Journal of
Renewable Energy Research and Applications (RERA)

Vol 1, No 1, 2020, 123-134 *DOI: 10.22044/rera.2020.9123.1017*

Energy and Exergy Assessment of a New Heat Recovery Method in a Cement Factory

M. Jalili[1], R. Cheraghi[1], M. M. Reisi[2] and R. Ghasempour[1*]

1. Renewable enegies and environmental engineering Department, Faculty of new Sciences and Technologies,University of Tehran, Tehran, Iran..
2. Faculty of Mechanical Engineering, Shahrood University of Technology, Shahrood, Iran.t

Receive Date 25 December 2019; Revised 20 January 2020; Accepted Date 28 Janury 2020
**Corresponding author: ghasempour.r@ut.ac.ir (R. Ghasempour)*

Abstract
Cement plants are one of the massive energy consumers and greenhouse gas producers. The processes carried out in a cement factory have considerable energy losses, and mostly happen due to the exhausted gases and airflow for cooling the clinker. The energy consumption in a regular plant is 25% electrical and 75% thermal. The main goal of this work is to represent a thermal recycling system in cement plants to generate power from high-temperature exhaust gases from the preheater and cooler high-temperature air. A thermodynamic analysis is carried out by the EES software, and the exergy efficiency and exergy destruction of each component of the system are obtained. Moreover, a parametric study on the suggested cycle is used, and the results obtained show that if the input temperature and pressure of turbines get closer to the critical point of the expanded working fluid in turbines, the rate of net output work increases, which leads to an increase in the exergy efficiency of the whole system. The increased network of the cycle is almost 20%, which would rise from 3497 kW to 4186 kW, and the exergy efficiency would rise from 38% to 45.94%.

Keywords: *Exergy efficiency; Exergy destruction; Cement industry; CO_2 emissions.*

1. Introduction

Energy consumption in the world has increased rapidly in the recent years. This increase has led to the reduction in fossil fuel resources, and many environmental issues have been put forward [1]. The world CO_2 emissions have increased from approximately 23,623 tonnes in 2000 to more than 33,443 in 2017. Since carbon dioxide emissions are mainly attributed to the use of fossil fuels and low technology efficiency, the importance of using clean energy systems such as fuel cells, and solar, wind, and geothermal energies, and also optimization of the energy systems has increased [2, 3]. One of the massive energy consumers is the cement industry. Figures 1 and 2 successively show the energy consumption of industries and CO_2 emissions in different parts of the industry including the cement industry.

The cement production in underdeveloping countries is increasing continuously as a result of the economic growth. China has produced about 1388 million tons of cement in 2008, which is half of the whole world's cement production [5]. Figure 3 shows the share of major cement producing countries in total production in 2017. As shown in this figure, China is the largest cement producer. The energy required to produce one ton of cement is between 3 to 5 gigajoules in a ton [6]. In advanced furnaces, the average energy consumption is about 2.95 gigajoules per ton of produced cement, while in some countries, it is more than 5 gigajoules per ton of produced cement. For example, the average energy consumption of major Chinese clinker plants is 5.4 gigajoules per ton [7]. Each ton of Portland cement produces about 1 ton of carbon dioxide [8]. Most carbon dioxide emissions in cement production are from calcination of limestone (50%) and fuel combustion (40%) [9].

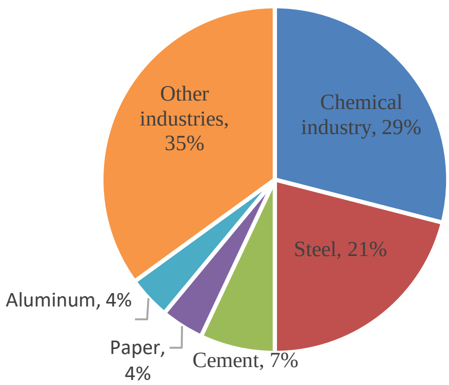

Figure 1. Energy consumption in different industries [4].

In a common cement plant, 25% of the whole required energy is electrical and the other 75% is thermal. Figure 4 shows the type and scale of energy consumption in a cement production process.

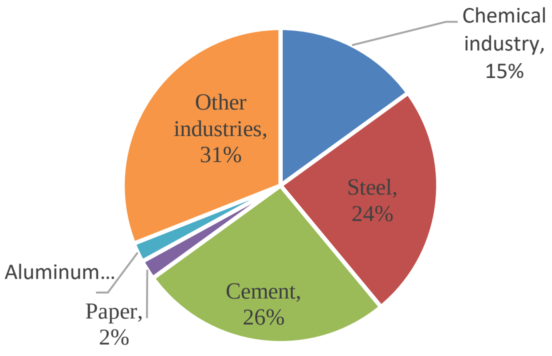

Figure 2. CO2 emissions from industrial activities [4].

However, this process has considerable heat losses, mostly caused by exhausted gases and environmental airflow, which is used for cooling the clinker that causes 35% to 40% of heat losses [11]. Almost 26% of inlet heat to the system would be wasted by dirt, clinker draining, radiation, and convection from furnace and preheaters [12]. Generally, energy casualties in a cement plant could be categorized as:

1) hot gases from stacks;
2) cooling stacks;
3) the shell of the furnace (radiation and convection).

A heat recovery system can be used to increase the efficiency of a cement plant and also reduce the number of emissions from pollutants [13]. Using a waste heat recovery system in a cement plant will reduce the dependence on the national electric grid, and will increase the profits of the plants. The wasted heat can also be used for preheating the raw materials before the clinker process [14]. Among the thermal cycles that generate electricity, the organic Rankine cycle is considered as an appropriate technology for converting low heat to power [15]. This power cycle is relatively simple and has a high flexibility of efficiency from different thermal resources [16].

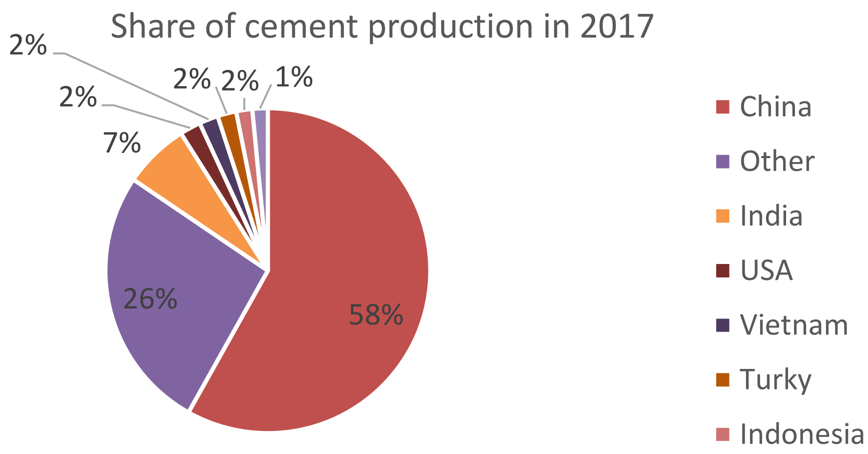

Figure 3. Share of cement production in different countries in 2017 [10].

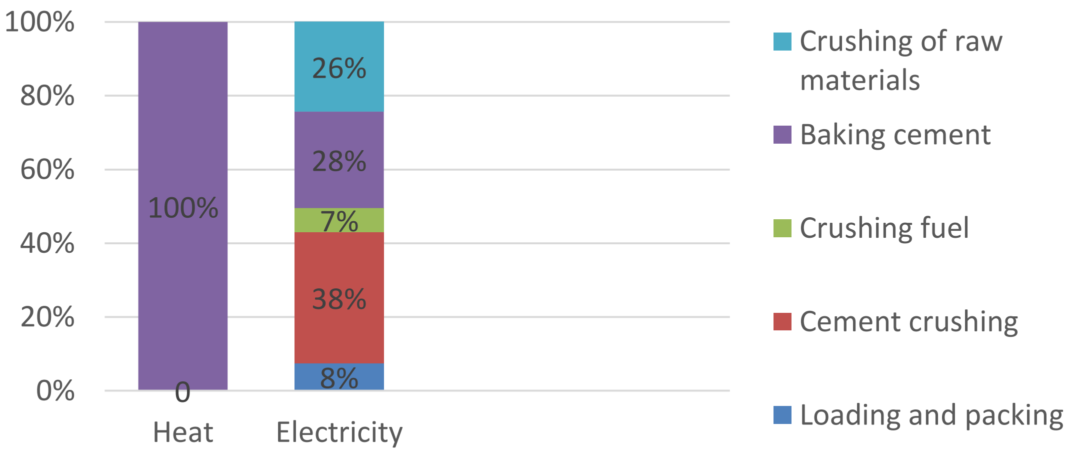

Figure 4. Type and scale of energy consumption in a cement production process [11].

Wang and partners have used four different cycles, i.e. organic Rankine cycle, Kalina cycle, and two types of the steam cycle, for recovering wasted heat from preheater's exhausted gas and cooler for producing power in cement plants. Exergy analysis and parametric optimization are carried out to achieve the maximum efficiency. The results obtained show that the Kalina cycle has a better performance in a cement plant [17]. Zeeshan *et al*. [18] have analyzed the effect of setting up a Waste Heat Recovery Power Plant in a cement plant on consuming energy and water and also cement production. The potential of installing a waste heat recovery system in Islam Abad (Pakistan) has been investigated. The annual production capacity of this system is approximately 38400 MW, which causes a 32.98% saving in electricity cost and also reducing greenhouse gases around 1049965 tons per year.

Madlool *et al*. [11] have studied energy consumption at a cement plant. They proved that the dry process had more efficiency than the wet process and using a fan that had an adjustable speed ability for clinker, about 30% of energy consumption could be saved in 2 to 3 years payback period. They also calculated the amount of CO_2 reduction and the payback period of capital for different energy-saving strategies. In a study [7], energy consumption has been investigated for a cement plant in India. A Rankine cycle was used for waste heat recovery, and the results obtained showed that the proposed system could generate 4.4 MW electricity (about 30% of the total electricity required by the plant). Also the energy efficiency of the plant improved by 10%, and a two-year payback period of capital was obtained.

Amiri and Vaseghi [19] have compared the performance of the Rankine steam cycle, organic Rankine, Kalina, and carbon dioxide supercritical cycle for recovering the waste heat. Abdollahpour *et al*. [20] have investigated theoretically the thermodynamic and economic analysis of a solar-assisted transcritical CO_2 power cycle with LNG sub-system. The exergoeconomic results indicated that the power production of the solar-assisted system was about 8.53% and that solar collector, evaporator, condenser, CO_2 turbine, and LNG turbine 20 had the highest total cost rate of exergy destruction. Karellas *et al*. [9] have compared the energy and exergy of wasted heat recovery in the Rankine cycle to the organic cycle at a cement plant. An economic analysis was performed to obtain the payback period time in a cement plant. The results obtained showed that at a temperature above 310 ºC, the Rankine cycle had a better performance than the organic cycle. At this condition, the efficiency of the Rankine cycle was 23.58 and that of the organic cycle was 17.56. Also when cooling air was used for preheating the consumed water, the efficiency of the Rankine cycle increased to 24.58%. The economic results showed that the payback period to the capital was 5 years. Chen *et al*. [21] have compared the performance of the carbon dioxide supercritical cycle with an organic Rankine cycle (ORC) with R123 operating fluid for using low-temperature waste heat. They found that the cycle output power in the CO_2 supercritical cycle was a little more than an ORC. Liu *et al*. [22] have studied the effects of different fluids in the ORC on the thermal efficiency and the efficiency of the heat recovery system. Hang [23] has studied different working fluids such as benzene, toluene, p-xylene, R113, and R123 for ORC to recover waste heat. Among the studied fluids, p-xylene had the most efficiency and benzene had the least. It was also shown that the reversibility was related to the type of thermal source. P-xylene had the least reversibility in heat recovery with high temperatures, while R113 and R123 had a better performance in heat recovery with low temperatures. Legmann [12] has used ORC for heat recovery that was obtained from the clinker cooler and produced electricity continuously during cement production. Ahmadi *et al*. [24], have performed an exergetic analysis of an integrated power system consisting of a solid oxide fuel cell (SOFC) with a gas turbine in an ORC. According to the efficiency and exergy destruction investigation, the highest exergy degradation was in SOFC, while its exergy efficiency was 75.7%. In addition, the exergy efficiency of ORC and the total power plant were 64.9 and 39.9%, respectively. In another paper, Ahmadi *et al*. [25] have coupled a parabolic collector with a regular Rankine cycle to increase the system's power output efficiency by supplying the required heat. Also a thermal storage system was added to the cycle to be used at night. The energy, exergy, and economic analyzes of the hybrid system showed that the net generating capacity of the power plant increased by 8.14%. Also the exergy analysis showed that the boiler had the highest rate of exergy degradation. In general, the exergy efficiency of the system was reduced due to high collector losses [26].

Since cement plants are one of the largest industries in energy consumption and emission of pollutants, focusing on this area is essential to reduce energy consumption. A proper scenario for achieving the mentioned goal is to use the thermal recovery system and produce power from exhausted gases from stacks and hot air from the cooler in cement plants. This reduces the dependency on the national grid and increases the profitability of the plant. In this work, two ORCs were used to generate power. Since the flue gases and hot were different temperatures and compositions, the thermal recycling proposed in this work used these two heat sources separately. Organic fluids such as isobutene (R600a) and butane (R600) have more molecular masses, less boiling points, and more steam pressure compared with water. However, it was considered that these fluids had high temperatures after expanding in turbines, and it was proper to use this potential for heat transfer in the cycle to maximize the output of both turbines. For this purpose, two components were added to these ORCs, one was the heat exchanger that connected two ORCs (which were designed based on preheating gases and hot air separately) and the other one was a regenerator for the cycle (numbered 2) with R600 working fluid being used.

2. System Description

In this work, a thermodynamic cycle was presented for reproducing power. Isobutan (R600a) and butane (R600) are low environmental impact refrigerants in a small refrigerator system. Excellent thermodynamic performance, non-toxic properties, zero ozone depletion potential, and low global warming potential of butane and isobutane are the main reasons of their application in ORC systems. Therefore, in this proposed heat recovery system, butane and isobutane were used as a refrigerant and working fluid. Thus based on figure 5, the hot exhausted gases from preheaters with approximate temperature of 340 $^{\circ}$C at point 1 were injected into the first power generator cycle evaporator that had R600a as the working fluid, and after heat transfer with isobutene at point 2 would be ejected from the evaporator. This heat transfer led R600a to reach the necessary temperature to enter the turbine, and after expanding in turbine and producing power before entering the condenser had a heat transfer with the working fluid of the asymptotic cycle, and then entered the condenser that had a heat transfer with the cold water of the condenser, and finally, got out of the condenser as a saturated liquid, and then entered the pump where its pressure increased as much as the pressure of the evaporator. The working fluid of the asymptotic cycle was R600, and after receiving heat, the temperature of the working fluid changed from 89.45 $^{\circ}$C at point 12 to 113.3 $^{\circ}$C at point 13, and then entered the evaporator that had heat transfer with the hot air of the cooler. By this heat transfer, the temperature of R600 reached 200 $^{\circ}$C at point 14, meanwhile the hot water of cooler reached 180 $^{\circ}$C at point 9 from 270 $^{\circ}$C at point 8. R600 after expanding in the turbine and creating a mechanical work before entering the condenser, lost the heat to the exiting fluid and then entered the condenser and the pressure increased as much as the pressure of the evaporator by the pump. Since the expanded working fluid had a high temperature, it could give some of the temperatures to the fluid before the evaporator. For this reason, the regenerator and heat exchanger were used to connect the two ORCs. The thermodynamic properties of different parts of the cycle are given in table 1.

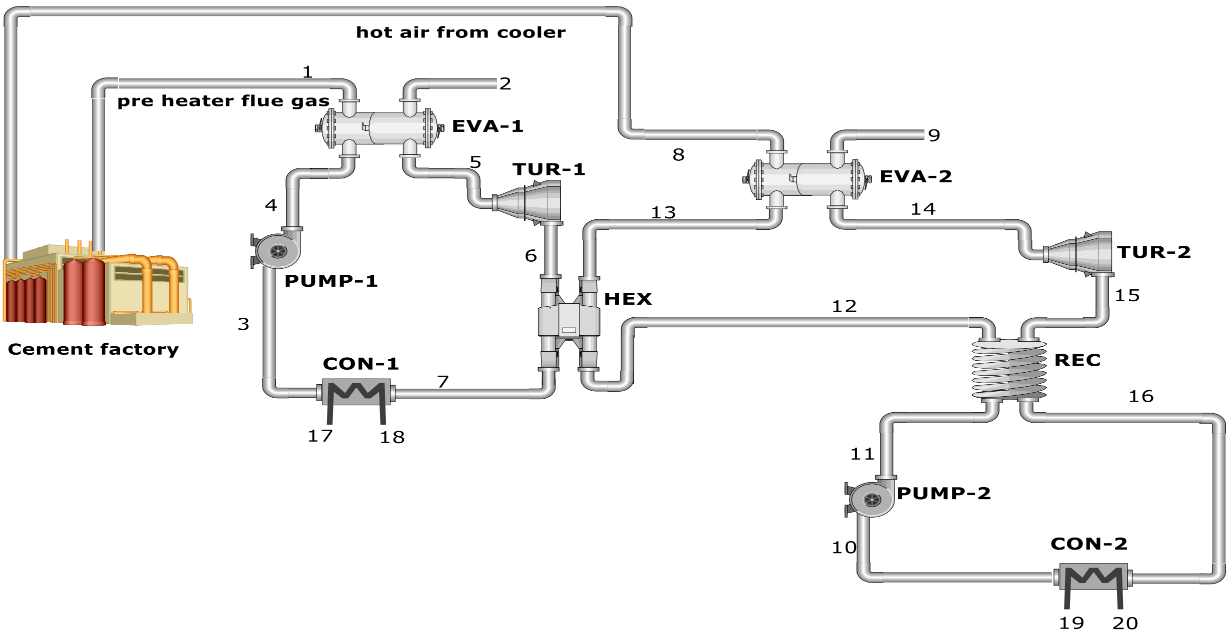

Figure ۵. Schematic representation of the thermal recycling cycle.

Table 1. Thermodynamic characters of the thermal recycling cycle.

state	Working fluid	T (℃)	P (kPa)	$\dot{m}\ (\frac{kg}{s})$	H (kJ/kg)	S (kJ/kg.K)	$\dot{E}\ (kW)$
0	------	25	101	---			
1	Pre-heater flue gas	340	101	87.6	-3222	6.931	15919
2	Pre-heater flue gas	180	101	87.6	-3393	6.607	9350
3	R600a	30.48	410	16.32	273	1.252	822.6
4	R600a	32.28	3280	16.32	278.9	1.254	908.8
5	R600a	300	3280	16.32	1200	3.434	5327
6	R600a	245.8	410	16.32	1077	3.494	3042
7	R600a	140.1	410	16.32	820.1	2.988	1303
8	Hot air	270	101	71	548.1	7.471	4808
9	Hot air	180	101	71	455.4	7.285	2175
10	R600	25	243.7	20.68	259.4	1.207	738.5
11	R600	25.62	1236	20.68	261.4	1.208	774.4
12	R600	89.45	1236	20.68	455.2	1.788	1209
13	R600	89.45	1236	20.68	658.3	2.348	1955
14	R600	200	1236	20.68	976.5	3.13	3711
15	R600	160.3	243.7	20.68	896.9	3.177	1780
16	R600	69.29	243.7	20.68	703.2	2.676	856.1
17	Water	25	101	426.9	104.8	0.3669	0
18	Water	30	101	426.9	125.8	0.4365	74.04
19	Water	20	101	438.8	104.8	0.3669	0
20	Water	30	101	438.8	125.8	0.4365	76.11

3. Modeling and Analysis

3.1. Thermodynamic analysis

The thermodynamic analysis involves mass conservation, energy conservation, and exergy balance; by applying these rules for each component of the cycle as a control volume, the amount of work and heat exchanged with the environment could be achieved.

The assumptions that were considered in this work were as follow [25]:

- The system works in steady-state.
- The outlet of the condensers is in the form of saturated liquid with the same pressure of the condenser.
- The pressure drop inside the heat exchangers and pipes is ignored.
- The isentropic efficiency of the turbine, pump, and heat exchangers is 0.8, 0.9, and 0.7.
- The changes in the kinetic energy and potential are ignored.

The general form of the mass conservation law (for an open system) is as equation (1).

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

In this equation, $\dot{m}_{in}$ indicates the input mass flow rate of the system and $\dot{m}_{out}$ indicates the output mass flow rate.

The general form of the first law of thermodynamics or the conservation of energy regardless of the kinetic energy and potential (for the open system) is as equation (2).

$$\dot{Q} - \dot{W} = \sum \dot{m}h_{out} - \sum \dot{m}h_{in} \quad (2)$$

In this equality, $\dot{W}$ and $\dot{Q}$, respectively, denote the rates of work and exchanged heat, and h is the enthalpy of flow.

In table 2, the equations derived from the first law of thermodynamics (the principle of conservation of energy) are represented for the components of the thermal recovery cycle.

Table 2. Energy balance of each component of system.

Components	First law of thermodynamics
Evaporator (1)	$\dot{m}_1h_1 + \dot{m}_4h_4 = \dot{m}_2h_2 + \dot{m}_5h_5$
Turbine (1)	$\dot{W}_{Tur(1)} = \dot{m}_5h_5 - \dot{m}_6h_6$
Heat exchanger	$\dot{m}_6h_6 + \dot{m}_{12}h_{12} = \dot{m}_7h_7 + \dot{m}_{13}h_{13}$
Condenser (1)	$\dot{m}_7h_7 + \dot{m}_{17}h_{17} = \dot{m}_3h_3 + \dot{m}_{18}h_{18}$
Pump (1)	$\dot{W}_{PUMP(1)} = \dot{m}_4h_4 - \dot{m}_3h_3$
Evaporator (2)	$\dot{m}_{13}h_{13} + \dot{m}_8h_8 = \dot{m}_{14}h_{14} + \dot{m}_9h_9$
Regenerator	$\dot{m}_{15}h_{15} + \dot{m}_{11}h_{11} = \dot{m}_{12}h_{12} + \dot{m}_{12}h_{12}$
Condenser (2)	$\dot{m}_{19}h_{19} + \dot{m}_{16}h_{16} = \dot{m}_{10}h_{10} + \dot{m}_{20}h_{20}$
Pump (2)	$\dot{W}_{PUMP(2)} = \dot{m}_{10}h_{10} - \dot{m}_{11}h_{11}$

3.2. Exergy analysis

Exergy analysis is a powerful tool for investigating the energy conversion systems in order to determine the original sources of irreversibility and highly destructive components in the system. The exergy of a matter is often divided into four sections: physical, chemical, kinetic, and potential. Due to the low velocity and low altitude changes, the kinetic and potential exergy are often neglected. The exergy equilibrium obtained by the first and second laws of thermodynamics is in the form of relation (3):

$$\dot{E}_Q - \dot{E}_W = \sum \dot{m}_{out}e_{out} - \sum \dot{m}_{in}e_{in} - \dot{E}_D \quad (3)$$

In this equation, $\dot{E}_Q$, $\dot{E}_{W,}$ and $\dot{E}_D$, respectively, show the exergy rates related to heat transfer, work, and destruction. The special exergy of flow is shown by e that contains both the physical and chemical parts.

$$e = e^{ph} + e^{ch} \quad (4)$$

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

$$e_i^{ch} = \sum_i^n x_i e_{0,i}^{ch} + RT_0 \sum_i^n x_i \ln(x_i) \quad (6)$$

$$\dot{E}_i = \dot{m}_i e_i \quad (7)$$

Table (3) summarizes the exergy destruction rates of each component of the system:

Table 3. Exergy destruction of each component of system.

Components	Exergy destruction
Evaporator (1)	$\dot{E}_{D,EVA(1)} = \dot{E}_4 + \dot{E}_1 - \dot{E}_2 - \dot{E}_5$
Turbine (1)	$\dot{E}_{D,TUR(1)} = \dot{E}_5 - \dot{E}_6 - \dot{W}_{Tur(1)}$
Heat exchanger	$\dot{E}_{D,HEX} = \dot{E}_6 + \dot{E}_{12} - \dot{E}_{13} - \dot{E}_7$
Condenser (1)	$\dot{E}_{D,CON(1)} = \dot{E}_7 + \dot{E}_{17} - \dot{E}_{18} - \dot{E}_3$
Pumps (1)	$\dot{E}_{D,PUMP(1)} = \dot{E}_3 - \dot{E}_4 - \dot{W}_{PUMP(1)}$
Evaporator (1)	$\dot{E}_{D,EVA(2)} = \dot{E}_{13} + \dot{E}_8 - \dot{E}_{14} - \dot{E}_9$
Regenerator	$\dot{E}_{D,RECUPERATOR} = \dot{E}_{15} + \dot{E}_{11} - \dot{E}_{12} - \dot{E}_{16}$
Condenser (2)	$\dot{E}_{D,CON(2)} = \dot{E}_{19} + \dot{E}_{16} - \dot{E}_{10} - \dot{E}_{20}$
Pumps (2)	$\dot{E}_{D,PUMP(2)} = \dot{E}_{10} - \dot{E}_{11} - \dot{W}_{PUMP(2)}$
Turbine (2)	$\dot{E}_{D,TUR(1)} = \dot{E}_{14} - \dot{E}_{15} - \dot{W}_{Tur(2)}$

To define the exergy efficiency for each one of the components, first, it is necessary to describe the concepts of fuel exergy and production exergy. In fact, fuel exergy is the consumed exergy by a component of the system that produces the desired and product exergy. Exergy efficiency for each component of the system is defined as the product exergy divided by the fuel exergy based on equation (8).

$$\varepsilon = \frac{\dot{E}_P}{\dot{E}_F} \quad (8)$$

where E_P^0 and E_F^0 are as the product exergy and the fuel exergy, respectively. Table (4) shows some expressions for fuel and production exergy for each component of the system.

Table 4. Fuel and production exergy for each component of system.

Components	Fuel	product
Turbine (1)	$\dot{E}_5 - \dot{E}_6$	$\dot{W}_{Tur(1)}$
Heat exchanger	$\dot{E}_6 - \dot{E}_7$	$\dot{E}_{13} - \dot{E}_{12}$
Condenser (1)	$\dot{E}_7 - \dot{E}_3$	$\dot{E}_{18} - \dot{E}_{17}$
Pumps (1)	$\dot{W}_{PUMP(1)}$	$\dot{E}_4 - \dot{E}_3$
Evaporator (1)	$\dot{E}_1 - \dot{E}_2$	$\dot{E}_5 - \dot{E}_4$
Turbine (2)	$\dot{E}_{14} - \dot{E}_{15}$	$\dot{W}_{Tur(2)}$
Regenerator	$\dot{E}_{15} - \dot{E}_{16}$	$\dot{E}_{12} - \dot{E}_{11}$
Condenser (2)	$\dot{E}_{16} - \dot{E}_{10}$	$\dot{E}_{20} - \dot{E}_{19}$
Pumps (2)	$\dot{W}_{PUMP(2)}$	$\dot{E}_{11} - \dot{E}_{10}$
Evaporator (2)	$\dot{E}_8 - \dot{E}_9$	$\dot{E}_{14} - \dot{E}_{13}$

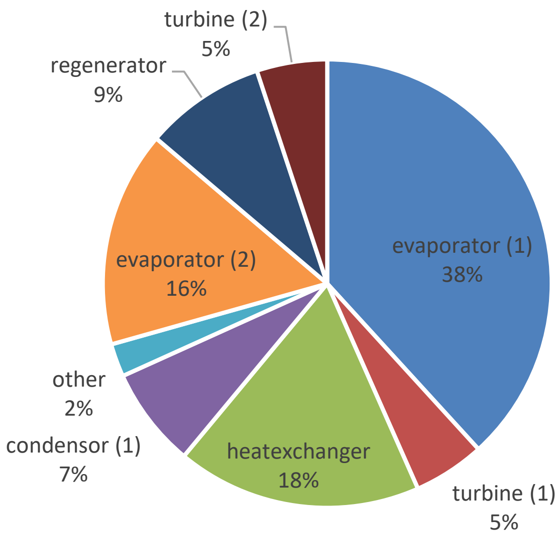

Figure 6. Exergy destruction of each component of system.

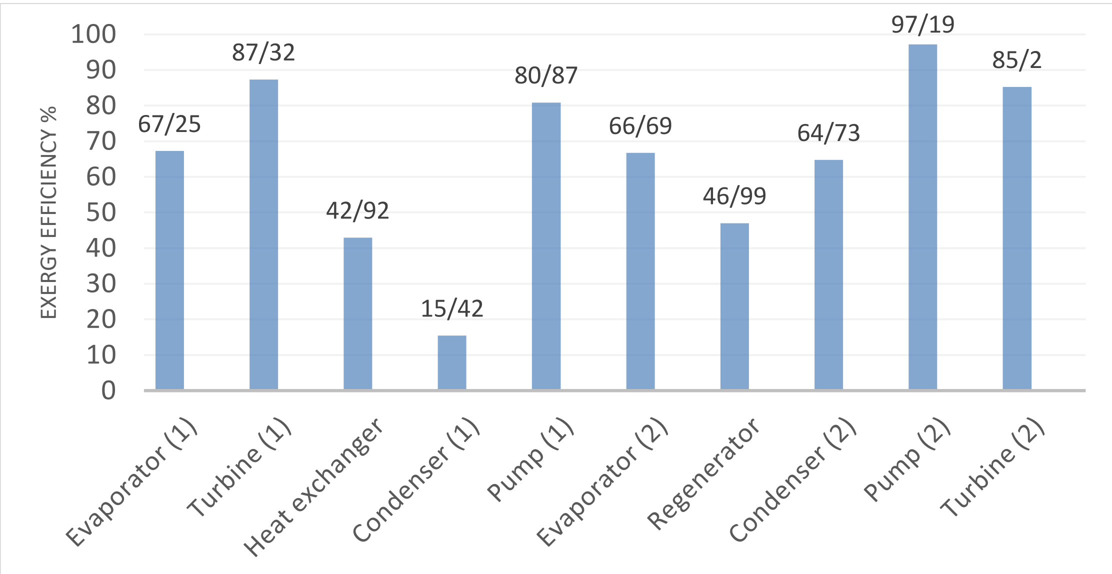

Figure 7. Exergy efficiency of each component of system.

The exergy efficiency and net work rate for the whole system are given by equations (9) and (10), respectively.

$$\psi = \frac{\dot{W}_{net}}{\dot{E}_1 + \dot{E}_8} \tag{9}$$

$$\dot{W}_{net} = \dot{W}_{TUR1} + \dot{W}_{TUR2} - \dot{W}_{PUMP1} - \dot{W}_{PUMP2} \tag{10}$$

4. Parametric Study

A comprehensive parametric study was performed on the basic parameters of the cycle in order to show their effects on the net output power and exergy efficiency. The basic parameters involved include temperatures and inlet pressures to the turbines. Figures (8) and (9) illustrate the influence of the inlet pressure of the first and second turbines on the turbine-generated work and net outlet work of the cycle.

According to figures (8) and (9), by increasing the inlet pressure of the turbines at the constant temperatures of 300 and 200 ºC, respectively, for the first and second turbines, the net work is increased.

In fact, for each turbine, as the inlet pressure of the turbine increases, the average temperature at which the heat is transferred to the working fluid is increased, and this increases the turbine production work. Figures (10) and (11) show the effect of the inlet temperature of the turbines at constant pressure of 3280 kPa for the first turbine and 1236 kPa for the second turbine.

According to these diagrams, by increasing the inlet temperature of the turbine, or in other words, the degree of superheat, the net work will be increased due to increase in the average temperature at which heat is transferred to the fluid.

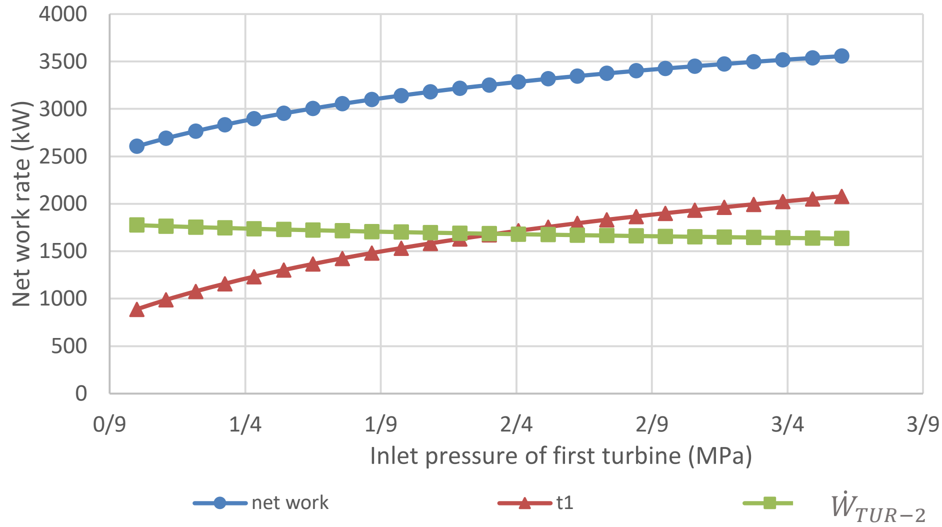

Figure 8. Influence of inlet pressure of first turbine on net work rate.

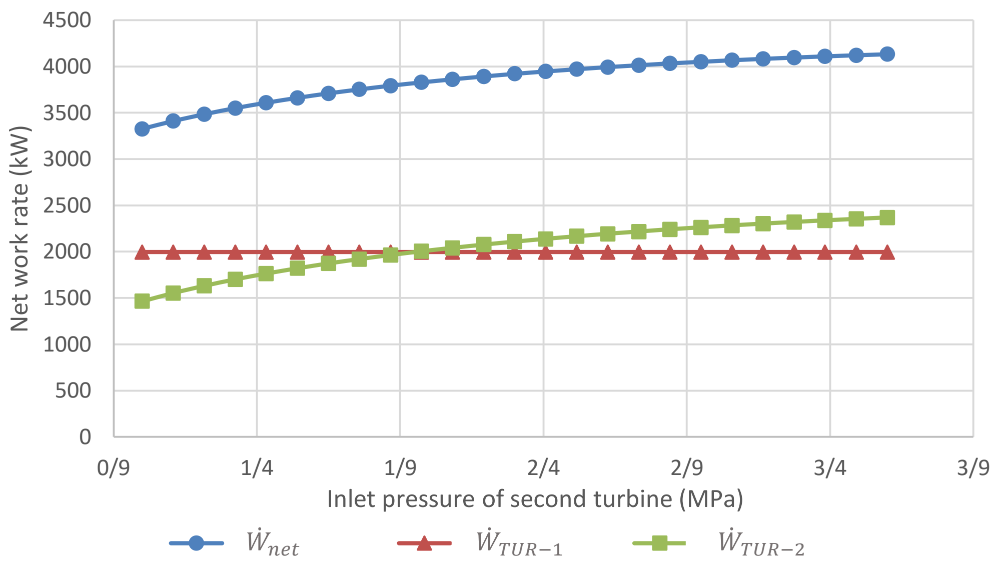

Figure 9. Influence of inlet pressure of second turbine on net work rate.

Figures (12) and (13) consider the impact of the inlet pressure on the turbines at different inlet temperatures. These diagrams assume that one of the turbines operate at a constant temperature and pressure, and the effect of changing the temperature and pressure of the other turbine on the net output work can be observed.

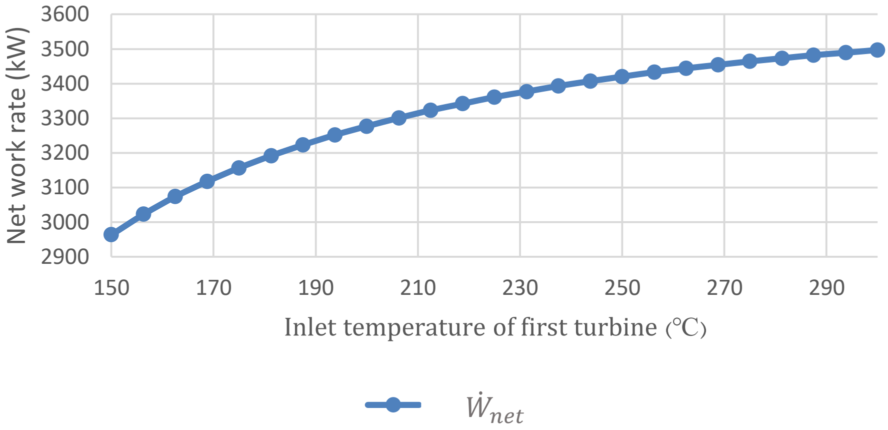

Figure 10. Influence of inlet temperature of first turbine on net work rate.

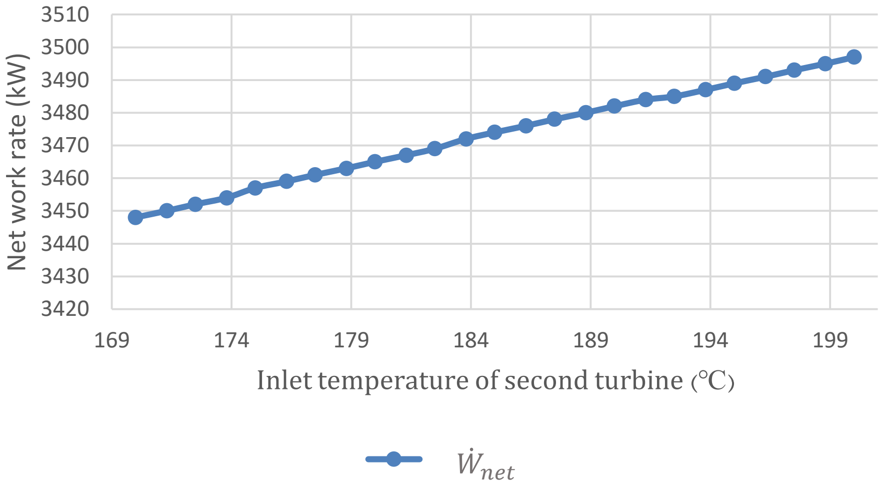

Figure 11. Influence of inlet temperature of second turbine on net work rate.

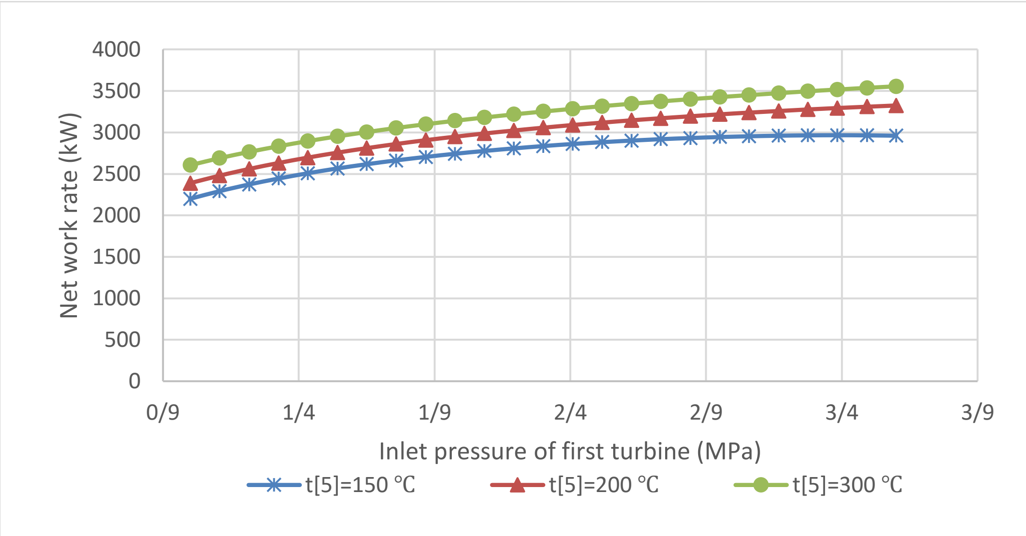

Figure 12. Influence of inlet pressure of first turbine on net work rate at different temperatures.

According to figure (12), when the first turbine operated at 300 ℃ and 3280 kPa, the net work increased dramatically by increasing the temperature and pressure of the second turbine. According to figures (12) and (13), as the pressure and inlet temperature for each turbine increase, the net output work is increased.

With increase in temperature and pressure, the average temperature at which heat transfers through the operating fluid and a hot source increases, and due to that, the efficiency of the cycle and net output work is increased. Figures (14) and (15) show the effect of the inlet temperatures of the first and second turbines on the exergy efficiency of the whole system. These graphs assume that each time the temperature and pressure of one of the turbines are constant (for the first turbine 300 ℃ and 3280 kPa, and the second turbine 200 ℃ and 1236 kPa) and the pressure and temperature values for the second turbine change. In figures (14) and (15), with increase in temperature and pressure, the net work increases, and according to equation (10), the increase in the net work is associated with an increase in the total exergy efficiency.

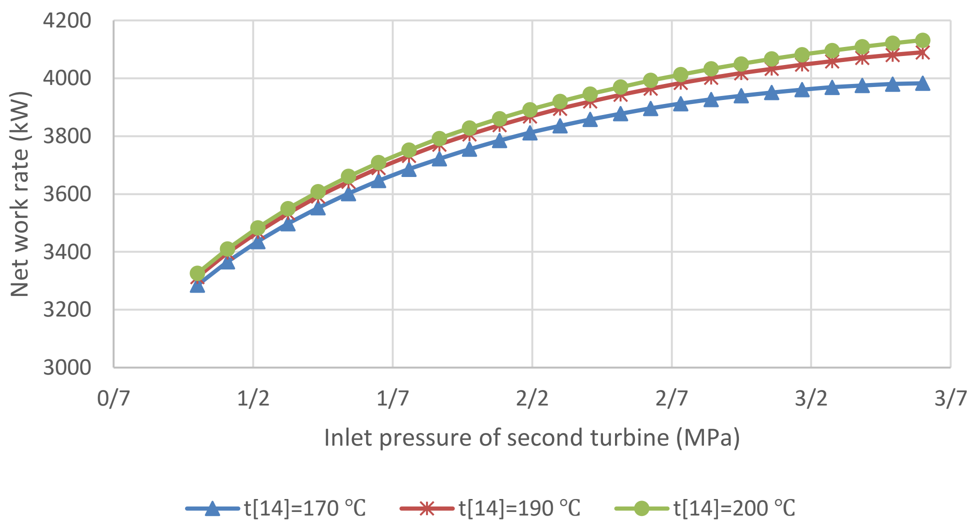

Figure 13. Influence of inlet pressure of second turbine on net work rate at different temperatures.

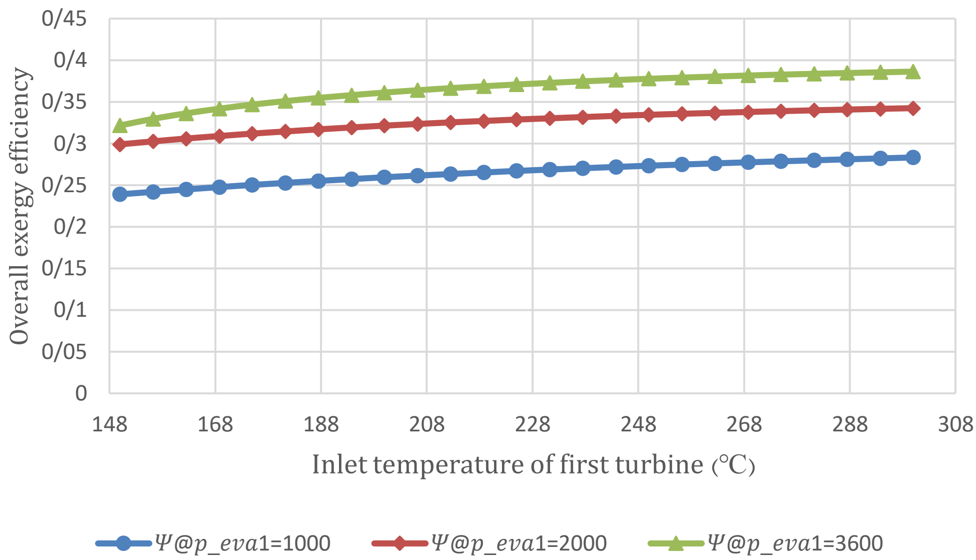

Figure 14. Influence of inlet temperature of first turbine on the exergy efficiency of the whole cycle.

According to the diagrams presented, as the temperature and pressure of the turbines increase, the work will increase. If the first and second turbines operate at a pressure of 3600 kPa, the first turbine operates at an inlet temperature of 300 ℃ and the second turbine at 200 ℃, the net operating rate of the system increases from 3497 kW to 4186 kW.

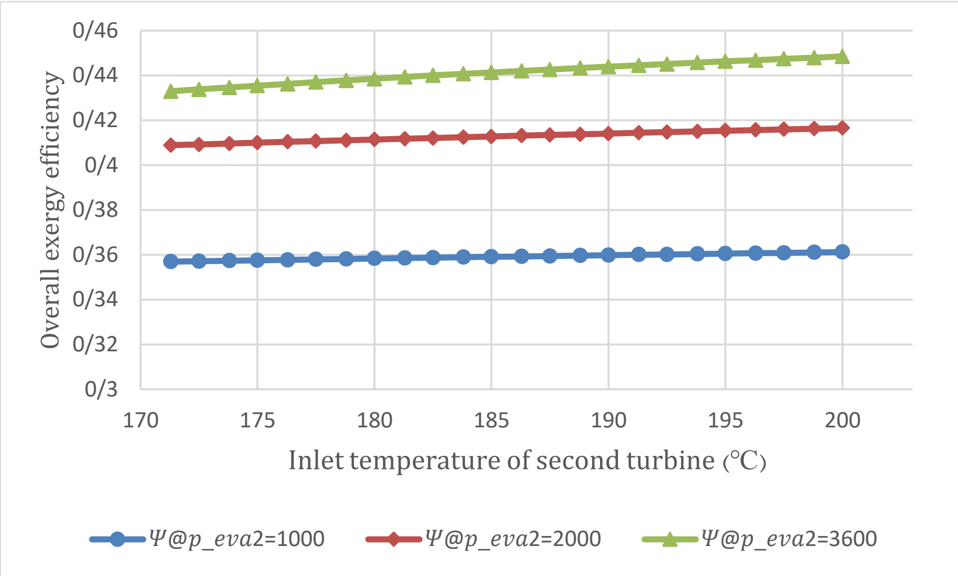

Figure 15. Influence of inlet temperature of second turbine on the exergy efficiency of the whole cycle.

5. Conclusion

Since cement plants are the major energy consumers in both the thermal and electric powers, implementing a thermal recycling system in these plants, in addition to generating power and reducing dependence on the national grid, would reduce greenhouse gas emissions. According to the parametric study performed on the proposed cycle, the following results were obtained:

- When the inlet temperature and pressure of the turbine were closer to the critical point of the expanded operating fluid in the turbine (the pressure of 3600 kPa was considered as the highest allowable pressure for both fluids and also their temperature was obtained by the temperature of the preheating gases and the hot air of cooler and the efficiency of the evaporators), the net output work rate of the cycle, and consequently, the exergy efficiency of the whole system increased.
- When the inlet pressure of the turbines was increased from the value listed in table 1 to 3600 kPa, the net work of the cycle reached from 3497 kW to 4186 kW and the exergy efficiency reached from 38% to 45.94%.

Nomenclature

e	Special exergy (kJ/kg)
$\dot{E}$	Exergy flow rate (kW)
$\dot{E}_D$	Exergy rate of destruction
$\dot{E}_Q$	Exergy associated with heat
$\dot{E}_W$	Exergy associated with work
h	Specific enthalpy (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (kPa)
$\dot{Q}$	Heat transferred (kW)
R	Gas constant (kJ/kmol K)
s	Specific entropy (kJ/kg K)
T	Temperature (°C)
$\dot{W}$	Work rate (kW)

Greek symbols

ψ	Exergy efficiency for whole system
ε	Exergy efficiency for each component

Subscripts

0	Reference environment condition
ch	Chemical
CON	Condenser
EVA	Evaporator
F	Fuel
HEX	Heat exchanger
in	Inlet stream
out	Outlet stream
P	Product
ph	Physical
TUR	Turbine

References

[1] Mohammadi, Amin, Muhammad Ali Ashjari, and Amirhassan Sadreddini. "Exergy analysis and optimisation of waste heat recovery systems for cement plants."International Journal of Sustainable Energy 37, No. 2 (2018): 115-133.

[2] Ramezanizadeh, Mahdi, Mohammad Alhuyi Nazari, Mohammad Hossein Ahmadi, and Lingen Chen. "A review on the approaches applied for cooling fuel cells." International Journal of Heat and Mass Transfer 139 (2019): 517-525.

[3] Ahmadi, Mohammad Hossein, Mahdi Ramezanizadeh, Mohammad Alhuyi Nazari, Giulio Lorenzini, Ravinder Kumar, and Ravindra Jilte. "Applications of nanofluids in geothermal: A

review." Mathematical Modelling of Engineering Problems 5, No. 4 (2018): 281-285.

[4] I. E. A. Peter Levi, “IEA Technology Roadmap, The global iron and steel sector,” (2019).

[5] Hasanbeigi, Ali, Lynn Price, Hongyou Lu, and Wang Lan. "Analysis of energy-efficiency opportunities for the cement industry in Shandong Province, China: A case study of 16 cement plants." Energy 35, No. 8 (2010): 3461-3473.

[6] Söğüt, Ziya, Zuhal Oktay, and Hikmet Karakoç. "Mathematical modeling of heat recovery from a rotary kiln." Applied thermal engineering 30, No. 8-9 (2010): 817-825.

[7] Khurana, Shaleen, Rangan Banerjee, and Uday Gaitonde. "Energy balance and cogeneration for a cement plant." Applied Thermal Engineering 22, No. 5 (2002): 485-494.

[8] Worrell, Ernst, Lynn Price, Nathan Martin, Chris Hendriks, and Leticia Ozawa Meida. "Carbon dioxide emissions from the global cement industry." Annual review of energy and the environment 26, No. 1 (2001): 303-329.

[9] Karellas, S., A-D. Leontaritis, G. Panousis, E. Bellos, and E. Kakaras. "Energetic and exergetic analysis of waste heat recovery systems in the cement industry." Energy 58 (2013): 147-156.

[10] Naqi, Ali, and Jeong Gook Jang. "Recent progress in green cement technology utilizing low-carbon emission fuels and raw materials: A review." Sustainability 11, No. 2 (2019): 537.

[11] Madlool, Naseer Abboodi, Rahman Saidur, M. Shouquat Hossain, and Nasrudin Abd Rahim. "A critical review on energy use and savings in the cement industries." Renewable and Sustainable Energy Reviews 15, No. 4 (2011): 2042-2060.

[12] Legmann, Hilel. "Recovery of industrial heat in the cement industry by means of the ORC process." In IEEE-IAS/PCS 2002 Cement Industry Technical Conference. Conference Record (Cat. No. 02CH37282), pp. 29-35. IEEE, 2002.

[13] Saneipoor, P., G. F. Naterer, and I. Dincer. "Heat recovery from a cement plant with a Marnoch Heat Engine." Applied Thermal Engineering 31, No. 10 (2011): 1734-1743.

[14] Ahamed, Jamal Uddin, N. A. Madlool, Rahman Saidur, M. I. Shahinuddin, A. Kamyar, and Haji Hassan Masjuki. "Assessment of energy and exergy efficiencies of a grate clinker cooling system through the optimization of its operational parameters." Energy 46, No. 1 (2012): 664-674.

[15] Kaşka, Önder. "Energy and exergy analysis of an organic Rankine for power generation from waste heat recovery in steel industry." Energy Conversion and Management 77 (2014): 108-117.

[16] Larsen, Ulrik, Leonardo Pierobon, Fredrik Haglind, and Cecilia Gabrielii. "Design and optimisation of organic Rankine cycles for waste heat recovery in marine applications using the principles of natural selection." Energy 55 (2013): 803-812.

[17] Wang, Jiangfeng, Yiping Dai, and Lin Gao. "Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry." Applied Energy 86, No. 6 (2009): 941-948.

[18] Zeeshan, Muhammad, and Muhammad Naeem Arbab. "Waste heat recovery and its utilization for electric power generation in cement industry." International Journal of Engineering and Technology 15 (2015).

[19] Amiri, Ali, and Mohammad Rahim Vaseghi. "Waste heat recovery power generation systems for cement production process." IEEE Transactions on Industry Applications 51, No. 1 (2014): 13-19.

[20] Abdollahpour Amir, Ghasempour Roghayeh, Kasaeian Alibakhsh, H. Ahmadi Mohammad , " Exergoeconomic analysis and optimization of a transcritical CO2 power cycle driven by solar energy based on nanofluid with liquefied natural gas as its heat sink" Journal of Thermal Analysis and Calorimetry 139, No. 1 (2020) : 451–473.

[21] Chen, Yang, P. Lundqvist, Anders Johansson, and P. Platell. "A comparative study of the carbon dioxide transcritical power cycle compared with an organic Rankine cycle with R123 as working fluid in waste heat recovery." Applied thermal engineering 26, No. 17-18 (2006): 2142-2147.

[22] Liu, Bo-Tau, Kuo-Hsiang Chien, and Chi-Chuan Wang. "Effect of working fluids on organic Rankine cycle for waste heat recovery." Energy 29, No. 8 (2004): 1207-1217.

[23] Hung, Tzu-Chen. "Waste heat recovery of organic Rankine cycle using dry fluids." Energy Conversion and management 42, No. 5 (2001): 539-553.

[24] Ahmadi, Mohammad, Mirhadi Sadaghiani, Fathollah Pourfayaz, Mahyar Ghazvini, Omid Mahian, Mehdi Mehrpooya, and Somchai Wongwises. "Energy and exergy analyses of a solid oxide fuel cell-gas turbine-organic Rankine cycle power plant with liquefied natural gas as heat sink." Entropy 20, No. 7 (2018): 484.

[25] Mohammadi, Amin, Mohammad Hossein Ahmadi, Mokhtar Bidi, Mahyar Ghazvini, and Tingzhen Ming. "Exergy and economic analyses of replacing feedwater heaters in a Rankine cycle with parabolic trough collectors." Energy Reports 4 (2018): 243-251.

[26] Wang, Huarong, Jinliang Xu, Xufei Yang, Zheng Miao, and Chao Yu. "Organic Rankine cycle saves energy and reduces gas emissions for cement production." Energy 86 (2015): 59-73.