New Journal and we have not received input yet 23 (2021) 100853

Contents lists available at ScienceDirect

Thermal Science and Engineering Progress

journal homepage: www.sciencedirect.com/journal/thermal-science-and-engineering-progress

Assessment of improvement potential of a condensed combi boiler via advanced exergy analysis

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ARTICLE INFO

Keywords:
Advanced exergy analysis
Combustion
Condensed combi boiler
Natural gas

ABSTRACT

The combi boilers are commonly used for the heating purposes in residences since the natural gas has started to be used prevalently as fuel. Depending on this common usage, the investigations on the development of these kinds of boilers accelerated in the last decades. In this aim, it is very important to determine the improvement limits from the thermodynamic point of view. In this study, a 24 kW commercial condensed combi boiler was experimentally performed under the real operating conditions to determine the improvement paths from the thermodynamics point of view. In this aim, the advanced exergy analysis was conducted to determine the improvement potential of the combi boiler by defining ideal and unavoidable conditions including the combustion process. The most improvable components were determined as the combustion unit and recuperator. In the combustion unit, it was determined that the gas radiation has a considerable rate with 40.16% in the heat transfer phenomena for the real operating conditions. As conclusion, an improvement potential of approximately 14.32% and 7.09% was determined for the combustion unit and recuperator, respectively. The overall improvement potential of the combi boiler system was determined as 22.17%.

1. Introduction

The consumption of natural gas used for residential heating purpose reached to 15 billion m^3 (32.29% of the total consumption) in 2019 [1]. In this regard, the investigations on the efficiency improvement of the natural gas-fired combi boilers come into prominence from both economic and environmental points of view. The studies on the efficiency improvements commonly focuse on the external improvements such as coupling with the thermo-photovoltaic (TPV) generators [2] the thermoelectric generator [3] and the micro power generator organic Rankine cycle (ORC) [4] or such as integration with solar collector systems [5] and integration with photovoltaic thermal (PVT) systems [6].

However, the studies on the internal improvements are limited, and frequently handled by product developers under commercial concerns. The current studies frequently focus on the waste heat recovery of the flue gases where a few focus on the combustion parameter such as excess air. Atmaca et al. [7] investigated two different models to analyze the performance of combi boilers with a conical heat cell unit. They aimed to decrease the number of experiments by their numerical model. Balanescu and Homutescu [8] experimentally investigated the efficiency of 25 kW condensed natural gas-fired boiler under real operating conditions. They reported efficiency of 77.43% for the reference boiler where it was reported as 93.93% for condensed type under the full loading case. They also reported a fuel saving of 17.56%. Men et al. [9] reported an efficiency increase of up to 106% for the boiler and 88% for the enthalpy-wheel recovery system. Lee et al. [10] in their study, reported an efficiency ranging between 82% and 84% with an exhaust gas temperature of 120–160 °C for traditional boilers. This value was reported as 86–88% with an exhaust gas temperature of approximately 60 °C. Wang et al. [11] investigated the boilers with heat recovery and with the water recovery. They concluded an efficiency increase by 3.0–23.8% for the case with the heat recovery and 5.1–41.4% % for the case with the water recovery when compared with the traditional ones. Lazzarin [12] reported that it was available to increase efficiency at a rate of 12% with a recovery of 65% of the latent heat of vapor. So, the efficiency of the non condensed boiler would rise to 104% from the value of 92%. Wang et al. [13] in their study in which the dehumidification and heat recovery was modeled, conducted that it was available to increase efficiency by about 3%. They also conducted that the system efficiency would decrease by about 2% when the temperature of heating circuit water increases by 1 °C. Yamashita and Utaka [14] proposed a new type of heat exchanger for the latent heat recovery and conducted that it was

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https://doi.org/10.1016/j.tsep.2021.100853
Received 20 October 2020; Received in revised form 15 January 2021; Accepted 16 January 2021

Nomenclature

Cr	condensed water ratio (%)
$\dot{E}$	energy rate (kW)
$\dot{E}x$	exergy rate (kW)
h	specific enthalpy (kJ/kg)
$\bar{h}$	specific molar enthalpy (kJ/kmole)
I	exergy destruction rate (kW)
IP	improvement potential (%)
$\dot{m}$	mass flow rate (kg/s)
$\dot{n}$	molar rate (kmole/s)
P	pressure (kPa, bar, or atm)
$\dot{Q}$	heat rate (kW)
s	specific entropy (kW/kg K)
T	temperature (K or °C)
$\dot{W}$	work rate (kW)

Greek symbols

ε	effectiveness (%)
η	energy efficiency (%)
η_{II}	exergy efficiency (%)
λ	coefficient of excess air
ψ	specific flow exergy (kJ/kg)

Subscripts

cc	combustion chamber
d	destruction
f	fuel
g	generated
i	inlet or ith component
k	kth component
o	outlet
p	product
r	reactant
s	isentropic
T	value at a specified temperature
0	value at the reference state

Superscripts

f	formation
fg	flue gas
o	absolute
w	water
AV	avoidable
EX	exogenous
EN	endogenous
Q	exergy term related to heat
UN	unavoidable
W	exergy term related to work

Abbreviations

CC	Combustion chamber
CU	Combustion unit
F	Fan hood
G	Generator
HE	Heat exchanger
P	Pump
R	Recuperator
OS	Overall system

available to gain the waste heat up to 93%. Lee and Kim [15] investigated heat recovery by water spray process and reported an efficiency increase by 6% in comparison to the conventional heat recovery process. Lee et al. [16] in their study, investigated various types of boilers including non-condensed, condensed, condensed with waste heat recovery, and condensed with exhaust gas recirculation and circulation of condensed water. They reported efficiency of 80.94% for non-condensed cases and an efficiency of 88.96% for the case of waste heat recovery. The efficiency value was reported as 93.91% for the case of condensed with exhaust gas recirculation and circulation of condensed water. They also reported that it was possible to decrease the excess air ratio down to 1.1 where it was about 1.4 for the non-condensed case. Liu et al. [17] studied two types of burners for a 24 kW condensed combi boiler. They conducted a reference excess air ratio of 1.35 for the complete combustion process, which is a very critical issue on efficiency. Satyavada and Baldi [18] indicated the efficiency is higher when the return temperature was below the dew point. They reported efficiency of approximately 91% under the combustion with 1.15 excess air.

Although the studies in the literature show that there is a huge opportunity to increase the efficiency of combi boilers by developing new designs, the wall-hanged combi boilers have limits on both sizes and operating conditions to develop new designs of heat recovery systems and/or combustion units. Therefore, the limits of improvement potential should be drawn carefully in the thermodynamic applicability. In this regard, this study, as a pathway for the product developers, aims to determine these limits from the thermodynamics point of view. In this aim, the advanced exergy analysis was conducted for a 24 kW commercial condensed combi boiler operating under real conditions of the space heating purpose. Thus, exogenous, endogenous, avoidable, and unavoidable parts of the exergy destruction for all components were investigated to determine the improvement potential of the each components and overall system.

2. Material and methods

In this study, a 24 kW wall-hanged combi boiler was handled for the space heating case. The flow diagram is given in Fig. 1.

The boiler is essentially composed of the burner (B), combustion chamber (CC), heat exchangers, fan hood (F), and pump (P). It has two heat exchangers namely the main heat exchanger (HE) and recuperator (R) in which the latent heat is recovered. It has also a third exchanger for domestic hot water treatment. The B, CC, and HE are in the integrated form in a case named combustion unit (CU) which is operating as a traditional (non-condensed) boiler. Natural gas is used in the boiler. The coefficients of the combustion reaction of natural gas were determined using experimental data for the real conditions. It was determined considering the technical possibility for unavoidable conditions where the ideal condition was an imaginary possible case. The data of combustion reaction, according to Eq. (1), is given in Table 1.

$$\overbrace{x(aCH_3+bC_2H_6+cC_3H_8)}^{Fuel}+\lambda y\overbrace{(3.76N_2+O_2)}^{Air}\rightarrow\overbrace{dCO+eCO_2+fNO+gNO_2+kO_2+mN_2+nH_2O_{(gaseous)}}^{Products} \quad (1)$$

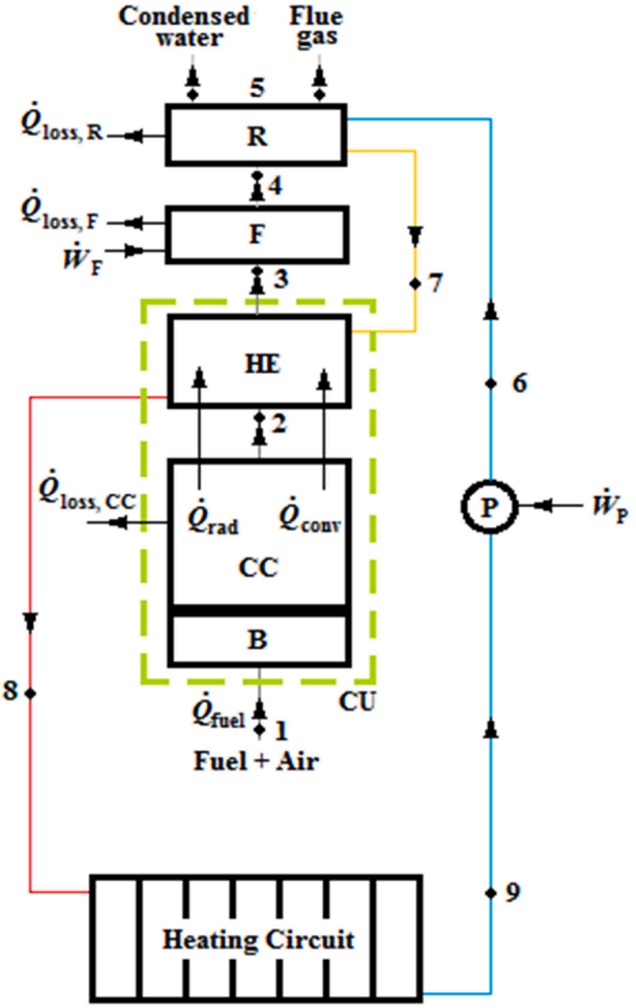

Fig. 1. Flow diagram of the wall-hanged condensed combi boiler.

Table 1
Coefficients of combustion reaction [19].

	(kmole/s)	Real Case	Ideal Case	Unavoidable Case
Fuel	x	$2.72\cdot10^{-5}$	$2.03\cdot10^{-5}$	$2.20\cdot10^{-5}$
	a	$9.95\cdot10^{-1}$		
	b	$4.10\cdot10^{-3}$		
	c	$1.40\cdot10^{-3}$		
Air	y	$5.47\cdot10^{-5}$	$4.08\cdot10^{-5}$	$4.42\cdot10^{-5}$
	λ	1.36	1.0	1.1
Product	d	$8.16\cdot10^{-9}$	–	–
	e	$2.74\cdot10^{-5}$	$2.05\cdot10^{-5}$	$2.21\cdot10^{-3}$
	f	$2.56\cdot10^{-7}$	–	–
	g	$4.35\cdot10^{-7}$	–	–
	k	$1.91\cdot10^{-5}$	–	$4.42\cdot10^{-6}$
	m	$2.79\cdot10^{-4}$	$1.54\cdot10^{-4}$	$1.83\cdot10^{-4}$
	n	$5.46\cdot10^{-5}$	$4.08\cdot10^{-5}$	$4.41\cdot10^{-5}$

Using the measured values, the thermodynamic properties were determined and energy and exergy values of points were calculated. For ideal and unavoidable cases, some main points and assumptions were made to determine the state points. The parameters used in the ideal and unavoidable cases are given in Table 2 [19–21].

Table 2
The parameters and assumptions for real, ideal, and unavoidable cases [19–21].

Component	Real case	Ideal case	Unavoidable case
CC	–	–	–
HE	$\mathcal{E} = 80.89$	$\mathcal{E} = 1$	$\mathcal{E} = 0.95$
F	$\eta = 0.65$	$\eta = 1$	$\eta = 0.70$
R	$\Delta P_w = 3\%$	$\Delta P_w = 0\%$	$\Delta P_w = 6\%$
	$\Delta P_{fg} = 3\%$	$\Delta P_{fg} = 0\%$	$\Delta P_{fg} = 2\%$
	$\Delta T = 32.6\,^{\circ}\mathrm{C}$	$\Delta T = 0\,^{\circ}\mathrm{C}$	$\Delta T = 15\,^{\circ}\mathrm{C}$
	$Cr = 44\%$	$Cr = 100\%$	$Cr = 80\%$
P	$\eta = 0.85$	$\eta = 1$	$\eta = 0.90$

Table 3
Thermodynamic properties of state points.

State	Fluid	Real Case		Ideal Case		Unavoidable Case	
		T (°C)	P (bar)	T (°C)	P (bar)	T (°C)	P (bar)
0	–	25.0	1.000	25.0	1.000	25.0	1.000
1	Natural gas	25.0	1.021	25.0	1.021	25.0	1.021
2	Flue gas	876.0	0.990	812.0	1.021	828.0	0.980
3	Flue gas	125.0	0.961	55.1	1.021	92.0	0.941
4	Flue gas	120.0	1.787	55.1	1.021	89.0	1.300
5	Water	62.0	1.735	29.4	1.021	44.4	1.248
	Flue gas	62.0	1.735	29.4	1.021	44.4	1.248
6	Water	29.4	2.739	29.4	2.500	29.4	2.656
7	Water	31.0	2.657	31.6	2.500	31.1	2.603
8	Water	51.0	2.577	51.0	2.500	51.0	2.551
9	Water	29.4	2.500	29.4	2.500	29.4	2.500

The pressure drops for the unavoidable case was accepted higher than those of real case since the heat exchangers area would increase. The dew point for the real case is 52 °C where it is 55.1 °C for the ideal case and 58 °C for the unavoidable case. The technical parameters of the cycle are given in Table 3.

2.1. *Conventional energy and exergy analyses*

Assuming of steady-state working condition, neglecting kinetic, and potential energy terms, the mass and energy balances for the components of the system are given as:

$$\sum \dot{m}_i - \sum \dot{m}_o = 0 \tag{2}$$

$$\dot{Q} - \dot{W} + \sum \dot{m}_i h_i - \sum \dot{m}_o h_o = 0 \tag{3}$$

where $\dot{Q}$ is heat rate term, $\dot{W}$is work term, $\dot{m}$ is the mass rate. i indicates the inlet conditions and o indicates the outlet conditions. For the combustion process, energy balance of boiler is given as:

$$\dot{Q}_f = \sum_i \dot{n}\left(\bar{h}_0^f - \bar{h}_T - \bar{h}_{298}\right) - \sum_o \dot{n}\left(\bar{h}_o^f - \bar{h}_T - \bar{h}_{298}\right) \tag{4}$$

Here, $\dot{Q}_f$ describes the heat of fuel. $\dot{n}$, $\bar{h}_0^f$, $\bar{h}_T$ and $\bar{h}_{298}$ are orderly the mole ratio, enthalpy of formation, enthalpy at the temperature T and enthalpy at 298 K. The exergy balances for the k^{th} component of the system, exergy efficiency, and exergy destruction ratios are respectively given as:

$$\dot{Ex}_k^Q - \dot{Ex}_k^W - \sum\left(\dot{m}_i\psi_i\right)_k - \sum\left(\dot{m}_o\psi_o\right)_k - \dot{Ex}_{d,k} = 0 \tag{5}$$

$$\eta_{II} = 1 - \frac{\dot{Ex}_{d,k}}{\dot{Ex}_{i,k}} \tag{6}$$

$$y_k = \frac{\dot{Ex}_{d,k}}{\dot{Ex}_{i,k}} \tag{7}$$

where $\dot{Ex}_k^Q$, $\dot{Ex}_k^W$ and ψ respectively describe the exergy of heat, the exergy of the work and the specific exergy of flow, and are given as:

$$\dot{E}x_k^Q = \left(1 - \frac{T_0}{T}\right) Q_k \tag{8}$$

$$\dot{E}x_k^W = \dot{W}_k \tag{9}$$

$$\psi = (h - h_0) - T_0(s - s_0) \tag{10}$$

Here, h and s indicate orderly the enthalpy and entropy at a certain state. Subscript of 0 indicates the reference state conditions. For the combustion process, exergy balance of boiler can be given as [22,23]:

$$\dot{E}x_f = \left(1 - \frac{T_0}{T_{cc}}\right) \dot{Q}_f + \dot{E}x^{Ch} - I \tag{11}$$

where subscript cc indicates the combustion chamber, $\dot{E}x^{Ch}$ and I respectively describe the chemical exergy of the fuel and exergy destruction rate, and are given as [22,24]:

$$\dot{E}x^{Ch} = \sum_i (x_i \bullet ex_i)_{product} - \sum_k (x_i \bullet ex_i)_{reactant} \tag{12}$$

$$I = \dot{E}x_{d,f} = T_0 S_g \tag{13}$$

where x_i and ex_i are the mole fraction of molar chemical exergy for the substance i respectively. The chemical exergy values of the substances can be achieved from the Ref. [22]. S_g is the entropy generation of the combustion process, and is given as [19]:

$$S_g = S_p - S_r + \frac{\dot{Q}_{loss,CC}}{T_{cc}} \tag{14}$$

Here $\dot{Q}_{loss,CC}$, S_p and S_r are heat losses from the combustion chamber, the entropy of products and the entropy of reactants respectively. S_p or S_r is given by the following equation [24]:

$$S_{p(\text{or}r)} = \sum \dot{n}_i \left(\overline{s}_i^{o}(T, P_0) - R_u ln(x_i P) \right) \tag{15}$$

Here, $\overline{s}_i^{o}$, R_u, x_i and P are respectively standard molar enthalpy of the *substance* i, universal gas constant, mole fraction of the *substance* i, and total pressure of the flow.

2.2. Advanced exergy analysis

The essential idea of advanced exergy analysis is based on the endogenous and exogenous exergy destruction terms of the k^{th} component. Endogenous exergy destruction term ($\dot{E}x_{d,k}^{EN}$) identifies the exergy destruction occurred within component depending on its irreversibilities. Exogenous exergy destruction term ($\dot{E}x_{d,k}^{EX}$) identifies the exergy destruction occurred within components depending on the irreversibilities of the other component working conditions. The following equation is used to give the relation between exergy destruction and these terms for the k^{th} component [25]:

$$\dot{E}x_{d,k} = \dot{E}x_{d,k}^{EN} + \dot{E}x_{d,k}^{EX} \tag{16}$$

Since the exergy destruction is directly related to working conditions, some of this destruction can be avoided by changing the working conditions. However, it is limited by some parameters such as the technological developments and environmental conditions. In this regard, the exergy destruction terms can be identified in terms of avoidable exergy destruction ($\dot{E}x_{d,k}^{AV}$) and unavoidable exergy destruction ($\dot{E}x_{d,k}^{UN}$) as following [26]:

$$\dot{E}x_{d,k} = \dot{E}x_{d,k}^{AV} + \dot{E}x_{d,k}^{UN} \tag{17}$$

Splitting the endogenous and exogenous parts into avoidable and unavoidable ones, Eq. (18) can be re-arranged as following [27]:

$$\dot{E}x_{d,k}^{EN} = \dot{E}x_{d,k}^{EN,AV} + \dot{E}x_{d,k}^{EN,UN} \tag{18}$$

$$\dot{E}x_{d,k}^{EX} = \dot{E}x_{d,k}^{EX,AV} + \dot{E}x_{d,k}^{EX,UN} \tag{19}$$

$$\dot{E}x_{d,k}^{AV} = \dot{E}x_{d,k}^{EN,AV} + \dot{E}x_{d,k}^{EX,AV} \tag{20}$$

$$\dot{E}x_{d,k}^{UN} = \dot{E}x_{d,k}^{EN,UN} + \dot{E}x_{d,k}^{EX,UN} \tag{21}$$

The analysis based on the ideal working conditions in which there are no irreversible processes. Therefore, if one changes one component with the real one, then the exergy destructions occurred in the overall system is endogenous exergy destruction of the component since the others work under reversible conditions. If the environmental and technological limitations are taken into account, then a component will work a situation between the real and ideal cases. This case is called the unavoidable case. If a component of the ideal system is changed with the unavoidable one, then the exergy destruction occurred in the overall system will be endogenous for the unavoidable component. Finally, depending on the splintered exergy destruction, the improvement potential can be calculated by the following equation:

$$IP_k = \frac{\dot{E}x_{d,k}^{AV}}{\dot{E}x_{i,k}} \tag{22}$$

A new parameter is useful to determine the exergetic performance of the improved real system. So, this modified exergy efficiency can be identified as follows:

$$\eta_{II}^{*} = 1 - \dot{E}x_{d,k} - \frac{\dot{E}x_{d,k}^{AV}}{\dot{E}x_{i,k}} \tag{23}$$

3. Results and discussion

The analyses were conducted in two steps. In the first step, the conventional energy and exergy analyses were conducted to determine the status of the system. In the second step, advanced energy and exergy analyses were conducted to determine the improvability of the system considering the technological limits.

3.1. Results of conventional energy and exergy analyses

The energy and exergy analyses were performed by using the values given in Tables 1 and 2. The obtained results are given in Tables 4–6 for real, ideal, and unavoidable conditions, respectively.

According to Table 4, for the real conditions, the highest exergy destruction occurs in CU with 37.9% of total destruction. The exergy destruction rate of R is 16.0% where it is about 1.0 for F. The exergy efficiency of the overall system was recorded as 44.8%. The energy efficiency was calculated as 75.9%. If the boiler had been operated as the traditional one, then the energy efficiency would be recorded as 80.6%.

Table 4
Conventional exergy analysis results for real conditions.

Component	$\dot{E}_i$ (kW)	$\dot{E}_o$ (kW)	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	η(%)
CU	20.736	16.703	−4.033	–	80.6
F	9.734	9.676	−0.106	0.048	98.9
R	2.987	1.433	−1.554	–	48.0
P	0.252	0.277	–	0.075	84.5
OS	***23.781***	***18.136***	***−5.692***	***0.123***	***75.9***
Component	$\dot{E}x_i$ (kW)	$\dot{E}x_o$ (kW)	$\dot{E}x_d$ (kW)	ε (%)	y (%)
CU	14.628	3.398	6.696	54.2	37.9
F	2.494	2.339	0.180	92.8	1.0
R	3.020	0.205	2.815	6.8	16.0
P	0.327	0.277	0.051	84.5	0.3
OS	***17.648***	***0.444***	***9.743***	***44.8***	***55.2***

Table 5
Conventional exergy analysis results for ideal conditions.

Component	$\dot{E}_i$ (kW)	$\dot{E}_o$ (kW)	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	η(%)
CU	16.189	16.181	−0.008	–	100.0
F	5.516	5.516	–	–	100.0
R	1.956	1.955	−0.001	–	100.0
P	0.253	0.253	–	–	100.0
OS	***18.145***	***18.136***	***−0.009***	–	***100.0***
Component	$\dot{E}x_i$ (kW)	$\dot{E}x_o$ (kW)	$\dot{E}x_d$ (kW)	ε (%)	y (%)
CU	11.847	0.404	3.203	73.0	24.0
F	0.506	0.506	–	99.9	0.0
R	1.491	0.046	1.445	3.1	10.8
P	0.253	0.253	–	100.0	0.0
OS	***13.336***	***0.444***	***4.649***	***65.1***	***34.9***

Table 6
Conventional exergy analysis results for unavoidable conditions.

Component	$\dot{E}_i$ (kW)	$\dot{E}_o$ (kW)	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	η(%)
CC	17.194	16.620	−0.574	–	96.7
F	6.346	6.322	−0.036	0.013	99.4
R	1.848	1.517	−0.331	–	82.1
P	0.252	0.268	–	0.046	90.0
OS	***19.440***	***18.136***	***−0.941***	***0.059***	***93.0***
Component	$\dot{E}x_i$ (kW)	$\dot{E}x_o$ (kW)	$\dot{E}x_d$ (kW)	ε (%)	y (%)
CC	12.312	0.828	4.169	66.1	29.5
F	1.178	1.118	0.066	94.4	0.5
R	1.620	0.055	1.565	3.4	11.1
P	0.298	0.268	0.030	90.0	0.2
OS	***14.149***	***0.444***	***5.830***	***58.8***	***41.2***

According to the commercial view in which the latent energy of condensed vapor is not taken into account, the energy efficiency with the consideration of condensed water was calculated as 84.0%.

According to Tables 5 and 6, the highest exergy destruction occurs in CU with 24.0% and 29.5% for the ideal and unavoidable conditions, respectively. The exergy destruction rate of R is 10.8% and 11.1% for the ideal and unavoidable conditions, respectively. The exergy efficiency of the overall system was recorded as 65.1% for the ideal case where it was recorded as 58.8% for the unavoidable case. The energy efficiencies for the ideal and unavoidable cases were calculated as 100.0% and 93.0%, respectively. The efficiencies of the non-condensed case were calculated as 100.0% and 96.7% where the commercial efficiencies were obtained as 110.9% and 102.9% for the ideal and unavoidable conditions, respectively. As a result of the analysis, it was also determined that the thermal radiation has a higher effect on the heat transfer mechanism of HE. The obtained results are given in Fig. 2 for the real conditions.

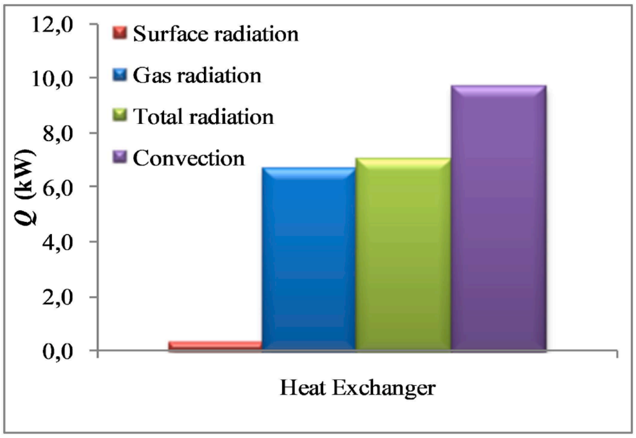

Fig. 2. The distribution of the heat transfer mechanisms in HE for the real (measured) condition.

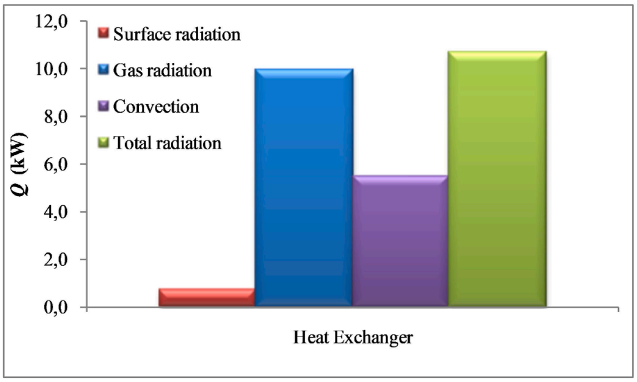

Fig. 3. The distribution of the heat transfer mechanisms in HE for the ideal condition.

According to Fig. 2, 57.9% of heat transferred to HE is occurred by convection where the remained part of 42.1% is occurred by radiation. The gas radiation is the effective mechanism in thermal radiation with a rate of 40.2%. The other part of 1.9% is surface radiation. In this case, the heat transfer area was measured as 0.1626 m^2. In this regard, gas radiation is the most important issue for the design of the main heat exchanger, and should be carefully taken into consideration in design problems. The related results for the ideal and unavoidable cases are given in Figs. 3 and 4, respectively.

According to Fig. 3, 34.1% of heat transferred to HE is occurred by convection where the remained part of 65.9% is occurred by radiation at a temperature of 812 °C. The gas radiation is the most effective mechanism in thermal radiation with a rate of 61.5%. The other part of 4.4% is surface radiation. In this case, the heat transfer area was calculated as 0.3036 m^2, which is 1.87 times higher than that of the real case.

According to Fig. 4, 38.0% of heat transferred to HE is occurred by convection where the remained part of 62.0% is occurred by radiation at a temperature of 828 °C. The gas radiation is the most effective mechanism in thermal radiation with a rate of 57.8%. The other part of 4.1% is surface radiation. In this case, the heat transfer area was calculated as 0.2764 m^2, which is 1.70 times higher than that of the real case.

3.2. Results of advanced exergy analysis

According to conventional exergy analysis, it is clear that the system can be improved in a considerable amount. However, it is impossible to understand how to do this by the conventional exergy analysis since the cause of the destruction is not known. So, it is a need to determine the internal and external causes. In this aim, the advanced exergy analysis method was conducted to understand the interaction of the components with each other. The results are given in Table 7.

Since the only avoidable part of exergy destruction can be reduced, that part of exergy destruction is taken into account first. Then, the exogenous and endogenous parts can be handled. In this regard, the highest avoidable exergy destruction belongs to CU with a value of 2.53 kW. R, F, and P follow CU with values of 1.25 kW, 0.11 kW, and 0.05 kW, respectively.

After scrutinizing the avoidable part, it should be handled endogenous and exogenous parts of avoidable exergy destruction to determine the internal and external sources. The internal sources depend on technological parameters on a large scale where the external ones depend on the working conditions. In this regard, the highest endogenous avoidable exergy destruction belongs to CU in an amount of 1.71 kW. These values are 0.10 kW, 0.20 kW, and 0.02 kW for F, R, and P, respectively. The percentage distribution of avoidable (AV) and unavoidable (UN) parts of exergy destruction is given in Fig. 5. The percentage distribution of endogenous (EN) and exogenous (EX) parts of

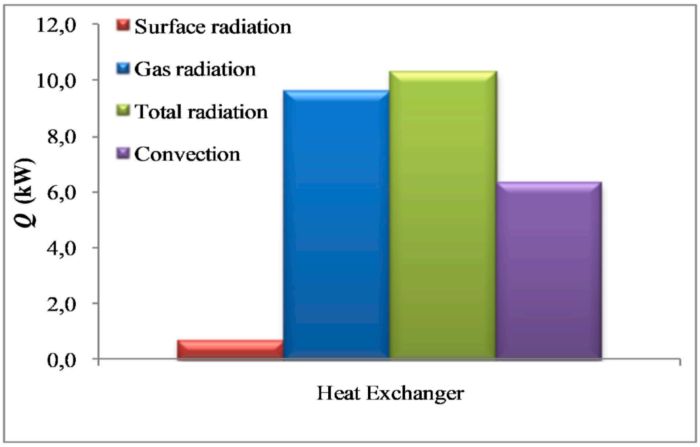

Fig. 4. The distribution of the heat transfer mechanisms in HE for the unavoidable condition.

Table 7
The results of advanced exergy analysis.

	Value (kW)								
	$\dot{Ex}_{d,k}$	$\dot{Ex}_{d,k}^{EN}$	$\dot{Ex}_{d,k}^{EX}$	$\dot{Ex}_{d,k}^{AV}$	$\dot{Ex}_{d,k}^{UN}$	$\dot{Ex}_{d,k}^{EN,UN}$	$\dot{Ex}_{d,k}^{EX,UN}$	$\dot{Ex}_{d,k}^{EX,AV}$	$\dot{Ex}_{d,k}^{EN,AV}$
CU	6.70	2.15	4.54	2.53	4.17	0.44	3.73	0.81	1.71
F	0.18	0.14	0.04	0.11	0.07	0.03	0.03	0.01	0.10
R	2.82	0.30	2.52	1.25	1.56	0.11	1.46	1.06	0.20
P	0.05	0.05	0.00	0.02	0.03	0.03	0.00	0.00	0.02
OS	***9.74***	***2.64***	***7.10***	***3.91***	***5.83***	***0.61***	***5.22***	***1.88***	***2.03***

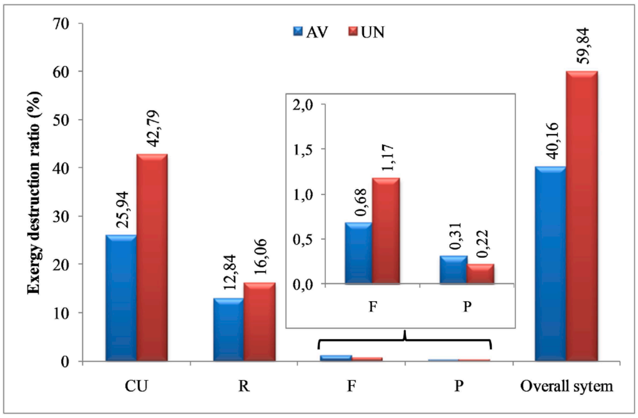

Fig. 5. Percentage distribution of avoidable and unavoidable exergy destructions in component basis.

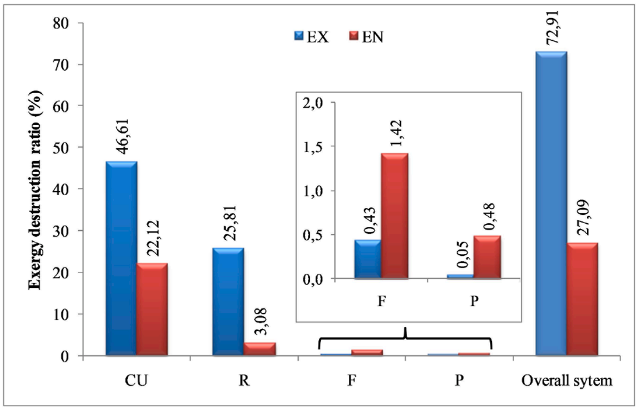

Fig. 6. Percentage distribution of endogenous and exogenous exergy destructions in component basis.

exergy destruction is given in Fig. 6. The percentage distribution of endogenous avoidable (EN,AV), endogenous unavoidable (EN,AV), exogenous avoidable (EX,AV), and exogenous unavoidable exergy (EX, UN) destructions are given in Fig. 7.

According to Fig. 5, 40.16% of total exergy destruction is the avoidable part. The highest ratio of this rate belongs to CU with a value of 25.94%. The unavoidable exergy destruction part of CU is 42.79%. The second-highest ratio belongs to R with a value of 12.84% where the third one belongs to F with 0.68%. The unavoidable parts are 16.06% and 1.17% for R and F, respectively.

According to Fig. 6, the highest ratio of endogenous exergy destruction to total destruction also belongs to CU with a value of about 22.12%. The exogenous exergy destruction part of CU is 46.61%. The second and third highest endogenous ratios belong to R with a value of 3.08% and F with a value of 1.42%. The exogenous parts are 25.81% and 0.43% for R and F, respectively. The total endogenous exergy destruction rate for the overall system is 27.09%.

According to Fig. 7, for CU, the endogenous avoidable, endogenous unavoidable, exogenous avoidable and exogenous unavoidable exergy destructions rates to total exergy destruction are 17.58%, 4.54%, 8.36%, and 38.25%, respectively. For R, these values are 2.00%, 1.08%, 10.83% and 14.98% in order. For F, these values are 1.07%, 0.35%, 0.10% and 0.33% in order.

As seen from the advanced exergy results, the wall-hanged boiler has a large potential to be improved. The improvement potential of the components is given in Fig. 8.

It is clear as seen in Fig. 8, CU has the highest improvement potential (*IP*) with a value of 14.32%. The improvement potential of R is 7.09%. This value is 0.65% for F. If all the enhancements, recoveries, and rehabilitations are made, it will be possible to increase the exergy efficiency of the overall system up to 66.97%. The modified exergy efficiency values of the improved components are given in Fig. 9.

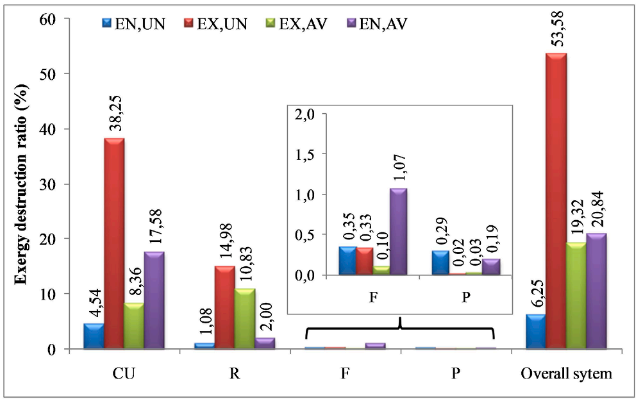

Fig. 7. Percentage distribution of endogenous avoidable, endogenous unavoidable, exogenous avoidable, and exogenous unavoidable exergy destructions on the component basis.

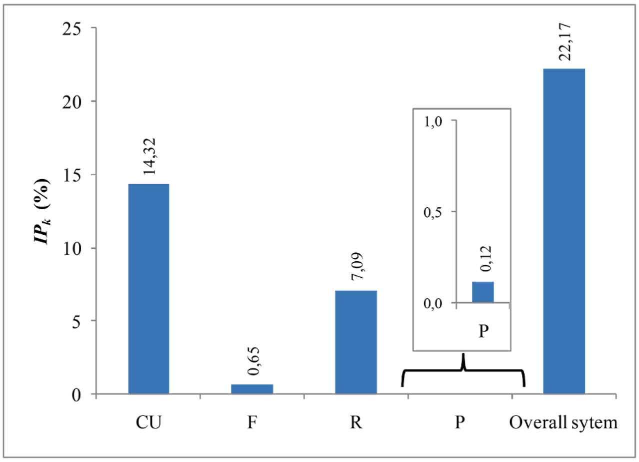

Fig. 8. Potential of improvement of the components.

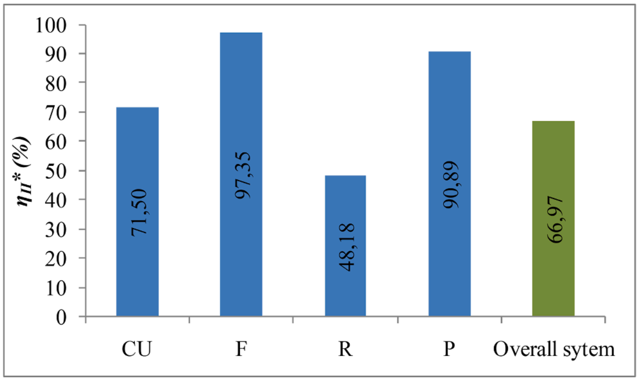

Fig. 9. Modified exergy efficiencies of the components.

According to Fig. 9, it is possible to increase the exergy efficiency of CU up to 71.50%. The modified exergy efficiency values are respectively 97.35%, 48.18%, 90.89%, for F, R, and P. The increase in the overall system is 49.5%. In the view of unavoidable conditions, it is available to decrease fuel consumption at a rate of 19.2%, which also means an emission decrease at the same amount.

4. Conclusions

This study investigates the improvement potential and limits of the improvability of a wall-hanged combi boiler used for residential heating. In this aim, advanced exergy analysis was conducted throughout the conventional exergy analysis. The real data was used obtained from the producer experiments. The ideal condition was formed imaginary according to the thermodynamics' laws. The unavoidable case was formed according to the technical possibilities. The highest exergy destruction was determined as 6.70 kW for the combustion unit, which also includes the main exchanger in itself. This also means a value of 37.9% of the total inlet exergy. The highest avoidable exergy destruction part of this destruction was determined as 25.94%. The endogenous part of avoidable destruction was calculated as 17.58% where the exogenous part was 8.36%. So, the efficiency of the overall system could be increased by up to 14.32% by the improved design of the combustion process and the main heat exchanger. For this aim, it was determined that the gas radiation form of heat transfer had been the main issue with an improvement potential of 43.3%. Therefore, the most important component was found as the main heat exchanger, which should be handled first. The burner also should be designed according to the shape of the heat exchanger to be formed of the larger radiation area with an increase up to 69.99% of the real case. The second component was as the recuperator with an improvement potential of 7.09%. In this aim, the best operation should recover the latent heat as available as close to a rate of recovery of 100%. However, this situation depends on the external conditions that determine the dew point and depends on the outlet conditions of the main exchanger. Although the fan hood has relatively lower destruction and improvement ratio, the destructions can be prevented since it originates from the heat losses at the wall surface.

CRediT authorship contribution statement

Oguz Arslan: Conceptualization, Methodology, Validation, Writing - review & editing, Investigation, Supervision. **Mehmet Ucar:** Data curation, Investigation, Resources.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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