SEPARATE AND COMBINED INTEGRATION OF KALINA CYCLE FOR WASTE HEAT RECOVERY FROM A CEMENT PLANT

by

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Original scientific paper
https://doi.org/10.2298/TSCI210418230M

This article reports on using Kalina cycle for waste heat recovery from a cement plant. Two design alternatives have been investigated using separate and combined waste heat recovery from the kiln, cooler, and preheater. Measurements and analysis have been performed to determine the waste heat from different stages of the cement manufacturing lines. The annual heat losses from the kiln surface, preheater, and the cooler are estimated as 79.23 GWh, 44.32 GWh, and 43.6 GWh at average temperatures of about 314 °C, 315 °C, and 254 °C, respectively. Analysis and optimization of using Kalina cycle for waste heat recovery from the kiln shell, cooler and preheater to produce electricity have been carried out using ASPEN software. Parametric study has been carried out to determine the design parameters for Kalina cycle including turbine inlet pressure, mass-flow rate, and NH_3-H_2O concentration. The value of net power output using combined waste heat recovery is about 7.35 MW as compared to 6.86 using separate waste heat recovery design with a total cost saving of about 23%.

Key words**:** *Kalina cycle, waste heat recovery, low-grade heat source, NH_3-H_2O mixture, power, cement industry, heat loss, power consumption*

Introduction

Energy cost average is about 55% of the total cost of cement production. Massive energy cost is due to both heat consumption in kiln operations and electrical power consumption for different operations of grinding mills, fans, and motors [1]. Energy consumption in a cement plant is divided into 25% in the form of electricity and 75% as heat. Waste heat recovery (WHR) for power generation is a way to reduce the total power consumption for cement production process [2]. In the whole process, about 35-40% of the heat is lost through different waste heat streams in the kiln, preheater, and cooler [3].

In cement plants, three points which can be used for WHR system. The first point is the exhaust gas of pre-heater with temperature of about 300-350 °C in 5-6 stages production lines. The second point is the cooler, where the clinker temperature at the exit of the kiln reaches 1000 °C. The clinker is air cooled to 100-120 °C producing waste hot air at about

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260-300 °C. The third point is waste heat from the kiln shell where the hot gases can reach temperatures more than 300 °C [4]. An energy audit analysis of a typical cement plant [5] showed that the kiln and cooler exhaust heat represent 19.15% and 5.61% of total heat input, respectively. For 5000 ton per day of kiln operation, the expected power generation was estimated to be approximately 6-9 MW [6]. Sogut *et al.* [7] estimated 217.31 GJ waste heat from the rotary kiln of cement plant in Turkey of which represents about 51% of the overall heat consumption of the process. Widuramina *et al.* [8] investigated the available waste heat in the cement kiln waste gas in a Norwegian cement plant. For a cement production of 1.3 million tons per year, it was found that 1.5-4.2 MW and 2.2-5.8 MW waste heat is available that can be used for low pressure steam and hot water generation.

Kalina cycle has been considered as an effective power cycles for low temperature WHR. It has many many practical advantages as compared to Rankine cycle [9, 10]. A Kalina based cogeneration system of power and refrigeration showed significantly higher energy efficiency as compared to the stand-alone Kalina cycle [9]. Numerical study of Kalina cycle utilizing low temperature geothermal heat source at 145 °C for power generating reported a cycle efficiency of 12.95% [10]. The cycle efficiency was shown to be improved by 2% using an auxiliary superheater in the system. Mehri *et al.* [11] proposed a new combined heat and power cogeneration system which is based on the Kalina cycle and uses geothermal energy as a heat source to produce electricity and pure water. The proposed system includes a Kalina cycle, a LiBr-H_2O heat exchanger and a water purification system. The First law and Second law efficiencies of the proposed system were found in the range of 16%-18.2% and 61.9%-69.1%, respectively.

Wasabi Energy estimated that the integration of the Kalina cycle technology into a cement plant in the Khairpur region of Pakistan for WHR in clinker cooler exhaust gases and preheater exhaust gases would reduce overall power consumption by 10-20% [12]. The FL Smith provided the Kalina cycle WHR to Star Cement L. L. C. in Ras Al Khaimah, United Arab Emirates. According to their system, the total power is expected to be reduced by 12% [13].

Sirko [14] studied a cogeneration plant using Kalina cycle. The net efficiency of the integrated Kalina plant was obtained between 12.3% and 17.1% and depends on the cooling water temperature and the ammonia content in the alkaline solution. A parametric study and optimization of Kalina cycle driven by solar energy has been reported by Wang *et al.* [15]. The net power output from the cycle can be maximized by proper choice of turbine inlet pressure and ammonia solution mass fraction with less sensitivity to changes in inlet temperature to the turbine. Carlos, *et al.* [16] performed thermodynamic analysis of ORC and Kalina cycles using different working fluids. Using R-290 as the working fluid of ORC and using a Kalina cycle composed of a mixture of 84% ammonia mass fraction and 16% water mass fraction, the best performance of the two cycles can be obtained. The net power of Kalina cycle was found to be 18% higher than ORC.

From the previous review, it can be concluded that the adaptation of Kalina cycle in cement plants needs more investigations and analysis. The size of the components or the selected conditions have rarely been taken into consideration. Also, the cycle configuration and integration in the cement plants should consider the differences in available amount and temperature levels of waste heat sources in the plant. As practical case study for a typical cement plant, the present article reports on WHR from Al Arish Cement plant in Egypt. Measurements and analysis have been performed to determine the waste heat from three points of the cement manufacturing lines. Analysis and optimization of using Kalina cycle to recover waste heat from the kiln shell, cooler and preheater to produce electricity have been carried out using ASPEN HYSYS software. Design parameters of system components and recovery heat exchangers of kiln shell, cooler and preheater are specified. The effects of turbine inlet pressure, ammonia concentration, and the evaporator exit temperature on Kalina cycle performance are investigated. Two design alternatives have been proposed and investigated for Kalina cycle integration in the cement plant. The first configuration uses separate recovery heat exchanger and Kalina cycle for the kiln, cooler, and preheater. The second combined WHR system combines the waste heat from the kiln, cooler, and preheater in a single cycle.

Plant description

The present study is carried out on a typical cement production plant in El Arish Cement Company in Egypt. The plant is located 70 km to the south of El Arish City in Sinai. It contains four production lines with average capacity 5800 ton per day. The plant started production with two lines in 2010 then added 2 lines in 2016. Table 1 shows technical data of major plant components as per the information available in January 2018.

Table 1. El Arish cement company production lines technical data, January 2018

Number of lines	Cement process	Preheater type	Preheater stage	Kiln average capacity ton per day	Raw mill type	Kiln diameter	Kiln length
4	Dry process	Double string	5	5800	Vertical mill	5 m	72 m
Cooler type	Fuel	Fuel consumption	Power consumption	Production availability	Cooling water	Raw mill cooling	Cooler type
Hydraulic grate cooler	Coal-heavy oil	900 kcal/kg	110 kWh per ton	345 day per year	Air cooling tower	Conditioning tower	Hydraulic grate cooler

Figures 1-3 show the flow diagram for the preheater, kiln area, and cooler area and the proposed positions for WHR. The WHR from the preheater depends on hot gases from the cyclone before raw mill process and ID fan. For the kiln, WHR is located around the kiln shell to collect heat loss by radiation and convection from kiln shell using secondary shell and insulation from ambient air. For the cooler area, WHR receives waste hot gas from cooler before entering the filter then to the stack.

Waste heat analysis and feasibility

Studying the heat source, the material flow direction, chemical composition, and hot gas characteristics are the first step to analyze waste heat from the plant. As can be seen in fig. 1, feeding material start firstly in preheater cyclones (C1-C5). A cyclone is a conical vessel shape in which fine material and gas stream pass tangentially by a vortex force within the vessel. The hot gas leaves the cyclone through a co-axial *vortex-finder* upward. The feeding material are thrown to the outside edge of the cyclone by centrifugal force action and leaves down through a flap gate valve. The feeding material passes from one cyclone to the other to enter the kiln. The average temperature in the first step of cyclones reaches 300-400 °C and increases gradually by going down to the next step of cyclones to reach about 800-900 °C at kiln inlet.

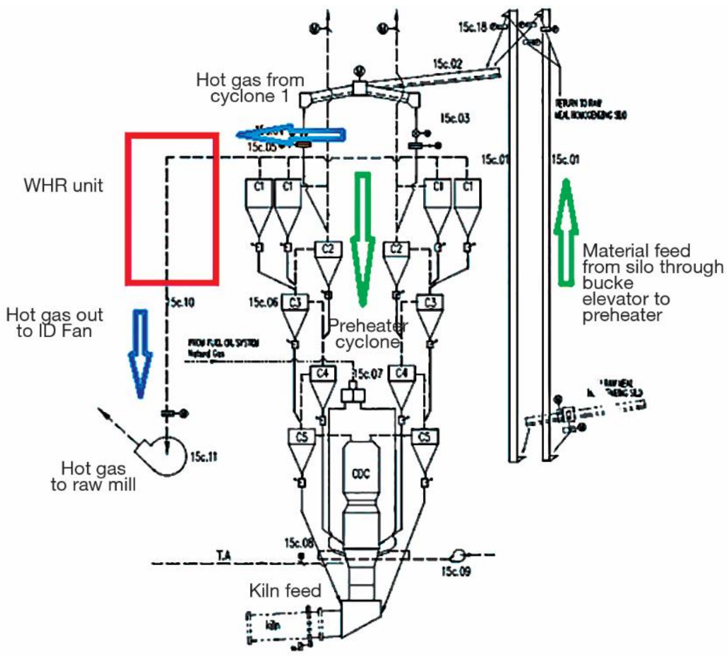

Figure 1. Flow diagram for preheater area and WHR system position, preheater

In the rotary kiln, see fig. 2, fuel is added to the system, using the main burner inside the kiln towards the outlet part and at the calciner part of preheater by using four burners. Typical fuels used in the plant include heavy oil, natural gas, coal, or a mixture of alternative fuels. The rotary kiln is made of a steel shell tube with number of sections welded together and is inclined to help material flow to next processes in the cooler. It has a layer of refractory bricks to withstand high operating temperature which may reach about 1500 °C during the calcination process. The kiln outer steel shell is exposed to the ambient and can reach a tem-

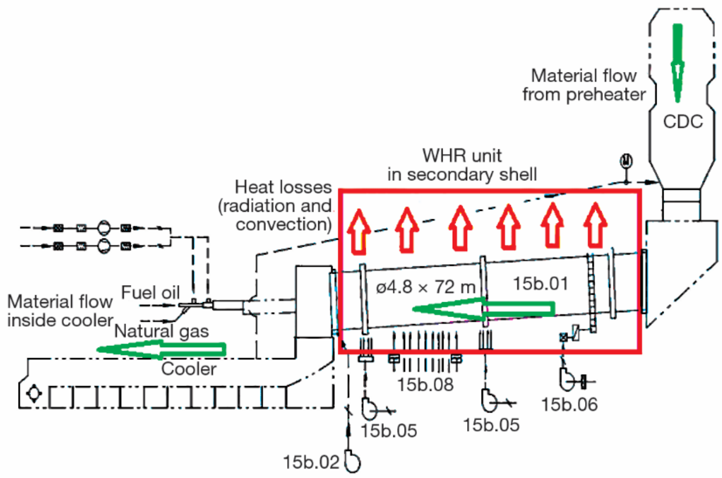

Figure 2. Flow diagram for kiln area and proposed WHR system position, kiln

perature of about 300-400 °C. The three live casted steel rings which support the rotary kiln, called *tyres*, rotate on the supporting rollers (two rollers for each tyre) and carries the heavy weight of the kiln. The kiln shell loses large amounts of heat by radiation and convection to the ambient. Additionally, air is pumped over specific areas over the shell surface using air nozzles to avoid shell deformation.

Feeding material finally leaves the rotary kiln and convert to clinker. It should be cooled down in clinker cooler, fig. 3. They move with special speed on grates cooled by external air fans. Hot gas with clinker dust with temperature of about 250-350 °C leaves the cooler to the filtering stage by using a centrifugal fan and then move to the stack. The major WHR sources from cement production lines are outlined in fig. 4. They include radiation and convection from the rotary kiln surface, cooler vent air, and hot gas exhaust from the cyclone preheater. They are analyzed in the following sections. They are analyzed in the following sections.

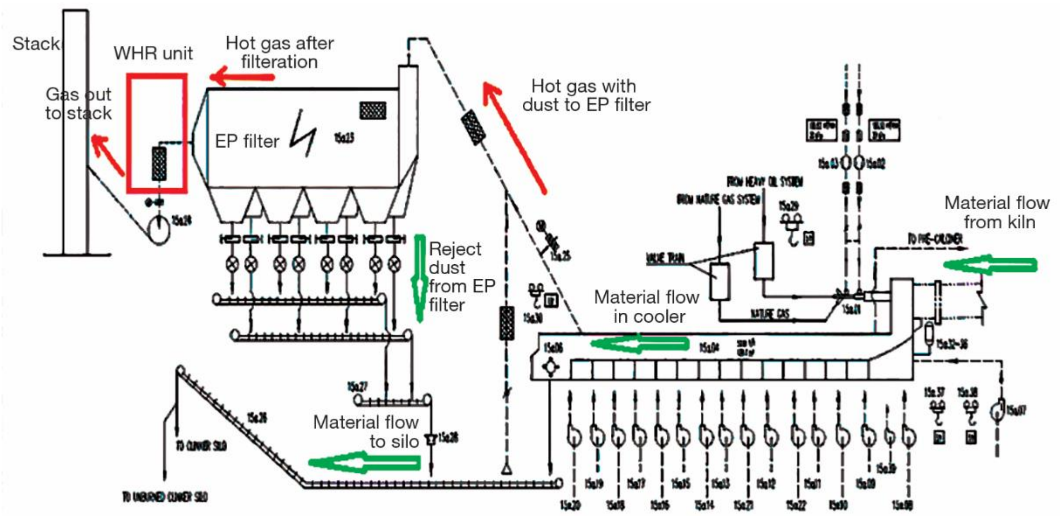

Figure 3. Flow diagram for the cooler area and proposed WHR system position, cooler

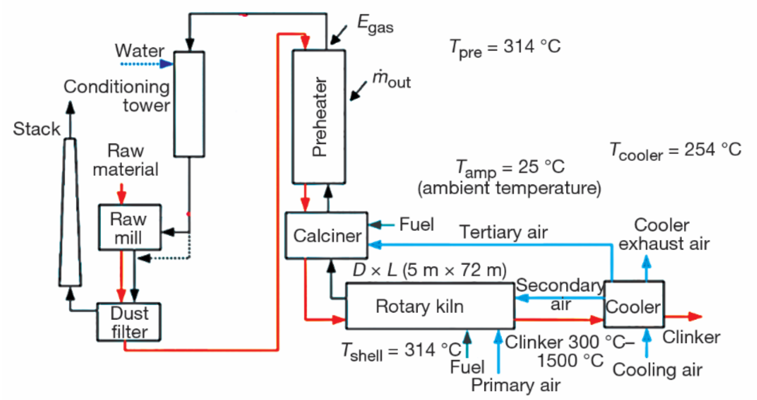

Figure 4. Schematic diagram showing major WHR sources from cement production lines

Waste heat from rotary kiln

Convection and radiation losses from the rotary kiln surface are function of the kiln surface temperature and forced air along its 72 meters length. It is known that the temperature of the surface is dependent on the type of fuel used, type of clinker manufactured, duration of operation from previous maintenance which influence refractory lining, and atmospheric conditions. The surface temperatures are monitored constantly by the plant control room during the normal operating conditions of the rotating kiln using IR image techniques.

Using IR measurements, the variation of average kiln surface temperature over a typical year is shown in fig. 5. The average shell temperature of rotary kiln is measured to be about 314 °C.

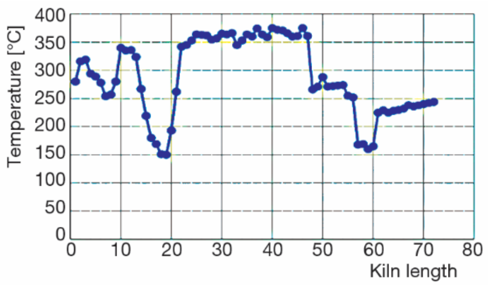

Figure 5. Variation of average measured temperature along kiln shell during normal operation

Convection heat loss from the kiln surface Q_{conv} is calculated using eq. (1).

$$Q_{\text{conv}} = h_{\text{c}} A_{\text{k}} (T_{\text{k}} - T_{\text{a}}) \tag{1}$$

where h_{c} is the convection heat transfer coefficient, A_{k} – the kiln surface area, T_{k} – the kiln surface temperature, and T_{a} – the average atmospheric temperature. The convection heat transfer coefficient h_{c} [Wm^{-2}K^{-1}] is calculated [6]:

$$h_{\text{c}} = 0.3D + 4 + 3.5\left(\frac{T_{\text{k}}}{100}\right) - 0.85\left(\frac{T_{\text{k}}}{100}\right)^2 + 0.076\left(\frac{T_{\text{k}}}{100}\right)^3 \tag{2}$$

The radiation heat losses from the kiln surface is calculated using eq. (3):

$$Q_{\text{rk}} = \varepsilon\sigma A_{\text{k}} (T_{\text{k}}^4 - T_{\text{a}}^4) \tag{3}$$

where T_{k} [K] is the kiln surface temperature, $\sigma = 5.67 \cdot 10^{-8}$ W/m^{2}k^4, A_{k} – the kiln surface area (πDL) estimated as 1130.4 m^2 and emissivity of steel is taken as $\varepsilon = 0.9$. For the calculation of total convection and radiation heat losses from the kiln surface and account for the variation of kiln surface temperature along its length, the kiln surface is divided into equal sections of one meter length each. The total annual convection and radiation losses from the kiln surface Q_{ckt} and Q_{rkt} [MWh] are obtained by summing together all values of convection and radiation losses from each meter of kiln and multiplying it by fraction of operating hours, y, in a year as given:

$$Q_{\text{ckt}} = 8760 y \sum_{i=1}^{n} Q_{\text{ck},i} \tag{4}$$

$$Q_{\text{rkt}} = 8760 y \sum_{i=1}^{n} Q_{\text{rk},i} \tag{5}$$

where n is the number of kiln sections, $n = 72$. The total annual heat loss from the kiln Q_{kiln} can be calculated by summing together convection and radiation losses:

$$Q_{\text{kiln}} = Q_{\text{ckt}} + Q_{\text{rkt}} \tag{6}$$

Waste heat from preheater

El Arish cement plant has four lines with kiln feed capacity of 5800 ton per day and preheater with double string design and 5 stages. After the hot gas from cement kiln is used to preheat the raw meal and calcination process, it is dissipated to the top of the preheater cyclones (cyclone 1 first stage) then to the conditioning tower before passing through the raw mill. The gas should be cooled before being sent to the raw mill. Some of the hot gas is used within the raw mill for drying and lifting process. The exhaust gas from the preheater can be used for WHR without influencing cement process with some limitations.

Figure 6 shows the variation of measured hot gas temperature form cyclone 1 over one year. To divert the hot gas through a heat exchanger for heat recovery, the cooling water will be removed from the cooling tower. The heat recovery system (heat exchanger) should be designed to maintain the required output temperature requirements for raw mill operation. The hot gas exit from heat exchanger should have the same temperature as the conditioning tower exit gas. In the present study, a heat exchanger for WHR is proposed to be installed in parallel to the conditioning tower after the preheater, see fig. 7.

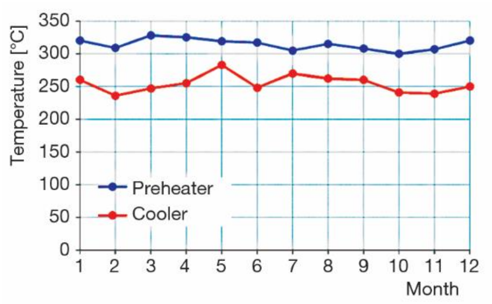

Figure 6. Monthly variation of exhaust gas temperature from the preheater and cooler

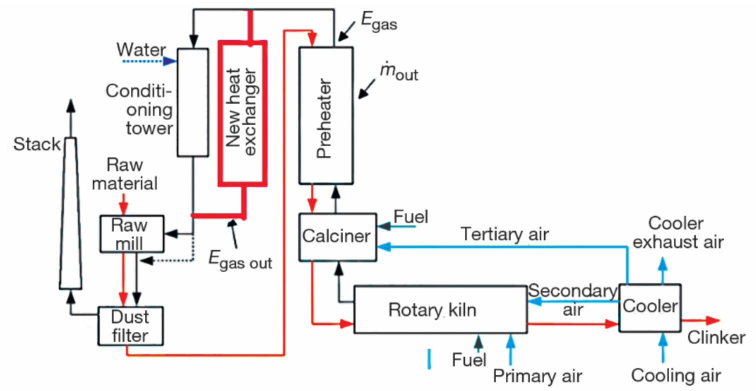

Figure 7. Proposed location of heat exchanger for heat recovery from preheater exhaust gas

The WHR from preheater exhaust gas, Q_p, can be calculated using the difference between the conditioning tower gas inlet and outlet parameters given:

$$Q_p = \dot{m}_i c_{p,i} T_i - \dot{m}_o c_{p,o} T_o \tag{7}$$

where $\dot{m}$, T, c_p are the gas the mass-flow rate, temperature [K], and specific heat at the inlet, i, and outlet, o, of cooling tower The specific thermal capacity of the hot gas, c_p can be estimated as function of mass fraction, x, of each component, k, in the exhaust gas and hot gas temperature, T, as reported in [6]. The volume fraction, x, for each gas component in the exhaust gas before and after the conditioning tower is estimated based on nominal data extracted from [6]. The mass-flow rate of gas entering the cooling tower is calculated:

$$\dot{m}_{\mathrm{i}} = \dot{V}_{\mathrm{i}} \sum_{k=\mathrm{CO_2,H_2O,N_2,O_2}} \rho_{k\mathrm{i}} x_{k\mathrm{i}} \tag{8}$$

where $\dot{V}_{\mathrm{i}}$ is the measured volume flow rate of gas at preheater outlet. The mass-flow rate of gas exit from the cooling tower is calculated considering the water vapor mass-flow rate added to the inlet gas by water nozzles in the conditioning tower, $\dot{m}_{\mathrm{w}}$, using:

$$\dot{m}_{\mathrm{o}} = \dot{V}_{\mathrm{o}} \sum_{k=\mathrm{CO_2,H_2O,N_2,O_2}} \rho_{k\mathrm{o}} x_{k\mathrm{o}} + \dot{m}_{\mathrm{w}} \tag{9}$$

The water vapor mass added to the gas in the cooling tower is equal to the make-up water rate supplied to the cooling tower. The measured value of make-up water is about as 55 m^3/h. It should be mentioned that this amount of make-up water shall be saved after applying the WHR from the preheater as shown in fig. 7. The total annual waste heat [MWh] from the preheater exhaust gas is calculated using:

$$Q_{pt=} \sum_{1}^{8760} (\dot{m}_{\mathrm{i}} c_{p,\mathrm{i}} T_{\mathrm{i}} - \dot{m}_{\mathrm{o}} c_{p,\mathrm{o}} T_{\mathrm{o}}) \tag{10}$$

Waste heat from cooler

The cooler waste hot gas, which is vented to the atmosphere, is waste heat from the system. Figure 6 shows average temperature of waste hot gas from cooler over one year. The temperature changes from one month to another due to shutdown times and process parameters change during normal operation.

The total annual waste heat by hot gas from the cooler is calculated using:

$$Q_{\mathrm{ct}} = \sum_{1}^{8760} \dot{V}_{\mathrm{c}} c_{p,\mathrm{c}} (T_{\mathrm{co}} - T_{\mathrm{a}}) \tag{11}$$

where $\dot{V}_{\mathrm{c}}$ is the volume flow rate of hot gas discharge from the cooler, T_{co} – the hot gas outlet temperature, and c_{pc} [kJm^{-3}K^{-1}] – the heat capacity of gas per unit volume. According to Terblanche [6], the specific heat of the clinker cooler hot gas can be approximated as a function of the gas temperature by using the specific thermal capacity of dry air calculation. Table 2 shows the annual average energy loss from the kiln shell, preheater and cooler. The largest source of heat loss is in the kiln shell and the clinker cooler gas.

Table 2. Waste heat analysis from kiln shell, preheater, and cooler

Item	Availability	Energy consumption	$Q_{\mathrm{kiln\ conv}}$	$Q_{\mathrm{kiln\ rad}}$	Q_{hourly} [MWh]	Q_{Annual} [GWh per year]	Average temperature [°C]	Carnot [η_{max}]
Kiln Shell	95%	900 kcal/kg clinker	28.44 GWh per year	50.79 GWh per year	9.1	79.23	314	50%
Preheater	95%				5	44.32	315	50.1%
Cooler	95%				4.98	43.7	254	44.4%
Total	95%				19.48	167.25		

Feasibility of waste heat recovery

The feasibility of WHR systems is controlled by several factors. These factors include heat temperature, heat quantity, and minimum allowed temperature. The overall efficiency of WHR power generation system increases with the increase of available heat temperature, T_H and the decrease of minimum allowed temperature, T_L. Using Carnot heat engine as the upper limit, the maximum possible efficiency of WHR power generation, η_{max}, system is given:

$$\eta_{max} = 1 - \frac{T_L}{T_H} \tag{12}$$

The quantity of heat determines the expected system power generation capacity. Taking the ambient temperature as the lower limit of minimum temperature (25-35 °C), η_{max} = 44.4 to 50% for WHR from cooler, kiln shell and preheater. On the other hand, the selection of minimum allowed temperature is related to the composition of exhaust heat streams. Depending on the combustion fuel used, they can contain CO_2, water vapor, and NO_x. Condensation of water vapor in the exhaust in the presence of these elements may result into material corrosion of heat exchangers. This limitation is present only in preheater and cooler exhaust gases. The kiln WHR system is not sensitive to this parameter.

Kalina cycle integration

Kalina cycle uses a binary working fluid consisting of a mixture of ammonia and water. The variation of boiling temperature of the mixture allows proper thermal integration with the waste heat source and cooling medium in the condenser. Several configurations of Kalina cycle have been reported depending on the application and heat source type. The configuration of Kalina cycle employed in the present study is shown in fig. 8. This configuration is usually used for low temperature applications (120-400 °C). The heat recovered from the kiln, preheater and cooler is used to evaporate the NH_3-H_2O mixture in a heat exchanger. The aqueous ammonia solution (83% ammonia mass fraction) leaves the evaporator and directly enters the separator. In the separator, ammonia-rich steam is directly sent to the turbine and the dilute solution enters the recuperator. The ammonia-rich vapor exits from turbine and is mixed with the dilute fluid passing through the recuperator. The mixed solution then enters a heat exchanger (recuperator) to exchange heat with the cold flow from the pump before entering to the condenser where it is condensed into saturated liquid. Cooling water available in the cement plant with an average temperature of 20 °C is used to cool Kalina cycle condenser.

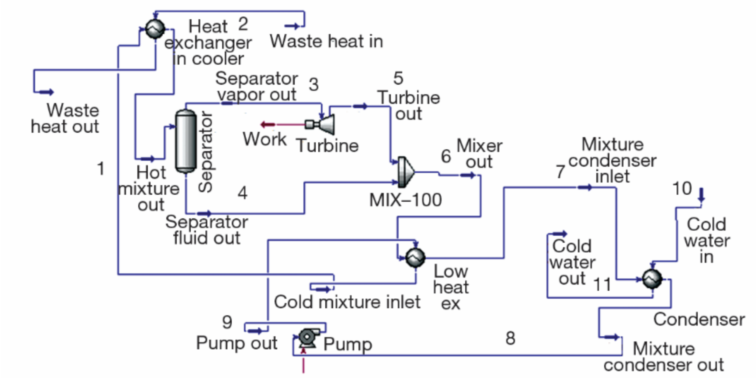

Figure 8. Kalina cycle integration and layout using ASPEN software

Kalina cycle thermodynamic analysis

For the analysis of the Kalina cycle, the following assumptions should be considered: steady-state operation of the cycle, saturated liquid at condenser outlet, saturated steam at the outlet of the turbine, condenser cooling water is at 20 °C, isenthalpic throttling process, complete liquid-vapor separation in the separator, pumps and turbines isentropic efficiency is 80%, negligible pressure and heat losses in the pipelines, heat exchanger efficiency of 80%, all equipment are adiabatic, and negligible changes in the kinetic and potential energies. Mass and energy balance is considered for each cycle component:

Evaporator: $$\dot{m}_1(h_2 - h_1) = \dot{m}_{\text{gas}}(h_{\text{out}} - h_{\text{in}}) \tag{13}$$

Separator: $$\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_4 h_4 \tag{14}$$

Recuperator: $$\dot{m}_6(h_7 - h_6) = \dot{m}_9(h_1 - h_9) \tag{15}$$

Turbine: $$W_{\text{T}} = \dot{m}_3(h_3 - h_5) \tag{16}$$

Pump: $$W_{\text{P}} = \dot{m}_8(h_9 - h_8) \tag{17}$$

Condenser: $$\dot{m}_7(h_8 - h_7) = \dot{m}_{\text{cw}} c_{p,\text{cw}}(T_{\text{cw,out}} - T_{\text{cw,in}}) \tag{18}$$

Mixer: $$\dot{m}_6 h_6 = \dot{m}_4 h_4 + \dot{m}_5 h_5 \tag{19}$$

The relationship between the mass-flow rate of the mixture and the amount of ammonia in the mixture related to fig. 8 are given by:

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_4 \tag{20}$$

$$\dot{m}_2 x_2 = \dot{m}_3 x_3 + \dot{m}_4 x_4 \tag{21}$$

$$\dot{m}_3 = \dot{m}_5 \tag{22}$$

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_4 \tag{23}$$

$$\dot{m}_6 x_6 = \dot{m}_5 x_5 + \dot{m}_4 x_4 \tag{24}$$

$$\dot{m}_6 = \dot{m}_7 \tag{25}$$

$$\dot{m}_7 = \dot{m}_8 \tag{26}$$

$$\dot{m}_9 = \dot{m}_1 \tag{27}$$

The performance of Kalina cycle coupled can be evaluated by estimating the thermal efficiency of the cycle:

$$\eta_{\text{I}} = \frac{W_{\text{T}} - W_{\text{P}}}{Q_{\text{c}}} \tag{28}$$

where W_{T}, W_{P}, and Q_{c} are turbine power, pump power, and heat rate input from the cooler exhaust gas. The second law efficiency of the cycle is calculated:

$$\eta_{\text{II}} = \frac{\eta_{\text{I}}}{\eta_{\max}} \tag{29}$$

Aspen simulation and model validation

Investigations of WHR in Al Arish cement plant has been carried out using ASPEN HYSIS software V8.4. The ASPEN HYSYS is used extensively in industry and academia for steady-state and dynamic simulation, process design, performance modelling, and optimization. It includes tools for estimation of physical properties and liquid-vapor phase equilibrium, heat and material balances, and simulation of many types of chemical engineering equipment. It also provides built-in blocks simulating basic process components such as heat exchangers, separators, turbines, and pumps required for the modelling of the cycles. In the present study, shell and tube heat exchangers are used for WHR. The inlet hot gas is fed into the shell side and the NH_3-H_2O mixture flow inside the tubes. The inlet conditions for hot gas such as temperature, pressure, flow rate, and mass fractions of gas components are set based on site measurements during normal operation. The flow rate of NH_3-H_2O through heat exchangers and mass fraction are set based on recommended values from previous research. On the other hand, the hot gas outlet temperature from the heat exchanger for the preheater is set to be 200 °C which is necessary for raw mill process. Drum type separator with minimum separator inlet vapor quality of 5% is selected. For turbines, the isentropic and mechanical efficiency are assumed as 90%. The mechanical efficiency of the pump is assumed to be 80%. The condenser cooling water is set at 20 °C and ammonia vapor quality is set to zero at the condenser outlet. Detailed design parameters fed to ASPEN software are shown in tab. 3. The calculated heat exchanger heat transfer area, mass-flow rate in each component, pump and turbine power are obtained for each case as will be discussed and analyzed in the following sections.

The numerical model of Kalina cycle is validated by comparison with previously published data of a base model of Kalina cycle power plant in Husavik, Iceland [14, 17]. Figure 9 shows the model developed in [14, 17] for the Kalina power plant of Husavik using hot water from a geothermal heat source at 124 °C. The cycle has high and low temperature recuperators (HTR and LTR). The HTR is the main generator of Kalina cycle and LTR is used for pre-heating of NH_3-H_2O mixture using hot ammonia exiting from the turbine. The cycle parameters reported in [14, 17] and the values of mass flow rate for each component are used as input data for the present ASPEN simulation as shown in fig. 9. The condenser is cooled using water at 5 °C. The turbine inlet pressure is 32.3 bar, ammonia mass fraction of 82%, turbine isentropic efficiency of 90%, and pump isentropic efficiency of 80%. As shown in fig. 9, the present ASPEN simulation results are in good agreement with the results reported by Sirko [14]. The maximum difference in the values of temperature does not exceed 3.5% and is attributed as due to uncertainty in the value of HTR and LTR heat exchangers efficiency. However, the mass fraction and mass balance are in excellent agreement. The net power output from the cycle is estimated to be about 2.26 MW in close agreement with the value of 2.37 reported in [14].

Based on the validation of the present model, it can be used as an analysis tool for studying the integration of Kalina cycle in the cement plant. In the present study two design alternatives for Kalina cycle integration in the cement plant are proposed. In the first proposal, separate cycle is integrated with each heat recovery component. In the second proposal, WHR from different components are combined to drive a single Kalina cycle. The results of this analysis are presented in the following sections.

Table 3. Design parameters for separate Kalina cycles driven by separate and combined WHR from cooler, preheater, and kiln

Component	Parameter	Separate WHR from cooler	Separate WHR from preheater	Separate WHR from kiln	Combined WHR
WHR (evaporator)	Shell and tube				
	Temperature of inlet mixture fluid, [C]	60	60	50	60
	Inlet temperature of hot gas, [C]	254	315	314	254, 315, 314 cooler, preheater, kiln, respectively
	Outlet temperature of hot gas (calculated), [C]	96	200		111, 242.4, 129.2 cooler, preheater, kiln, respectively
	Heat exchanger arrangement	Counter-flow	Counter-flow		Counter-flow
	Ammonia mass fraction, [%]	83	83	83	83
	Mass flow rate of fluid mixture, [kg/s]	17	17	8	27
Separator	Drum				
	Minimum separator inlet vapor quality, [%]	5	5	5	5
Recuperator	Drum type				
Turbine					
	Type	Axial multistage condensation back pressure turbine [14]			
	Rated speed, [rpm]	8000	8000		
	Isentropic efficiency, [%]	90	90	90	90
	Mechanical efficiency, [%]	90	90	90	90
	Outlet pressure, [bar]	7	7	7	7
	Inlet pressure, [bar]	40	40	40	40
	Turbine inlet temperature (simulation result), [C]	151.8	144.4	103.4	242.4
	Minimum turbine outlet vapor quality, [%]	90	90	90	90
Condenser	Shell and tube type				
	Condenser cooling water inlet temperature, [C]	20	20	20	20
	Cooling water flow rate, [kgs^{-1}]	300	300	144	500
Pump	Pump efficiency, [%]	80	80	80	80
	Pump power (calculated), [kW]	106	106	53	53

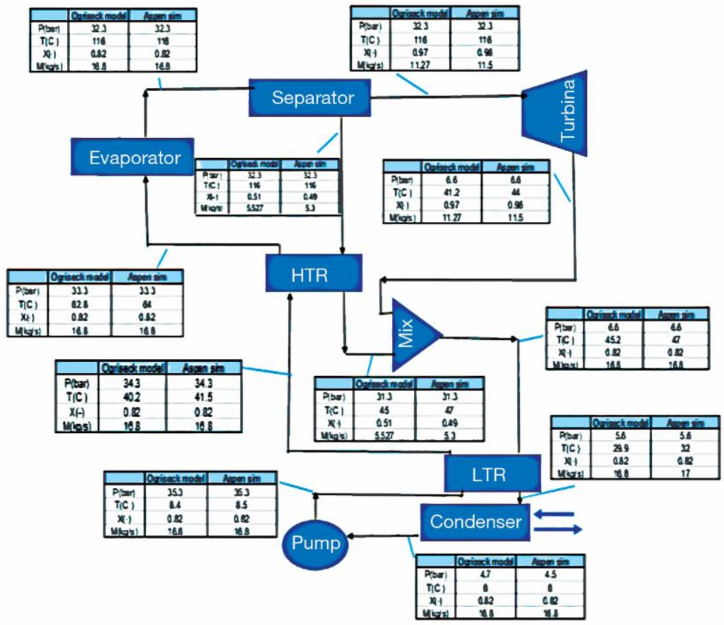

Figure 9. Kalina model validation, comparison with Sirko [14]

Kalina cycle driven by separate WHR from cooler, preheater, and kiln

Design parameters for separate Kalina cycles driven by separate WHR from cooler, preheater, and kiln are summarized in tab. 3. Design parameters shown in tab. 3 are selected based on a parametric analysis of the effect of turbine inlet pressure, mass-flow rate, and NH_3-H_2O concentration on Kalina cycle performance. The case of WHR from cooler is considered for this analysis. During a certain study, other design parameters are kept constant at the values shown in tab. 3. Figure 10 shows the effect of turbine inlet pressure on turbine power of the Kalina cycles. It can be observed that, the turbine power and cycle efficiency increase with the increase of turbine inlet pressure. Figure 11 shows that the turbine power and cycle efficiency increase with the increase of NH_3-H_2O concentration. In practice, 90% ammonia fraction is the break point of this behavior and the efficiency starts to decrease sharply [15]. Value of NH_3-H_2O concentration adopted in the present study is 83% as used in [14, 17]. On the other hand, as expected, the increase of turbine mass-flow rate results in the decrease of turbine inlet and outlet temperatures. However, high values of mass-flow rate would result in difficulty to in the condensation process using same water-cooling source from cooling tower and require a large condensation area. Also, low condensation pressures, may result in incomplete condensation at the end of the condenser and would cause damages to the circulation pump. Based on the previous results, design values of 40 bar, 7 bar, and 17 kg/s for turbine inlet pressure, outlet pressure, and mass-flow rate are adopted in the present study based on recommended turbine manufacturer data of axial multi-stage turbine and optimization study reported in [13].

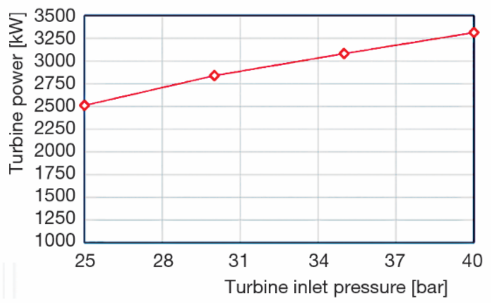

Figure 10. Effect of turbine inlet pressure on turbine power

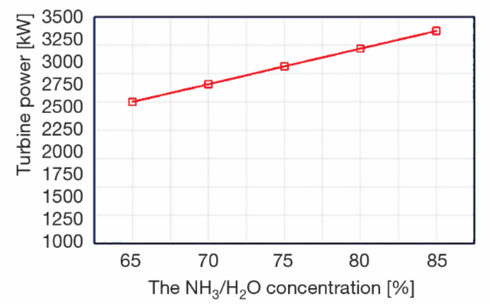

Figure 11. Effect of NH_3-H_2O concentration on turbine power

Design parameters for integration of Kalina cycle for WHR from cool, preheater, and kiln are shown in tab. 3. The ASPEN plus flow sheet for all the cases is shown in fig. 8. Figure 12 shows Kalina cycle simulation using ASPEN software in running mode for the case of WHR from the kiln surface. The results of ASPEN simulation are summarized below in tab. 4. The net power is the difference between turbine power and pump power. The values of turbine output power and cycle efficiency using WHR from the cooler and preheater are significantly higher than those obtained using the kiln.

Table 4. The ASPEN simulation results of Kalina cycles using separate WHR from cooler, preheater, and kiln

Case	Pump power [kW]	Turbine power [kW]	Net power [kW]	Cycle efficiency
Cooler WHR	106	3313	3207	32.4 %
Preheater WHR	106	3064	2958	28.5 5%
Kiln WHR	53	806	753	23.2%
Total	265	7130	6865	

Kalina cycle driven by combination of WHR from cooler, preheater, and kiln

In the previous sections, three separate Kalina cycles have been implemented to recover waste heat from the cooler, preheater, and kiln. In the present section, three heat exchangers are proposed to be implemented in series to recover the waste heat from the cooler, preheater, and kiln to heat NH_3-H_2O mixture before entering the separator and turbine of single Kalina cycle. Figure 13 shows the configuration of the proposed Kalina cycle. Design parameters for the proposed system is shown in tab. 3. Figure 14 shows simulation results of Kalina cycle driven by combination of waste heat from cooler, preheater, and kiln. Table 5 summarizes the performance parameters for combined WHR. As compared to separate WHR shown in tab. 4, combined WHR shows an improved thermal performance. The combined WHR also offers the advantage of a smaller number of system components as compared to separate cycles. An economic analysis would highlight the benefit of this issue.

Table 5. The ASPEN simulation results of Kalina cycle driven by combined WHR from cooler, preheater, and kiln

Pump power, [kW]	Turbine power, [kW]	Net power, [kW]	Cycle efficiency
185	7537	7352	30%

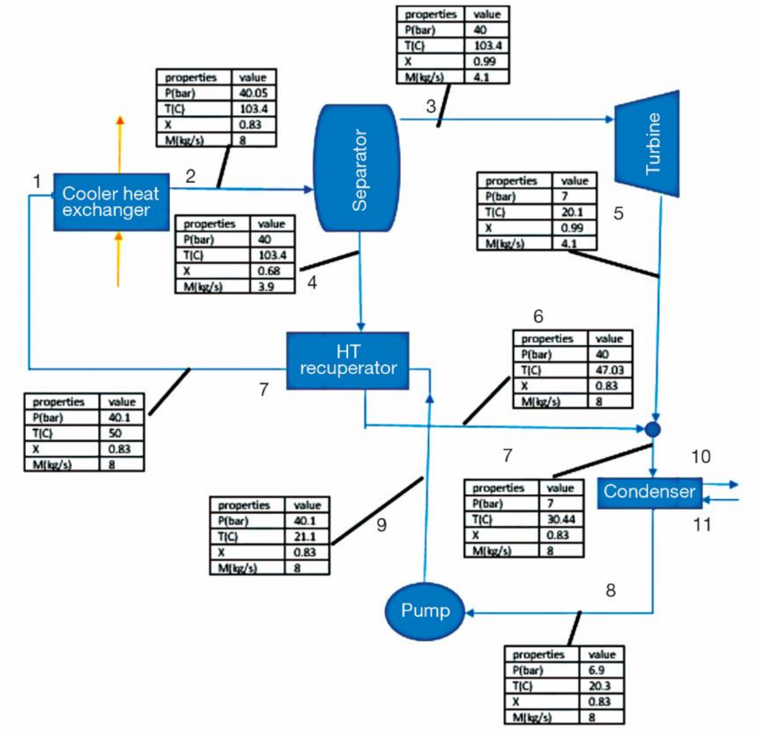

Figure 12. The ASPEN simulation results for WHR from kiln using Kalina cycle

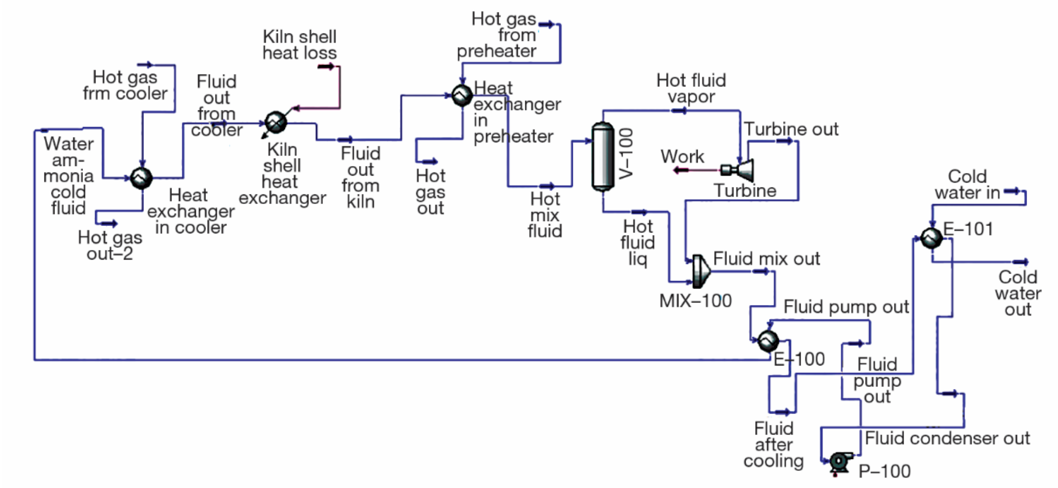

Figure 13. Kalina cycle driven by combination of waste heat from cooler, preheater, and kiln

Economic analysis and comparisons

Kalina cycle main cost include the cost of heat exchangers, recuperators, condensers, pumps, and turbines. The cost of heat exchanger, recuperator, and condenser are function of the surface area, A, for heat transfer which can be estimated as function of the logarithmic mean temperature difference, ΔT_m, and the overall heat transfer coefficient, U. The ASPEN software has been used to estimate the heat transfer surface areas for all components in Kalina cycle. Detailed calculations of required components surface areas, pump and turbine power, and cost analysis for separate and combined heat recovery have been performed. The cost function for each heat exchanger CHE is written [16]:

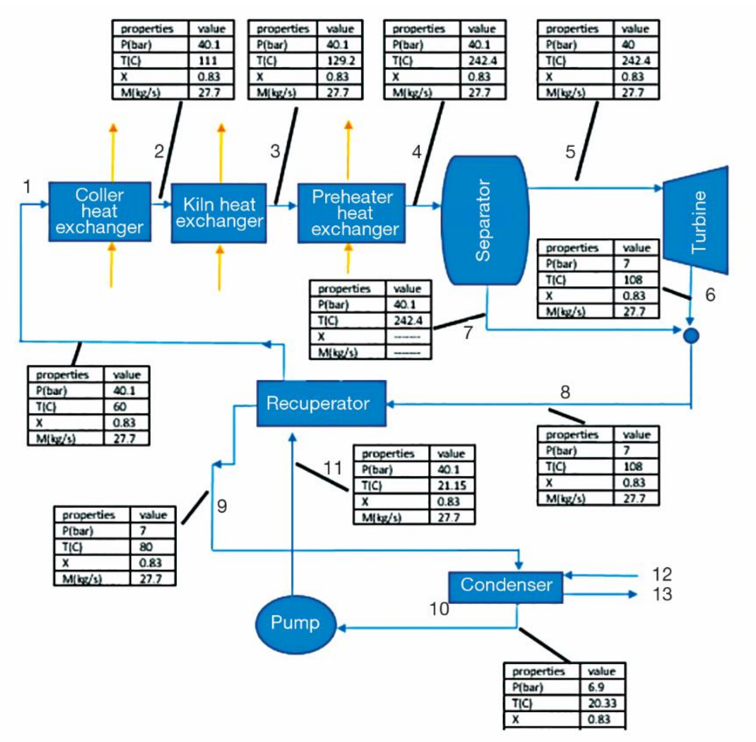

Figure 14. The ASPEN simulation results for combined WHR from cooler, preheater, and kiln using Kalina cycle

$$C_\mathrm{HE} = C_o(A)^n \tag{30}$$

where the base cost C_o is taken as 588 US\$/m^2 and n = 0.8 according to quotations from experienced professional manufacturing companies. The cost of pumps and turbines can be calculated as function of pump or turbine power [kW] using:

$$C_\mathrm{PT} = C_o(\mathrm{Power})^n \tag{31}$$

where the base cost, C_o, for the turbine is 4405 US\$/kW and 1120 US\$/kW for pump. The exponent n is taken as 0.7 and 0.8 for turbine and pump, respectively. The total cost is obtained by adding 20% of equipment cost for operation and maintenance and 20% for pipelines installations and infra-structure. Table 6 shows a summary of required components surface areas, pump and turbine power, and cost analysis for separate and combined WHR. The results show that a cost saving of about 23% with about 7% increase of total produced electric power have been obtained using Kalina cycle in combined WHR as compared to separate WHR design. Considering the cost of 1 kWh in Egypt, the payback periods for separate and combined WHR are 30, and 21 months, respectively.

Table 6. Comparison of heat transfer surface area, pump and turbine power, and cost analysis for combined and separate WHR

Case	Component	Cooler WHR	Preheater WHR	Kiln WHR	Total
Separate WHR	WHR exchanger area, [m^2]	1500	1378	338	3216
	Condenser area, [m^2]	3102	2880	707	6689
	Recuperator area, [m^2]	462	428	105	995
	Total heat transfer surface area, [m^2]				10900
	Pump power, [kW]	106	106	53	265
	Turbine power, [kW]	3313	3064	753	7130
	Total cost, [$]				9059046
Combined WHR	WHR Exchanger area, [m^2]	1850	1690	406	3946
	Condenser area, [m^2]	4411			4411
	Recuperator area, [m^2]	646			646
	Total heat transfer surface area				8273
	Pump power, [kW]		185		
	Turbine power, [kW]		7537		
	Total cost, [$]				6959720

Conclusions

Detailed waste heat analysis and recovery from a typical cement plant using Kalina cycle have been carried out using ASPEN software. The annual heat losses from the kiln surface, preheater, and the cooler are estimated as 79.23 GWh, 44.32 GWh, and 43.6 GWh at average temperatures of about 314 °C, 314 °C, and 254 °C, respectively. The present analysis indicates WHR for power generation with a maximum efficiency of 44% to 50% can be integrated with the cement plant.

Two design schemes for Kalina cycle integration in the cement plant using separate and combined WHR from the kiln surface, cooler, and preheater have been investigated. The design parameters for each configuration have been determined following a parametric study for the effect of turbine inlet pressure, mass-flow rate, and NH_3-H_2O concentration. The efficiency of the Kalina cycle increases as the ammonia concentration at the evaporator outlet increases and the turbine inlet pressure increases.

The results show that, for separate WHR, turbine output electric power from cooler, preheater and kiln shell are 3.31 MW, 3.06 MW, and 753 kW, respectively with total net output power of approximately 6.865 MW. Values of the cycle efficiency are 32.4%, 28.55%, and 23.2% for WHR from cooler, preheater, and kiln, respectively. The low efficiency of WHR from the kiln is attributed to the use or secondary shell with limitations on surface heat transfer due to mechanical parts rotation and maintenance requirements as well as low convection heat transfer.

The value of net power output using combined WHR is about 7.35 MW as compared to 6.86 using separate WHR design. A cost saving of about 23% with about 7% increasing of total produced electricity power have been obtained using Kalina cycle in combined WHR as compared to separate WHR design.

Nomenclature

A – area, [m^2]
A_k – kiln surface area, [m^2]
C_{HE} – cost for each heat exchanger, [US$]
C_o – base cost function, [US$/m^2]
c_p – specific heat, [Jkg^{-1}K^{-1}]
C_{PT} – cost of pumps and turbines, [US$]
h_c – convection heat transfer coefficient, [Wm^{-2}K^{-1}]
j – number of kiln length
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
Q_{conv} – convection heat losses, [kW]
Q_r – radiation heat losses, [kW]
T – preheater temperature, [C]
U – overall heat transfer coefficient, [Wm^{-2}K^{-1}]
$\dot{V}_i$ – volume flow rate, [m^{3}s^{-1}]
W_{out} – turbine output power, [kW]
W_{pump} – electrical power needed for pump, [kW]
x – NH_3-H_2O concentration

Greek symbols

ε – emissivity, [–]
η – Kalina cycle efficiency, [–]
η_{max} – maximum possible efficiency of WHR, [–]
ρ – density, [kgm^{-3}]
σ – Stefan Boltzmann constant, [Wm^{-2}K^{-4}]

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Paper submitted: April 18, 2021
Paper revised: May 26, 2021
Paper accepted: May 28, 2021