consideration. These main flaws of first law efficiency, exergy efficiency avoids by taking final state in process under consideration.

Appendix C

Following table shows calculated data of streams in co-generation plant according to fig. 1. The last row in the table represents referenced state *i. e.* state of the environment. Data were calculated using previously mentioned model [12].

Table 7. Stream's data and calculated exergy

Stream	q_m [kg/s]	T [°C]	p [bar]	e [kJ/kg]	E [kW]
c in	28.545	39	0.07	170.995	4881.05
c out	43.615	39	0.07	3.706	161.64
1	43.615	60	4.248	15.632	681.79
2	43.615	94	4.248	40.130	1750.27
3	57.838	146	4.248	97.246	5624.51
4	57.838	225	95	235.569	13624.84
5	57.838	545	95	1543.154	89252.94
e	10.278	106.7	1.3	583.007	5992.15
PH1	1.702	58.6	0.19	297.618	506.55
PH2	3.091	99	0.978	516.803	1597.44
De	2.744	182.7	4.248	780.526	2141.76
PH3	11.479	229.4	27.763	1212.471	13917.96
PH1 out	4.793	58.6	0.19	12.445	59.65
PH2 out	3.091	99	0.978	42.839	132.42
PH3 out	11.479	229.4	27.763	240.298	2758.38
Flue gas	98.726	200	1	201.335	19877.00
Fuel	4.95	15	1	35368.421	175073.68
air	0.0972	15	1	0.000	0.00
e com	10.278	15	0.07	–0.065	–0.66
cw in	1640.286	23	1	0.807	1323.71
cw out	1640.286	32	1	3.228	5294.84
0		15	1	0.000	0.00

Appendix D

A short calculation of flue gas exergy will be provided based upon certain assumptions. Fuel used in calculation is a hard coal with standard composition [13]. Combustion is complete with 70% of air excess and outgoing flue gas temperature is 200 °C.

(1) Chemical composition

$$c = 0.85, \, h = 0.05, \, s = 0.01, \, o = 0.08, \, n = 0.01$$

(2) Stoichiometric air to fuel ratio

$$\lambda = 1.7$$

(3) Flue gas temperature

$$\theta_{fg} = 200 \, °C$$

(4) Lower heating value [14]

$$\Delta h_L \quad 33900c \quad 117000 \left(h \quad \frac{o}{8} \right) \quad 10500s \quad 2500w \quad 33600 \, kJ/kg$$

(5) Stoichiometric air quantity

$$O_{min} \quad \frac{c}{12} \quad \frac{h}{4} \quad \frac{s}{32} \quad \frac{o}{32} \quad 0.0811 \, kmol/kg$$

(6) Flue gas composition

$$n_{CO_2} \quad \frac{c}{12} \quad 0.0708 \, kmol/kg$$

$$n_{H_2O} \quad \frac{h}{2} \quad 0.025 \, kmol/kg$$

$$n_{SO_2} \quad \frac{s}{32} \quad 0.0003 \, kmol/kg$$

$$n_{N_2} \quad \frac{n}{28} \quad 0.79\lambda \frac{O_{min}}{0.21} \quad 0.5193 \, kmol/kg$$

$$n_{O_2} \quad (\lambda \quad 1) \, O_{min} \quad 0.0568 \, kmol/kg$$

$$n_{fg} \quad n_{CO_2} \quad n_{H_2O} \quad n_{SO_2} \quad n_{N_2} \quad n_{O_2} \quad 0.6722 \, kmol/kg$$

(7) Flue gas mole mass and mole heat capacity

Gas	n_i	M_i	$[C_{mpi}]_0^{200}$
CO_2	0.0708	44	40.059
O_2	0.0568	32	29.931
N_2	0.5193	28	29.228
SO_2	0.0003	64	42.239
H_2O	0.025	18	34.118
Σ	0.6722		

$$[C_{mp \, fg}]_0^{200}$$

$$\frac{n_{CO_2}[C_{mpCO_2}]_0^{200} \quad n_{H_2O}[C_{mpH_2O}]_0^{200} \quad n_{SO_2}[C_{mpSO_2}]_0^{200} \quad n_{N_2}[C_{mpN_2}]_0^{200} \quad n_{O_2}[C_{mpO_2}]_0^{200}}{n_{fg}}$$

$$[C_{\mathrm{mp\ fg}}]_0^{200} = 30.617 \ \mathrm{kJ/kmolK}$$

$$M_{\mathrm{fg}} = \frac{n_{\mathrm{CO_2}} M_{\mathrm{CO_2}} + n_{\mathrm{HO_2}} M_{\mathrm{HO_2}} + n_{\mathrm{SO_2}} M_{\mathrm{SO_2}} + n_{\mathrm{N_2}} M_{\mathrm{N_2}} + n_{\mathrm{O_2}} M_{\mathrm{O_2}}}{n_{\mathrm{fg}}} = 29.67 \ \mathrm{kg/kmol}$$

(8) Fuel mass flow rate

$$q_{\mathrm{mF}} = \frac{q_{\mathrm{m4}}(h_5 - h_4)}{\Delta h_{\mathrm{L}} - n_{\mathrm{fg}}[C_{\mathrm{mp\ fg}}]_0^{200}\theta_{\mathrm{fg}}} = 4.95 \ \mathrm{kg/s}$$

(9) Flue gas mass flow rate

$$q_{\mathrm{m\ fg}} = q_{\mathrm{mF}} n_{\mathrm{fg}} M_{\mathrm{fg}} = 98.726 \ \mathrm{kg/s}$$

(10) Flue gas exergy rate

$$E_{\mathrm{fg}} = q_{\mathrm{mF}} n_{\mathrm{fg}} \left[[C_{\mathrm{mp\ fg}}]_{15}^{200}(\theta_{\mathrm{fg}} - \theta_0) - T_{\mathrm{en}} \ln\frac{T_{\mathrm{fg}}}{T_{\mathrm{en}}} \right] = 19.877 \ \mathrm{MW}$$

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