Energy 58 (2013) 147–156

Contents lists available at SciVerse ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Energetic and exergetic analysis of waste heat recovery systems in the cement industry

S. Karellas*, A.-D. Leontaritis, G. Panousis, E. Bellos, E. Kakaras

National Technical University of Athens, Laboratory of Steam Boilers and Thermal Plants, Heroon Polytechniou 9, 15780 Zografou, Athens, Greece

ARTICLE INFO

Article history:
Received 27 August 2012
Received in revised form
27 February 2013
Accepted 30 March 2013
Available online 13 June 2013

Keywords:
Waste heat recovery
ORC
Water-steam
Exergy analysis
Cement industry

ABSTRACT

In a typical cement producing procedure, 25% of the total energy used is electricity and 75% is thermal energy. However, the process is characterized by significant heat losses mainly by the flue gases and the ambient air stream used for cooling down the clinker (about 35%–40% of the process heat loss). Approximately 26% of the heat input to the system is lost due to dust, clinker discharge, radiation and convection losses from the kiln and the preheaters. A heat recovery system could be used to increase the efficiency of the cement plant and thus contribute to emissions decrease. The aim of this paper is to examine and compare energetically and exergetically, two different WHR (waste heat recovery) methods: a water-steam Rankine cycle, and an Organic Rankine Cycle (ORC). A parametric study proved that the water steam technology is more efficient than ORC in exhaust gases temperature higher than 310 °C. Finally a brief economic assessment of the most efficient solution was implemented. WHR installations in cement industry can contribute significantly in the reduction of the electrical consumptions operating cost thus being a very attractive investment with a payback period up to 5 years.

1. Introduction

The cement industry is one of the major industrial emitters of greenhouse gases and particularly CO_2 [1]. Cement production is an energy-intensive process and each tone of portland cement produced releases approximately 1 tone of CO_2 [2]. The major part of the CO_2 emissions from the production of cement is released from the calcination of limestone (50%) and from the combustion of fuels (40%). In addition, the EU has made a commitment to increase the 20% emissions target to 30% for the post Kyoto period if there are comparable targets from other developed countries and adequate actions by developing countries [3]. This prospect is expected to impose a further burden to the EU industry and the cement industry in particular which, representing more than 10% of the world production [4] is quite vulnerable to the issue of carbon leakage [5].

The cement clinker production sector is a substantially energy intensive industry accounting for 50–60% of the production costs [6] and consuming the 80% of the energy used in cement production [7], while is currently contributing to about 5% of the global anthropogenic emissions [8]. Thermal energy demands depend on the age of the plant and on the specific process but range between 3000 and 6500 MJ/tone clinker. The average specific energy consumption is about 2.95 GJ per ton of cement produced for well-equipped advanced kilns, while in some countries the consumption exceeds 5 GJ/ton. The electric energy demand ranges from 90 to 150 kWh per cement tone [4] .The 65% of this is used for the grinding of coal, raw materials and clinker [9].

In a typical cement plant, 25% of the total energy used is electricity and 75% is thermal energy. However, the process is characterized by significant heat losses mainly due to the flue gases and the ambient air stream used for cooling down the clinker. About 35%–40% of the process heat is lost from those waste heat streams [10]. Approximately 26% of the heat input to the system is lost by dust, clinker discharge, radiation and convection from the kiln and pre-heaters [11–13]. A heat recovery system could be used to increase the efficiency of a cement plant and thus contribute to emissions decrease. Moreover, it would reduce the amount of waste heat to the environment and lower the temperature of the exhaust gases [14]. Waste heat can be captured from combustion exhaust gases, heated products, or heat losses from systems [15]. Otherwise, the waste heat can be utilized in order to preheat the raw material before the clinkering process [16].

Waste heat recovery systems are already in operation in various industries with success. In Canada, the Gold Creek Power Plant [17] has a heat recovery system that produces 6.5 MW power using ORC technology. In India, the A.P. Cement Works with 4 MW and ORC technology. Another cement industry that uses waste heat recovery

* Corresponding author. Tel.: +30 210 7722810; fax: +30 210 7723663.
E-mail address: sotokar@mail.ntua.gr (S. Karellas).

0360-5442/$ – see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.energy.2013.03.097

Nomenclature

E	total exergy rate of material stream, W
E^W	exergy value of power output, W
h	specific Enthalpy of stream, J mol^{-1}
h_0	standard specific enthalpy at environmental conditions, J mol^{-1}
IR	irreversibilities rate of process, W
$\dot{m}_{air}$	mass flow rate of cooling air, kg s^{-1}
N	mole flow rate, mol s^{-1}
P_{el}	power from generator, W
p	pressure, bar
p_0	standard pressure under environmental conditions, bar
$\dot{Q}_{fluid}$	heat rate transferred to working fluid, W
$\dot{Q}_{gas}$	heat rate from the flue gases of the rotary kiln, W
$\dot{Q}_{HS}$	heat source rate, W
s	specific entropy of stream, J mol^{-1} K^{-1}
s_0	standard specific entropy at environmental conditions, J mol^{-1} K^{-1}
T	temperature, K
T_0	standard temperature under environmental conditions, K
x_i	mole fraction of component *i*
WHR	waste heat recovery

Greek symbols

ε_{ph}	specific physical exergy of material stream, J mol^{-1}
η	efficiency
η_{HEx}	heat-exchangers efficiency
η_{system}	system efficiency
η_{th}	thermal efficiency
η_{ex}	exergetic efficiency
ζ	exergy efficiency of component

Subscripts and superscripts

ex	exergetic
in	input
out	output
i	stream component

is Heidelberger Zement AG Plant in Lengfurt (Germany) [17] with 1.5 MW power and ORC technology. In addition to these industries, a new waste heat recovery system is under construction in Rohrdorf (Germany) [18] with 6.8 MW power and water-steam cycle technology.

This study aims at the identification of a best practice example for energy recovery in an existing commercial cement production plant with waste heat utilization as a new component. Two different methods will be examined in order to find which is more beneficial and more efficient for a cement production plant. Firstly, a water-steam Rankine cycle will be analyzed. The basic characteristics of this cycle are the two drums with 19 bar pressure and a maximum cycle temperature of 350 °C. The other is an Organic Rankine Cycle (ORC) using an indirect cycle with pressurized water at 30 bar. Part of this study was the evaluation of several organic fluids. It was concluded that isopentane has the optimum performance. Thus, any further analysis was carried out considering isopentane as the organic working fluid of the ORC.

Many parameters were investigated in order to design the thermodynamic cycles, in terms of energetic and exergetic efficiency. Pressure and temperature are the most important parameters regarding the efficiency of those systems. The arrangement of the cycle and its different components were optimized in order to improve the efficiency and determine the design of an optimum system. Aiming to define the cycle with the best performance, an energy and exergy analysis was done in order to find the cycle with the highest thermal and exergetic efficiency.

2. Waste heat recovery

The identification of the waste heat sources in the cement industry is of high importance for the improvement of the process efficiency. This study examines an old cement industry in Greece which produces 6700tn/day clinker and the two main waste heat sources are [19]:

- The exhaust gases from the rotary kiln, which after passing through the raw material preheater, are at a temperature of about 380 °C

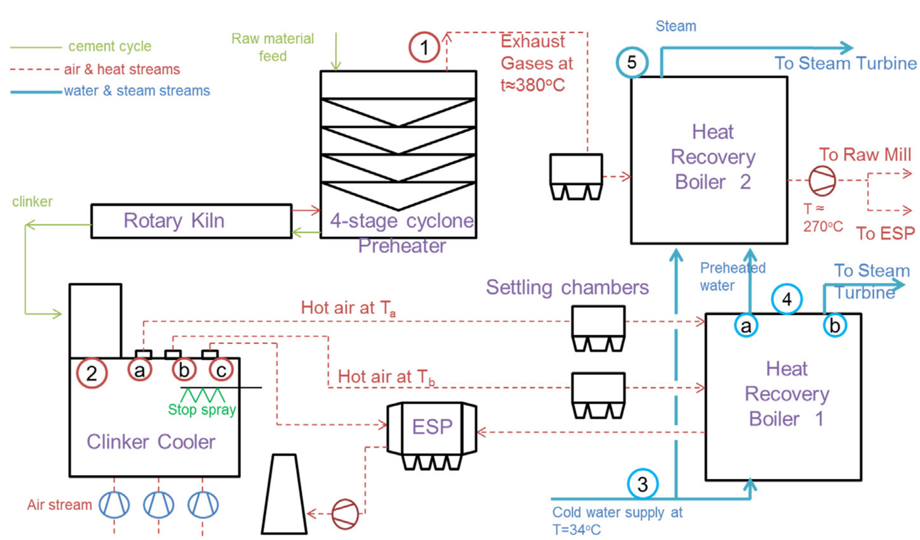

Fig. 1. Heat recovery system of a typical cement plant.

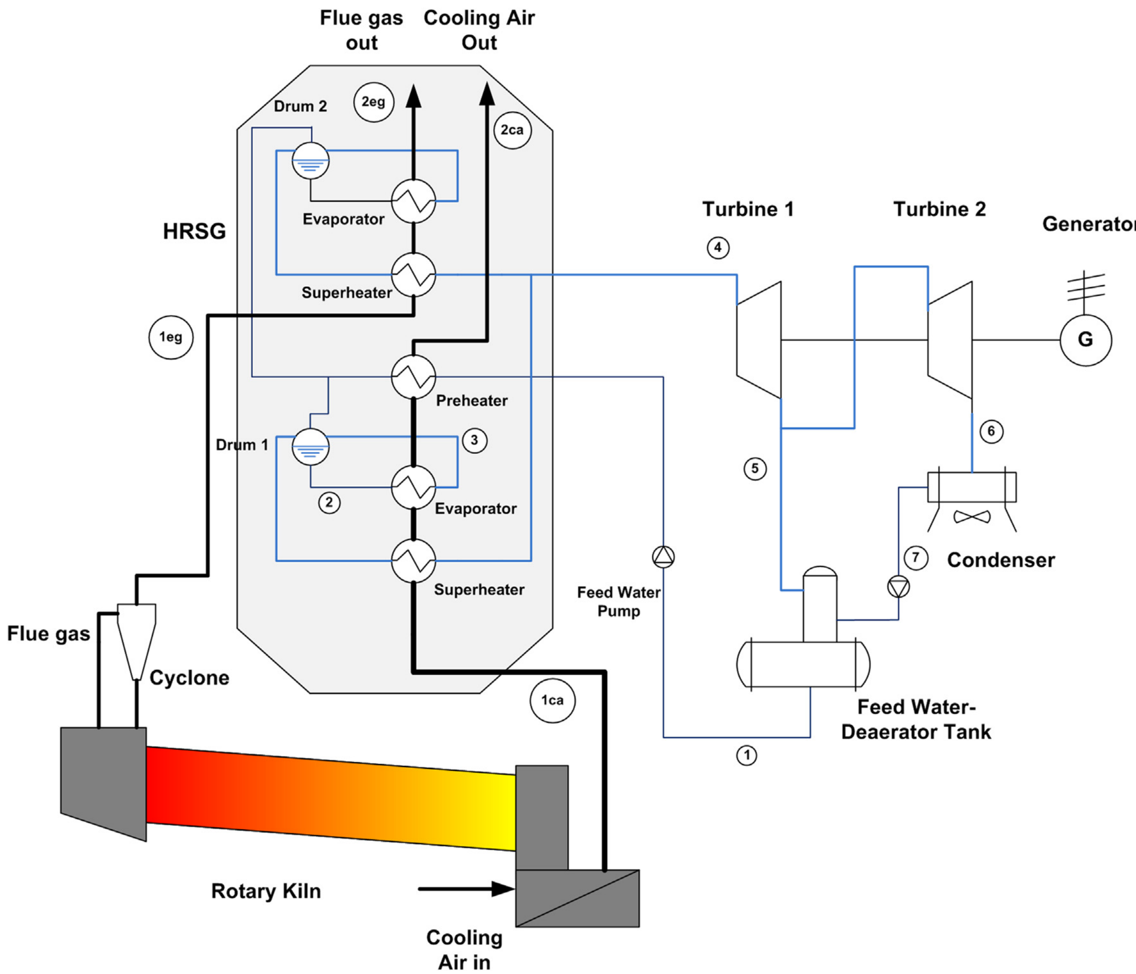

Fig. 2. Heat recovery system with water-steam cycle.

- The waste heat from the clinker cooler, in the form of hot air, at an average temperature of about 360 °C.

These waste heat sources can be efficiently used in a waste heat recovery system in order to produce electricity. Usually, a waste heat recovery boiler is used to produce steam which drives a steam turbine to generate electric power. The plant is then considered as a cogeneration plant, since two products (electricity and clinker) are provided through the same process.

The proposed heat recovery system is schematically shown in Fig. 1. As already discussed, there are two waste heat sources that can be used for the production of steam. The exhaust gases from the rotary kiln (point 1), after preheating and pre-calcinating the raw material, are available at a temperature of about 380 °C. This temperature depends on the number of the stages of the preheater. A 4 stage preheater provides exhaust gases at 300–380 °C, whilst a 5–6 stage preheater provides exhaust gases at 200–300 °C [20]. After passing through the settling chamber for the necessary dust removal, it enters the heat recovery boiler 2 and superheated steam is produced. That is a typical procedure in a cement plant heat recovery system. However, this is not the case with the second heat source. The hot air from the clinker cooler (point 2) is available at an average temperature of about 360 °C. During the cooling process of the clinker, the air can be taken from different points of the cooler and thus at different temperatures. For example the exit **2a** and 2**b** can be at a temperature of 500 °C and 300 °C respectively. This offers a number of advantages and can lead to higher system efficiency. The high temperature stream can be used for the superheating of the steam and then it can be mixed with the low temperature stream for the preheating and evaporation of the water. This means that a higher final temperature can be reached and a higher efficiency of the process can be achieved. Stream 2**a** can be mixed with stream **2b**. Exit **2c** is used for by-passing the heat exchanger when the heat recovery system is not in operation. The selection of the points that the hot air will be drawn from the cooler as well as the respective mass flows is of great importance for the design of the system. It is expected that the mass flow in exit **2b** will be much higher than in exit **2a**. It is noted that the hot air stream goes through an ESP (Electrostatic Precipitator) system before being released in the atmosphere, in order to remove the particles.

Concerning the exhaust gas source (point 1), for the current work, the exhaust gases with 96.71 kg/s mass flow rate and 380 °C temperature are investigated as a first heat source. This flow has to

Table 1
System characteristics.

System parameters	
Turbine isentropic efficiency (%)	85
Turbine mechanical efficiency (%)	99
Pump isentropic efficiency (%)	70
Generator electrical efficiency (%)	98
Generator mechanical efficiency (%)	98
Environmental temperature (°C)	25
Environmental pressure (bar)	1.013

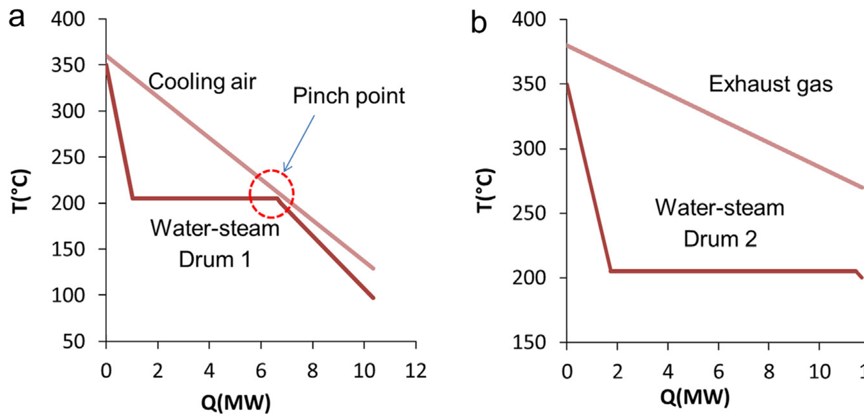

Fig. 3. Q–T diagram for the water-steam cycle. a) Drum 1, b) Drum 2.

leave the heat recovery system at a minimum temperature of 270 °C as it has to be reused in the raw material mill. The other heat source, the hot air from the clinker cooler (point 2), has an air mass flow rate of 42.91 kg/s at a temperature of 360 °C (Mixture of points 2a and 2b).

Those aspects will be thoroughly investigated with the help of thermodynamic models and simulations in order to assess the performance of the system. For analyzing this system, thermal efficiency is defined as:

$$\eta_{th} = \frac{P_{el}}{\dot{Q}_{fluid}}, \tag{1}$$

where P_{el} is the electric power produced by the generator and $\dot{Q}_{fluid}$ is the heat that the working fluid absorbs from the heat sources. The efficiency of the heat-exchange system is defined by the following equation:

$$\eta_{HEx} = \frac{\dot{Q}_{fluid}}{\dot{Q}_{HS}}, \tag{2}$$

where $\dot{Q}_{HS}$ is the heat source energy.

The heat source consists of the exhaust gas and the hot air and is calculated as the maximum energy that the heat source can provide to the working fluid. This is the sum of the available heat from the exhaust gas and the hot air assuming that both streams in the exit are at ambient temperature and pressure. However, this is not the case for the exhaust gas stream, as there is a 270 °C limit in the exit temperature in order to be reused in the raw material mill. Therefore, the exhaust gas heat is fully utilized taking into account the exit temperature requirements. So $\dot{Q}_{HS}$ is calculated as:

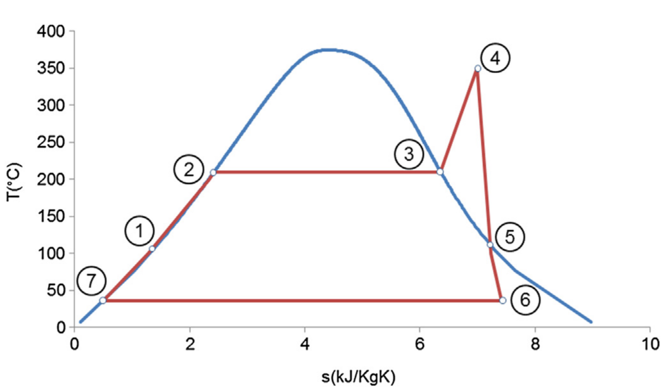

Fig. 4. Water-steam thermodynamic cycle.

$$\dot{Q}_{HS} = \dot{Q}_{gas} + \dot{m}_{air} \cdot (h_{in} - h_{ambient}), \tag{3}$$

Finally, the system efficiency can be calculated as:

$$\eta_{system} = \eta_{HEx} \cdot \eta_{th}, \tag{4}$$

3. Water-steam cycle

The first cycle that will be examined in this paper is a water-steam Rankine cycle in order to recover the waste heat from the cement industry process. The examined system and its specifications are presented in Fig. 2. The cycle's upper pressure and temperature is 19 bar and 350 °C respectively at the inlet of the turbine and a pressure of 0.06 bar at the exit of the turbine. The exhaust steam of the turbine is condensed in the condenser and then pumped to the deaerator tank. Simultaneously, some of the steam is extracted from the turbine at 1 bar in order to be used in the deaeration process. After that, the condensate goes through the feed pump and enters the air preheater where it is preheated to 200 °C. From that point, the feed water is separated into two streams. The first stream is evaporated and superheated utilizing the energy from the cooling air heat source. The other stream follows the same process utilizing the exhaust gas heat source. Each steam generator system consists of a drum and two heat exchangers. Finally, the two streams of superheated steam enter the steam turbine and the process is repeated. The main system characteristics of the water cycle are summarized in Table 1.

At this point, it is necessary to explain the parameters of the Rankine cycle, taking into account that the efficiency of the system increases with the increase of temperature and pressure at the inlet of the turbine. Firstly, the maximum temperature of the superheated steam is set at 350 °C, 10 K lower than the cooling air inlet temperature. The other important parameter, which has a great influence on the efficiency of the system, is the pressure at the inlet

Table 2
Water-steam cycle points.

Point	P (bar)	T (°C)	h (kJ/kg)
1	1.00	99.6	417.4
2	19.00	209.8	896.9
3	19.00	209.8	2797.3
4	19.00	350.0	3139.7
5	1.00	99.6	2626.1
6	0.06	36.2	2286.0
7	0.06	36.2	143.2

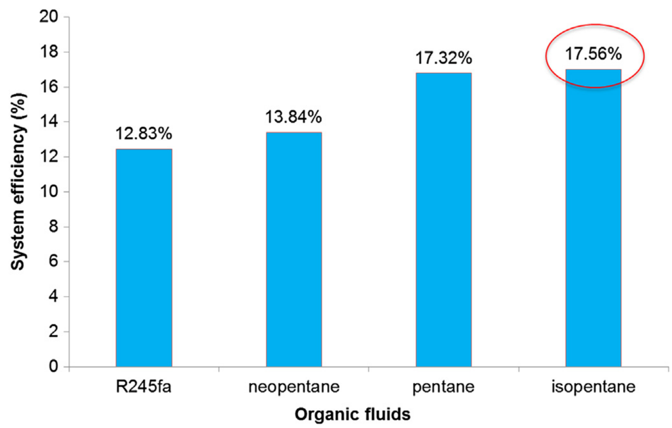

Fig. 5. Comparison of different organic fluids.

of the turbine which is set at 19 bar, taking into consideration the pinch point in the Q–T diagram (Fig. 3).

The intermediate pressure of the steam extraction from the turbine is set at 1 bar in order to be used for the deaeration and preheating process in feed water tank. The steam turbine vacuum is at about 0.06 bar. Finally it should be noted that the temperature after the air preheater is 200 °C, 10 K lower than the boiling point at 19 bar, which is 210 °C. The thermodynamic diagram T–S (Fig. 4) presents the complete cycle while its parameters are summarized in Table 2.

4. Organic Rankine cycle

Another way to recover the waste heat from a cement plant is an indirect ORC. The ORC is used in low-temperature energy sources, because of the low critical point of the organic fluids. In this paper, four different organic fluids were examined in order to choose the most appropriate working fluid regarding the thermodynamic performance for the given temperature limits. As it can be seen in Fig. 5, isopentane is the working fluid with the maximum system efficiency and thus it was selected as the working fluid for the ORC. Other parameters such as the price of the organic fluid and the energy consumption for its production were not taken into consideration.

In an ORC heat recovery system there is an intermediate heat transfer fluid in order to transfer the heat from the heat sources to the working fluid through heat exchangers. This is necessary for safety reasons, as many organic fluids are flammable and in case of failure of the heat exchanger the hot medium of the heat source and the organic fluid would get in contact resulting in an explosion. The heat transfer fluid should remain in liquid state and thus pressurized water at 30 bar is ideal for this use. It is important not to have steam, because steam is not able to transfer the heat to the organic fluid as effectively as water. The system is presented in Fig. 6.

There are two different circuits, one with pressurized water and one with the working fluid. The first one absorbs heat from the exhaust gas and from the cooling air, with two heat exchangers, in order to transfer this heat to the organic fluid. The water circuit operates between 220 °C and 125 °C, which is lower than the exit of the cooling air in the atmosphere. The energy is transferred from water to the working fluid through the heat exchangers, which are the preheater, the evaporator and the superheater. At the inlet of the turbine, the organic medium has a maximum temperature and pressure of 185 °C and 30 bar respectively. The turbine exhaust steam enters the regenerator, before the condenser, in order to preheat the working fluid. That way the system rejects less energy to the environment through the condenser. The system operating parameters, concerning the mechanical and electrical efficiencies and ambient conditions, remain the same as in the water steam cycle (Table 1). These thermodynamic procedures can be seen in the T–S diagram (Fig. 7).

The parameters of points 1–7 are given in Table 3. It is important to note that the mass flow rate of the water is 52.67 kg/s and of

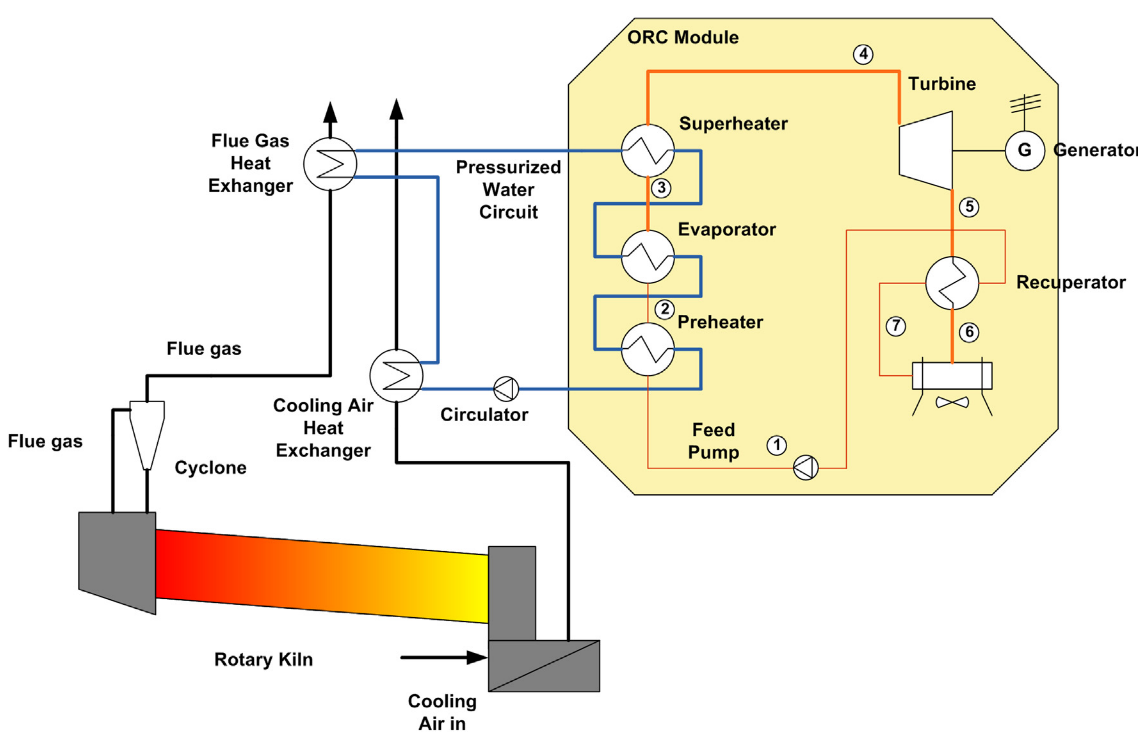

Fig. 6. Heat recovery system with ORC.

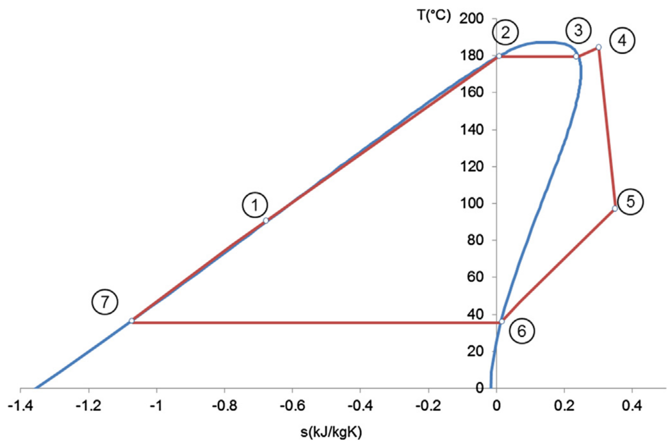

Fig. 7. *T–S* process diagram for isopentane.

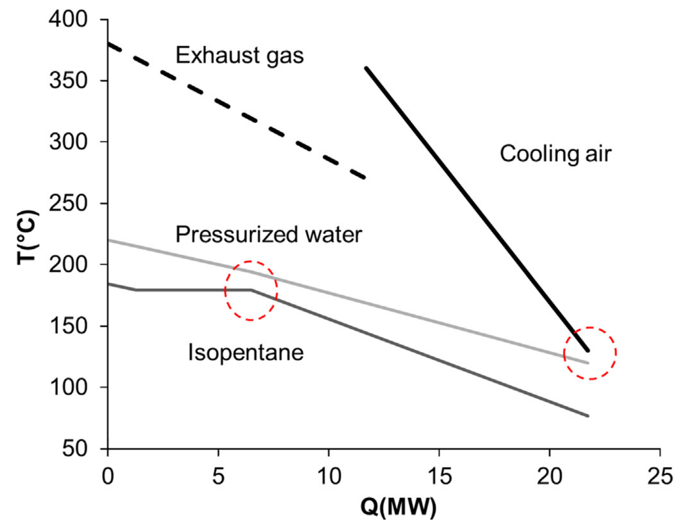

Fig. 8. *Q–T* diagram for the ORC.

isopentane 48.71 kg/s. The *Q–T* diagram that shows the heat exchange procedure is presented in Fig. 8.

As it is shown in Fig. 8, there are two pinch points in the *Q–T* diagram, the first one between the pressurized water and isopentane at 15 K and the second between the pressurized water and the cooling air at 5 K. The second pinch point is the most important among them, because it determines the exit temperature of hot air, which was set at 130 °C as in steam cycle case.

5. Exergetic analysis

After the energy analysis of the systems, an exergy analysis was performed. Exergy analysis is a very useful tool for analyzing thermodynamic systems, as it is possible to determine the maximum performance of the system and to find the components in which exergy loss takes place. So, the performance of the system can be optimized by minimizing the exergy losses [21], taking also into consideration economic factors.

At first, some theoretical points will be stated. Exergy is the maximum amount of work that can be produced by a system when a heat stream is brought to equilibrium in relation to a reference environment which is at reference conditions ($p_0 = 1.013$ bar, $T_0 = 298$ K). In this paper, the molar physical exergy is considered [22], [23]:

$$E = N\cdot\varepsilon_{ph}, \tag{5}$$

where N (mol s^{-1}) is the molar flow. Other forms of exergy, such as potential, kinetic and chemical were ignored in this work. The calculation of molar physic energy is the result of the use of 4 thermodynamic properties which are the temperature (T), the pressure (p), the enthalpy (h) and the entropy (s) and the expression that gives the physical exergy is the following:

$$\varepsilon_{ph} = (h - h_0) - T_0(s - s_0), \tag{6}$$

All the parameters of this equation are calculated for every stream with the use of the simulation program IPSEpro™.

Moreover, the exergy value of power output E^W is equal to the power. The exergy losses due to mechanical and electrical inefficiencies were taken in consideration but the heat losses of the system units were ignored. Both energy losses and exergy destruction have been summed under the term of irreversibilities, symbolized as IR. Taking an exergy balance in a control volume, gives the following expression:

$$\sum E_{i,in} = \sum E_{i,out} + \sum E_{i,losses} + \text{IR}, \tag{7}$$

For the system, exergetic efficiency can be defined as:

$$\eta_{ex} = \frac{\sum E_{i,out}}{\sum E_{i,in}}, \tag{8}$$

Table 3
ORC points.

Point	*P* (bar)	*T* (°C)	*h* (kJ/kg)
1	30	77.1	−223.0
2	30	179.5	94.9
3	30	179.5	201.4
4	30	184.4	228.0
5	1.4	97.4	127.4
6	1.4	46.9	31.0
7	1.4	35.5	−326.1

Table 4
Exergy balance for the heat recovery systems a) water-steam cycle, b) ORC.

a)Water-steam				b) ORC			
Component	ζ (%)	IR (kW)	IR (%)	Component	ζ (%)	IR (kW)	IR (%)
Turbine 1	87.6	595	3.10	Turbine 1	87.4	671	3.48
Turbine 2	85.5	428	2.23	Regenerator	71.6	185	0.98
Gas superheater	79.6	259	1.35	Preheater	83.3	834	4.34
Gas evaporator	63.2	2296	11.95	Evaporator	90.7	184	0.97
Air preheater	74.9	274	1.43	Superheater	87.9	64	0.33
Air evaporator	63.3	1317	6.85	Gas heat exchanger	60.7	2765	14.38
Air superheater	77.1	318	1.65	Air heat exchanger	55.8	2373	12.34
Pump	75.3	12	0.06	Pump	70.6	128	0.67
Deaerator		160	0.83	Condenser		670	3.48
Drum 1		25	0.13				
Drum 2		30	0.16				
Condenser		515	2.67				
Exergy losses		(kW)	(%)	Exergy losses		(kW)	(%)
Gas exhaust		6292	32.74	Gas exhaust		6292	32.74
Air exhaust		440	2.29	Air exhaust		440	2.29
Products exergy		(kW)	(%)	Products exergy		(kW)	(%)
Power		6258	32.56	Power		4613	24.00
Total		19219	100.00	Total		19219	100.00

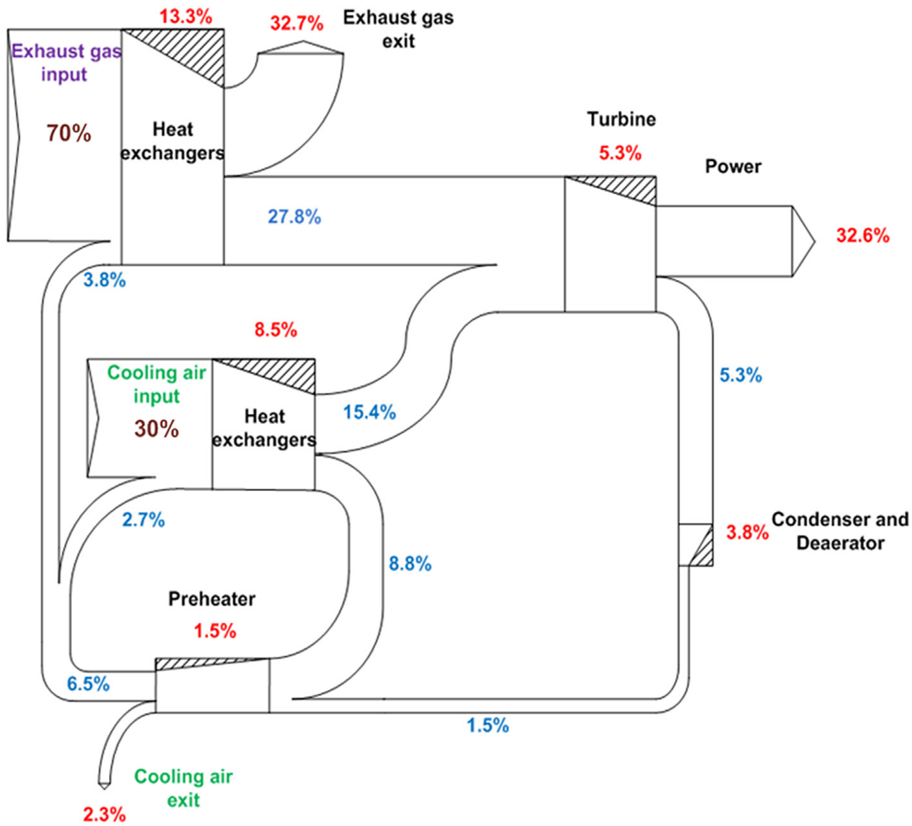

Fig. 9. Grassmann diagram for water-steam cycle.

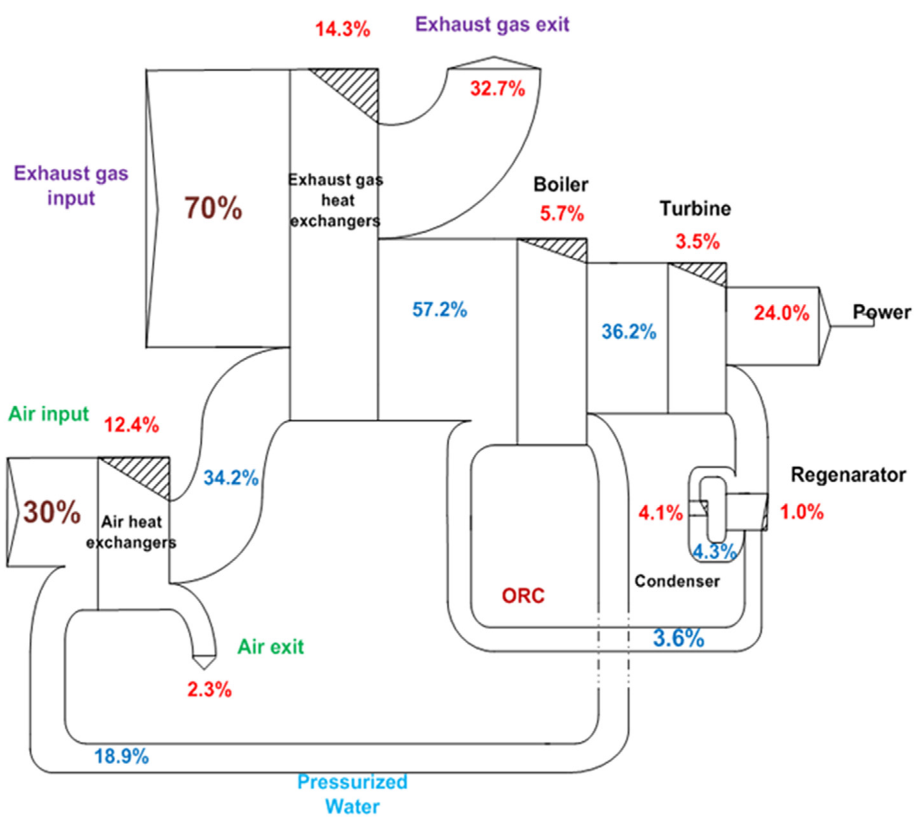

Fig. 10. Grassmann diagram for ORC.

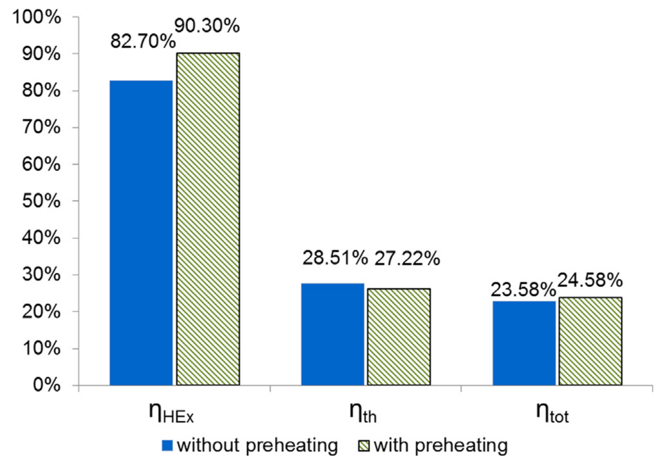

Fig. 11. Comparison of water-steam cycles with and without additional preheating.

with $\sum E_{i,in}$ and $\sum E_{i,out}$ being defined properly for each system describing exactly the amount of exergy that the system consumes to produce useful products. More specifically, $\sum E_{i,out}$ in this system concerns only the power of the generator.

6. Results

The two cycles were simulated with IPSEpro™ .The optimum water-steam cycle has a system efficiency of 23.58% producing 6.26 MW electric power, whilst the optimum ORC has a thermal efficiency of 17.56% producing 4.66 MW electric power. The other thermodynamic tool that is used for the comparison of the cycles is the exergy analysis. After the analysis with IPSEpro™, the exergetic efficiency was calculated at 32.56% for the water-steam cycle and at 24.00% for the ORC. Once more, it is shown that the water-steam cycle has a better performance in these conditions. The main reason for this result is firstly the higher maximum temperature of the water-steam cycle which results in more work produced by the turbine and secondly the existence of the intermediate pressurized water circuit in the ORC, which causes additional exergy destruction.

Another way to determine the most efficient system is to compare the *Q*–*T* diagrams (Figs. 3 and 8). This can be done by comparing the area between the lines of the heat source and the working fluid. In the ORC case the exergy destruction which is related to this area is higher and the performance of the system deteriorates. The reason for this is the use of an intermediate medium (pressurized water) and the physical properties of isopentane, which works in lower temperature than water-steam. All the results of the exergy analysis are summarized in Table 4. For the water-steam cycle it is clear that the main exergy loss is from the exhaust gas (Table 4a) because of its high exit temperature (270 °C), which is a restriction imposed by the production process of the cement plant. The next major exergy destruction is located in the two heat exchangers that operate as evaporators. For the ORC, the main exergy loss is also located in the gas exhaust (Table 4b) and in the two heat exchangers.

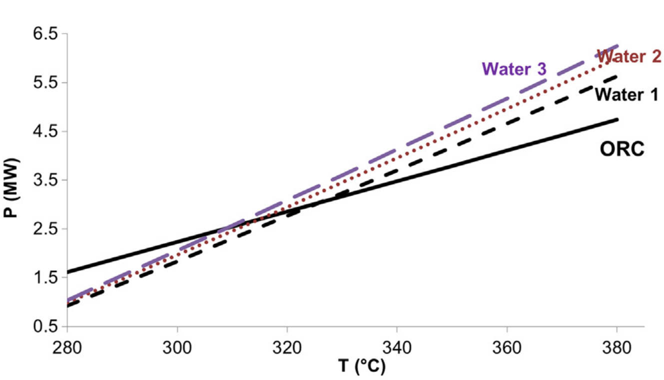

Fig. 12. Comparison of water-steam cycles with ORC cycles.

Table 5
Specific emissions factors for Greek energy mix (Reference year 2010).

CO_2 (g/kWh$_{el}$)	NO_x (g/kWh$_{el}$)	SO_x (g/kWh$_{el}$)	PM10 (g/kWh$_{el}$)
848	0.939	2.829	0.223

Another useful tool for the exergetic comparison of the two systems are Grassmann diagrams which are presented in Figs. 9 and 10 for the water-steam and the ORC system respectively. From these diagrams it is clear that the main reason which makes the water-steam system more efficient is the lower exergy losses in the heat exchangers. More specifically, the water-steam system has 21.8% losses compared to 26.7% of the ORC system.

After the comparison of those two heat recovery systems, it is useful to suggest further possible solutions and changes that could increase the efficiency but may have larger capital cost. For the water-steam cycle, which is the most efficient solution, a further preheating is possible. More specifically, the hot air from point 2ca (Fig. 2) can be utilized to preheat the water at point 7 in order to provide more heat to the system. This change improves the η_{HEx} but reduces the η_{th}. However, the total system efficiency is improved. The comparison of the efficiencies is shown in Fig. 11.

Another way to compare the water Rankine cycle and the Organic Rankine Cycle is shown in Fig. 12. Four different systems are compared, three with water Rankine cycle and an ORC with isopentane as working fluid. The first water system (water 1) is a simple cycle with turbine, condenser, feed pump and boiler. The second system (water 2) is the system of Fig. 2 which has been analyzed and the third (water 3) is the optimum system with preheater which was described in the previous paragraph. The four systems were examined for various heat source temperatures. The results are summarized in Fig. 12, which shows which system performs better in every case. It is essential to say that the horizontal axis shows the temperature of the exhaust gases, while for all cases the temperature of the cooling air is considered 20 K lower than the temperature of the exhaust gas.

Fig. 12 shows that when the temperature of the exhaust gases is lower than 310 °C, the ORC system is more efficient while for higher exhaust gas temperatures the water cycle system performs better. So for a cement production plant that uses old technology and its exhaust gases are at a high temperature level, a water-steam Rankine cycle heat recovery system is preferable. On the other hand, for a modern cement plant which is more efficient and thus its exhaust gases are at a relatively low temperature, a heat recovery system using the ORC is more suitable.

After analyzing the performance of the two systems, it is possible to make an estimation of the avoided emissions including

Table 6
Avoided annual emissions of the two systems.

	Steam cycle	ORC
Power (MW)	6.26	4.66
Avoided CO_2 (t/a)	37,159.00	27,661.00
Avoided NO_x(t/a)	41.14	30.63
Avoided SO_x (t/a)	123.96	92.28
Avoided PM_{10} (t/a)	9.77	7.27

Table 7
Energy production and avoided cost.

Produced electricity	43,820 MWh/y
WHR plant own consumptions (~3% of power production)	1314 MWh/y
Net electrical energy for use	42,505 MWh/y
Avoided cost of electricity (gross profit)	4.25×10^6 €/y

Table 8
Financing plan.

Own capital	50%
Capital from long term loan	50%
Loan interest	5%
Loan payback period	10 years
Gross annual profit	4.25×10^6 €/y
Annual expenses (O&M cost)	3% of Capital Cost
Discount rate	12%

CO_2, NO_x, SO_x, PM_{10} particles. Taking into account the energy mix of Greece [24], emissions data for each plant technology [25–27] and the total electricity production [24] for the reference year 2010, the average emission factor per each pollutant is shown in Table 5. In addition it is assumed that the cement plant has an annual operation time of 7000 h and that all the energy produced by the heat recovery system is either consumed by the plant itself or delivered to the national power grid. By multiplying the above mentioned specific emissions factors with the produced energy in annual basis the avoided emissions for the WHR (waste heat recovery) plant operation can be calculated.

The results for the ORC and for the water-steam cycle are shown in Table 6. It is clear that a significant amount of emissions can be annually avoided, for any of the two cases considered.

For the best solution in the examined case the benefit in plant operation can be occurred by the avoided cost of the produced electricity. A 6.26 MW WHR plant, with a minimum operation time of 7000 h in annual basis can produce 43.82 GWh of electrical energy. By substituting the purchases of the respective electricity amount with its own power production, cement plant can save considerable amount of money by avoiding the purchase cost of this energy from the grid. The net profit of cement plant occurs considering the operational cost of the WHR plant, which is subtracted from the avoided cost of electricity. Considering the price of the industrial electrical energy about 100 €/MWh cost savings from WHR plant operation are shown in Table 7.

Considering the above calculated gross profit a brief parametrical investment assessment was implemented. The investment's attractiveness was examined by calculating the Discounted Payback Period. Payback calculation compares revenues with costs and determines the length of time required to recoup the initial investment.

The sensitivity of the investment under consideration is examined with regard to the parameter of Specific Capital Cost, which is expected to have a major impact on the project's viability. Specific Capital Cost was considered in the range of 1500–3000 €/kW. Analysis was based on the following financing plan (Table 8). Fig. 13 presents the discounted payback period in relation to the Specific Capital Cost.

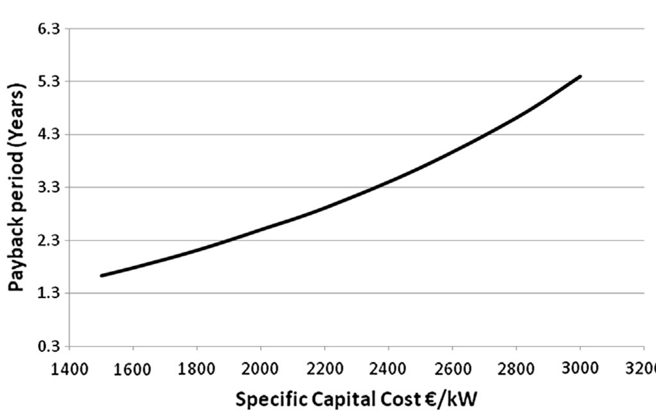

Fig. 13. The payback period of the investment for various specific capital cost estimations.

7. Conclusions

Waste heat recovery is feasible for a cement industry and it can offer about 6 MW of electric power for a typical cement plant. The preheater and clinker cooler exhaust gases are the heat sources for the heat recovery systems. Two different cycles were investigated; a water-steam cycle and an ORC with isopentane as working fluid. The energy and exergy analysis proved that the water steam-cycle has better performance with a system efficiency of 23.58% compared to 17.56% of the ORC. For high heat source temperatures steam cycle is more appropriate heat recovery solution. By the exergy analysis conducted it is shown that the cycle with the lower exergy efficiency has the worst performance. The main reason for the lower efficiency of the ORC is its lower maximum temperature compared to the maximum temperature of the water steam-cycle. It is important to clarify why the exergy efficiency is higher than system efficiency. The reason for this is the way that the system efficiency has been determined in this paper. Finally the water–steam cycle can be further improved reaching 24.58% system efficiency by utilizing the high exhaust temperature of the cooling air in order to preheat the condensates before the inlet of the feed tank.

To sum up, the present study concludes that the water-steam cycle is the more efficient solution. In this case, due to the relatively high temperature level of the heat source (over 150°C), the higher operating temperature and pressure of water-steam provide increased efficiency compared to organic fluids. In this study, the temperature of the waste heat sources is high due to the low efficiency of the production procedure of the cement plant considered. It is obvious, that in the case of a newer (state of the art) cement plant with higher efficiencies, the ORC may be more advantageous than the water steam recovery cycle. Higher efficient cement plants have lower exhaust gas temperature and calculations performed showed that if the exhaust gas temperature is lower than 310 °C, ORC heat recovery systems are more efficient solutions.

It should be stressed that energy and exergy analysis have provided important results for the evaluation of the two heat recovery systems.

From economical point of view WHR plant in cement industry seems to be a very attractive investment with a payback period up to five years, by saving considerable amount of money that otherwise should be spent to purchase the produced electricity from the grid.

References

[1] Mehta PK. Greening of the concrete industry for sustainable development. Concrete International 2002;24(7):23–8.
[2] Worrell E, Price L, Martin N, Hendriks C, Meida LO. Carbon dioxide emissions from the global cement industry. Annual Review of Environment and Resources 2001;26:303–29.
[3] Climate change, cement and the EU. CEMBUREAU; July 1998.
[4] BAT reference document for the cement, lime and magnesium oxide manufacturing industries. Seville, Spain: EIPPCB; May 2010.
[5] Carbon leakage: European cement industry at risk. Boston Consulting Group; 2008.

[6] Wang J, Dai Y, Gao l. Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry. Applied Energy 2009;86:941–8.
[7] Ziya SÖĞÜT M. A research on exergy consumption and potential of total CO2 emission in the Turkish cement sector. Energy Conversion and Management April 2012;56:37–45.
[8] Cement technology roadmap 2009. WBCSD-IEA; 2009.
[9] Atmaca Adem, Kanoglu Mehmet. Reducing energy consumption of a raw mill in cement industry. Energy June 2012;42(1):261–9.
[10] Madlool NA, Saidur R, Hossain MS, Rahim NA. A critical review on energy use and savings in the cement. Renewable and Sustainable Energy Reviews 2011: 2042–60.
[11] Legmann Hilel, ORMAT Group. Recovery of industrial heat in the cement industry by means of the ORC processes. In: Cement industry technical conference 2002.
[12] Khurana Shaleen, Banerjee Rangan, Gaitonde Uday. Energy balance and cogeneration for a cement plant. Applied Thermal Engineering 2002;22: 485–94.
[13] Engin T, Ari V. Energy auditing and recovery for dry type cement rotary kiln systems – a case study. Energy Conversion Management 2005;46:551–62.
[14] Saneipoor P, Naterer GF, Dincer I. Heat recovery from a cement plant with a Marnoch heat engine. Applied Thermal Engineering 2011;31:1734–43.
[15] Johnson I, Choat B, Dillich S. Waste heat recovery: opportunities and challenges. Minerals, Metals and Materials Society; 2008. EPD Congress.
[16] Ahamed JU, Madlool NA, Saidur R, Shahinuddin MI, Kamyar A, Masjuki HH. Assessment of energy and exergy efficiencies of a grate clinker cooling system through the optimization of its operational parameters. Energy October 2012;46(1):664–74.
[17] Bronicki Lucien Y, Chairman. Organic Rankine cycle power plant for waste heat recovery, presented at 13th Symposium.on industrial applications of gas turbines. Alberta, CANADA: BANF; October 1999.
[18] www.siemens.com [accessed 16.01.12].
[19] Madlool NA, Saidur R, Rahim NA, Islam MR, Hossian MS. An exergy analysis for cement industries: an overview. Renewable and Sustainable Energy Reviews January 2012;16(1):921–32.
[20] European Commission, Integrated Pollution Prevention and Control (IPPC). Reference document on best available techniques in the ferrous metals processing industry December 2001.
[21] Wang Jiangfeng, Dai Yiping, Gao Lin. Exergy analyses and parametric optimizations for different cogenerationpower plants in cement industry. Applied Energy 2008;86(6):941–8.
[22] Kotas TJ. The exergy method of thermal plant analysis, vol. 64. Malabar (FL): Krieger Publishing Company; 1995. p. 212–29. Chemical Engineering Research and Design.
[23] Karellas S, Panopoulos KD, Panousis G, Kakaras E, Boukis I. Energetic and exergetic investigation of integrated RDF gasification energy systems, ECOS 2010. In: Proccedings the 18th international conference on efficiency, costs, optimization. Lausanne, Switzerland: Simulation and Environmental Impact of Energy Systems; 2010 Jun 14–17.
[24] www.admie.gr.
[25] European pollutant release and transfer register (E-PRTR). http://prtr.ec.europa.eu/; 5/2/2013.
[26] European Union. Climate action, European union transaction log. http://ec.europa.eu/environment/ets/; 5/2/2013.
[27] IEA ETSAP energy technology systems analysis programme technology brief E02. www.etsap.org; April 2010.