

INVESTIGATION OF USING KALINA CYCLE FOR WASTE HEAT RECOVERY IN A CEMENT PLANT

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ABSTRACT

The cement industry is considered one of the most energy intensive industrial processes in the world. The 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. Waste heat recovery is a way to reduce the total power consumption for the cement production process by using a heat recovery system to generate electrical energy with no more fuel or electrical power consumption. In a typical cement plant, 25% of the total energy is electrical energy, and 75% is thermal energy. About 35-40% of the total process heat is lost through waste heat streams. In a cement plant, 26% of heat input is lost through the kiln, preheater surfaces, convection from the kiln, and preheaters. This article reports on waste heat recovery from a typical cement plant in Egypt. Measurements and analyses 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, 44.32, and 43.6 GWh at average temperatures of about 314, 314, and 254°C, respectively. Analysis and optimization of using the Kalina cycle for Waste Heat Recovery (WHR) from the kiln shell, cooler, and preheater to produce electricity have been carried out using ASPEN software. A parametric study has been carried out to determine the design parameters for the Kalina cycle including turbine inlet pressure, mass flow rate, and ammonia water concentration. Two design alternatives have been investigated using separate and combined WHR from the kiln, cooler, and preheater. The value of net power output using combined WHR is about 7.35 MW as compared to 6.86 using a separate WHR design with a total cost saving of about 23%.

Keywords: Kalina cycle, waste heat recovery, ammonia-water mixture, Cement industry, heat loss.

دراسة استخدام دورة كالينا لاستعادة الحرارة المفقودة في مصنع الأسمنت

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الملخص:

تعتبر صناعة الأسمنت من أكثر العمليات الصناعية كثافة في استخدام الطاقة في العالم. يبلغ متوسط تكلفة الطاقة حوالي ٥٥٪ من التكلفة الإجمالية لإنتاج الأسمنت. ترجع تكلفة الطاقة الهائلة إلى كل من استهلاك الحرارة في عمليات الفرن واستهلاك الطاقة الكهربائية للعمليات المختلفة لعمليات الطحن للمواد الخام والتبريد عن طريق المراوح نقل الخامات. يعد استرداد الحرارة المهذرة طريقة لتقليل إجمالي استهلاك الطاقة لعملية إنتاج الأسمنت باستخدام نظام استرداد الحرارة لتوليد الطاقة الكهربائية دون مزيد من استهلاك الوقود أو الطاقة الكهربائية. في مصنع الأسمنت النموذجي، يتم فقدان حوالي ٣٥-٤٠٪ من إجمالي الطاقة الحرارية خلال تيارات الحرارة المهذرة. يفقد الغبار، وتصريف الكلنكر، والإشعاع من الفرن، وأسطح التسخين المسبق، والحمل الحراري من الفرن، وهذا يمثل ما يقرب من ٢٦٪ من مدخلات الحرارة.

نتناول في هذا البحث طرق استرداد الحرارة المهذرة من مصنع أسمنت نموذجي في مصر عن طريق دورة كالينا. تم إجراء القياسات والتحليلات لتحديد الحرارة المفقودة من المراحل المختلفة لخطوط تصنيع الأسمنت. تقدر الخسائر الحرارية السنوية من سطح الفرن والتسخين المسبق والمبرد بـ ٧٩,٢٣ و ٤٤,٣٢ و ٤٣,٦ جيجاوات ساعة بمتوسط درجات حرارة حوالي ٣١٤ و ٣١٤ و ٢٥٤ درجة مئوية على التوالي. تم إجراء تحليل وتحسين استخدام دورة كالينا لاستعادة الحرارة المفقودة من الفرن والمبرد والتسخين المسبق لإنتاج الكهرباء باستخدام برنامج أسبين تم إجراء دراسة بارامترية لتحديد معايير تصميم دورة كالينا بما في ذلك ضغط مدخل التوربينات، ومعدل التدفق الكتلي، وتركيز نسبة الماء والأمونيا. تم فحص بديلين للتصميم لاستعادة الحرارة المفقودة باستخدام برنامج أسبين من خلال فصل ودمج الطريقتين وتطبيقهم في مرحلة الفرن والمبرد والتسخين المسبق. تبلغ قيمة صافي خرج الطاقة باستخدام مجتمعة حوالي ٧,٣٥ ميغاوات مقارنة بـ ٦,٨٦ باستخدام تصميم منفصل لتوليد الطاقة مع توفير إجمالي في التكلفة يبلغ حوالي ٢٣٪.

الكلمات المفتاحية: دورة كالينا، استعادة الحرارة المفقودة، خليط الماء والأمونيا، صناعة الأسمنت، فقد الحرارة

1. INTRODUCTION

The cement industry is considered one of the most energy intensive processes in the world. The energy cost average is about 55% of the total cost of cement production and this is due to both heat consumption in kiln operations and electrical power consumption for different operations [1]. Waste heat recovery is considered to reduce the total power consumption in the cement production process by using a heat recovery system to generate electrical energy [2]. In a cement plant, 75% of the total energy is thermal energy and 25% is electrical energy. About 35-40% of the total process heat is lost through waste heat streams. Approximately 26% of heat input is lost by dust, clinker discharge, and radiation from the kiln, preheater surfaces, convection from the kiln, and preheaters [3].

In cement plants, all the output gases from rotary kilns, pre-heater, and calciners are used to heat up feed material. In cement plants, we have three points which can be used for waste heat recovery system. The first point is the exhaust gas temperature of pre-heater which is about 300 – 350 °C in the case of 5 – 6 stages. The second point in the cooler where the clinker temperature coming out from the kiln about 1000 °C. The clinker is cooled to 100-120 °C using ambient air and this generates hot air of about 260-300 °C. The third point is through kiln shell where the hot gases can reach temperatures more than 300 °C [4]. An energy audit analysis of a typical cement plant showed that the major heat loss sources include the kiln exhaust (19.15% of total input) and cooler exhaust (5.61% of total input). Heat recovery typically depends on, the flow rate of exhaust gases and their temperature. For 5000 ton/day of kiln operation, the expected power generation was estimated to be approximately 6-9 MW [5].

Kalina cycle has been considered one of the most efficient power cycles for low-grade waste heat recovery [6]. The Kalina cycle has been considered one of the efficient systems for low-grade heat power cycle generation since various modifications. Recently, the Kalina power

cycle has attracted much attention. Kyoung, Hoon Kim [7] studies show that a cogeneration cycle of power and absorption refrigeration based on the Kalina cycle system is proposed. cycle combines a Kalina cycle and aqua-ammonia absorption refrigeration cycle with a once-through configuration. Compared to the stand-alone Kalina cycle, the results showed significantly higher energy efficiency about 60% more than without the use of rectifier, superheater, or sub-cooler system. Parametric analysis approved that the ammonia fraction, separator pressure and source temperature have a higher impact on the system performance including fluid mass flow rates, power generation, energy efficiencies, and optimum ammonia fraction for the maximum energy efficiency, Kolar Deepak [8] developed Kalina basic cycle which has practical advantages over the Rankine cycle when we use low enthalpy geothermal heat source. The use of an additional superheater increases energy and exergy efficiencies by 2% and 3% respectively by increasing the enthalpy of the vapor mixture by adding additional heat from the source fluid at a temperature higher than the operating temperature. Ziya Sogut et al. [9] showed that the obtained heat recovery from the kiln for a cement plant in Turkey preserves 217.31 GJ per year, which is 51% of the overall heat of the cement plant process. An exergy analysis was carried out based on the operational data. According to the developed mathematical for waste heat recovery exchanger. It was found that heat recovery decreases the consumption of natural gas and coal by 62.62% and 51.55% and reduce CO₂ emissions by 1816.90 kg/h (natural gas) and 5901.94 kg/h (coal). Mehri Akbari [10] proposed a combination of Kalina cycle with an ammonia-water working fluid and a heat transformer cycle with lithium bromide–water working fluid. The production of pure water by the proposed combination is an additional advantage. The first and second law efficiencies of the proposed cycle are around 24% and 13% higher than the corresponding values for the Kalina cycle.

Widuramina Sameendranath [11] studied the available heat in the cement kiln exhaust gas in a Norwegian cement plant that produces 1.3 million tons of cement per year. A mass and energy balance were developed for the raw mill area, and process data was obtained from the plant process database, also, manually measurements for gas flow rates were used to calculate the available heat. The available heat can be used by a combination of low pressure (LP) steam generation and hot water generation. Waste heat which was calculated was 1.5–4.2 MW for LP steam generation and 2.2–5.8 MW for hot water generation. Hedman, Bruce [12] estimated that the integration of the Kalina cycle technology into a cement plant for waste heat recovery in clinker cooler exhaust gases and preheater exhaust gases would reduce overall power consumption by 10 - 20% in addition to the Market and Supplier analysis study. FL Smith supplied a Kalina cycle waste heat recovery system to Star Cement Co. L.L.C. (Aditya Birla Group of Companies) for its cement plant in Ras Al Khaimah, United Arab Emirates. Based on their system it is expected 12% reduction of power from total power [13]. Research and development should focus mainly on the optimization of the cycle efficiency or the output power concerning the cycle configuration and the available working fluids. On the other hand, the size of the components or the selected conditions have rarely been taken into consideration. Parametric analysis and optimization of a Kalina cycle driven by solar energy have been reported by Wang et al. [14]. The exergy and economic analysis of Kalina cycle for low-temperature geothermal sources as a practical case study in Brazil by implementation of the cost function for each component in Kalina cycle such that heat exchanger, pump and turbine was represented by Carlos, Campos [15]. This study can be considered as a preliminary reference guideline for implementation and adaptation in cement plants. The adaptation in cement plants needs more investigations and analysis that consider the differences in the available amount and temperature levels of waste heat sources in the plant. As a practical case study for a typical cement plant, the present article reports on waste heat recovery from the Al Arish Cement plant in Egypt. Measurements and analyses have been performed to determine the waste heat from three points of the cement manufacturing lines. Analysis and optimization of using the Kalina cycle to recover waste heat from the kiln shell, cooler, and preheater to produce electricity have been carried out using ASPEN software. It can be noted that it is the first time to study three points together in the same plant. Design parameters for the flow rate of Ammonia-Water mixture through the heat exchanger of kiln shell, cooler, and preheater shall be determined. Also, this study investigates the effects of turbine inlet pressure, ammonia concentration, and the evaporator exit temperature on Kalina cycle performance.

The present article is organized as follows. Section 2 provides plant description and waste heat analysis. Section 3 reports on waste heat analysis and feasibility from the cement plant. Section 4 is devoted to Kalina cycle integration, thermodynamic analysis, modeling, and ASPEN simulation. Sections 5, and 6 report on the design parameters and discuss the results of separate and combined WHR from cooler, preheater, and kiln. Economic study and comparisons are presented in Section 7.

2. 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 4 production lines with an average capacity of 5800 tons/day. The plant started production with two lines in 2010 then added 2 lines in 2016. Table 1 shows the technical data of major plant components.

Figures 1 to 3 show the flow diagram for the preheater, kiln area, and cooler area. Also, the proposed positions for waste heat recovery (WHR) are indicated in these figures. WHR from the preheater is proposed to be located to receive hot gases from cyclone a before raw mill process and ID fan. For the kiln, WHR is proposed to be located around the kiln shell to collect heat loss by radiation and convection from the kiln shell using a secondary shell and insulation from ambient air. For the cooler area, WHR receives waste hot gas from the cooler before entering the filter then to the stack.

3. 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 starts 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 is 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 OC and increases gradually by going down to the next step of cyclones to reach about 800-900 OC at kiln inlet.

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 also at the calciner part of the preheater by using 4 burners. Typical fuels used in the plant include heavy oil, natural gas, coal, or a mixture of alternative fuels. The rotary kiln is made out of a steel shell tube with the number of sections welded together and is inclined to help material flow to the next processes in the cooler. It has a layer of refractory bricks to withstand high operating temperature which may reach about 1500 OC during the calcination process. The kiln outer steel shell is exposed to the ambient and can reach a temperature of about 300-400 OC. The three live rings which support the rotary kiln rotates are called "tires". They are constructed from special casted steel and are loosely but guided. Also, they rotate on the supporting rollers (two rollers for each tire) and carry the heavy weight of the kiln. The kiln shell loses large amounts of heat by radiation and convection to the ambient. Additionally, the 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 a clinker cooler, see Fig. 3. They move with special speed on grates cooled by several external air fans. Hot gas with clinker dust with a temperature of about 250 – 350 OC leaves the cooler to the filtering stage by using the big centrifugal fan and then move to the stack.

The major waste heat recovery 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.

3.1 WASTE HEAT FROM ROTARY KILN

Convection and radiation losses from the rotary kiln surface are a 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 influences 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 infrared image techniques.

Table 1. El Arish Cement Company production lines technical data

Number of lines	Cement process	Preheater type	Preheater stage	Kiln average capacity ton/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	
Hydraulic grate cooler	Coal-heavy oil	900 kcal/kg	110 kwh/ton	345 day/year	Air cooling tower	Conditioning tower	

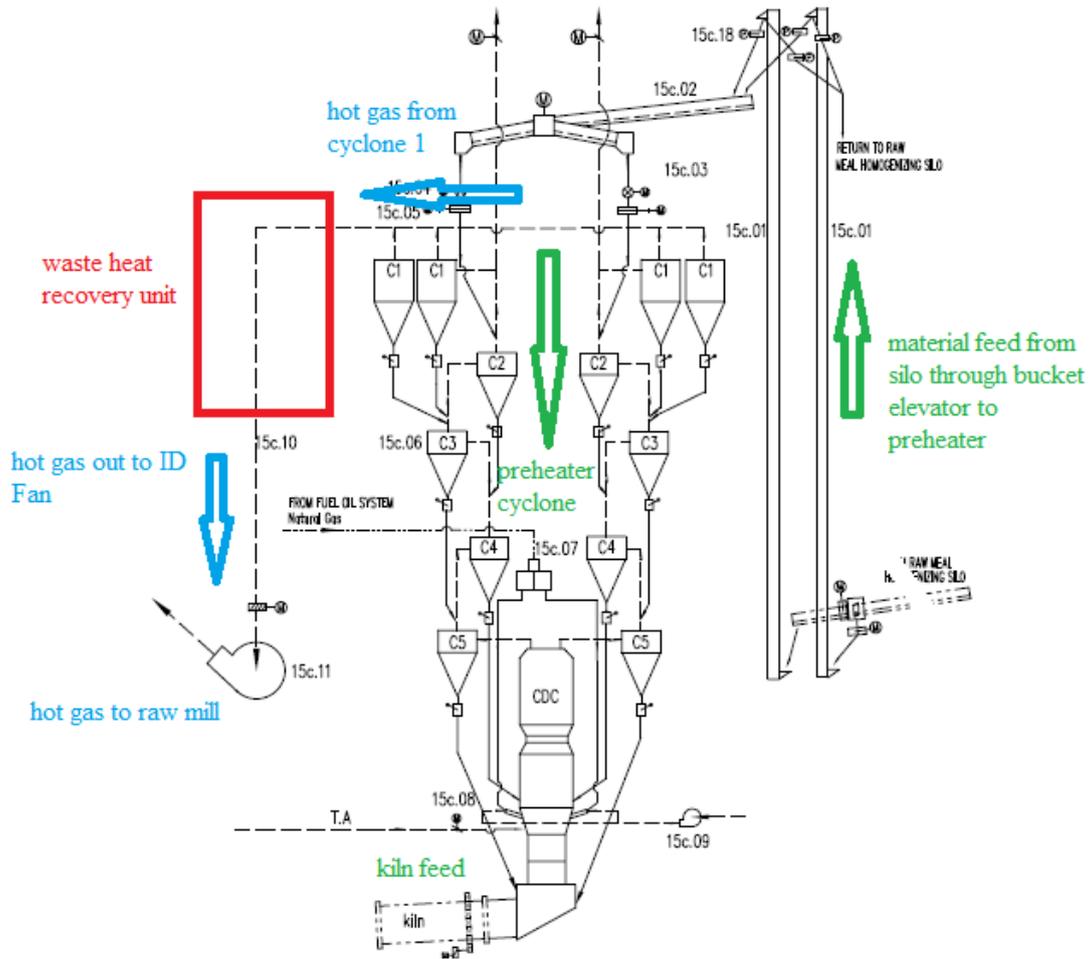


Figure 1. Flow diagram for preheater area and waste heat recovery system position

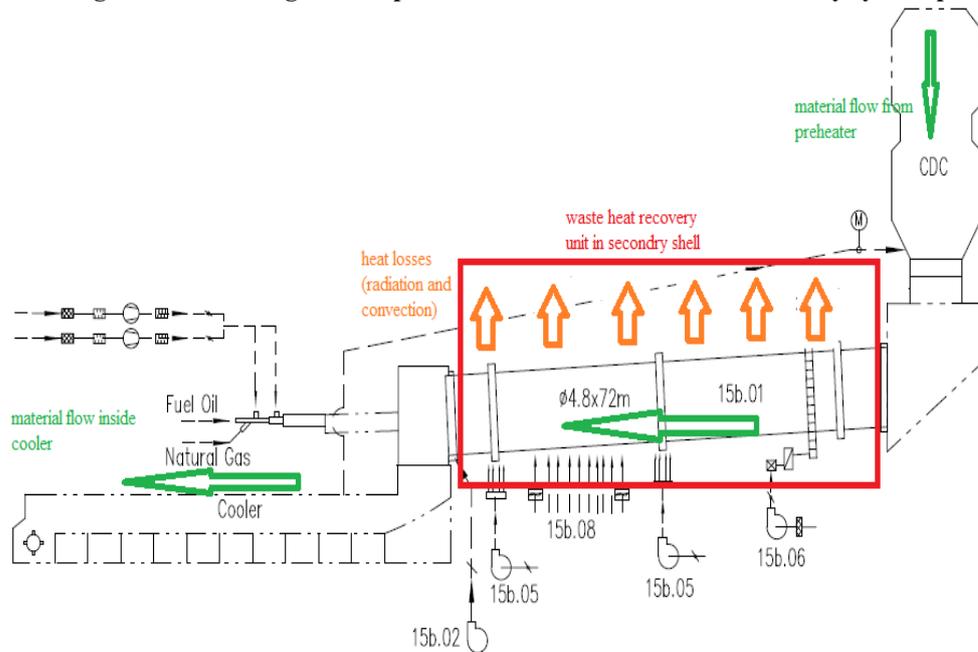


Figure 2. Flow diagram for kiln area and proposed waste heat recovery system position

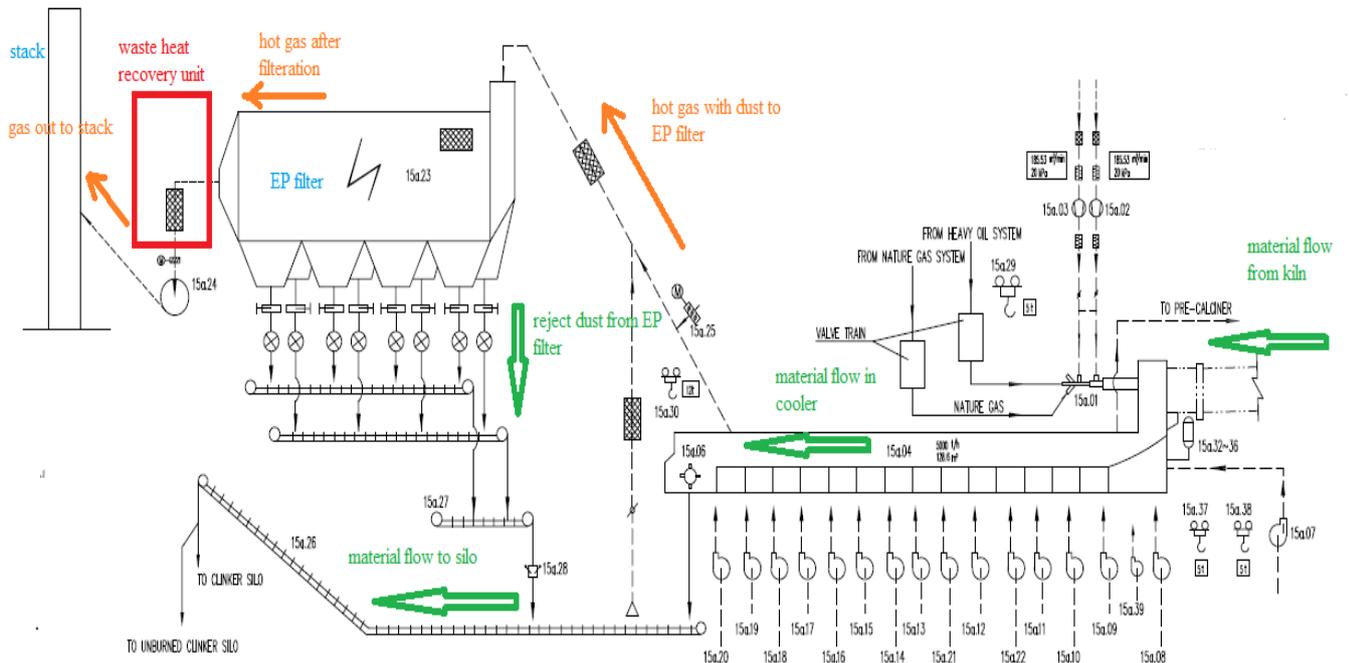


Figure 3. Flow diagram for the cooler area and proposed waste heat recovery system position

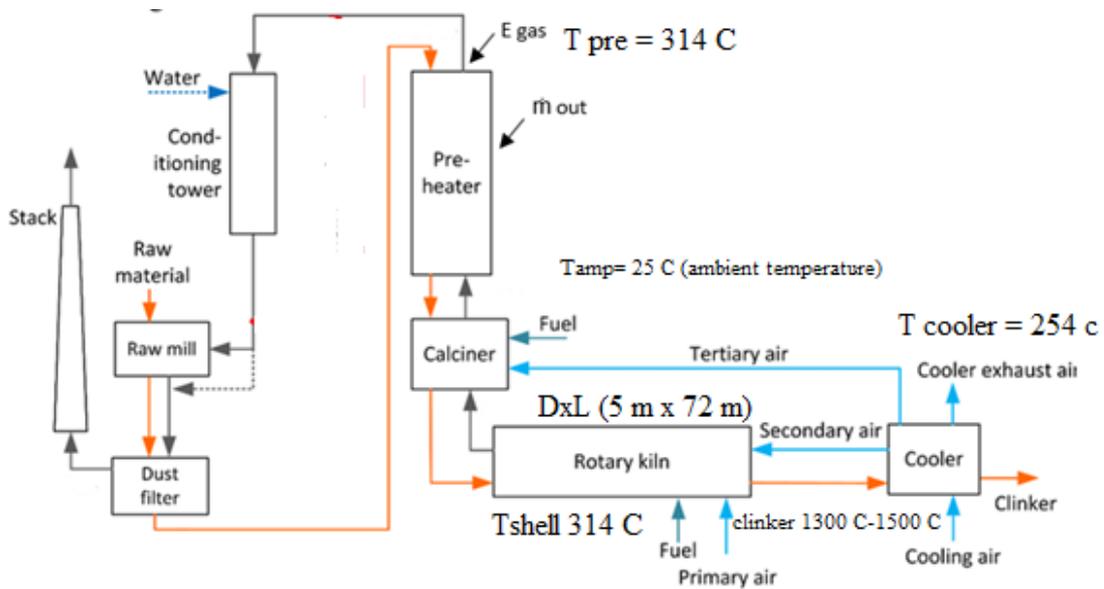


Figure 4. Schematic diagram showing major waste heat recovery sources from cement production lines

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

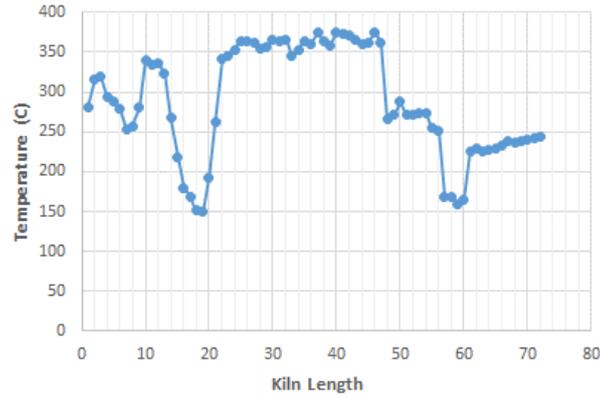


Figure 4. Variation of average measured temperature along kiln shell during normal operation

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

$$Q_{conv} = h_c A_k (T_k - T_a) \quad (1)$$

Where h_c is the convection heat transfer coefficient, A_k is the kiln surface area, T_k is the kiln surface temperature in kelvin, and T_a is the average atmospheric temperature in kelvin. The approximation given by Eq. (2) is used with accuracy throughout the cement industry to calculate the convection heat transfer coefficient h_c in W/m^2K which can be calculated using equation (2) [6].

$$h_c = 0.3D + 4 + 3.5\left(\frac{T_k}{100}\right) - 0.85\left(\frac{T_k}{100}\right)^2 + 0.076\left(\frac{T_k}{100}\right)^3 \quad (2)$$

The radiation heat losses from the kiln surface is calculated using Eq. (3) [6].

$$Q_{rk} = \varepsilon \sigma A (T_k^4 - T_a^4) \quad (3)$$

Where T_k is the kiln surface temperature in kelvin, $\sigma = 5.67 \times 10^{-8} W/m^2k^4$, A_k is the kiln surface area ($\pi D L$) estimated at $1130.4 m^2$ 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 surface areas 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 losses from each meter of the kiln and multiplying it with a fraction of operating hours (y) in a year as given by Eq. (4), and Eq. (5).

$$Q_{ckt} = 8760 \cdot y \sum_{i=1}^n Q_{ck,t} \quad (4)$$

$$Q_{rkt} = 8760 \cdot y \sum_{i=1}^n Q_{rk,t} \quad (5)$$

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

$$Q_{kilm} = Q_{ckt} + Q_{rkt} \quad (6)$$

3.2 WASTE HEAT FROM PREHEATER

El Arish cement plant has four lines with a kiln feed capacity of 5800 ton/day and preheater with double string design and 5 stages. After the hot gas from the 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 the drying, and lifting process. The exhaust gas from the preheater can be used for waste heat recovery without influencing the cement process with some limitations.

Figure 6 shows the variation of measured hot gas temperature from 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 the heat exchanger should have the same temperature as the conditioning tower exit gas. In the present study, a heat exchanger for waste heat recovery is proposed to be installed in parallel to the conditioning tower after the preheater, see Fig. 7.

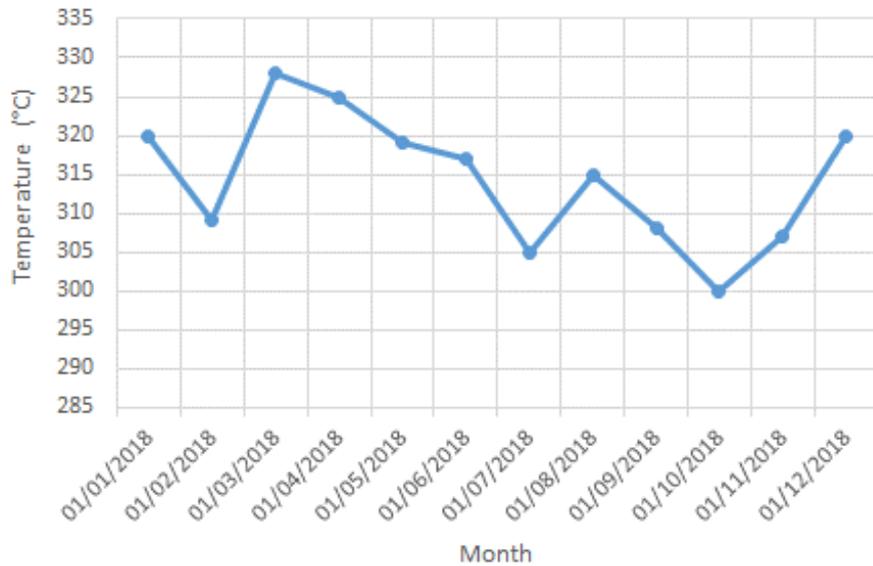


Figure 7. Annual variation of exhaust gas temperature from the preheater

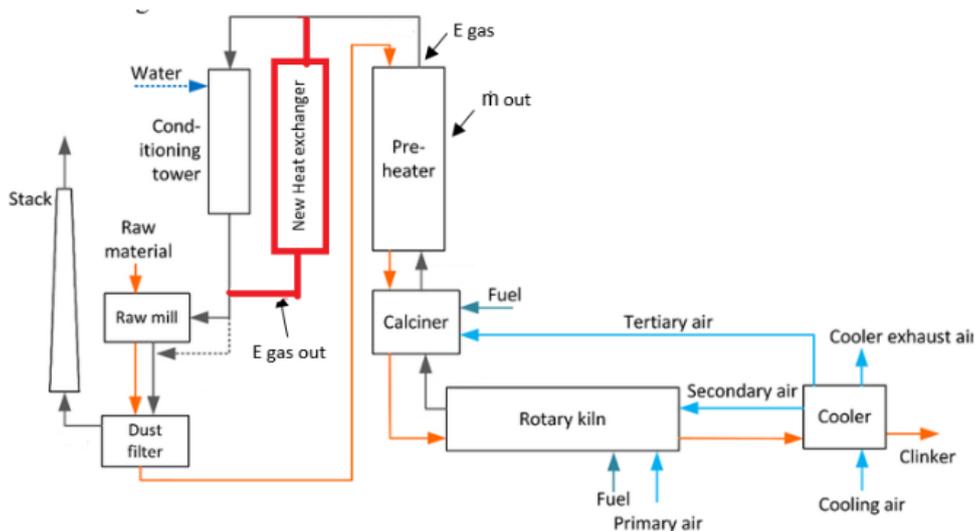


Figure 7. Proposed location of heat exchanger for heat recovery from preheater exhaust gas.

The waste heat recovered from the preheater exhaust gas (Q_p) can be calculated using the difference between the conditioning tower gas inlet and outlet parameters given by Eq. (7).

$$Q_p = \dot{m}_i c_{pi} T_i - \dot{m}_o c_{po} T_o \quad (7)$$

Where \dot{m} , T , c_p are the gas the mass flow rate, temperature in kelvin, and specific heat at the inlet (i) and outlet of cooling tower (o). 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], see Table 2.

Table 2. Volume fraction of preheater exhaust gas components before entering conditioning tower [6]

Gas component	CO ₂	H ₂ O	O ₂	N ₂
<i>x inlet</i>	0.2355	0.0910	0.0388	0.6350
<i>x outlet</i>	0.2263	0.1264	0.0373	0.6101

The mass flow rate of gas entering the cooling tower is calculated using Eq. (8)

$$\dot{m}_i = \dot{V}_i \cdot \sum_{k=CO_2;H_2O;N_2;O_2} \rho_{kt} x_{kt} \quad (8)$$

Where \dot{V}_i is the measured volume flow rate of gas at the 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}_w) using Eq. (9)

$$\dot{m}_o = \dot{V}_o \cdot \sum_{k=CO_2;H_2O;N_2;O_2} \rho_{ko} x_{ko} \quad (9) + \dot{m}_w$$

The water vapor mass added to the gas in the cooling tower is estimated based on the measured values of the water rate supplied to the conditioning tower. The total annual waste heat in MWh from the preheater exhaust gas is calculated using Eq. (10).

$$Q_{pt} = \sum_1^{8760} (\dot{m}_i c_{pi} T_i - \dot{m}_o c_{po} T_o) \quad (10)$$

3.3 WASTE HEAT FROM COOLER

The cooler waste hot gas which is vented to the atmosphere is considered to be waste heat from the system. Figure 8 shows the 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.

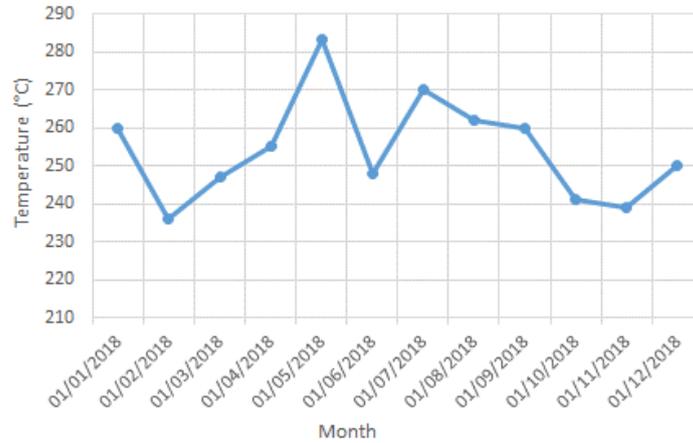


Figure 5. Waste hot gas average temperature from cooler

The total annual waste heat by hot gas from the cooler is calculated using Eq. (11).

$$Q_{ct} = \sum_1^{8760} \dot{V}_c c_{pc} (T_{co} - T_a) \tag{11}$$

Where \dot{V}_c is the volume flow rate of hot gas discharge from the cooler, T_{co} is the hot gas outlet temperature, and c_{pc} is the heat capacity of gas per unit volume $\text{kJ/m}^3 \text{K}$. According to Ulrich 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 3 shows the annual average energy loss from the kiln shell, preheater, and cooler. The largest source of heat loss is considered to be in the kiln shell and the clinker cooler gas.

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

Item	Availability	Energy consumption	$Q_{\text{kiln conv}}$	$Q_{\text{kiln rad}}$	$Q_{\text{hourly MWh}}$	$Q_{\text{Annual GWh/year}}$	Average temperature °C	Carnot η_{max}
Kiln Shell	95%	900 kcal/kg clinker	28.44 GWh/year	50.79 GWh/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		

3.4 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 the WHR power generation system increases with the increase of available heat temperature (T_H) and the decrease of the 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 by Eq. (12)

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

The quantity of heat determines the expected system power generation capacity. Taking the ambient temperature as the lower limit of minimum temperature (25 to 35 °C), $\eta_{max} = 44.4$ 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 CO₂, water vapor, and NOX. Condensation of water vapor in the exhaust in the presence of these elements may result in material corrosion of heat exchangers. This limitation is present only in the preheater and cooler exhaust gases. The kiln waste heat recovery system is not sensitive to this parameter.

4. KALINA CYCLE INTEGRATION

Kalina cycle uses a mixture of ammonia and water as a binary working fluid with different boiling temperatures. Therefore, contrary to the Rankine cycle and ORC, the temperature profile during boiling and condensation is not constant. This allows us to obtain good thermal matching with the waste heat source and cooling medium in the condenser. Several configurations of the Kalina cycle have been proposed depending on the end-use. The configuration of the Kalina cycle employed in the present study is shown in Fig.9. It consists of a turbine (TUR), a generator (GEN), a recuperator (RE), a condenser (CD), a pump (PU), a separator (SEP), a throttle valve (THV), a mixer (MX), and an evaporator (EV) with a waste heat stream as the heat source. This configuration is commonly used for low-temperature applications (120-400 C). Heat recovered from kiln, preheater, and cooler is used to convert ammonia-water mixture in the evaporator into a vapor state. The ammonia-water solution (with an ammonia mass fraction 83%) exits the evaporator and directly enters the separator, state 2. In the separator, the working fluid mixture is separated into an ammonia-rich vapor and a weak solution. The ammonia-rich vapor with an ammonia mass fraction of 83 % from the separator outlet is directly fed to the turbine. The weak solution leaves the separator as a saturated liquid state and passes to the recuperator. The ammonia-rich vapor after expansion through the turbine enters the mixing point, where it is mixed with the working fluid passing through the recuperator.

The mixed solution enters the recuperator, where heat is exchanged with the cold stream from the pump. The hot stream leaving the recuperator passes through the condenser where it becomes a saturated liquid. Cooling water available in the cement plant with an average temperature of 20 oC is used to cool the Kalina cycle condenser.

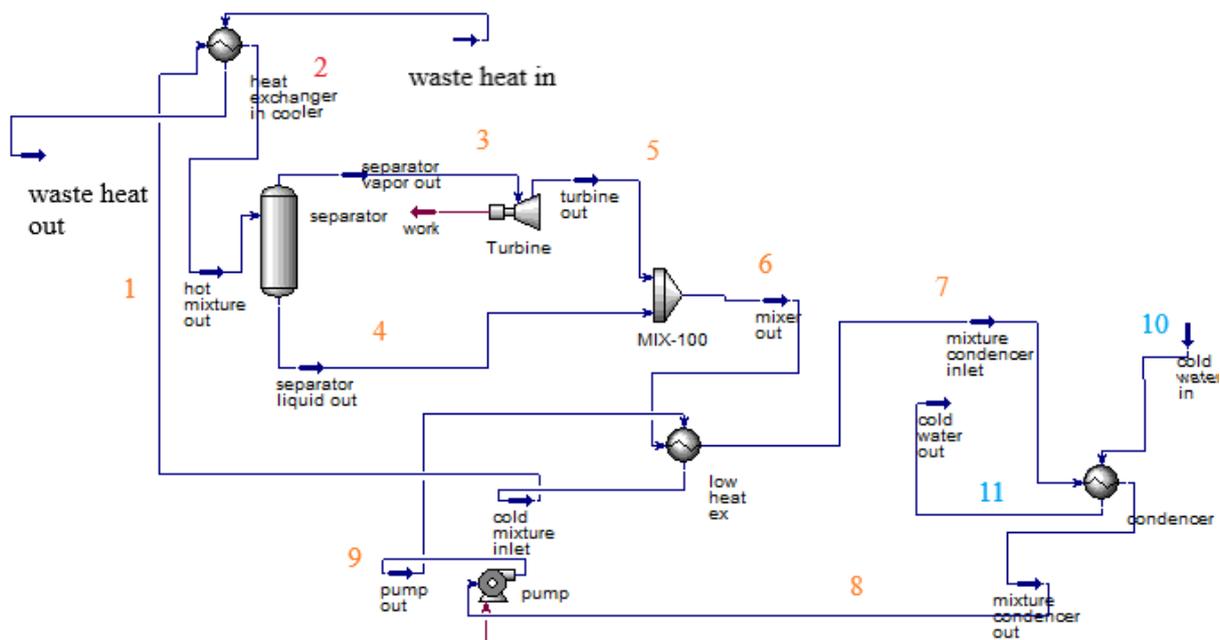


Figure 6. Kalina cycle integration and layout using Aspen software

4.1 KALINA CYCLE THERMODYNAMIC ANALYSIS

The following assumptions are considered for the analysis of the Kalina cycle: steady-state operation of the cycle, working fluid at the outlet of condenser is saturated liquid, working fluid at the outlet of the turbine is saturated vapor, the temperature of the water source from the cooling tower to the condenser inlet is 20 °C, throttling process is isenthalpic, separator completely separates the liquid and vapor, the isentropic efficiency of pump and turbine is 80 %, pressure losses and heat losses in pipes are neglected, the effectiveness of the heat exchanger is 80%, all the devices are adiabatic, the kinetic energy and potential energy changes in the devices are neglected. Mass and energy balance is considered for each cycle component, as follows:

$$\text{Evaporator: } \dot{m}_1 \cdot (h_2 - h_1) = \dot{m}_{gas} \cdot (h_{out} - h_{in}) \quad (12)$$

$$\text{Separator: } \dot{m}_2 \cdot h_2 = \dot{m}_3 \cdot h_3 + \dot{m}_4 \cdot h_4 \quad (13)$$

$$\text{Recuperator: } \dot{m}_6 \cdot (h_7 - h_6) = \dot{m}_9 \cdot (h_1 - h_9) \quad (14)$$

$$\text{Turbine: } W_{tur} = \dot{m}_3 \cdot (h_3 - h_5) \quad (15)$$

$$\text{Pump: } W_{pul} = \dot{m}_8 \cdot (h_9 - h_8) \quad (16)$$

$$\text{Condenser: } \dot{m}_7 \cdot (h_8 - h_7) = \dot{m}_{cw,cd} \cdot c_{p,cw} \cdot (T_{cw,out} - T_{cw,in}) \quad (17)$$

$$\text{Mixer: } \dot{m}_6 \cdot h_6 = \dot{m}_4 \cdot h_4 + \dot{m}_5 \cdot h_5 \quad (18)$$

The equations relating the mass flow rates for the mixture and the quantity of ammonia in the mixture for the streams related to Fig. 9 are given by:

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_4 \quad (20)$$

$$\dot{m}_2 x_2 = \dot{m}_3 x_3 + \dot{m}_4 x_4 \quad (21)$$

$$\dot{m}_3 = \dot{m}_5 \quad (22)$$

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_4 \quad (23)$$

$$\dot{m}_6 x_6 = \dot{m}_5 x_5 + \dot{m}_4 x_4 \quad (24)$$

$$\dot{m}_6 = \dot{m}_7 \quad (25)$$

$$\dot{m}_7 = \dot{m}_8 \quad (26)$$

$$\dot{m}_9 = \dot{m}_1 \quad (27)$$

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

$$\eta_I = (W_T - W_P) / Q_c \quad (28)$$

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 as:

$$\eta_{II} = \eta_I / \eta_{max} \quad (29)$$

4.2 ASPEN SIMULATION AND MODEL VALIDATION

Investigations of waste heat recovery in the Al Arish cement plant has been carried out using Aspen Hysis software V8.4. Aspen HYSYS is a chemical simulator software used to model and simulate chemical and thermal processes include mass balance, vapor-liquid equilibrium, energy balance, chemical kinetics, heat transfer, mass transfer, and pressure drop.

The numerical model of the Kalina cycle is validated by comparison with previously published data of a base model of the Kalina cycle using the Kalina power plant in Husavik, Iceland [15]. The cycle has LT and HT recuperators for the pre-heating of the ammonia-water mixture. The generated electricity is calculated as 2.2 MW. As shown in Figure 10, the present ASPEN simulation is in good agreement with the results reported by Ogriseck [16].

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

5. KALINA CYCLES DRIVEN BY SEPARATE WHR FROM COOLER, PREHEATER AND KILN

5.1 BASIC DESIGN PARAMETERS

Design parameters for the integration of the Kalina cycle for WHR from cooler, preheater, and kiln simulation using Aspen software for each case. We can notice that the values of turbine output power consequently cycle efficiency using WHR from the cooler and preheater are significantly higher than those obtained using the kiln. Fig 19 shows ASPEN simulation results for WHR from preheater using Kalina cycle with fluid mixture mass flow rate 17 kg/s.

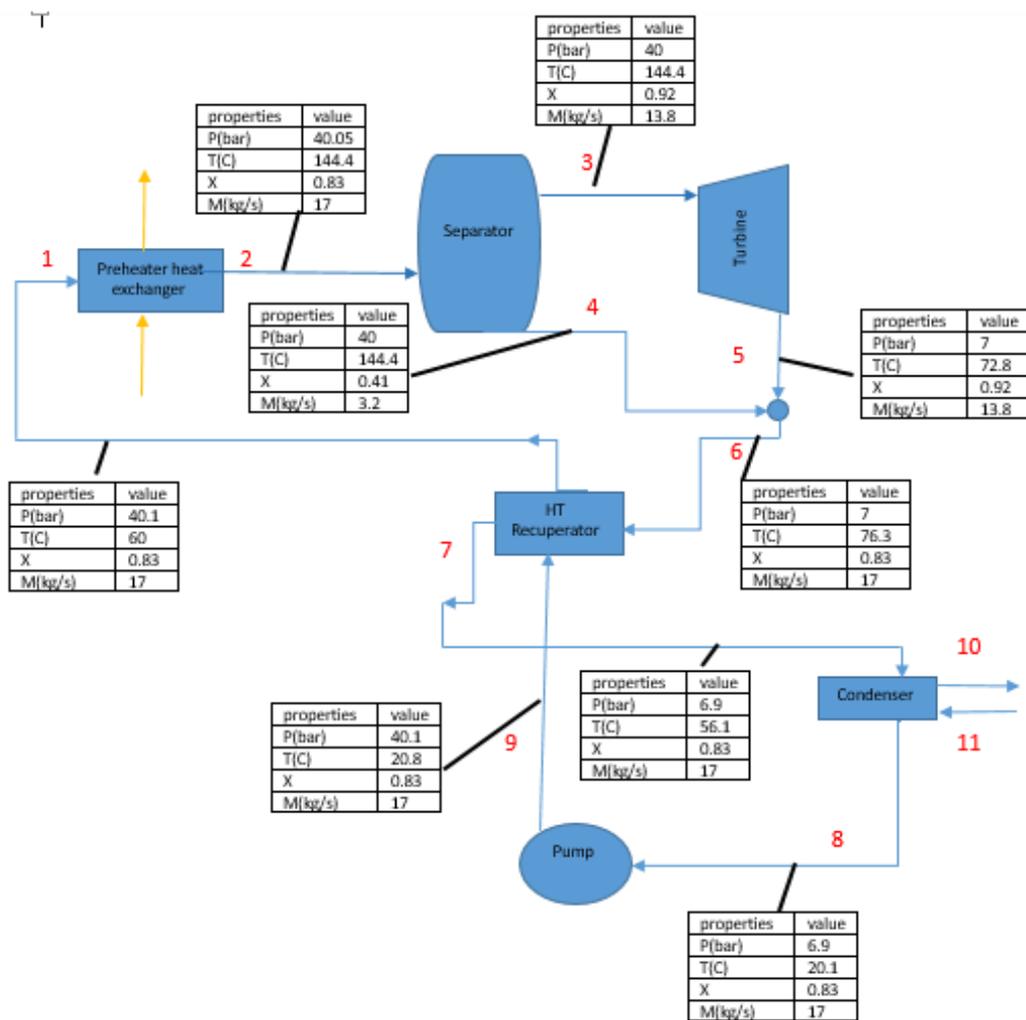


Figure 7: ASPEN simulation results for WHR from preheater using Kalina cycle.

simulation results for WHR from the kiln using Kalina cycle with fluid mixture mass flow rate 8 kg/s shown in Fig 11. We can notice that the available waste heat from the kiln which less than the cooler and preheater due to rotating mechanical equipment of the rotary kiln which effects on heat exchanger design consequently limitation of waste heat recovery area.

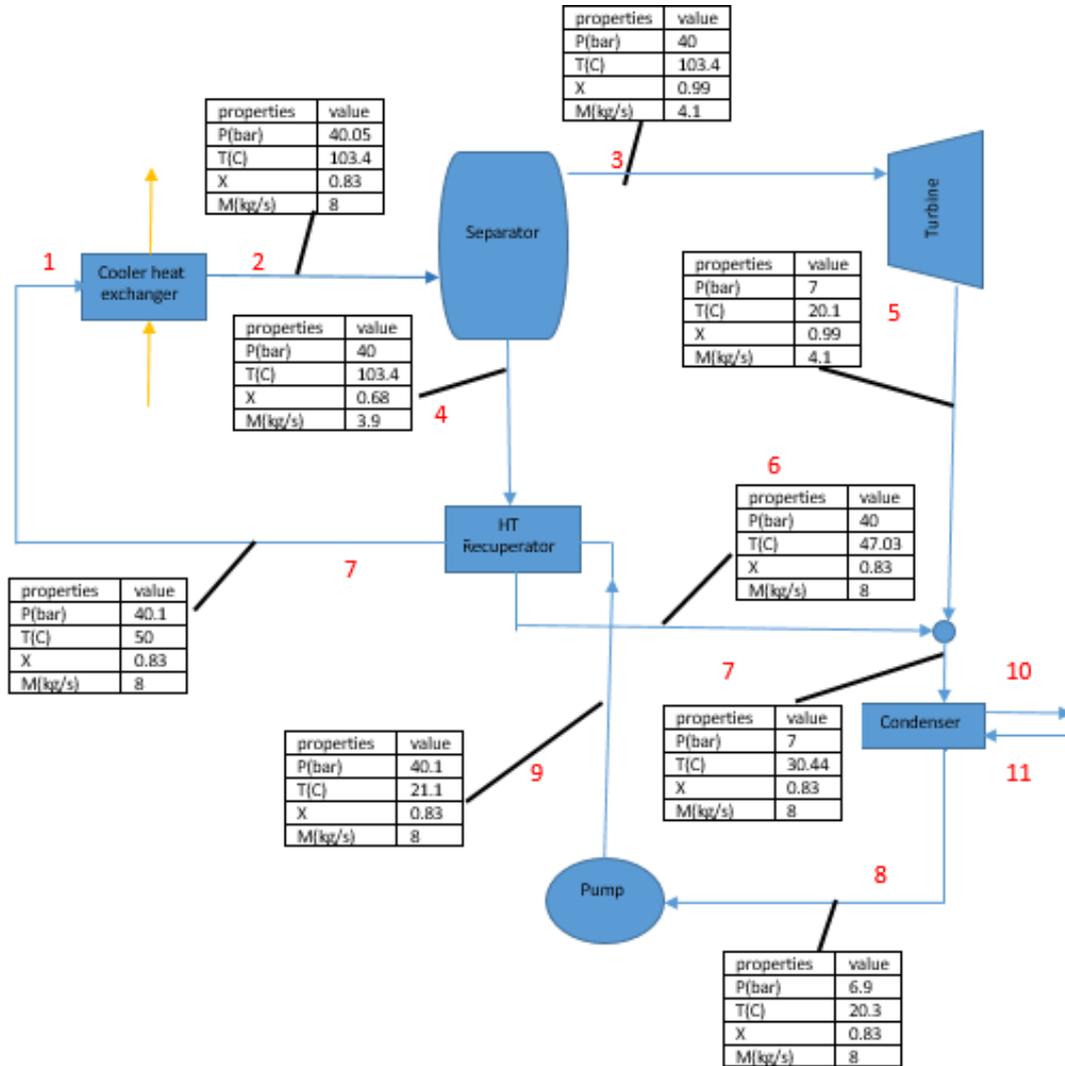


Figure 8: Aspen simulation results for WHR from kiln using Kalina cycle

5.2 EFFECTS OF KALINA CYCLE DESIGN PARAMETERS

Design parameters shown in Table 4 are selected based on parametric analysis of the effect of turbine inlet pressure, mass flow rate, ammonia water concentration on Kalina cycle performance. Kalina Model validation, comparison with Ogriseck [17] shows in Figure 10. 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 Table 4. Figure 11 shows the effect of turbine inlet pressure on the 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 12 shows that the turbine power and cycle efficiency increase with the increase of ammonia water concentration. In practice, 90% ammonia fraction is the breaking point of this behavior and the efficiency starts to decrease sharply [15]. On the other hand, as expected, the increase in 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 in the

condensation process using the same water cooling source from the 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 above results, design values of 40 bar and 17 kg/s for turbine inlet pressure and mass flow rate are adopted in the present study. The turbine outlet pressure of 7 bar is selected based on recommended turbine design.

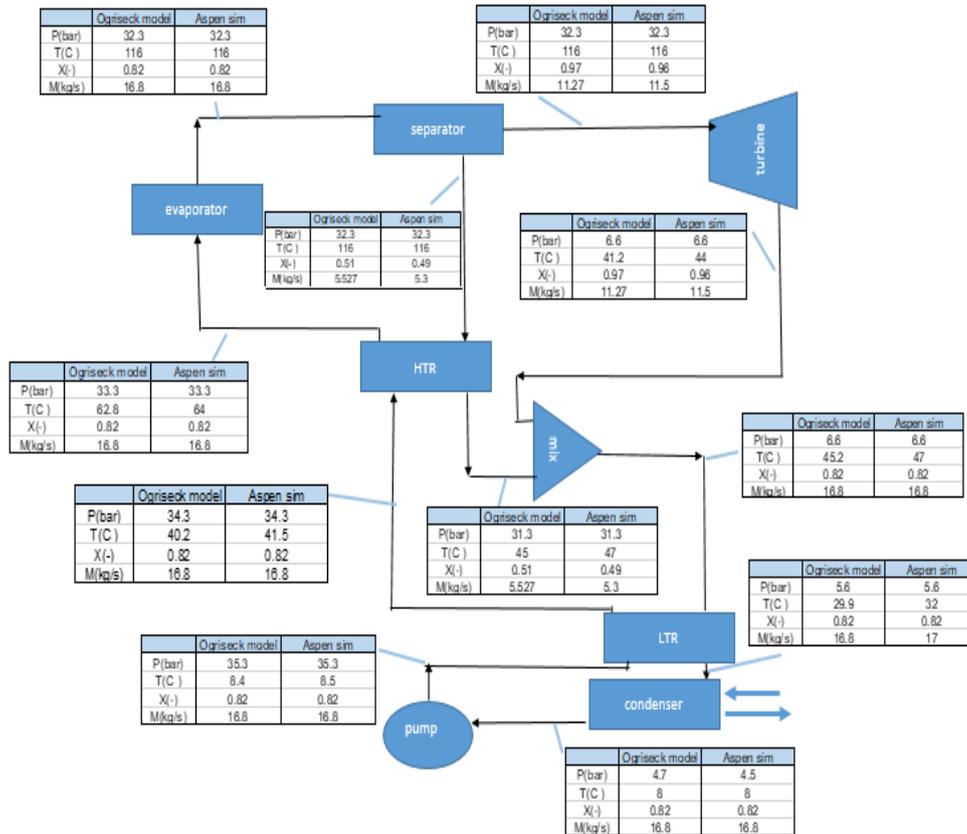


Figure 9. Kalina Model validation, comparison with Ogriseck [17].

We can notice from Table 4 that the values of turbine output power consequently cycle efficiency using WHR from the cooler and preheater are significantly higher than those obtained using the kiln. And when using combined WHR system we can obtain more power and higher efficiency due to reheating process which effect on fluid inlet turbine temperature after every stage also using only one separator and one pump for all system comparing with using three pumps and separator for separate Kalina single cycle.

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	60 C	60 C	50 C	60 C
	Inlet temperature of hot gas	254 C	314 C	314 C	254 C, 314 C, 314 C cooler, preheater, kiln, respectively
	Outlet temperature of hot gas (Calculated)	96 C	200 C		111 C, 242.4 C, 129.2 C 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	17 kg/s	17 kg/s	8 kg/s	27.7 kg/s
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	8000 rpm	8000 rpm		
	Isentropic efficiency	90%	90%	90%	90%
	Mechanical efficiency	90%	90%	90%	90%
	Outlet pressure	7 bar	7 bar	7 bar	7 bar
	Inlet pressure	40 bar	40 bar	40 bar	40 bar
	Turbine Inlet Temperature (simulation result)	151.8 C	144.4 C	103.4 C	242.4 C
	Minimum turbine outlet vapor quality	90%	90%	90%	90%
Condenser	Shell and tube type				
	Condenser cooling water inlet temperature	20 C	20 C	20 C	20 C
	Cooling water flow rate	300 kg/s	300 kg/s	144 kg/s	500 kg/s
Pump	Pump efficiency	80%	80%	80%	80%
	Pump power (Calculated)	106 (kW)	106 (kW)	53 (kW)	53 (kW)

Figure 13 shows the effect of turbine inlet pressure on the 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 14 shows that the turbine power and cycle efficiency increase with the increase of ammonia water which shows the increase of turbine power with ammonia water concentration increases. In practice, 90% ammonia fraction is the breaking point of this behavior and the efficiency starts to decrease sharply [15].

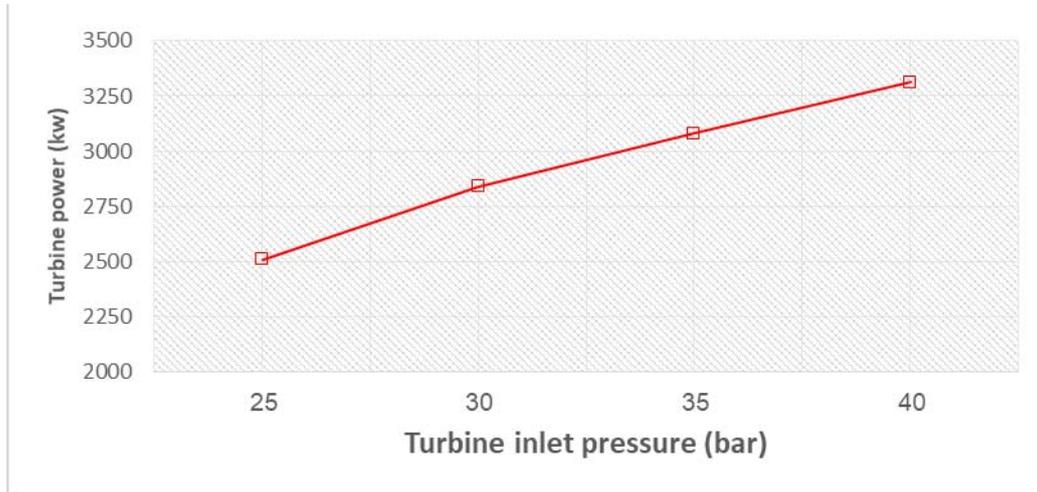


Figure 13. Effect of turbine inlet pressure on turbine power

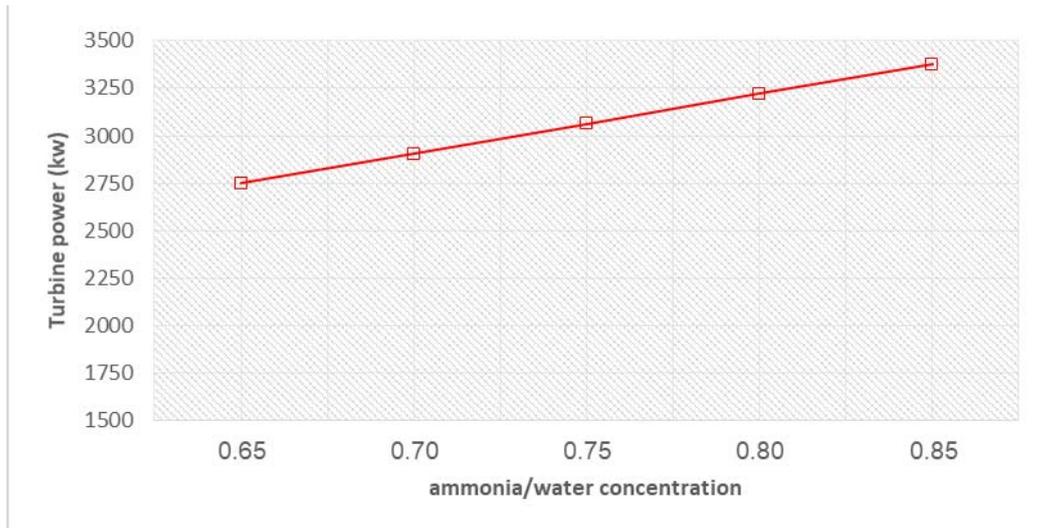


Figure 14. Effect of ammonia/water concentration on turbine power

5.3 PERFORMANCE OF KALINA CYCLES DRIVEN BY SEPARATE WHR FROM COOLER, PREHEATER AND KILN

Design parameters for the integration of the Kalina cycle for WHR from cool, preheater, and kiln are shown in Table 5. The ASPEN plus flow sheet for all the cases is shown in Fig. 9. Figure 13 shows the Kalina cycle simulation using Aspen software in running mode for the case of WHR from the kiln surface. The results of the ASPEN simulation are summarized below in Table 5. 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 5. 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	

6. 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 ammonia-water mixture before entering the separator and turbine of a single Kalina cycle. Figure 15 shows the configuration of the proposed Kalina cycle.

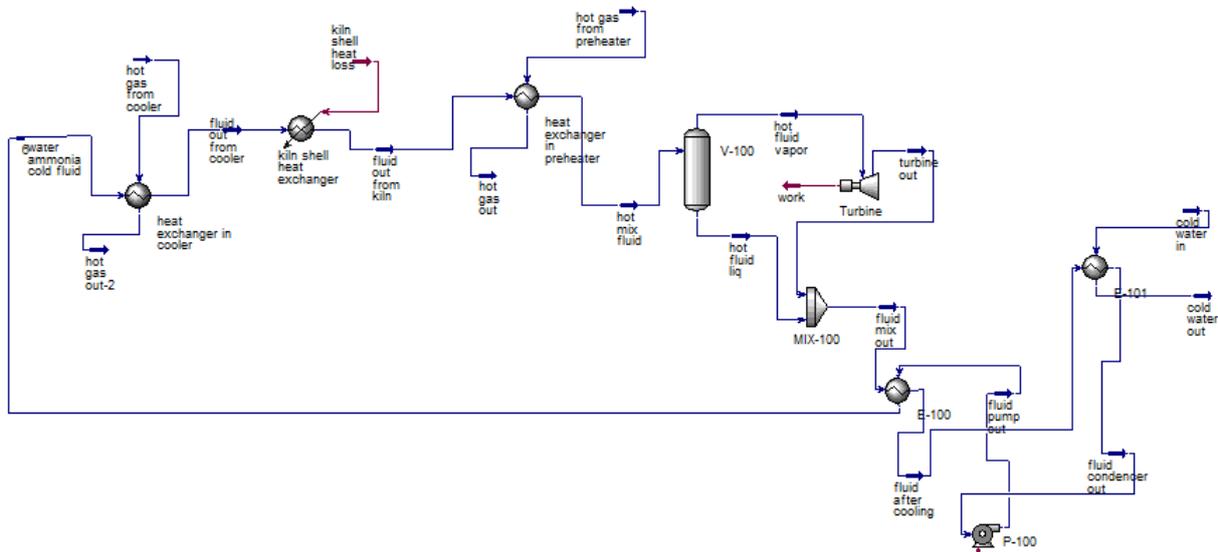


Figure 15. Kalina cycle driven by combination of waste heat from cooler, preheater, and kiln

Figure 16 shows simulation results of the Kalina cycle driven by a combination of waste heat from cooler, preheater, and kiln. The combined waste heat recovery also offers the advantage of a fewer number of system components as compared to separate cycles. An economic analysis would highlight the benefit of this issue.

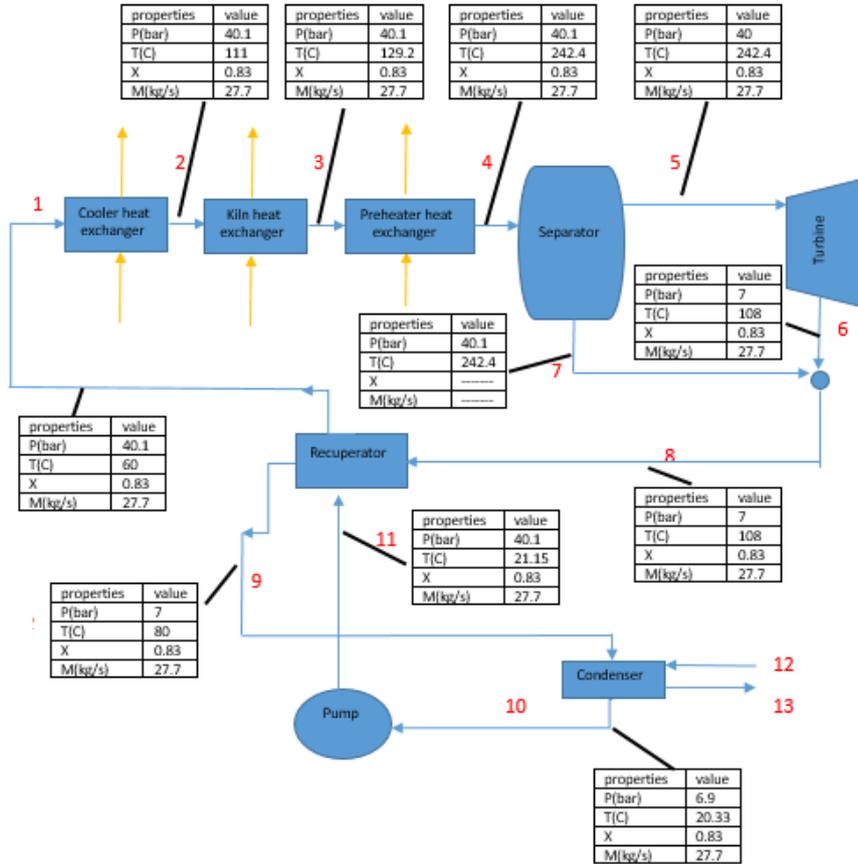


Figure 16. Aspen simulation results for combined WHR from cooler, preheater, and kiln using Kalina cycle

Table 6 summarizes the simulation results of Kalina cycle for combined WHR as compared to separate WHR using Eq (28) and Eq (29) which illustrate higher efficiency.

Table 6: 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%

7. ECONOMIC ANALYSIS AND COMPARISONS

Economic comparisons of separate and combined energy recovery using Kalina cycles are presented in this Section.

Kalina cycle's main cost includes the cost of heat exchangers, recuperators, condensers, pumps, and turbines. The cost of the heat exchanger, recuperator, and condenser are a function of the surface area (A) for heat transfer which can be estimated using Eq. (30) as a function of the logarithmic mean temperature difference ΔT_m and the overall heat transfer coefficient U.

$$q = U A \Delta T_m = U A \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln \frac{(T_{h2} - T_{c2})}{(T_{h1} - T_{c1})}} \tag{30}$$

Aspen software has been used to estimate the heat transfer surface areas for all components in Kalina cycle. The cost function for each heat exchanger C_{HE} is written as [15].

$$C_{HE} = C_o(A)^n \quad (31)$$

Where the base cost C_o is taken as 588 US\$/m² and $n=0.8$ according to economic analysis data reported in [15]. The cost of pumps and turbines can be calculated as function of pump or turbine power in kW using Eq. (32).

$$C_{PT} = C_o(\text{Power})^n \quad (32)$$

Where the base cost (C_o) for the turbine is 4405 US\$/kW and 1120 US\$/kW for a pump. The exponent (n) is taken as 0.7 and 0.8 for the turbine and pump, respectively [15]. Since, the above values are based on available data in 2012, a 60 % cost increasing is assumed to account for current equipment costs. Also, the total cost is obtained by adding 20% of equipment cost for scheduling and operation maintenance and 20% for pipelines insulation and infrastructure. Detailed calculations of required components surface areas, pump and turbine power, and cost analysis for separate and combined heat recovery have been performed.

The results show that a cost saving of about 23% with about 7% increasing of the total produced electricity power has been obtained using Kalina cycle in combined WHR as compared to separate WHR design. Considering the cost of 1 kWh to be 1 LE, the payback periods for separate and combined WHR are 30, and 21 months, respectively.

8. CONCLUSIONS

Detailed waste heat analysis and recovery from a typical cement plant using the 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, 44.32, and 43.6 GWh at average temperatures of about 314, 314, and 254 oC, 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 alternatives 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 ammonia water concentration. The efficiency of the Kalina cycle increases with increasing ammonia concentration at the evaporator outlet and increasing turbine inlet pressure.

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 a total net output power of approximately 6.865 MW. The 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 of 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 a separate WHR design. A cost-saving of about 23% with about 7% increasing of the total produced electricity power has been obtained using Kalina cycle in combined WHR as compared to separate WHR design. Considering the cost of 1 kWh to be 1 LE, the payback periods for separate and combined WHR are 30, and 21 months, respectively.

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NOMENCLATURE

A	Area, m ²
A_k	kiln surface area, m ²
C_{HE}	The cost for each heat exchanger, US\$
C_{PT}	The cost of pumps and turbines, US\$
C_{eq}	Equipment Cost, US\$
C_o	the base cost function, US\$/ m ²
c_p	specific heat, J/(K kg)
h_c	convection heat transfer coefficient, W/(m ² .K)
j	Number of Kiln length,
\dot{m}	mass flow rate, kg/s
Q_{conv}	Convection heat losses, kW
Q_r	The radiation heat losses, kW
T	Preheater temperature , C
U	the overall heat transfer coefficient, W/(m ² K)
\dot{V}_t	Volume flow rate, m ³ /s
W_{out}	Turbine output power, kW
W_{pump}	Electrical power needed for pump, kW
x	Ammonia water concentration
ρ	Density, kg/m ³
ε	Emissivity, dimensionless
σ	Stefan Boltzmann constant, W/m ² K ⁴
η_{max}	maximum possible efficiency of waste heat recovery, dimensionless
η	Kalina cycle efficiency, dimensionless