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TECHNICAL PAPER

Improving waste incineration CHP plant efficiency by waste heat recovery for feedwater preheating process: energy, exergy, and economic (3E) analysis

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Abstract

In this work, waste heat recovery, as a passive technique with low cost, is used for increasing the efficiency of a waste incineration CHP plant via an innovative way. Here, the recovered thermal energy is suggested to be used for the feedwater preheating process instead of steam extractions of turbines or any other auxiliary heat input for this purpose. For investigating the effects of the proposal, three different scenarios are considered and compared to each other. The first scenario is an old design waste CHP plant with a fuel-burning preheating process; the second scenario is a CHP plant with steam extractions from turbines for feedwater preheating; and the last scenario is when both of the previous preheating tools are removed from the cycle and instead, the energy of the exhaust gas of the incinerator is recovered and utilized for feedwater preheating. The energy, exergy, and economic analyses of the three scenarios are carried out, and the main performance factors of these power plants are compared. The results show that the efficiency of the heat recovered case considering both heat and power outputs can reach about 94% while the plants with old-fashion design and with steam extraction lines could give only 77% and 82% efficiencies. The exergetic efficiency of the proposed solution reaches the top value of 81% among the three scenarios, and its unit product cost will be lowest among all with a value of below 11 \$/GJ. In the end, the effects of several operating parameters on the efficiency of the proposed system are investigated and reported.

Keywords Waste heat recovery · Passive energy enhancement · Waste incineration CHP plant · Feedwater preheating · Thermodynamic analysis

List of symbols

C	Unit product cost (\$/GJ)
E	Exergy (kJ)
$\dot{E}$	Exergy rate (kW)
$\overline{\Delta G}^0$	Change in Gibbs function at 1 atm (kJ/kmol)
h	Specific enthalpy (kJ/kg)
$\bar{h}_f^\circ$	Enthalpy of formation (kJ/kmol)
HHV	Higher heating value (kJ/kg)
i_r	Interest rate
LHV	Lower heating value (kJ/kg)
M	Molar mass (kg/kmol)
$\dot{m}$	Mass flow rate (kg/s)
MC	Moisture content (%)
P	Pressure (kPa)
$\dot{Q}$	Heating load (kW)
w	kmol of moisture per kmol of MSW
$\dot{W}$	Power (kW)
X_i	Mole fraction of the ith component
Z	Investment cost of components (\$)
$\dot{Z}$	Investment cost rate of component (\$/h)
z_H	Weight fraction of hydrogen
z_C	Weight fraction of carbon
z_O	Weight fraction of oxygen

Subscript and abbreviations

CC	Combustion chamber
Ch	Chemical
CI	Capital investment
COND	Condenser
CFWH	Closed feedwater heater
CRF	Capital recovery factor
D	Destruction

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HPT	High-pressure turbine
IPT	Intermediate pressure turbine
i	Inlet
L	Loss
LPT	Low-pressure turbine
WI	Waste incinerator
MSW	Municipal solid waste
o	Outlet
OFWH	Open feedwater heater
OM	Operation and maintenance
P	Product
Ph	Physical
T	Turbine
SG	Steam generator
tot	Total

Greek symbols

η_{II}	Exergy efficiency
η_{I}	Energy efficiency
η_{is}	Isentropic efficiency

1 Introduction

Waste heat recovery (WHR) can be classified in the category of passive performance improvement in energy systems as it is usually can be done via simple and low-cost methods [1]. Such processes, i.e., WHR in different systems and via different approaches, not only could enhance the energy performance of the systems but also lead to large benefits in terms of greenhouse emission reduction. Given the true meaning of sustainability, WHR, especially when done in critically important energy units, could be considered as a big step forward to the fulfillment of the SDGs of the UN [2].

There are many articles in the literature proposing, investigating, and analyzing WHR in energy systems. Tian et al. [3] analyzed the impacts of a WHR process from the motor of an electric vehicle on the performance of a thermal management system. Yang et al. [4] tried to find an optimal operating strategy for a green supply chain based on WHR quality and compared some decision models for the supply chain. Cheng et al. [5] studied the challenges and also opportunities for WHR from solid granular materials. Araiz et al. [6] did a techno-economic analysis of WHR from a system using thermoelectric generators and concluded that a very low levelized cost of electricity could be achieved by using this technology. Jannatkhah et al. [7] did an energy and exergy analysis of WHR for an integrated power plant, including Organic Rankine Cycle–Ejector Refrigeration Cycle systems. Garcia and Vargas [8] presented an investigation of WHR from fluid streams with thermoelectric elements and calculated the overall efficiency of two cases (steam and air fluids) for a variety of operating conditions. Neshat and Asghari [9] studied WHR from the exhaust gases of a Homogeneous Charge Compression Ignition engine and analyzed the impacts of different effects of reformer gas on its capacity. Feng et al. [10] thermodynamically analyzed and also optimized the effects of a WHR unit on the performance of a Brayton–Kalina hybrid cycle. Shi et al. [11] assessed and compared the impacts of weight on the performance of a WHR unit in four different configurations of automotive engines.

On the other hand, waste-feed power plants are so popular in many of the developed countries [12]. The main reasons for this are that incineration of waste not only is a good and environmentally friendly alternative for traditional waste disposal methods but also it can prepare free energy for power and heat production [13]. Such power plants mostly come in hybrid heat and power production form (CHP plants). Studies on the importance of waste power plants, methods for improving their efficiency or performance, etc. are frequent in the literature. For example, some of these works in the last few years are reviewed hereunder. Arabkoohsar and Nami [14] increased the power output of a waste-fired CHP plant parallelizing it with an Organic Rankine cycle. Arabkoohsar and Sadi [15] combined a waste firing CHP plant with a solar thermal field for uniform power production. They also investigated the possibility of integrating waste CHP plants with large absorption chillers for power, heat, and cold production [16]. WHR of a 3 MW waste-fired power plant via an ORC unit was investigated by Behzadi et al. [17] from energy, exergy, and economic points of view. They concluded that the total produced power and exergy efficiency increases by about 370 kW and 1.77%, respectively. Considering the produced electricity and environmental impact as objectives, the optimization of a waste to energy CHP power plant was performed by Safarian et al. [18] to clarify its importance than the direct combustion of MSW. In another study, different scenarios of single- and two-stage waste CHP plant was studied by Singh and Hachem-Vermette [19]. They reported the CO_2 emission of 0.05 ton/h to 0.6 ton/h for various scenarios. Yang et al. [20] presented a technical and economic investigation of energy recovery from organic part of waste sources in waste CHP plants. Vida and Lelia [21] studied various possibilities for waste treatment, including recovery or disposal in Romania, covering the development of waste CHP plants. Sahlin et al. [22] highlighted the importance of waste firing CHP plants in Sweden and investigated the impacts of increasing the capacity and number of these plants on the district heating systems of this country.

This work proposes WHR from waste firing CHP plants for the specific objective of feedwater preheating and, by that, increasing the efficiency of the cycle. Here, three different scenarios, including the proposed system, an old-fashion CHP plant with fuel firing preheater, and a cycle with steam feedwater preheaters are investigated from energy, exergy, and economic points of view. Then, the obtained results of the simulations are compared and interpreted. In summary, the novelties and contributions of this work are:

- Performing a very thorough parametric investigation on the impacts of different parameters on the achievable benefits from the proposed WHR technique in the waste CHP plant.
- Applying the exergy analysis to find the best model from the quality of the energy conversion aspect.
- Implementing a comparative economic evaluation to ascertain the best model from an economic point of view.

2 Description of Scenarios

Figure 1 shows the schematic diagrams of the three different scenarios considered on waste firing CHP plants to be investigated of the proposed plants consisting of two parts: the waste incineration unit and the Rankine cycle.

In model (a), which is an old-fashion waste CHP plant with a fuel-burning preheater, the input waste (state 1) is

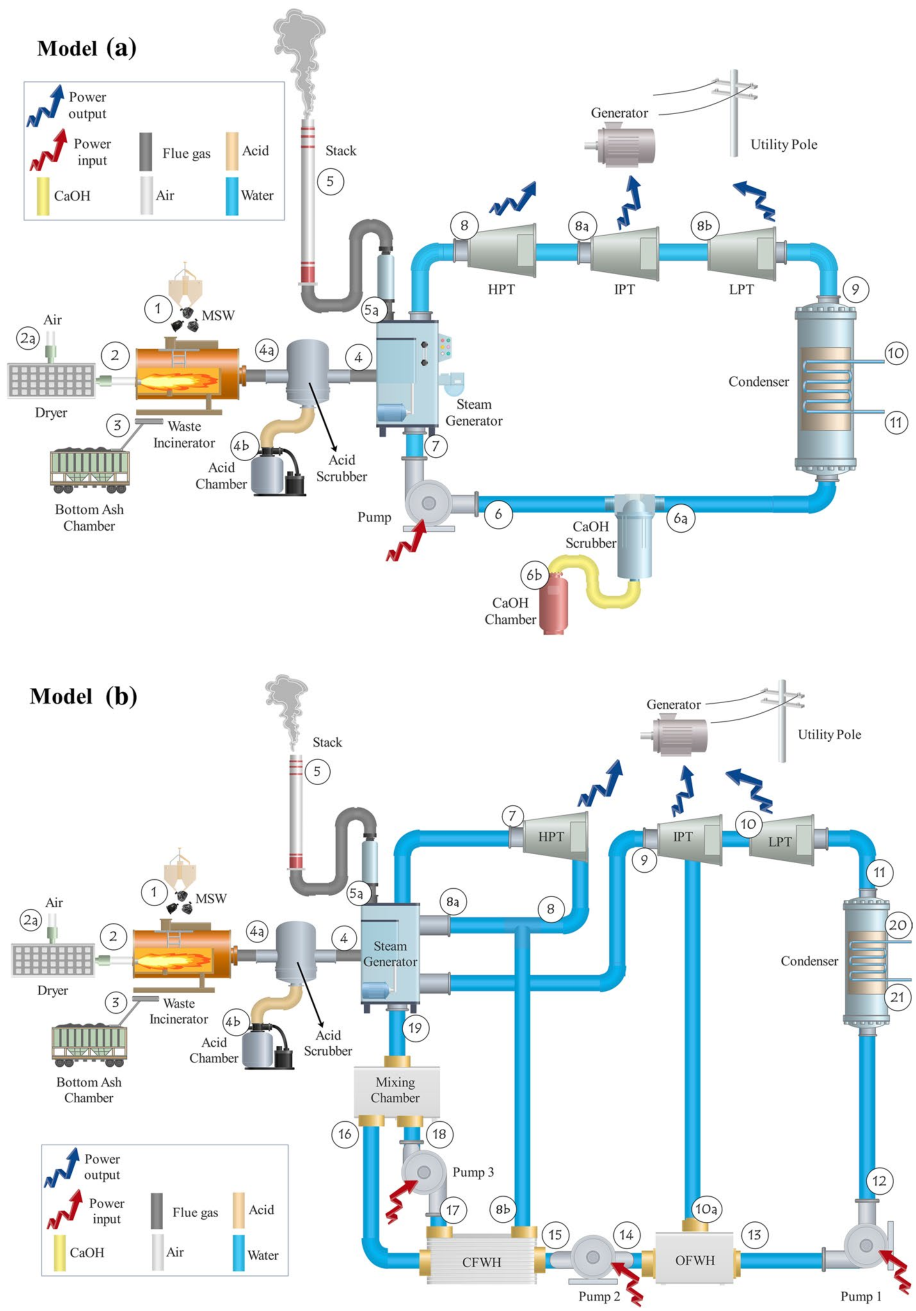
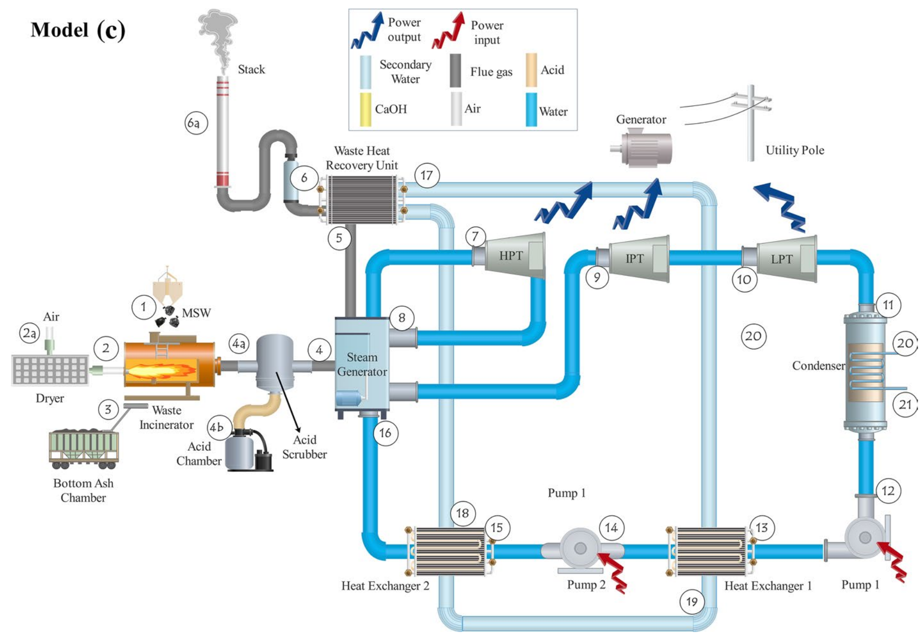


Fig. 1 Schematic of waste firing CHP plant in different scenarios: **a** old-fashion cycle, **b** reheat and regeneration lines with steam extractions, and **c** reheat and preheat lines with the recovered waste heat

Fig. 1 (continued)



fed into the incinerator, and simultaneously air (state 2) is injected. The incinerator is where waste is combusted and heat is released for being used to drive the Rankine cycle (state 4). The generated high-temperature combustion productions drive the steam generator (SG), and after doing the heating process and decreasing in the temperature leaves the boiler from the stack (state 5).

In model (b), which is a CHP cycle burning waste with preheating and regeneration lines, the preheating lines are supplied by some steam flows coming from the turbines. According to model (b), after the first expansion stage in the first turbine, steam comes back to the boiler for reheating process (state 8a) and then is expanded in the next turbine stages (states 9 and 10). There are two steam extraction lines from the turbines for preheating the feedwater before state 10a, which is the open feedwater tank, the extracted steam is mixed with feedwater while in the closed feedwater tank (state 8b) no mixing takes place.

Finally, in the proposed model, i.e., model (c), in which the action of withdrawing is replaced with a preheat line. The required energy of the preheating line is supplied by the waste recovery process on which the waste heat of the outlet flue gas (state 5) is recovered to warm the secondary water (state 17). The exploited heat is transferred to the water of the Rankine cycle via heat exchanger 1 and heat exchanger 2.

Note that in all of the three models, the heat rejected from the condenser is treated for heat supply applications.

Table 1 lists the main operating/design parameters of the considered CHP plants.

Information about the compositions and heating value of the used waste for the power plants are listed in Table 2.

Table 1 Operating parameters of each model

Parameters	Value
Mass rate of inlet waste (ton/day)	500
Combustion product temperature (°C)	1100
Stack flue gas temperature (°C)	200
Inlet pressure for first, second and third turbines (MPa)	10, 3, 0.25
Pressure of the condenser (kPa)	10
Inlet temperature of first, second and third turbines (°C)	500, 500, 350
Thermal efficiency of steam recovery unit (%)	95
Turbines and pumps isentropic efficiency (%)	85
Electricity generator efficiency (%)	95
y, extracted ratio for first preheating line	0.12
x, extracted ratio for second preheating line	0.12

Table 3 reports the molar fraction of the inorganic components of the considered waste source.

3 Modeling

3.1 Energy model

Mass and energy balance equations of each of the control volumes are required to simulate the system performance energetically. For model (a), model (b), and model (c), the mass and energy balances (Eq. 1 and Eq. 2) are applied as listed in the following tables (Tables 4, 5 and 6) [24–27]:

Table 2 The compositions and lower heating value of the waste [14]

C	H	O	N	Ash	LHV (kJ/kg)	HHV (kJ/kg)
47.36	6.25	39.57	0.91	5.91	12,500	19,601

Table 3 The characteristics and molar fractions of the inorganic components [23]

Element in ash	Mole ratio (%)	$\bar{h}_f^0$ (kJ/kmol)
Al_2O_3	7.24	23,258.44
SiO_2	40.41	13,248.07
Fe_2O_3	9.05	30,935.66
Cao	38.54	12,537.88
P_2O_5	4.76	46,904.24

Table 4 Mass and energy balance for every component of model (a)

Component	Mass balance	Energy balance
Waste incineration	$\dot{m}_1+\dot{m}_2=\dot{m}_3+\dot{m}_4$	$\dot{m}_1\ h_1+\dot{m}_2\ h_2=\dot{m}_3\ h_3+\dot{m}_4\ h_4$
Steam generator	$\dot{m}_7=\dot{m}_8,\ \dot{m}_4=\dot{m}_5$	$\dot{m}_4\ (h_4-h_5)=\dot{m}_7\ (h_8-h_7)$
Turbine	$\dot{m}_8=\dot{m}_9$	$\eta_{T,is}=\frac{\dot{W}_T}{\dot{W}_{T,is}},\ \dot{W}_T=\dot{m}_8\ (h_8-h_9)$
Condenser	$\dot{m}_6=\dot{m}_9,\ \dot{m}_{10}=\dot{m}_{11}$	$\dot{m}_6\ (h_9-h_6)=\dot{m}_{10}\ (h_{11}-h_{10})$
Pump	$\dot{m}_6=\dot{m}_7$	$\eta_{P,is}=\frac{v_6(P_7-P_6)}{\dot{W}_P},\ \dot{W}_P=\dot{m}_6\ (h_7-h_6)$

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out}h_{out} - \sum \dot{m}_{in}h_{in} \tag{2}$$

3.2 Exergy and economic models

For exergy analysis, neglecting the potential and kinetic terms, for any control volumes one has [29]

$$\dot{E}_Q - \dot{E}_W = \sum \dot{m}_{out}e_{out} - \sum \dot{m}_{in}e_{in} + \dot{E}_D \tag{3}$$

where the specific exergy is [29]:

$$e_i = e_i^{ph} - e_i^{ch} \tag{4}$$

$$e_i^{ph} = \left(h_i - h_0\right) - T_0\left(s_i - s_0\right) \tag{5}$$

To evaluate the performance of each model from an economic standpoint, the theory of specific costing theory (SPECO) is used. The cost of capital and O&M costs are written as [30]:

Table 5 Mass and energy balance for every component of model (b) [28]

Component	Mass balance	Energy balance
Waste incineration	$\dot{m}_1+\dot{m}_2=\dot{m}_3+\dot{m}_4$	$\dot{m}_1\ h_1+\dot{m}_2\ h_2=\dot{m}_3\ h_3+\dot{m}_4\ h_4$
Steam generator	$\dot{m}_{19}=\dot{m}_7,\ \dot{m}_4=\dot{m}_5,\ \dot{m}_{8a}=\dot{m}_9$	$\dot{m}_4\ (h_4-h_5)=\dot{m}_{19}\ (h_7-h_{19})+\dot{m}_{8a}\ (h_9-h_{8a})$
HPT	$\dot{m}_7=\dot{m}_8$	$\eta_{HPT,is}=\frac{\dot{W}_{HPT}}{\dot{W}_{HPT,is}},\ \dot{W}_{HPT}=\dot{m}_7(h_7-h_8)$
IPT	$\dot{m}_9=\dot{m}_{10}+\dot{m}_{10a},\ \dot{m}_{10a}=y\ \dot{m}_{10}$ $\dot{m}_8=\dot{m}_{8a}+\dot{m}_{8b},\ \dot{m}_{8b}=x\ \dot{m}_8$	$\eta_{IPT,is}=\frac{\dot{W}_{IPT}}{\dot{W}_{IPTT,is}},\ \dot{W}_{IPT}=\dot{m}_{10}(h_9-h_{10})+\dot{m}_{10a}(h_9-h_{10a})$
LPT	$\dot{m}_{10}=\dot{m}_{11}$	$\eta_{LPT,is}=\frac{\dot{W}_{LPT}}{\dot{W}_{LPT,is}},\ \dot{W}_{LPT}=\dot{m}_{10}(h_{10}-h_{11})$
Condenser	$\dot{m}_{11}=\dot{m}_{12},\ \dot{m}_{20}=\dot{m}_{21}$	$\dot{m}_{11}\ (h_{11}-h_{12})=\dot{m}_{20}\ (h_{21}-h_{20})$
Pump 1	$\dot{m}_{12}=\dot{m}_{13}$	$\eta_{P,is}=\frac{v_{12}(P_{13}-P_{12})}{\dot{W}_{P1}},\ \dot{W}_{P1}=\dot{m}_{12}\ (h_{13}-h_{12})$
Pump 2	$\dot{m}_{14}=\dot{m}_{15}$	$\eta_{P,is}=\frac{v_{14}(P_{15}-P_{14})}{\dot{W}_{P2}},\ \dot{W}_{P2}=\dot{m}_{14}\ (h_{15}-h_{14})$
Pump 3	$\dot{m}_{17}=\dot{m}_{18}$	$\eta_{P,is}=\frac{v_{17}(P_{18}-P_{17})}{\dot{W}_{P3}},\ \dot{W}_{P3}=\dot{m}_{17}\ (h_{18}-h_{17})$
OFWH	$\dot{m}_{13}+\dot{m}_{10a}=\dot{m}_{14},\ \dot{m}_{10a}=y\ \dot{m}_{10}$	$\dot{m}_{13}\ h_{13}+\dot{m}_{10a}\ h_{10a}=\dot{m}_{14}\ h_{14}$
CFWH	$\dot{m}_{8b}=\dot{m}_{17},\ \dot{m}_{15}=\dot{m}_{16}$	$\dot{m}_{8b}\ (h_{8b}-h_{17})=\dot{m}_{15}\ (h_{16}-h_{15})$
Mixing chamber	$\dot{m}_{18}+\dot{m}_{16}=\dot{m}_{19}$	$\dot{m}_{16}\ h_{16}+\dot{m}_{18}\ h_{18}=\dot{m}_{19}\ h_{19}$

Table 6 Mass and energy balance for every component of model (c)

Component	Mass balance	Energy balance
Waste incineration	$\dot{m}_1+\dot{m}_2=\dot{m}_3+\dot{m}_4$	$\dot{m}_1\, h_1+\dot{m}_2\, h_2=\dot{m}_3\, h_3+\dot{m}_4\, h_4$
Steam generator	$\dot{m}_{16}=\dot{m}_7$, $\dot{m}_4=\dot{m}_5$, $\dot{m}_8=\dot{m}_9$	$\dot{m}_4\,(h_4-h_5)=\dot{m}_{16}\,(h_7-h_{16})+\dot{m}_8\,(h_9-h_8)$
HPT	$\dot{m}_7=\dot{m}_8$	$\eta_{\mathrm{HPT,is}}=\frac{\dot{W}_{\mathrm{HPT}}}{\dot{W}_{\mathrm{HPT,is}}}$, $\dot{\mathrm{W}}_{\mathrm{HPT}}=\dot{m}_7(h_7-h_8)$
IPT	$\dot{m}_9=\dot{m}_{10}$	$\eta_{\mathrm{IPT,is}}=\frac{\dot{W}_{\mathrm{IPT}}}{\dot{W}_{\mathrm{IPTT,is}}}$, $\dot{\mathrm{W}}_{\mathrm{IPT}}=\dot{m}_9(h_9-h_{10})$
LPT	$\dot{m}_{10}=\dot{m}_{11}$	$\eta_{\mathrm{LPT,is}}=\frac{\dot{W}_{\mathrm{LPT}}}{\dot{W}_{\mathrm{LPT,is}}}$, $\dot{\mathrm{W}}_{\mathrm{LPT}}=\dot{m}_{10}(h_{10}-h_{11})$
Condenser	$\dot{m}_{11}=\dot{m}_{12}$, $\dot{m}_{20}=\dot{m}_{21}$	$\dot{m}_{11}\,(h_{11}-h_{12})=\dot{m}_{20}\,(h_{21}-h_{20})$
Pump 1	$\dot{m}_{12}=\dot{m}_{13}$	$\eta_{P,is}=\frac{v_{12}(P_{13}-P_{12})}{\dot{W}_{P1}}$, $\dot{\mathrm{W}}_{\mathrm{P1}}=\dot{m}_{12}\,(h_{13}-h_{12})$
Pump 2	$\dot{m}_{14}=\dot{m}_{15}$	$\eta_{\mathrm{P,is}}=\frac{v_{14}(P_{15}-P_{14})}{\dot{W}_{\mathrm{P2}}}$, $\dot{\mathrm{W}}_{\mathrm{P2}}=\dot{m}_{14}\,(h_{15}-h_{14})$
Heat exchanger 1	$\dot{m}_{13}=\dot{m}_{14}$, $\dot{m}_{17}=\dot{m}_{19}$	$\dot{m}_{13}\,(h_{14}-h_{13})=\dot{m}_{17}\,(h_{19}-h_{17})$
Heat exchanger 2	$\dot{m}_{15}=\dot{m}_{16}$, $\dot{m}_{18}=\dot{m}_{19}$	$\dot{m}_{15}\,(h_{16}-h_{15})=\dot{m}_{19}\,(h_{18}-h_{19})$
WHR unit	$\dot{m}_{17}=\dot{m}_{18}$, $\dot{m}_5=\dot{m}_6$	$\dot{m}_5(h_5-h_6)=\dot{m}_{17}\,(h_{18}-h_{17})$

$$\dot{Z}_k=\dot{Z}_k^{\mathrm{CI}}+\dot{Z}_k^{\mathrm{OM}} \tag{6}$$

$$\dot{Z}_k^{\mathrm{CI}}=\left(\frac{\mathrm{CRF}}{\tau}\right)Z_k \tag{7}$$

$$\mathrm{CRF}=\frac{i_{\mathrm{r}}\left(1+i_{\mathrm{r}}\right)^n}{\left(1+i_{\mathrm{r}}\right)^n-1} \tag{8}$$

$$\dot{Z}_k^{\mathrm{OM}}=\gamma_k Z_k \tag{9}$$

where n is the number of the years (20 years) and i is the interest rate (12%). For $\dot{Z}_k$ to be accounted in the present year, one has [31]:

$$\dot{Z}_k^{\mathrm{PY}}=\dot{Z}_k\times\frac{\mathrm{CI}_{\mathrm{PY}}}{\mathrm{CI}_{\mathrm{RY}}} \tag{10}$$

The related purchase cost (Z_k) is listed in Table 7.

3.3 Performance assessment

Finally, to investigate and compare the performance of the proposed models, net generated power and power, and CHP efficiencies are, respectively, written as follows [33–36]:

$$\dot{W}_{\mathrm{T}}=(\dot{W}_{\mathrm{HPT}}+\dot{W}_{\mathrm{IPT}}+\dot{W}_{\mathrm{LPT}})\times\eta_{\mathrm{generator}} \tag{11}$$

$$\dot{W}_{\mathrm{P}}=\dot{W}_{\mathrm{Pumps}}/\eta_{\mathrm{pump}} \tag{12}$$

$$\dot{W}_{\mathrm{net}}=\dot{W}_{\mathrm{T}}-\dot{W}_{\mathrm{P}} \tag{13}$$

Table 7 Cost functions for every component of each model [23, 29, 32]

Component	Z_k (\$)
Waste incinerator	$Z_{\mathrm{cc}}=c_1\dot{m}_{\mathrm{air}}\left(1+\exp\left(c_2T_{\mathrm{out}}-c_3\right)\right)\frac{1}{0.995-\frac{P_{\mathrm{out}}}{P_{\mathrm{in}}}}$ $c_1=48.64$ \$/(kg/s), $c_2=0.018\ \mathrm{K}^{-1}$, $c_3=26.4$
Pump	$Z_{\mathrm{P}}=c_4\dot{W}_{\mathrm{Pm}}^{0.71}$ $c_4=3540$ \$/(kW)$^{0.71}$
Turbine	$Z_{\mathrm{T}}=c_5\dot{W}_{\mathrm{T}}^{0.7}$ $c_5=6000$ \$/(kW)$^{0.7}$
Steam generator	$Z_{\mathrm{SG}}=C_6\left(\left(\frac{Q_{\mathrm{ec}}}{\Delta T_{\mathrm{lm,ec}}}\right)^{0.8}+\left(\frac{Q_{\mathrm{ev}}}{\Delta T_{\mathrm{lm,ev}}}\right)^{0.8}\right)+C_7\dot{m}_{\mathrm{water}}+C_8\left(\dot{m}_{\mathrm{syngas}}\right)^{1.2}$ $c_6=6570$ \$/(kW/K)$^{0.8}$, $c_7=21{,}276$ \$/(kg/s), $c_8=1184.4$ \$/(kg/s)$^{1.2}$
Condenser	$Z_{\mathrm{COND}}=c_9\dot{m}_{\mathrm{water}}$ $c_9=1773$ \$/(kg/s)
Feedwater heater	$Z_{\mathrm{FWH}}=66\dot{Q}\left(\frac{1}{T_{\mathrm{TTD}}+c_{10}}\right)^{0.1}$ $c_{10}=4$ for OFWH and $c_{10}=6$ for CFWH
Heat exchanger	$Z_{\mathrm{HEX}}=C_{11}\left(\frac{A_{\mathrm{AHX}}}{0.093}\right)^{0.78}$ $c_{11}=390$ \$

$$\eta_{\mathrm{I,Power}}=\frac{\dot{W}_{\mathrm{net}}}{\dot{m}_{\mathrm{MSW}}\times\mathrm{LHV}_{\mathrm{MSW}}}\times 100 \tag{14}$$

$$\eta_{\mathrm{I,CHP}}=\frac{\dot{W}_{\mathrm{net}}+\dot{Q}_{\mathrm{cond}}}{\dot{m}_{\mathrm{MSW}}\times\mathrm{LHV}_{\mathrm{MSW}}}\times 100 \tag{15}$$

$$\eta_{II,CHP} = \frac{\sum_{i=1}^{n_p} \dot{E}_{p_i}}{\dot{E}_{in}} \tag{16}$$

$$c_{P,total} = \frac{\sum_{k=1}^{n_k} \dot{Z}_k + \dot{C}_{fuel}}{\sum_{i=1}^{n_p} \dot{E}_{p_i}} \tag{17}$$

Here $\dot{E}_{in}$ is:

$$\dot{E}_{in} = e_{MSW}^{ch} + w \times e_{water}^{ch} + 4.76 \times m \times \bar{e}_{air}^{ch} \tag{18}$$

$$e_{MSW}^{ch} = LHV_{MSW} \frac{1.044 + 0.016\frac{z_H}{z_C} - 0.3493\frac{z_O}{z_C}\left(1 + 0.0531\frac{z_H}{z_C}\right)}{1 - 0.4124\frac{z_O}{z_C}} \tag{19}$$

where z_H, z_C, and z_O are the weight fraction of H, C, and O_2 atoms in the waste. Also, m and w are the kmol of air and moisture per kmol of MSW, which enters the waste incinerator as follows [37]:

$$w = \frac{M_{MSW}MC}{18(1 - MC)} \tag{20}$$

4 Results and discussion

A comparison of the net produced power, power efficiency, CHP energy, and exergy efficiencies as well as unit product costs for different scenarios is presented in Fig. 2. As seen, model (c) is the best model with the highest net produced power and efficiencies and lowest unit product cost. Figure 2 indicates that the WHR process in model (c) leads to 870 kW higher net produced power, 1.17% higher power efficiency, 11.02% and 9.06% higher CHP energy and exergy efficiencies, respectively, and 0.56 \$/GJ lower unit product cost compared to the conventional Rankine cycle (Model (b)). Therefore, the proposed model is a promising technique to enhance the performance of the waste firing CHP plant from energy/exergy/economic standpoints.

4.1 Parametric study

After making an overall assessment of the three scenarios, a parametric study is carried out to investigate the effects of changing in various operating parameters of the system on the efficiencies, work production, and unit product cost of different scenarios. These parameters include waste mass flow rate, waste moisture, combustion temperature, inlet temperature and pressure of turbines, and condenser pressure, and ambient temperature. Furthermore, the variation of extracted ratio for the first and second preheating lines as significant decision parameters on performance and economic indicators of model (b) is investigated. Eventually, the variation of performance and economic objectives with flue gas condensation temperature of model (c) is examined.

The influence of waste mass flow rate on performance and economic indicators of each model is presented in Fig. 3. As seen, the waste mass flow rate has almost no effects on the efficiencies of the cycles, and of course, expectedly, increasing the supplied waste flow rate linearly increases the rate of work production of the cycles. Figure 3 further indicates that the increase in waste mass flow rate results in a decrease in unit product cost for each model, which is rational based on Eq. (17).

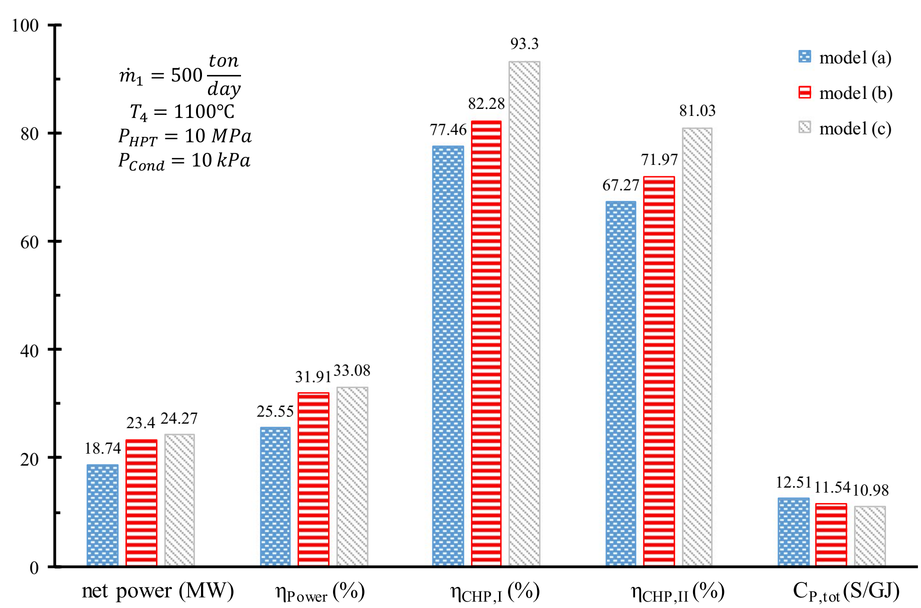


Fig. 2 The comparison of the net produced power, power and CHP energy/exergy efficiency values, and unit product cost

An effective factor on the performance and economic aspects of an incinerator is the moisture content of the input fuel, waste. From Fig. 4, it can be inferred that the net produced power decreases as the moisture content increases. This is reasonable because a higher moisture content results in a decrease in enthalpy of produced syngas, so the value of heat transferred to the water will decrease. When the moisture content increases, the unit

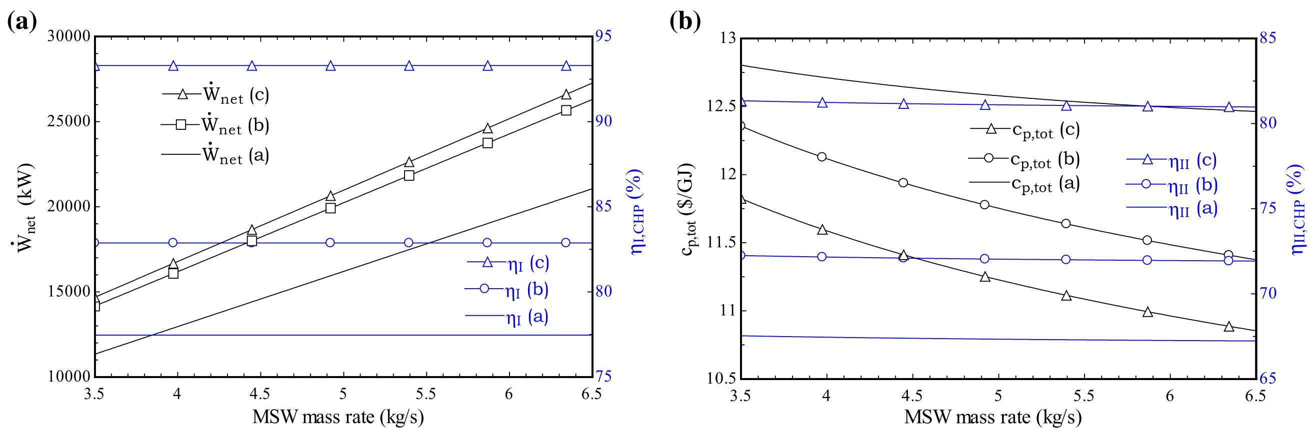


Fig. 3 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with waste mass rate

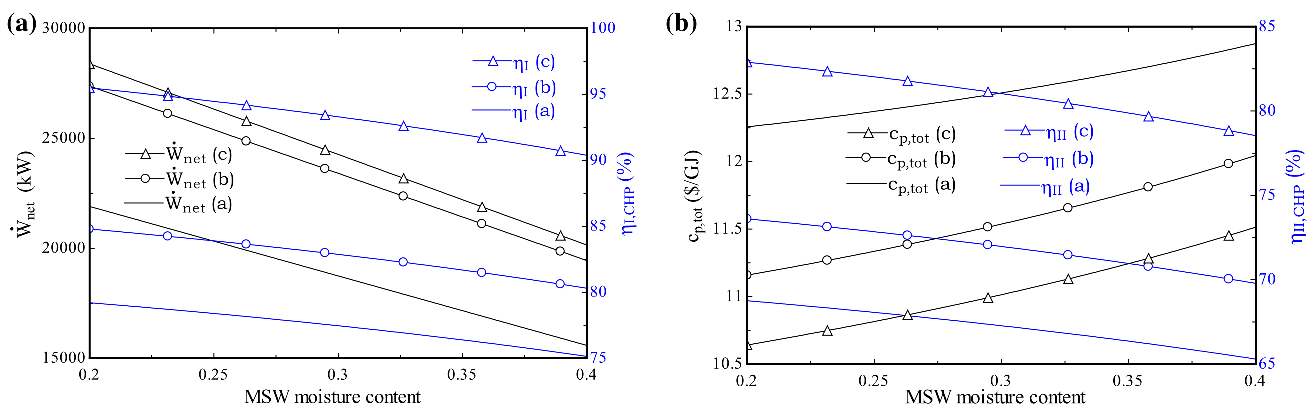


Fig. 4 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with waste moisture content

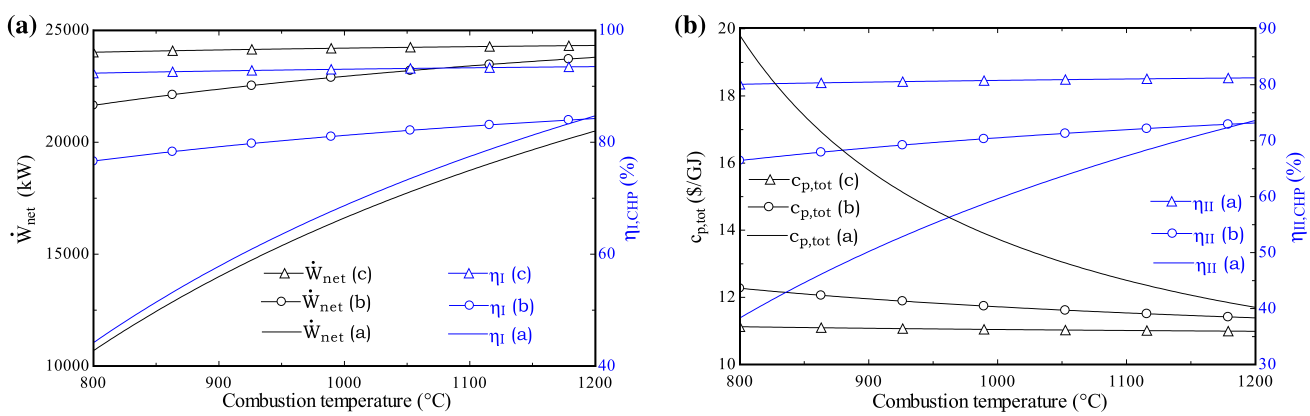


Fig. 5 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with combustion temperature

product cost increases too, which is unfavorable. It is also shown that in the selected domain of moisture content, model (c) is the most appropriate model because of not only the highest output power and efficiencies but also the lowest unit product cost.

Figure 5 shows the effect of combustion temperature on the performance and economic points of view of each model. By increasing the combustion chamber temperature from 800 to 1200 °C, while for model (a) the net produced power is increased considerably, it is increased insignificantly for models (b) and (c). Also, according to Fig. 5, picking up the combustion temperature results in a higher enthalpy in the combustion products stream and, thus, a higher input energy rate into the boiler, improving the overall power and energy and exergy efficiencies of the CHP plant in different cycles. Figure 5 illustrates that in opposite to the trend of performance indicators, unit product cost is decreased as the combustion temperature increases.

Figure 6 illustrates how the efficiencies, net power output, and economic aspect of the power plants in different scenarios will change as a result of changing the inlet pressure of the high-pressure turbine. As seen, picking up the pressure of the inlet steam into the turbine leads to the growth of enthalpy, and thus, at a constant mass flow, the power output goes up. Increasing the output power of the turbine, the performance of the cycles improves in all the three scenarios. Also, the positive effect of the increase in the inlet pressure of the high-pressure turbine on unit product cost can be inferred from Fig. 6b.

In Fig. 7, the variation of power and CHP efficiencies, unit product cost, and the power output of the high-pressure turbine inlet temperature for each model are demonstrated. By increasing the high-pressure turbine inlet temperature, a higher rate of work could be generated by the turbine set, and this means higher rates of power productions by the power cycles in all the three scenarios. Figure 7 further depicts that while the more high-pressure turbine inlet temperature

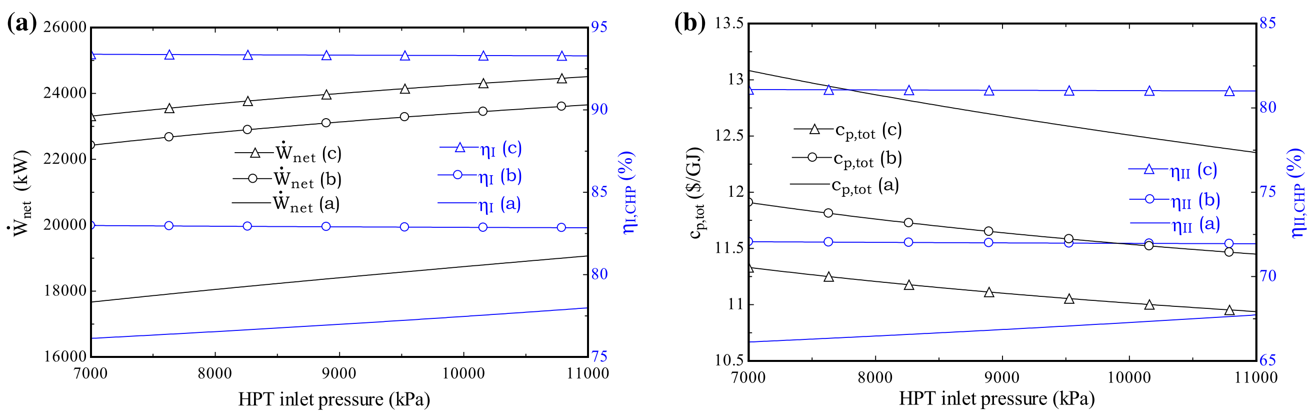


Fig. 6 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with an inlet pressure of high-pressure turbine

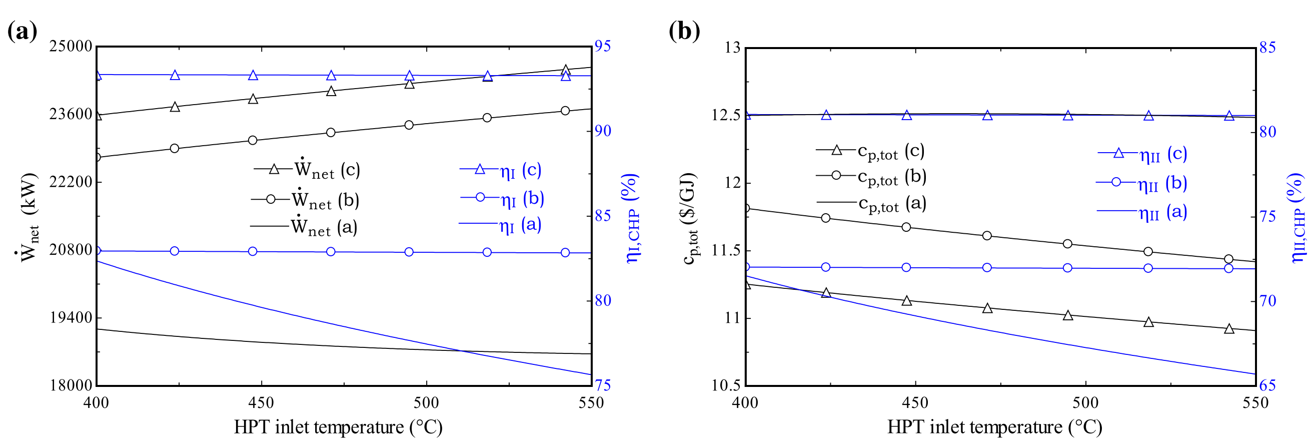


Fig. 7 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with the inlet temperature of high-pressure turbine

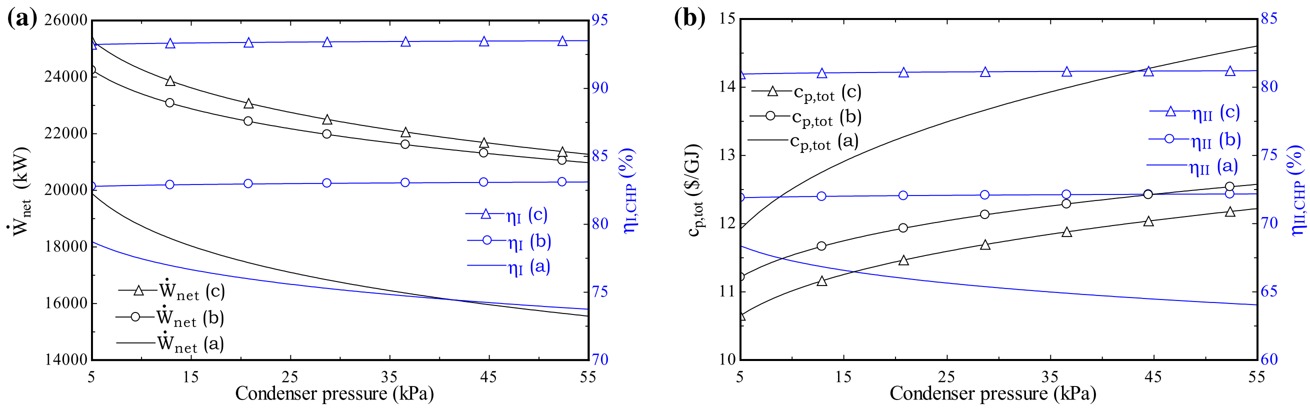


Fig. 8 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with condenser pressure

results in the rather constant efficiencies of models (b) and (c), it leads to a considerably lower energy and exergy efficiencies than model (a). According to Fig. 7b, increasing the high-pressure turbine inlet temperature is economically beneficial in each model.

The more condenser pressure leads to lower enthalpy differences in the low-pressure turbine; thus, the output power value drops (according to Fig. 8a). It could also be seen that the growth of condenser pressure has negative impacts on the models' operation proficiencies from the exergy efficiency and economic standpoints. In contrast with exergy efficiency, the value of energy efficiency increases with an increase in the condenser pressure for models (b) and model (c). It is noteworthy that same as the previous figures, in the whole range of condenser pressure, model (c) offers the highest power production rate as well as energy/exergy efficiency values along with the lowest unit product cost.

Figure 9 illustrates the changes in the values of exergy/energy efficiencies, work production, and the unit cost of product versus the changes in the ambient temperature. Referring to this figure, by increasing the ambient temperature from − 10 to 40 °C, the value of exergy efficiency of models (a), (b), and (c) increases by about 0.3%, 0.32%, and 0.36%, respectively. Also, the net produced power, energy efficiency, and unit product cost of each model do not depend on ambient temperature, as shown in Fig. 9.

The changes in energy/exergy/economic indices of model (b) with the extracted ratio for the second preheating line (x) is presented in Fig. 10. When x increases from 0.08 to 0.22, energy/exergy efficiency values decrease with the same rate, and net produced power increases. According to Fig. 10b, the unit product cost decreases with an increase in x up to a certain value (0.13) then increases.

The influence of the extracted ratio for the first preheating line (y) of model (b) is illustrated in Fig. 11. Referring to Fig. 11, by increasing y from 0.08 to 0.22, the power production rate picks up about 31 kW, and energy/exergy efficiency values improve by, respectively, about 1.59% and 1.38%. This is justified since a more extracted ratio means a higher heat transferred to steam entering the waste incinerator, so the value of steam temperature increases, and based

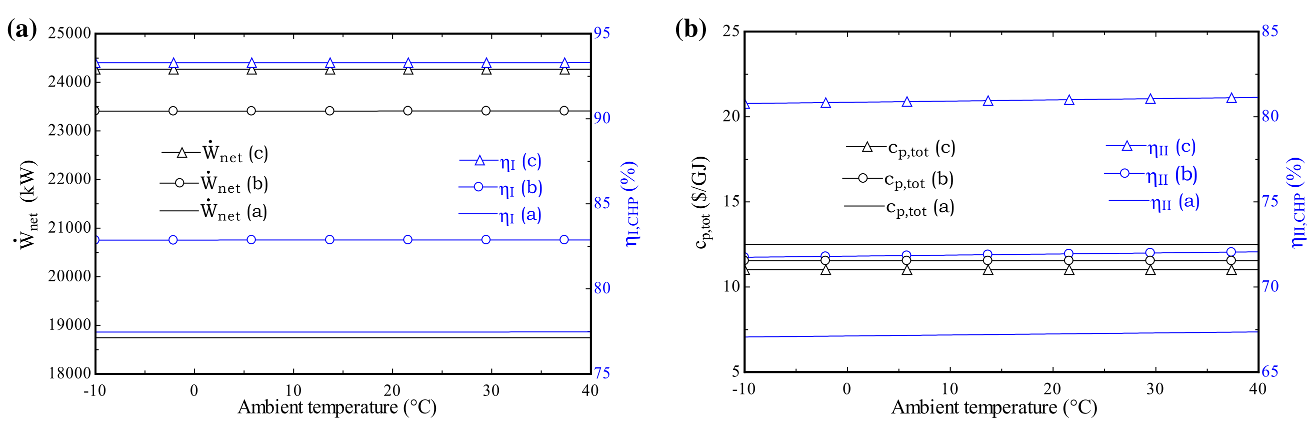


Fig. 9 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency with ambient temperature

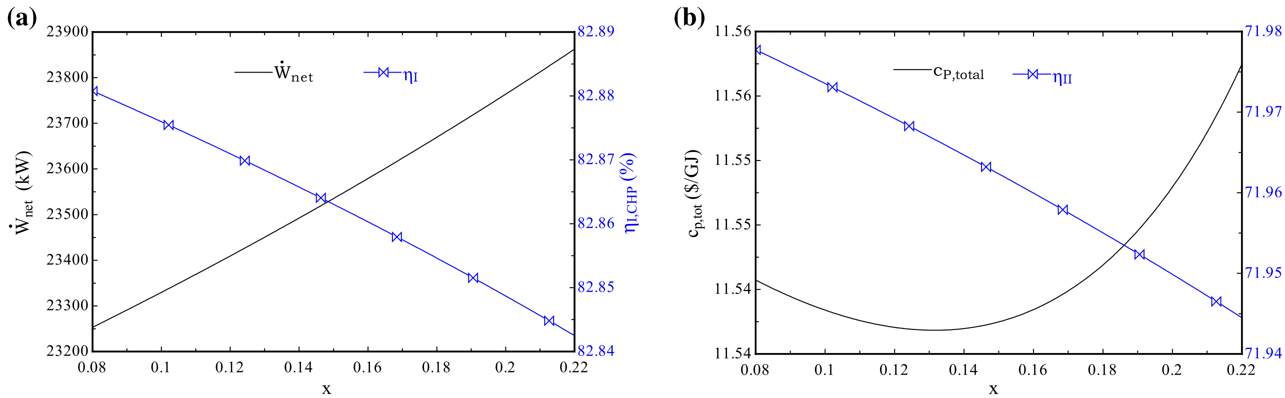


Fig. 10 Variation of the **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency of model (b) with the extracted ratio for the second preheating line (x)

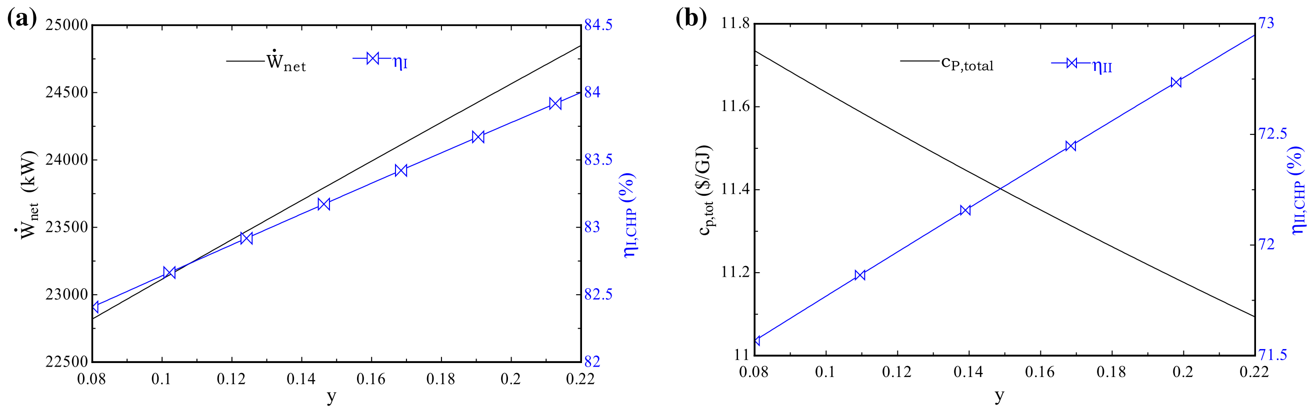


Fig. 11 Variation of **a** net produced power and energy efficiency, **b** product unit cost and exergy efficiency of model (b) with the extracted ratio for the first preheating line (y)

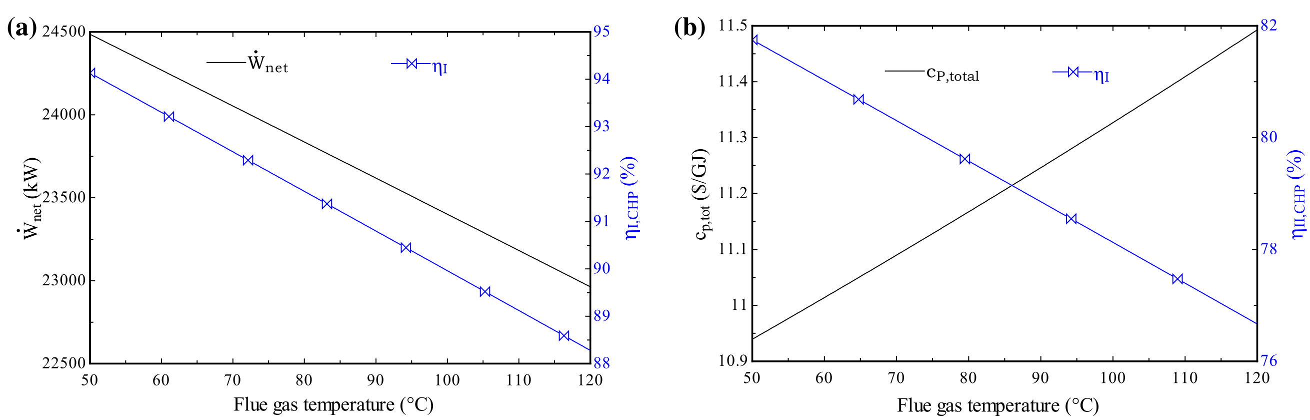


Fig. 12 Variation of **a** net produced power and energy efficiency, **b** unit cost of product and exergetic efficiency value of model (c) with flue gas condensation temperature

on energy balance in the incinerator, a higher mass flow rate goes into the high-pressure turbine. By increasing y, the total produced exergy increases; thus, the unit product cost decreases as can be inferred from Fig. 11b.

Flue gas condensation temperature of model (c) is another substantial decision parameter in which its effect on system performance from energy, exergy, and economic standpoints is investigated in Fig. 12. Based on the figure, a more flue gas temperature leads to a lower waste heat exploited via WHRU; hence, lower heat is transferred to steam in heat exchangers. In this regard, as the flue gas condensation temperature increases from 50 to 120 °C, the net produced power and energy and exergy efficiencies are decreased, and a higher unit product cost is obtained. Consequently, it is an appropriate option to decrease the flue gas condensation temperature up to the dew point temperature of flue gas entering the chimney.

4.2 Results of exergy analysis

Rates of irreversibilities of different components are presented in Fig. 13. Mainly, chemical reactions, some irreversible processes like mixing, and large temperature differences are the key sources of irreversibility in exergy analysis. As seen, the waste incinerator has the largest irreversibility rate in each model because of the existence of all of the aforementioned sources. Also, due to the high value of temperature difference between cold and hot side streams, HRSG is the second source of irreversibility in each model. Figure 13 further shows that in models (b) and (c), respectively, condenser and WHRU are the third vital components from an exergy point of view with a destruction rate of 8127 kW and 5623 kW. Similarly, the rate of irreversibility of OFWH in model (b) is 407.6 kW greater than CFWH as a result of the mixing of cold and hot streams. Furthermore, the exergy loss rate in the model (c) (2857 kW) is much lower than its value in model (a) (6394 kW) and model (b) (5098 kW). This shows the importance of the flue gas condensation process from exergy analysis point of view. Finally, what stands out from Fig. 13 is that the irreversibility rate of the pumps is negligible, because they can be assumed having constant temperature without any mixing and chemical reaction.

5 Conclusions and Recommendations

This study presents a passive proposal (i.e., WHR from the exhaust gases) for energy efficiency enhancement in municipal solid waste firing CHP plants. Here, it is suggested that the recovered heat be used for the preheating of the pressurized water pumped into the boiler. In this way, there will be no need to extract steam from the turbines, and this will increase the efficiency of the cycle for both electricity and

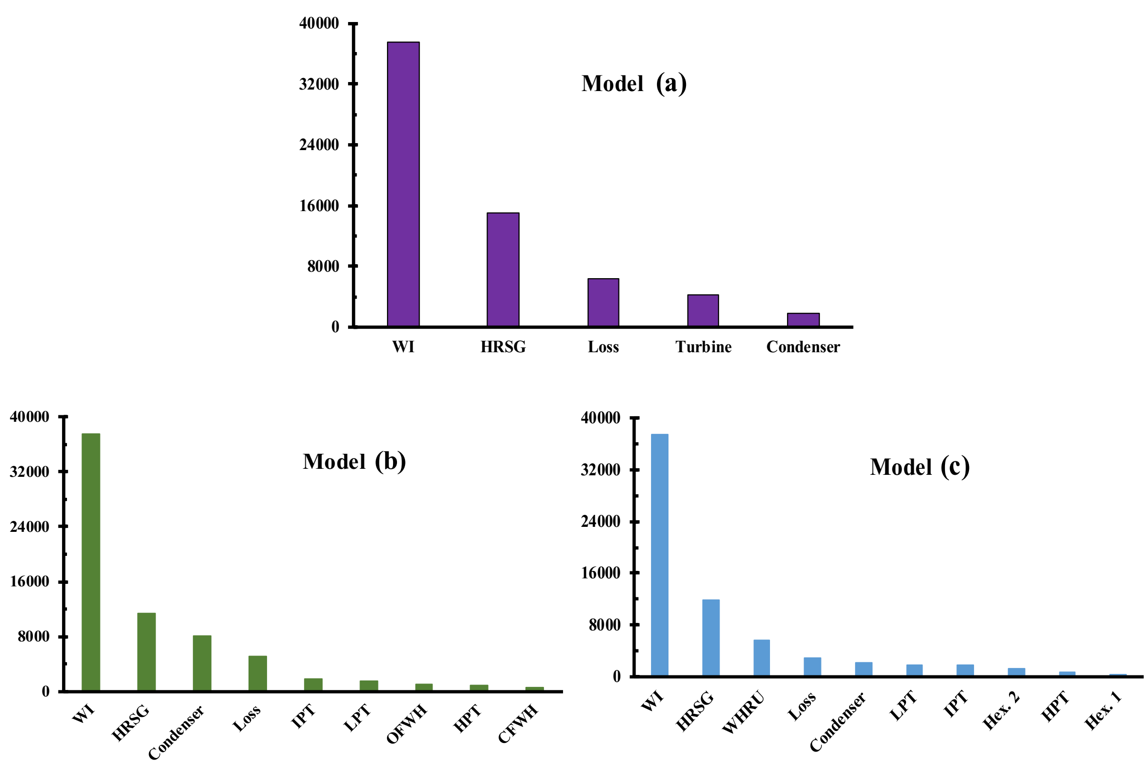


Fig. 13 Exergy destruction (kW) in various system components for each model

heat productions. For a good judgment of the proficiency of the proposal compared to other possibilities, three different scenarios are considered and compared to each other. The energy, exergy, and economic analysis of the three scenarios is carried out, and the main performance objectives of these power plants are compared. In addition, a very detailed parametric study was carried out on the suggested, and the competing power plants and the effects of various operating parameters on the rate of power production, energy and exergy efficiencies, and unit product cost of these cycles were investigated. The important conclusions are as follows:

- The results indicate that the proposed system is better than the other two systems and with such a small change in the configuration of such plants (adding a WHR unit form the exhaust), significant technical and economic benefits could be achieved.
- The results show that the energy efficiency of the heat recovered case considering both heat and power outputs can reach about 94% while the plants with old-fashion design and with steam extraction lines could give only 77% and 82% efficiencies.
- In addition to superiority from the thermodynamic aspect, the heat recovered case is also environmentally superior with 0.56 $/GJ and 1.53 $/GJ lower value of unit product cost than the plants with old-fashion design and with steam extraction lines, respectively.
- In each model, the waste incinerator and steam generator are the main sources of exergy destruction.

Finally, a series of recommendations for the future extension of this study can be outlined as below:

- Applying single and multi-objective optimization to find the optimal points from a performance/economic standpoint.
- Replacing feedwater heater in the plant with steam extraction lines (model (b)) with solar-driven systems.
- Applying an ORC as a low-grade WHR unit to exploit the waste of the plants with old-fashion design and with steam extraction lines to decrease the total exergy loss rate.

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