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D4.1 Waste heat recovery system for oxyfuel cement plants

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Abstract

In an oxyfuel cement plant combustion is performed in a CO_2 and O_2 rich environment. The CO_2 rich flue gas allows relatively easy purification in a CO_2 purification unit. Both the air separation and CO_2 purification unit require additional power. Part of this power might be generated by waste heat recovery systems generating power from waste heat. In this report a detailed review of several waste heat recovery systems for the temperature levels and available waste heat of a BAT reference plant, the Lägerdorf plant and the Slite plant is presented. The most viable options for the oxyfuel cement plants are identified and compared in a thermodynamic analysis. Based on the analysis it can be recommended to use an organic Rankine cycle for the heat-to-power generation in the oxyfuel cement plant.

Table of Contents

List of Abbreviations

ASU	Air separation unit
BAT	Best available technology
BC	Brayton cycle
CPU	CO_2 purification unit
ECRA	European Cement Research Academy
HRSG	Heat Recovery Steam Generator
ORC	Organic Rankine cycle
PEC	Partially Evaporating Cycle
RC	Rankine cycle
tCO_2	supercritical CO_2
TLC	Trilateral cycle
WHR	Waste heat recovery
WHRS	Waste Heat Recovery System

1. Introduction

This document presents an overview of possible waste heat recovery systems (WHRS) for heat-to-power production in the oxyfuel process design of a cement plant for the AC^2OCem project. In addition to steam cycles and organic Rankine cycles other solutions, like the Kalina and trilateral cycles, are investigated. The heat sinks and heat sources of three oxyfuel plants, a Best Available Technique (BAT) reference plant, the Lägerdorf plant and the Slite plant are identified. The temperatures of heat sources and heat sinks are especially important for the evaluation of different heat recovery systems.
The most promising heat recovery systems are identified based on a comprehensive literature review and a thermodynamic analysis of these systems is performed.

The document is organised as follows: In Section 2 an overview of the oxyfuel cement plant configurations and the heat sources and heat sinks of the plants is given. Different heat to power cycles are introduced in Section 3. An overview of proposed heat recovery systems for cement plants is given in Section 3. In Section 4 several alternative configurations of the most promising heat recovery systems are investigated in more detail. A thermodynamic analysis for the heat recovery system is discussed in Section 5. The document ends with a conclusion in Section 6.

2 Oxyfuel cement plant configurations and power generation from waste heat

In the oxyfuel technology, combustion is performed with an oxidizer consisting mainly of oxygen and CO_2, to produce a CO_2 rich flue gas which allows a relatively easy purification with a CO_2 purification unit (CPU). The kiln system itself is modified when the oxyfuel technology is integrated (Figure 1). Additional power is needed for an air separation unit (ASU) and for the CPU. Some of the additional power demand may however be covered with a waste heat to power cycle generating power from waste heat in the oxyfuel cement plant.

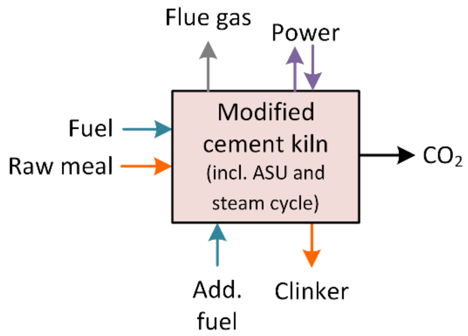


Figure 1. Schematic overview of the integration of the oxyfuel process with a cement kiln.

The potential for power generation from waste heat recovery (WHR) is dependent on the amount of available waste heat and the temperature levels of the waste heat, which differ between plants mainly depending on the plant capacity, raw material moisture, and plant configuration. Below is a description of the oxyfuel process integrated in a BAT plant. In addition, short descriptions of Holcim's Lägerdorf cement plant and HeidelbergCement's Slite plant are given, and the main differences envisioned regarding waste heat availabilities and temperature levels when oxyfuel process is integrated are summarised.

2.1 Oxyfuel BAT plant

The BAT plant is a reference cement plant defined by the European Bref document [1] and adopted by the European Cement Research Academy (ECRA) and the CEMCAP project. It is based on a dry kiln process, consisting of a five-stage cyclone preheater, calciner with tertiary air duct, rotary kiln, and grate cooler (Figure 2). It has a capacity of 3000 tonne clinker per day, which corresponds to ca. 1.0 million tonne clinker per year, or 1.4 million tonne cement per year, with a run time of >330 days per year. The raw material, which consists of 77 wt% CaCO3, 14 wt% SIO2, 6% wt% water and small amounts of Al2O3, FE2O3, MgCO3 is first ground the raw mill, where it is also dried by hot flue gas from the preheater. A more detailed description of this plant is given by Voldsund, et al. [2].

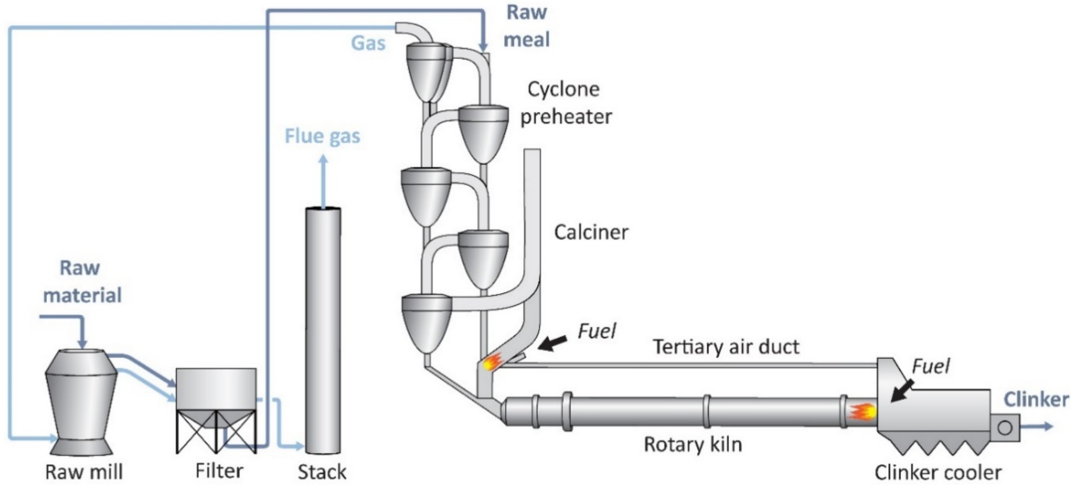


Figure 2. The clinker burning line of the BAT cement plant.

The integration of the oxyfuel process to the BAT plant was investigated and developed through the ECRA projects [3-5] and the CEMCAP project [2, 6]. A schematic overview of the process is shown in Figure 3. Oxygen from the ASU is mixed with recirculated CO_2 rich flue gas and forms the so-called oxidizer gas stream. The oxidizer is first fed to the clinker cooler where it is preheated while it also cools the clinker. From the cooler one part of the preheated air is sent to the rotary kiln main burner and the calciner while another part is used as a heat source to preheat air before it is recycled back to the clinker cooler. The flue gas produced from combustion in the main burner and the calciner and from the calcination reactions that takes place in the calciner has a high concentration of CO_2 since it in this process is not diluted with air. The slightly cooled hot flue gas from the preheater is dedusted in a filter and then water is removed in a condenser. Afterwards one part of the CO_2 rich flue gas is recirculated and mixed with CO_2, while the other part is purified in a CPU and sent for storage. Preheated air is sent to the raw mill for raw meal drying, instead of using the preheater exhaust gas as is done in conventional cement plants.

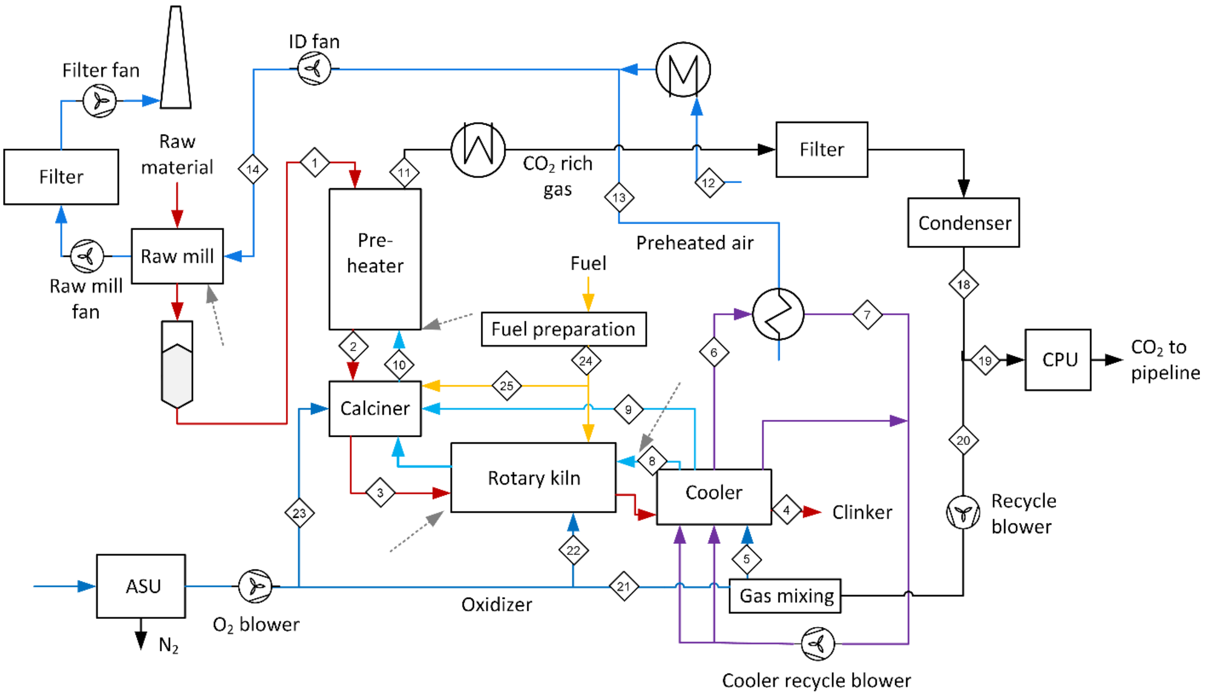


Figure 3. Schematic overview of the oxyfuel BAT cement plant configuration investigated in CEMCAP.

2.1.1 Heat sources and heat sinks

The most important heat sources and sinks of the oxyfuel BAT plant are indicated in Figure 4. The potential heat sources indicated by red squares are:

- Exhaust gas leaving the preheater at 394 °C
- Hot CO_2 leaving the clinker cooler at 338 °C
- Compression heat from CO_2 compression train
- Compression heat from the ASU air compression

The heat sink indicated by a blue square is:

- Drying of raw material in the raw mill

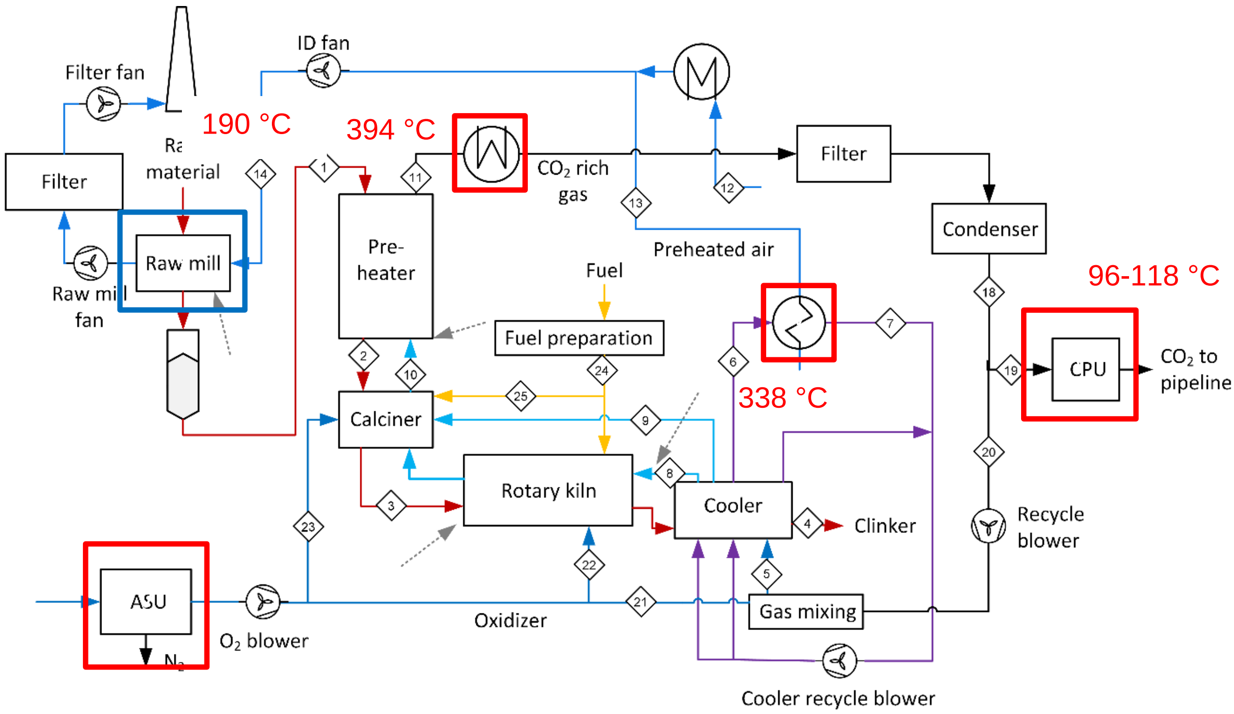


Figure 4. Schematic overview of the Oxyfuel BAT plant with indicated heat sources and sinks.

2.1.2 CEMCAP waste heat recovery and power generation

In the CEMCAP project, an organic Rankine cycle (ORC) was considered for power generation from waste heat. It was assumed that a hot oil loop was required for heat recovery from the preheater flue gas. The integration of the heat recovery system and the ORC into the BAT oxyfuel cement plant is illustrated in Figure 5. From the preheater exit heat is recovered from the hot flue gas in a two-stage heat exchanger, with the hot-oil as an intermediate working fluid. The heat in the hot-oil is used to preheat hot air that is send to the raw mill, and to heat vent gas in the CPU before expansion, before the remaining heat is used for electric power generation in the ORC. Low temperature compression heat from the CPU is partly used for preheating some of the air to be sent to the raw mill and partly used for power generation in the ORC. The compression heat from the ASU was not included in the heat integration, since the ASU was not modelled in detail, and the temperature levels was expected to be relatively low. Considering this system, the highest temperature of the waste heat available for the ORC was the temperature of the hot oil after the preheating of the air sent to the raw mill, which was 314 °C.

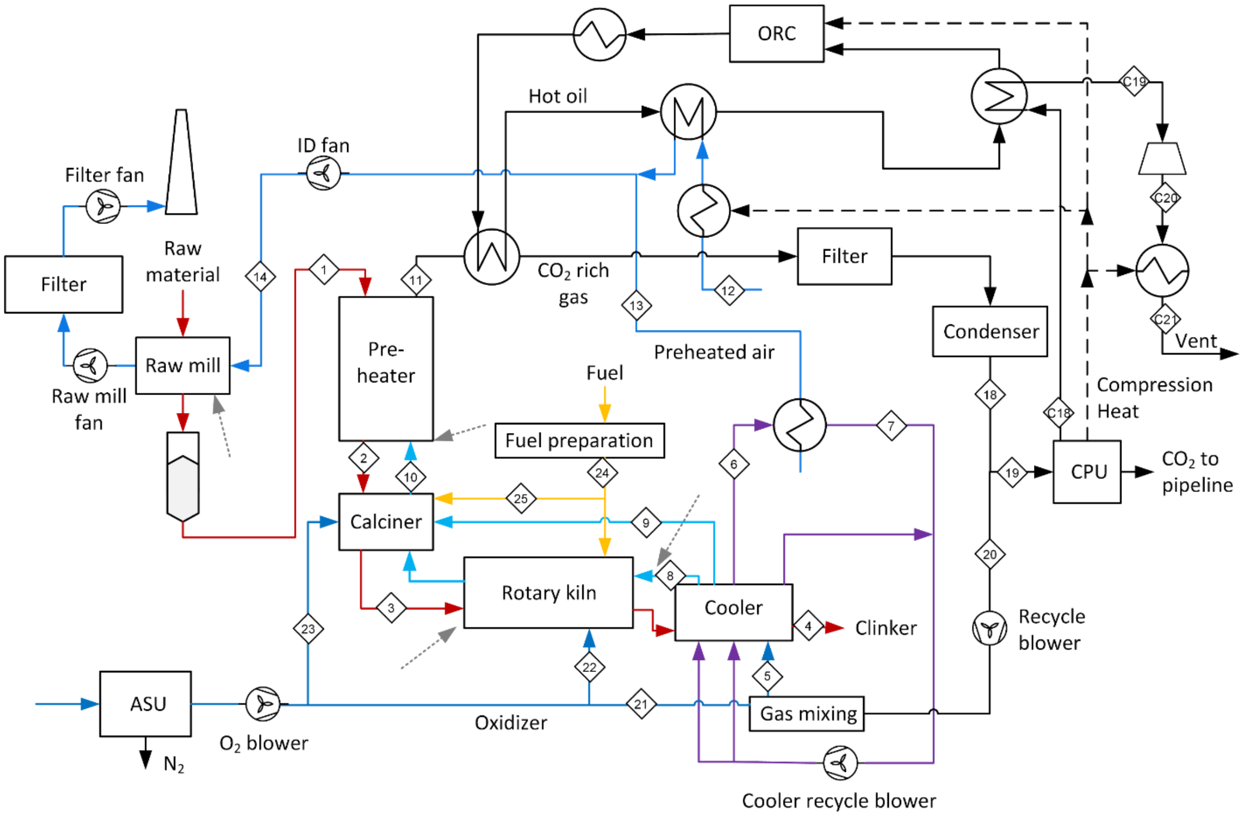


Figure 5. Schematic overview of the BAT oxyfuel cement plant with integration of ORC.

2.2 Oxyfuel Lägerdorf and Slite plants

The detailed oxyfuel configurations of the Lägerdorf and Slite plants are not finalised, but some assumptions can be made regarding the expected waste heat availabilities with corresponding temperature levels.

2.2.1 Plant characteristics in normal operation

The **Lägerdorf plant** is a more than 150 years old cement plant located north of Hamburg which is owned by Holcim. The kiln investigated in AC²OCem is Kiln 11 that was constructed in 1995, and a schematic overview of this kiln is given in Figure 6. It has an original design production capacity of 4500 t/day, and the raw meal is chalk (with moisture content 23%) instead of limestone. The chalk is suspended with water and to reduce heat consumption in the furnace, the slurry is mechanically dewatered in a chamber filter process. The resulting filter cake has a moisture content of 21%. Sludge from the chamber filter is mixed with fly ash and sent to a hammer mill and flash dryer and further to a double string 3-stage preheater with inline calciner.

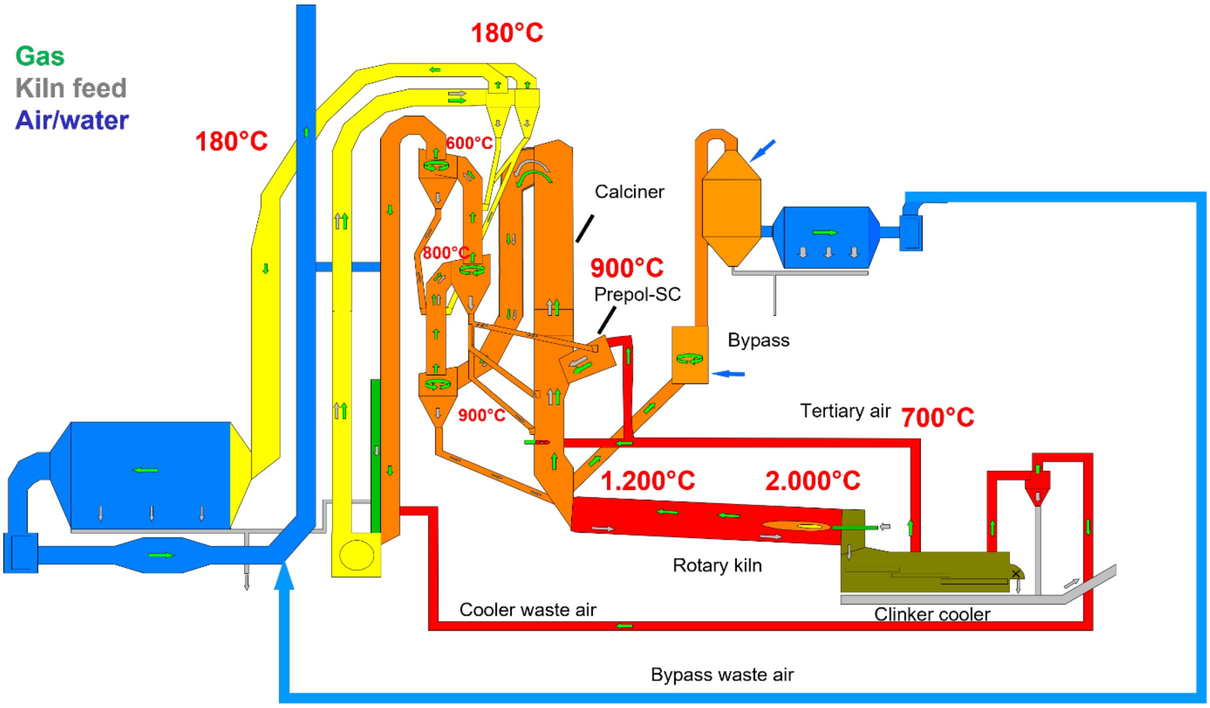


Figure 6. Schematic overview of the Lägerdorf plant.

The **Slite plant** is a large cement plant owned by Cementa (HeidelbergCement) and located at Gotland in Sweden. The kiln investigated in AC²OCem is Kiln 8 which was built in 1995. This kiln has a clinker production capacity of 5600 t/day which is almost doubled compared to the BAT plant. The calciner has two parallel combustion chambers and the preheater has two strings with 5 stages each. The plant has a scrubber to remove sulfur and avoid a sulfur cycle.

Key process data for the Lägerdorf and Slite plants compared to the BAT plant in normal operation are summarised in Table 1.

Table 1. Key process data for the Lägerdorf and Slite plants in normal operation compared to the BAT plant.

	BAT plant	**Lägerdorf plant**	**Slite plant**
Clinker production capacity [t/day]	3000	4500	5600
Raw material	Limestone	Chalk	Limestone
Kiln process	Dry	Semi wet	Dry
Type of mill	Raw mill	Hammer mill dryer	Vertical roller mill
Preheating tower	5 stages	3 stages	5 stages
Plant fuel consumption [kJ/kg clinker]	3100	4500	3720

2.2.2 Oxyfuel preliminary designs

After clinker cooler pilot experiments were performed in CEMCAP with a one stage cooler, it was found that air leakage in such a cooler could be a problem, and therefore it was decided to assume two stage coolers in the designs of the Lägerdorf and Slite plants.

For the **Lägerdorf plant**, two main options for plant layout have been discussed. The hammer mill is normally heated with flue gas from the preheater. However, this type of mill is associated with high air leak, which dilutes the CO_2 if the oxyfuel flue gas is sent through it. In the first option the flue gas is nevertheless sent through the hammer mill, which means that the CO_2 purification afterwards is more demanding (Figure 7). In the second option the heat for the hammer mill is supplied by air that is heated in the clinker cooler and by the preheater flue gas (Figure 8).

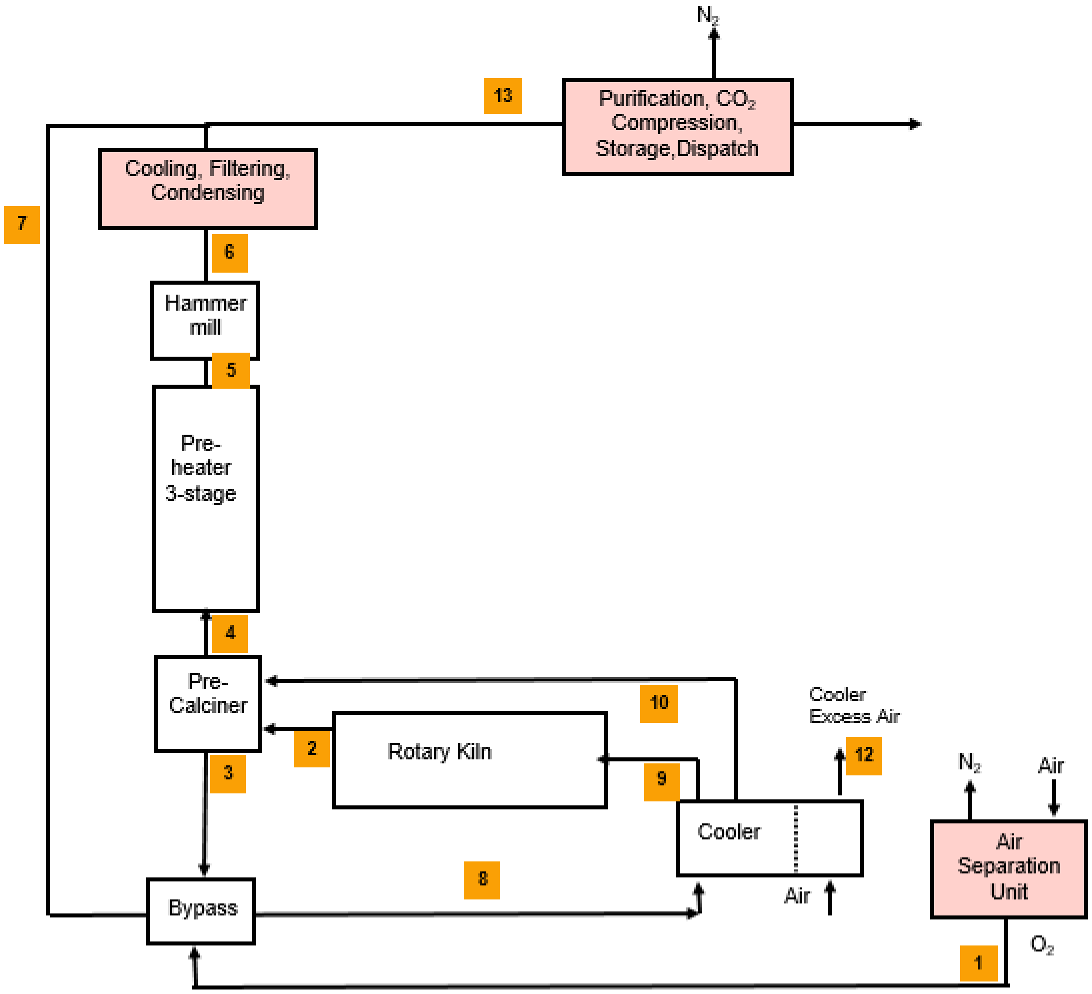


Figure 7. Lägerdorf oxyfuel design – Version 1.

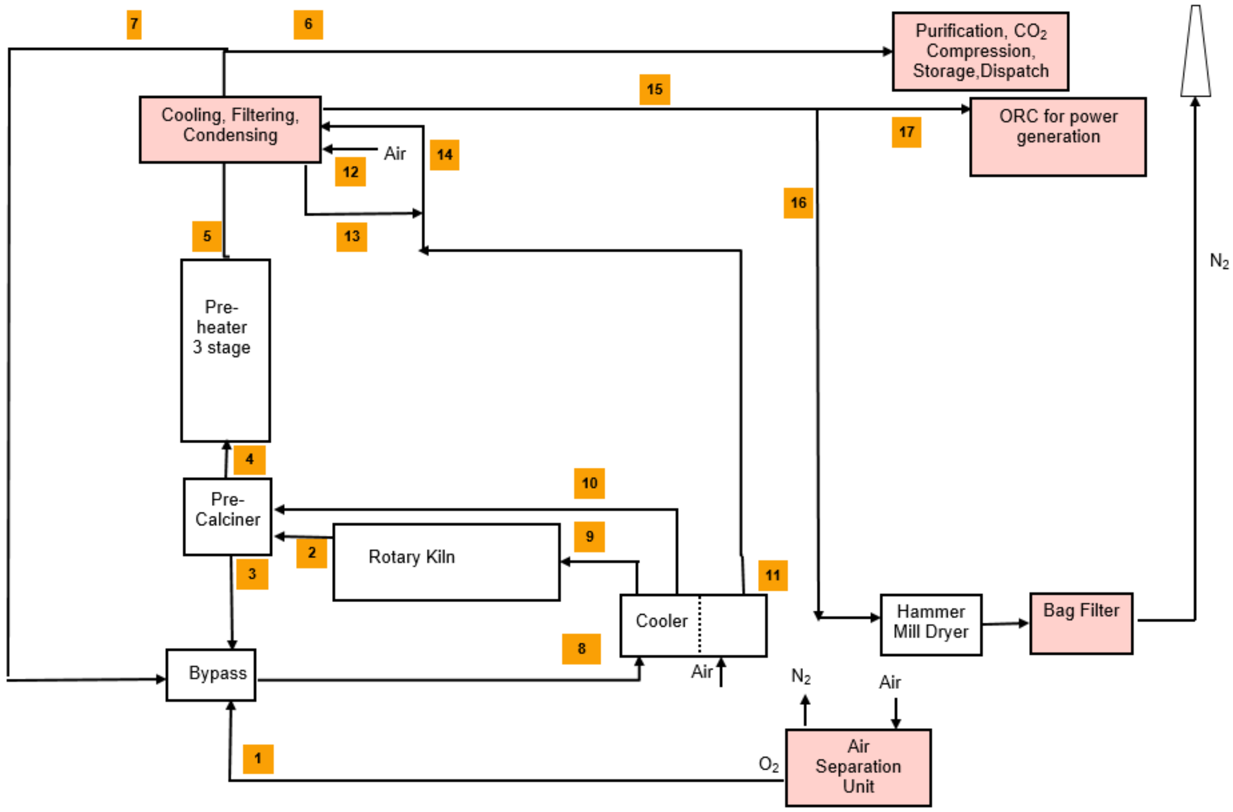


Figure 8. Lägerdorf oxyfuel design – Version 2.

Preliminary models of the core oxyfuel process of the Lägerdorf plant have been developed by VDZ, and preliminary heat integration has been performed by SINTEF. This work indicates that the temperature levels of the waste heat available for a heat to power cycle for these plants are:

- Lägerdorf plant V1:
  - Air from clinker cooler at 220 °C (stream 12)
  - Flue gas from the hammer mill dryer at 130 °C (stream 6)
- Lägerdorf plant V2:
  - Preheated air at 460 °C (stream 17)

The **Slite plant** is more similar to the BAT plant, and for this plant the oxyfuel configuration similar to the one considered for the BAT plant in CEMCAP will be considered, with the exception that a two stage cooler will be considered instead of a one stage cooler. Preliminary models of the Slite plant in oxyfuel mode show that the temperature of the flue gas at the preheater exit is 380-400 °C and the temperature of the cooler exhaust is 280-290 °C. This means that the temperature of the heat available after the heat demand of the raw mill is covered is likely to be within the range spanned by the two Lägerdorf options (220 °C and 460 °C).

It should be noted that a hot oil loop as considered in CEMCAP may be required, but it would be interesting to also investigate the effect of omitting this if this is feasible for the power cycle considered. Further, heat exchanger pinch point temperature approach for dusty gas was 80 °C.

3 Heat to power cycles

In this section an overview of different heat to power systems is given. The focus will be on the Rankine cycle with water/steam as working fluid, the organic Rankine cycle (ORC), the trilateral cycle (TLC) and the Kalina cycle. Several different configurations and the influence of different working fluids are discussed.

3.1 Steam cycle

A steam cycle is a Rankine cycle (RC) where water/steam is the working medium. A schematic overview of a basic steam cycle is given in Figure 9. The working fluid is in liquid state compressed to a high pressure by a pump and heated, evaporated, and superheated in a heat recovery steam generator (HRSG). The high-pressure steam is then used to generate power by a turbine. The resulting low-pressure steam is condensed by a condenser before it is sent back to the pump (Figure 9).

Lowest feasible condenser pressure is saturation pressure at ambient temperature. Temperature of steam entering a turbine is restricted by metallurgical limitations by the materials. High pressure requires piping that can withstand great stresses at elevated temperatures. It is possible to design plants to operate with HRSG pressures exceeding critical pressure of water of 221 bar and temperatures exceeding 600°C.

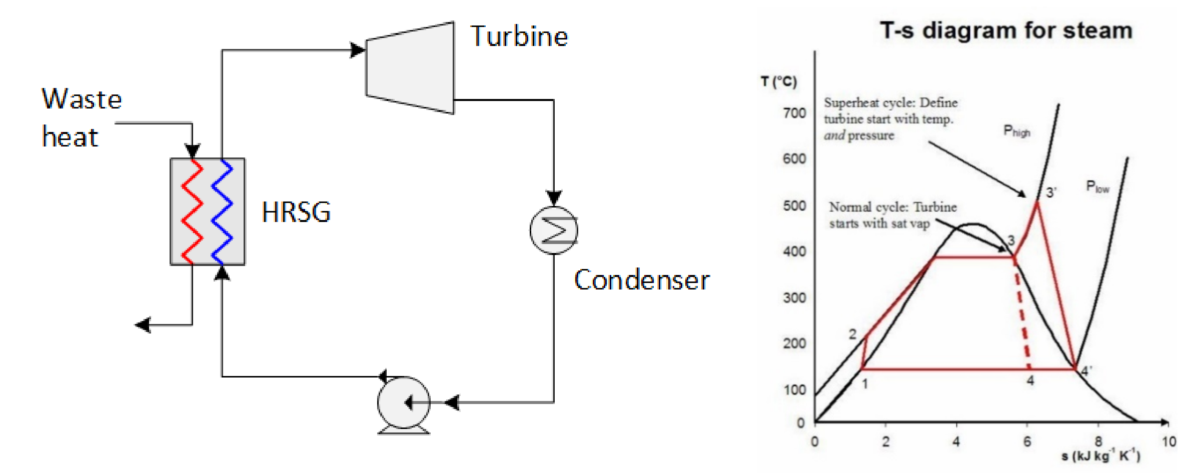


Figure 9. Schematic diagram and T-s diagram of a steam Rankine cycle.

Water, the working fluid of the steam cycle, is a "wet" fluid. This is because the slope of its dew point curve in the temperature-entropy diagram (Figure 9) is negative, and consequently superheating is required to avoid expansion into the two-phase region. This superheating requirement is a disadvantage with steam cycles for low to medium temperature heat sources compared to other working fluids.

Quoilin, et al. [7] recommended operating steam RCs just at temperatures higher than 450℃ to avoid droplet formation during the expansion. Otherwise, higher costs can be expected due to higher thermal stresses in the boiler and on the turbine blades [7]. On the other hand, Karellas, et al. [8] compared the steam RC and an Organic Rankine cycle (ORC) for a cement plant and found that the steam RC is more efficient in exhaust gas temperatures higher than 310℃. Steam RC can reach thermal efficiencies higher than 30% while high temperature ORC do not exceed efficiencies of 24% [7]. Additional advantages with the steam Rankine cycle are the low-cost and environmentally friendly working fluid. Disadvantages are the large pressure drop in the turbine, low condensation pressure, superheating requirements, and higher volume flows [7]. A large pressure drop will cause a large volume flow rate, which increases the turbine size and might require more complicated multi-stage turbines. Subatmospheric pressure causes leakage into the cycle, which requires cleaning of the working fluid.

The basic steam RC (Figure 9) can be modified in various ways to increase the performance of the cycle. Many possible configurations of the ORC, discussed in Section 3.6, can be employed also for steam RC. In this document we will not focus further on these modifications of the steam RC since it is expected that the ORC is more suited for the temperatures of the heat source in a cement plant.

3.2 Organic Rankine cycle

Organic Rankine cycles (ORCs) apply the principle of the steam RC (Figure 9), but use organic working fluids with low boiling points and can therefore be used to recover heat from lower temperature heat sources. Further advantages of the ORC compared to the steam RC are the lower turbine inlet temperature, which reduces thermal stresses in the boiler and on the turbine blades, higher fluid density, which results in smaller volume flows and smaller components, the lower evaporation pressure, and the higher condensing pressure [7]. Moreover, an additional water treatment system, which is usually necessary for the steam RC, is avoided. The organic working fluids of the ORC can be isentropic or "dry". The slopes of these fluids' dew point curves are zero or positive (Figure 10). Therefore, superheating can be avoided.

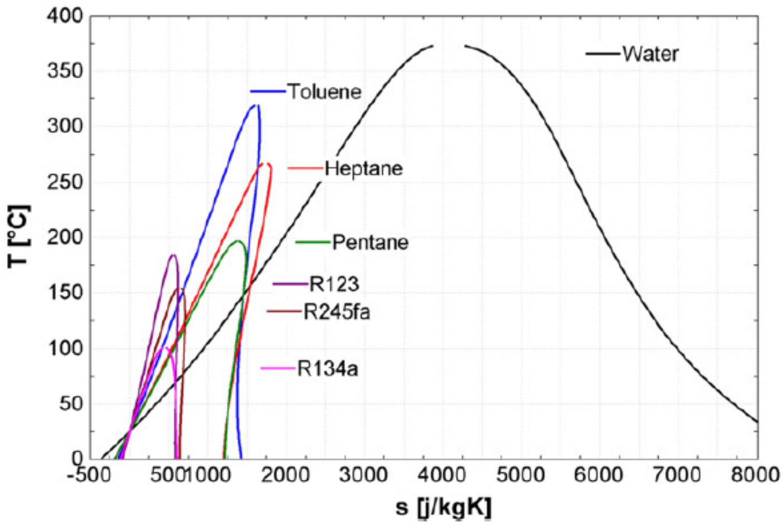


Figure 10. T-s diagram of water and various typical ORC working fluids [7].

While superheating can be avoided with some of the working fluids of the ORC they are often flammable, toxic and have a high global warming potential. In case of failure of the heat exchanger the contact of the hot medium and the organic fluid can cause an explosion. Therefore, the working fluids of an ORC are for safety reasons usually heated by an indirect heat cycle using thermal oils at ambient pressure. This causes further heat transfer loses and reduces the efficiency of the ORC [8].

A challenge in the design of an ORC is to find a suitable working fluid for the specific case. In addition, mixtures can sometimes be a viable option increasing the complexity of this design step. Moreover, the ORC can be operated subcritically, transcritically and supercritically and modified to different configurations. These possibilities increase the complexity of the design of the heat to power cycle and makes it challenging to find the "best" system.

In the following a brief overview of the choice of working fluids is given.

3.2.1 Working fluids

The choice of working fluid is often crucial in the design of an ORC. Consequently, each scientific article designing an ORC investigates at least some working fluid candidates. Usually pure fluids are considered, but mixtures are possible. An important distinction between working fluids can be made between "wet", isentropic and "dry" fluids, where "wet" fluids have negative slope of the saturation curve in the vapor region, isentropic fluids have infinitive slope (the curve is a vertical line), and the "dry" fluids have positive slope. Wet fluids require superheating. As mentioned in Section 3.2 the working fluid of ORCs can be isentropic or "dry".

Drescher and Brüggemann [9] evaluated 700 substances of the Design Institute for Physical Properties to find suitable fluids for ORC in biomass power and heat plants. Important screening criteria were critical temperature and pressure, melting point and autoignition temperature. The maximum and minimum process temperature should be compatible with the fluid stability and the melting temperature of the fluid should be below ambient temperature. Otherwise the fluid may solidify during shutdown time.

In Drescher and Brüggemann [9] the highest efficiency was reached for fluids when superheating was avoided, and the fluid was expanded directly from the dew line. For the biomass plants with a maximum process temperature of 300℃ the family of alkylbenzenes showed highest efficiency.

Aziz, et al. [10] covered heat sources in the range of 500 to 700℃ and assumed a thermal oil for the indirect heat transfer from the flue gas in their analyses. As a working fluid they chose M-xylene, Decane and Propylcylohexane. The temperature of the thermal oil at the evaporator inlet was fixed to 370℃ and the lowest temperature in the flue gas was fixed to be 180℃ to avoid acid condensation. A genetic algorithm was used to maximise the exergetic efficiency while minimising the total heat transfer requirement of the cycle (UA). M-xylene as a working fluid was the best option for the considered process.

Zeotropic mixtures

Zeotropic mixtures are interesting working fluids. Mixing of two fluids with different boiling points, results in a working fluid that boils over a range of temperatures. The amount of energy that can be recovered is higher than for single component fluids since the energy recovery can start at one temperature level and finish at a much lower temperature level. The thermodynamic losses are lower since a better match between the temperature profiles is achieved. The boiling temperature range can be modified by adjusting the composition and the pressure level of the working fluid. The advantage is an increased efficiency in the evaporator and condenser. However, ORCs with mixtures as working fluids require larger heat exchanger areas than the basic ORCs because of the reduced temperature difference and lower heat transfer coefficients.

Guo, et al. [11] carried out a detailed comparison between the use of pure fluids and zeotropic mixtures as ORC working fluids in terms of efficiency. They examined one pure fluid, one zeotropic mixture matching the heat source, and one zeotropic mixture matching the heat sink. The greatest efficiency was obtained for the zeotropic mixture matching the heat sink. Braimakis, et al. [12] and Chen, et al. [13] also considered the supercritical operation mode with pure fluids and zeotropic mixtures as working fluids. They figured out that second law efficiency of pure fluid subcritical ORC can be increased by supercritical ORC with zeotropic mixtures by up to 60%. Braimakis, et al. [12] considered in their article heat source temperatures ranging from 150 to 300℃ while Chen, et al. [13] used cycle high temperatures of 100 to 200℃. Dong, et al. [14] considered zeotropic mixtures as working fluids with a heat source inlet temperature of 280℃. An ORC system with recuperator was used. A mixture of siloxanes dexamethyldisiloxane and octamethyltrisiloxane in the composition 0.4/0.6 increased the cycle efficiency compared to ORCs with pure fluids. Eller, et al. [15] analysed the second law performance of novel working fluid pairs. They found that the second law of efficiency of an ORCs with zeotropic mixtures in sub- and supercritical operation is up to 13% higher than for the Kalina cycle. They, therefore, conclude that an ORC with zeotropic miztures has a greater potential than the Kalina cycle for waste heat recovery. At a heat source in the range 350-400℃ the highest efficiency is obtained with a benzene/toluene (36/64) mixture.

3.3 Trilateral cycle

The trilateral cycle (TLC) closely resembles both steam RC and ORC (Figure 11). The difference lies in that the working fluid is directly expanded from the saturated to the two-phase region. This improves the temperature match in the heater. Intrinsically the TLC has a lower thermal efficiency than the ORC [16]. Nevertheless, it has a higher potential to recover heat because of the better match between the temperature profiles of the heat carrier and working fluid.

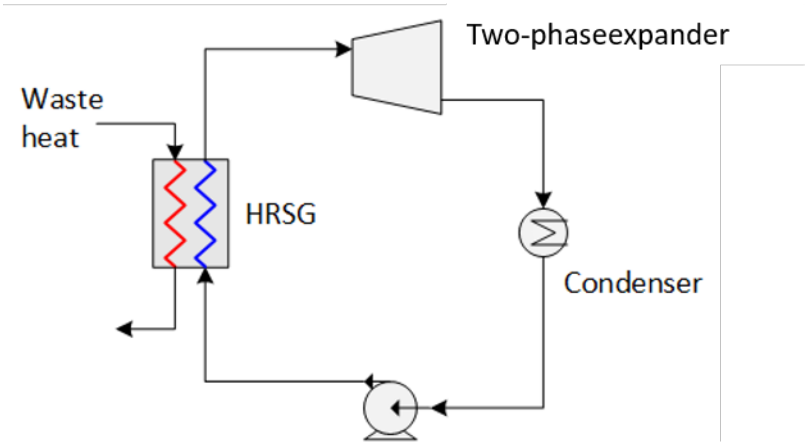


Figure 11. Trilateral cycle.

Fischer [17] compared the trilateral cycle with water as working fluid with an ORC. He found that for a heat source of 350℃ the total exergy efficiency is larger by 3% for the TLC than for the ORC. Moreover, the TLC has always a better performance than the transcritical cycle (see Section 3.5.2). However, the volume flow change at the expander outlet compared to the inlet increases for some cases by a factor of 710. Moreover, the outlet volume flows at the expander are about 3-70 times larger for the TLC with water as working fluid than for the ORC depending on the temperature. The large volume flows is a consequence of the low vapor pressure of water at ambient temperatures. These large volume flows are problematic in real plants.

Another challenge of the TLC is the availability of two-phase expanders with high isentropic efficiency [16].

Fischer [17] mentioned the use of mixtures as working fluids since the temperature glide could improve condenser performance. However, he recommended the use of pure water as the advantage of a better temperature glide is small compared to the disadvantage of the corrosive behaviour of water/ammonia mixture, which had been used in another work by Zamfirescu and Dincer [18].

The power flash cycle, a generalisation of the TLC, was investigated Lai and Fischer [19]. Here the compressed liquid was heated up to its boiling point and then a flash expansion into the superheated region was performed. This reduced the volume flows at the expander outlets. At a temperature of 350℃ of the heat carrier water produced the most power. For lower temperatures cyclopentane was recommended, to avoid large volume flows [19].

At extremely low temperatures (<80℃) the trilateral cycle is superior to ORC [20]. However, in Rohde, et al. [21] the TLC achieved up to 10 % higher work output than the basic ORC for a heat source at 100°C, but the performance of both cycles were comparable for heat sources at 150°C and 200°C.

Another adaption of the TLC to reduce the heat exchanger area and pumping power is the partially evaporating cycle (PEC). Here the working fluid is partially evaporated in the evaporator before expanded in the two-phase expander. While the PEC comes with the mentioned advantages, the heat recovery is reduced due to poorer temperature match and the complexity of the heat exchanger increases. Moreover, PEC showed no improvement compared to the basic RC for heat sources at 150 and 200℃ [22].

3.4 Kalina cycle

The Kalina cycle (KC) is an alternative to the Rankine cycle and was proposed in 1984 [23]. In the KC mixtures of fluids with different boiling points are used as working fluid, normally zeotropic mixtures of ammonia and water, which increase the evaporator and condenser efficiency (see Zeotropic mixtures under Section 3.2.1). In the Kalina cycle the working fluid is split into streams with different concentrations, which gives flexibility to optimize heat recovery and condensation.

A schematic overview of a basic KC is given in Figure 12. The working fluid is heated and partly evaporated by waste heat in a heat recovery vapour generator. Liquid is removed from the vapour in a vapour-liquid separator. The vapour is sent to a turbine that generates power. The liquid is used for preheating of the working fluid before the pressure is decreased and it is mixed with the vapour exiting the turbine. The vapour mixture is condensed and pumped to the preferred pressure level, before being preheated and sent back to the evaporator.

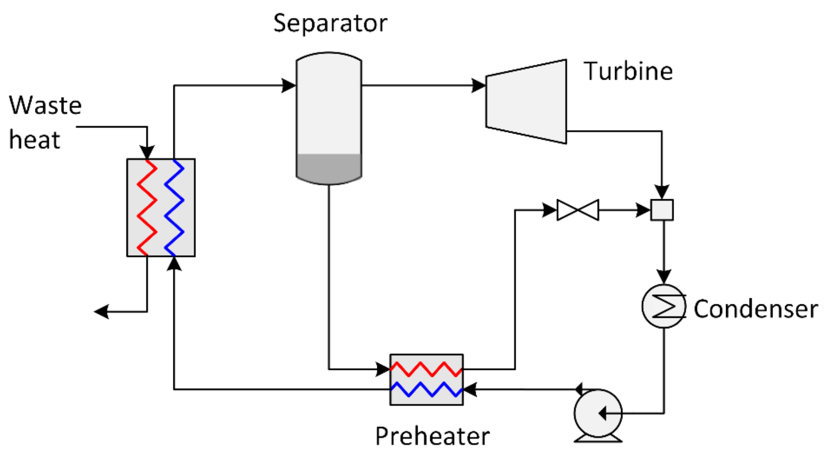


Figure 12. Schematic diagram of a basic Kalina cycle.

A review of research on the KC was given by Zhang, et al. [24]. They pointed out that the main technical concern for applications of the KC focuses on environmental and safety features of the ammonia-water mixtures, which, however, can be addressed by appropriate design of the cycle. Nevertheless, it was advised to not exceed 400℃ with these mixtures, since ammonia can become unstable. Although KCs considered for small power plants and also for utilization of high temperature heat sources are able to generate more power, it was concluded that Kalina cycles have low opportunity to be economically justified as compared to its much more simple ORC cycles.

Wang, et al. [25] compared a single flash steam cycle, dual-pressure steam cycle, ORC and KC for a cement plant. The exhaust gas from the preheater, clinker cooler and suspension preheater boiler were used with temperatures at 340℃, 320℃ and 210℃, respectively. The KC achieved the best performance in the exergy analysis.

Bombarda, et al. [26] compared the Kalina and organic Rankine cycle for Diesel engines. The ORC worked with hexamethyldisiloxane and the KC with a water ammonia mixture. The net electric power was almost equal for both cycles for a heat source of 346℃ and a logarithmic mean temperature difference in the heat recovery exchanger of 50℃.

Nguyen, et al. [27] investigated a Kalina split cycle, a more complex configuration of the KC. The heat source inlet temperature was 346℃. The split-cycle was more efficient than the simple KC, but it was also pointed out that the costs of the Kalina split-cycle was about 30-40% higher compared to the organic Rankine cycle. This excludes many KC as candidates for cement plants, since the efficiency gains are small compared to the additional investment costs due to higher complexity of the KC. Similarly, Bahrampoury, et al. [28] compared different double pressure KC with the simple base KC. Again, the energy efficiency of double pressure KC was higher than of the base case. However, in the thermoeconomic evaluation the simple KC outperformed all other considered configurations.

Eller, et al. [15] showed that higher efficiency in the KC can be obtained with an alcohol/alcohol mixture compared with the common ammonia/water mixtures for heat sources above 250℃. However, even the best KC was outperformed by ORC with zeotropic mixtures.

Rostamzadeh, et al. [29] studied a cooling, heating and power system with a heat source temperature of 325℃. The KC-based system had a higher thermal efficiency and total and unit cost of product, while the ORC-based system showed a higher exergy efficiency.

3.5 Subcritical, transcritical and supercritical operation of heat to power cycles

A brief overview of the different operation conditions of heat-to-power cycles is given in this section. The focus is on ORCs. However, also the other cycles have the possibility to be operated sub-, trans- or supercritically.

3.5.1 Subcritical operation

Subcritical operation is the standard way of operating the ORC (Figure 13). The pressure is increased by a pump (1-2) by external work. The high-pressure fluid passes through a heat exchanger, which increases the temperature and evaporates the fluid (2-2b). Depending on the fluid superheating is necessary (2b-3). The fluid expands through a turbine which generates power (3-4). At the end of the expansion a superheated fluid is still in the superheated state. A condenser cools the fluid and converts the fluid to its liquid state (4-1).

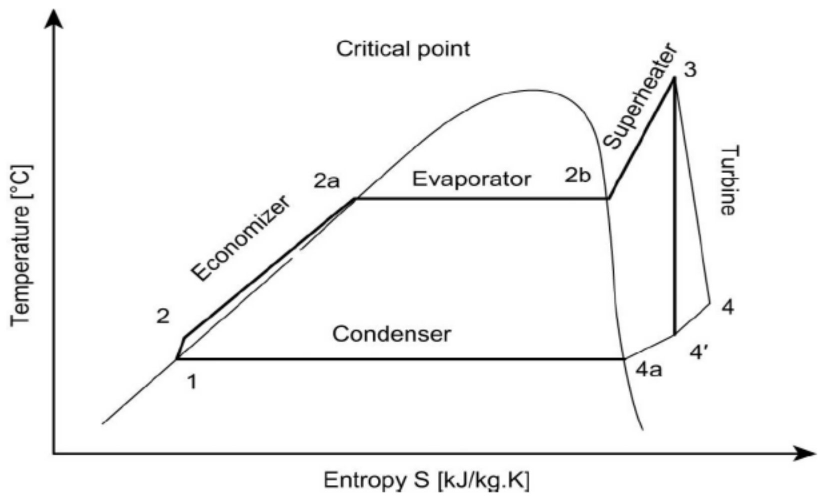


Figure 13. T-s diagram for R134a [30].

The subcritical operation is the standard operation of the ORC and is usually used as a reference case to compare the performance with more advanced operations [31].

3.5.2 Transcritical cycle

In the subcritical cycle the constant temperature in the evaporation causes an imperfect heat transfer and exergy destruction. A way to improve the heat transfer is to operate the heat exchanger above critical pressure. Then the fluid does not undergo a phase transition during heating and a supercritical fluid enters the turbine at state 3 (Figure 14). While the heat exchanger exergy losses are reduced a challenge with operating cycles in a supercritical model is the large compression power and high operating pressures [32].

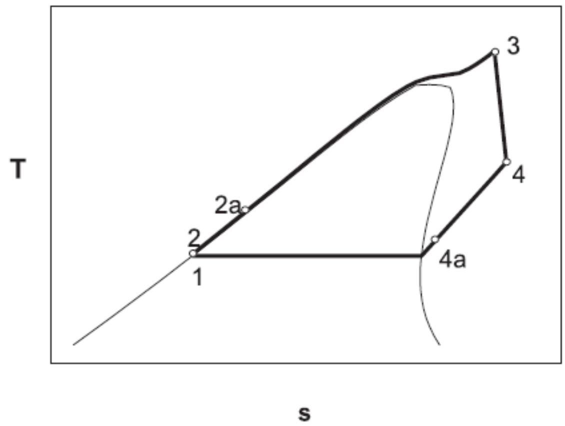


Figure 14. T-s diagram where the maximum pressure is supercritical [32].

A cycle operating in a transcritical regime performs the best in comparison to a subcritial and several different Kalina cycles in the study conducted by Becquin and Freund [31] for heat source temperatures ranging from 80℃ to 200℃. However, the performance benefit of the transcritical operation decreases at high temperatures. Moreover, the transcritical cycle entails higher component costs than the other cycles discussed in the paper because of the high heat transfer surface area, larger required mass flows and high pressures in which the cycles are operated.

Lai, et al. [32] investigated high temperature ORC processes with heat carrier inlet temperatures of 280℃ and 350℃ and supercritical and subcritical maximum pressures. Depending on the operation conditions and the working fluid allowing supercritical maximum pressure can be beneficial because of the better heat transfer in the heater. For the high temperature case cyclopentane ranked the best of the considered working fluids. Shengjun, et al. [33] also compared a transcritical and subcritical cycle for geothermal power generation. The heat source temperature was just 90℃. Here the transcritical cycle was preferable since it could increase the utilisation of the heat at low costs. Nevertheless, it was pointed out that the costs of heat exchanger increase rapidly with the operating pressure, which can be a disadvantage for transcritical cycles operating at high pressure levels. Higher exhaust gas temperatures with indirect heating via thermal oil were discussed by Algieri and Morrone [34]. The working fluid temperature was constrained to be 400℃. The transcritical cycle with internal regeneration and cyclohexane as working fluid increased the efficiency compared to the subcritical cycle. However, the higher operating pressure might rise safety concerns and might have a negative effect economically.

3.5.3 Supercritical cycle

A fully supercritical cycle is operated above critical pressure for both heat injection and rejection. A fully supercritical cycle is also referred to as a Brayton cycle. In this cycle the evaporator is replaced with a heater and the condenser with a cooler since no phase transition occurs. The main difference between the Brayton cycle and the Rankine cycle is that the Rankine cycle is a vapor cycle, while the Brayton cycle operates supercritically between liquid and vapor phase. The large compression power, high operating pressures, and unstable amounts of liquid and vapor in the supercritical phase are challenging. The advantage is an improved heat exchange. However, for the cement plant a supercritical cycle might be not economically competitive with simpler solutions with smaller initial investment costs. Nevertheless, Kizilkan [35] compared a sCO_2 Brayton cycle (BC) and a steam RC in a cement plant thermodynamically. He found that the sCO_2 BC outperformed the steam RC considering energy and exergy efficiency.

3.6 Different configurations of heat to power cycles

In this section a brief overview of common advanced configurations of heat to power cycles is given. The focus is on configurations for the ORCs even though they are not exclusive for ORCs and can be applied to the steam and the trilateral cycles as well.

3.6.1 Dual pressure cycles

The dual pressure cycle, also called a "two-stage" cycle splits the ORC cycle into several pressure levels (Figure 15). This increases the efficiency of the heat transfer. Moreover, introducing a second pressure level also creates a new pinch point, which allows more heat transfer into the cycle. Astolfi [36] believes that applications where two or more pressure level cycles might be profitable are deep geothermal reservoirs with high exploration and drilling costs, and industrial WHR from plants like cement and steel production industries. A double pressure and simple Kalina cycle are compared by Bahrampoury, et al. [28]. The double pressure Kalina cycles are more exergy efficient, but in the thermoeconomic evaluation the simple Kalina cycle is superior because of lower investment costs.

Becquin and Freund [31] compared several advanced cycles. The dual pressure cycle increased the efficiency for low-temperature exhaust gas, but at higher temperatures around 200℃ it showed no improvement to the simple subcritical ORC.

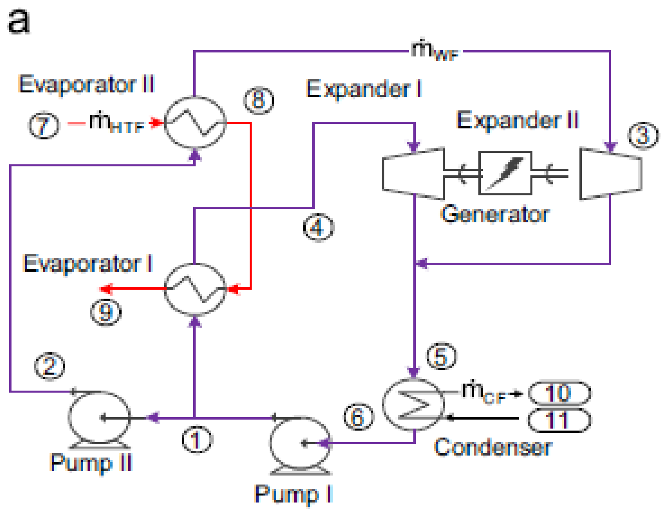


Figure 15. Dual pressure ORC [16].

Another two-stage process was proposed by Meinel, et al. [37]. Here a two-stage turbine was used, where in the first turbine the condensate was extracted and mixed with the output of the condenser behind the second turbine. This saturator cycle was compared to a subcritical ORC and an ORC with recuperator for a heat source of 490℃. A thermal oil loop at 240℃ was used to transfer the heat to the working fluid. The saturator cycle was beneficial for isentopic fluids, while the recuperator cycle was the best for dry fluids.

While the heat exchanger surface area per unit power produced is smaller for the dual pressure cycle at a heat source of 200℃, the cycle becomes more complex with additional mixer, separators, controls and possibly two expanders.

3.6.2 ORC with recuperator

A recuperator reuses the heat after the expander to preheat the working fluid. This increases the thermal efficiency since it improves the temperature match between the working fluid and the heat source and sink. A recuperator is only advisable if a superheated state is necessary after the expander. For dry fluids the net power does not increase by adding a recuperator [16]. Moreover, a recuperator increases the pressure drop and increases the investment costs since additional components are required. A recuperated ORC with pentane as working fluid was chosen in Price and Hassani [38] for solar power plant. The working fluid temperature was around 305-390℃.

3.6.3 Other configurations

The regenerative Rankine cycle is an ORC cycle with turbine bleeding coupled to a direct contact heat exchanger. Similar to the cycle with recuperator the working fluid is pre-heated before entering the evaporator. Mago, et al. [39] compared the regenerative cycle with a basic ORC. The regenerative cycle had a higher thermal efficiency and lower irreversibility. However, the performance difference between the two cycles depended highly on the working fluid. The best efficiency was shown by R123 at a turbine inlet temperature about 150-180℃.
In the organic flash cycle (OFC) vapor and liquid are separated in a flash tank behind the evaporator. The vapor fraction is fed to a turbine while the liquid phase is directly returned to the condenser. The advantage is a better match of temperature profiles of the heat carrier and the working fluid [16]. Ho, et al. [40] compared the organic flash cycle with an ORC for a heat source of 300℃. The OFC had a comparable performance to the optimized ORC and aromatic hydrocarbons outperformed siloxanes as working fluids.

3.7 Recent studies on waste heat recovery solutions for conventional cement plants

In Table 2 an overview of different recent studies on WHR in conventional cement plants is given. In a cement plant, in addition to the energy saving, the heat to power system reduces the limitations by the ID fan, reduces water consumption, removes the danger of fan build ups caused by sticky particulates due to evaporating cooling and reduces carbon footprint [41].

Table 2. Overview of studies on WHR in cement plants. The table is similar to the one presented in Kizilkan [35].

Reference	Exhaust Gas temperature [℃]	Cycle [1]	Working Fluid	Energy Efficiency [%]	Exergy Efficiency [%]	Power production [kW]	Payback time [yr]
Karellas, et al. [8]	360; 380	RC ORC (with	Water Isopentane	23.6 17.6	32.6 24	~6200 ~4700	~5

[1] RC = steam Rankine Cycle; ORC = Organic Rankine Cycle; KC = Kalina Cycle; BC= Brayton cycle; subcrit = subcritical; Reg. = regenerative; Rec = recuperated;

		thermal oil)					
Mohammadi, et al. [42]	270; 340	ORC DP RC Reg. DP ORC	Cyclohexane Water Cyclohexane	- - -	44.8 50.9 59.4	~5200 ~5700 ~6500	-
Sanaye, et al. [43]	305; 290	RC ORC	Water Toluene	- -	52.1 46.5	9100 6600	3.4 5.1
Fergani, et al. [44]	350	ORC (with thermal oil)	**Cyclohexane** Benzene Toluene	12.8	27.1	~1570	-
Moreira and Arrieta [45]	310; 440	Subcrit. ORC; **Subcrit reg. ORC**	**R142b** R11 R123	22.9	50.6	~5600	<2
Ahmed, et al. [30]	200	ORC	R134a	18 (lowest flue gas temp + pinch point 10K)	58.2	~1000	-
Fierro, et al. [46]	327	ORC Rec. ORC	Cyclopentane Cyclopentane	16 17.3	37.5 40.5	3800 4100	8 6
Han, et al. [47]	381	RC	Water	-	-	4600	-
Naeimi, et al. [48]	270; 340	RC	Water	22.2-23.5	73.5-74.9	4400-5200	-
Amiri Rad and Mohammadi [49]	315; 380	RC	Water	16	39	4000	-

Júnior, et al. [50]	390	KC	Ammonia/water	24.0	48.8	~2400	-
Kizilkan [35]	376; 451	RC BC	Water CO_2	24.2 27.6	51.4 58.2	8300 9400	-
Olumayegun and Wang [51]	380	BC	CO_2	29.2	-	5000	-
Jamali and Noorpoor [52]	150	ORC	R123	15.5-26.6*	13.2-15.9* (multi-generation system)	17400-18400	-
Nami and Anvari-Moghaddam [53]	250	RC ORC + absorption chiller	Water hexamethyldisloxane	5.4 7.1 (just power)	53 63 (entire system)	450 590	4.7 5.1

Karellas, et al. [8], Mohammadi, et al. [42] and Sanaye, et al. [43] compared the RC and ORC for the WHR in cement plants. In all cases the exhaust gas from the rotary kiln and the grate cooler are used.

Karellas, et al. [8] compared energetically and exergetically a steam RC and an ORC for a cement plant. A direct heat exchange was used for the steam RC while an indirect heat exchange using pressurised water was used for the ORC. It was assumed that the steam RC operates just 10K below the exhaust gas temperature. However, the indirect heat cycle for the ORC used a pressurised water circuit operated between 220℃ and 125℃. Moreover, the maximum inlet temperature and pressure to the turbine in the ORC were 185℃ and 30 bar, respectively. Consequently, the ORC operated 175℃ lower than the exhaust gas temperature. In both cases the exit temperature of hot air was 130℃. Isopentane was used as working fluid in the ORC since it resulted in the highest system efficiency compared to R245fa, neopentane and pentane. The steam RC was more efficient when the exhaust gas temperature exceeded 310℃.

Similarly Mohammadi, et al. [42] compared heat recovery systems for high and low temperature ranges in cement plants. A regenerative ORC had the best exergy efficiency compared to a dual pressure Rankine cycle and a simple ORC for the high temperature range.

However, only the regenerative ORC performed better than the steam RC while the simple ORC was inferior to the steam RC. The remaining heat of the exhaust gas, which was not captured by the heat recovery cycle, was used to preheat the raw material. A second heat recovery cycle was used to recover heat from the outlet of the grate cooler in a low temperature cycle. For the low temperature cycle, where the energy carrier had a temperature of just 120℃, the ORC showed better performance than a CO_2 cycle to which it was compared.

Steam Rankine, ORC, Kalina and supercritical CO_2 cycles for cements plants are compared by Amiri and Vaseghi [54]. A waste heat boiler with settling chamber is used for the waste heat recovery. The settling chamber removes dust from the exhaust gas. The heat is extracted from several spots of the clinker cooler to produce a higher final temperature of the steam. The steam RC, ORC and supercritical CO_2 cycles have similar costs while the Kalina cycle has significant higher costs per kW generated power.

Sanaye, et al. [43] designed a WHR and power generation system for two parallel lines of cement production. The raw mill section required a temperature of about 214℃ for drying the raw material. Therefore, in the design process it was required that the temperature from the suspension boilers was more than 202℃. The process was optimized where the objective function was maximising the annual benefit and minimising the exergy destruction. A Pareto front was created using these two objective functions. The steam RC produced with 9.14 MW considerably more power than the ORC using toluene as working fluid, which just produced 6.56 MW. Moreover, the steam RC had a faster payback period and a larger CO_2 production decrease.

In several articles an ORC is designed for the WHR. Fergani, et al. [44] studied an ORC for a cement plant comparing the three working fluids cyclohexane, benzene, and toluene. A thermal oil was used to transfer the heat from the exhaust gas to the working fluid. The thermal oil operated in a temperature range of 310℃ to 115℃. A multi-objective particle swarm optimization was performed to optimize the process. Cyclohexane allowed the highest turbine inlet pressure and gave thermodynamically and exergoeconomically the best performance. Benzene, on the other hand, had the best from an exergoenvironmental point of view.

Moreira and Arrieta [45] studied the economic performance of a regenerative ORC under subcritical conditions. The preheater exhaust gas exited the evaporation unit at 228℃ to allow drying of the raw material before its inlet into the suspension preheater. The process was optimized with a genetic algorithm. The organic fluids with the highest power outputs were R142b, R11 and R123. Financially the regeneration ORC was not beneficial in comparison to the simple ORC because of higher investment costs. Superheating, however, reduces investment costs. A design methodology for an ORC based on actual data from a cement plant was presented in Ahmed, et al. [30]. The ORC was combined with a gas turbine to convert the gas turbine waste heat into electrical power. As working fluid R134a was chosen and approximately 1 MW power was produced. In the case study the exhaust gas temperature was just 200℃.

In Fierro, et al. [46] cyclo-pentane was used as a working fluid in the ORC, which delivered the highest net work with an outlet temperature of the hot gas of 180℃ (an ORC that operates at 180℃). The payback time is about 8 years and a power of 3.77 MW is produced in the cement plant. Moreover, a recuperated ORC design increases the work and economic performance of 8.75% and 13.5%, respectively. This decreases the payback period about two years.

Ramshaw, et al. [55] use an ORC since it has compared to the steam RC significantly higher operational flexibility when heat source conditions vary, it has a very high cycle efficiency also at partial loads, water treatment is avoided, dry expansion and lower turbine revolutions per minute increases the turbine's life time, automatic start up and shut down without specific technical knowledge needed for the operation is possible, is has a higher flexibility in layout arrangement, and it has lower requirements of maintenance. Similarly, Börrnert [56] mentioned that the ORC has compared to the steam RC a simpler design of the heat exchangers and turbine, can operate at lower temperatures, has a high turbine efficiency with excellent part load behaviour, has shorter start up times, can operate automatically without personal, has low operating and maintenance costs and has moderate capital expenditure due to standard components and compact design.

Han, et al. [47] designed a WHRPGS for a cement plant containing two HRSGs. One is installed to recover heat from the cooler the other to recover heat from the preheater system. A medium and low-pressure steam flows are supplied to the turbine of the Rankine cycle. Moreover, hot air from the cooler is used in a coal mill, which needs a hot air temperature of 220℃. A combined pinch and exergy analyse is conducted. The retrofitting of the cement plant with the WHRPGS, which was mainly modifications of the air cooler boiler, installations of hot gas pipes from the boiler to the coal mill and valves, had a pay-back period of about six months.

How to recovery heat in different scenarios in a cement plant are studied in Naeimi, et al. [48]. It is recommended mixing the hot streams from the cooler and the preheater and using just one boiler rather than having two separate boilers for these two hot streams.

Amiri Rad and Mohammadi [49] optimized the energy and exergy efficiency of a steam Rankine cycle for WHR in a cement plant. The exhaust gas has a temperature of 200℃ when it enters the raw mill for dehumidification of the raw materials. As minimum pressure of the cycle 20kPa was chosen. A two-stage turbine with reheating is used instead of a single-stage turbine to

reduce the irreversibility of the process. The maximum power generation is achieved with a maximum pressure of 1398 kPa.
Hunter and Ray [41] pointed out that a barrier for implementing the system is the long payback time of three to five years. Therefore, their main objective is to create a system with short amortisation period, which might be though not the most energy efficient system possible. They recommended using a steam RC.

Júnior, et al. [50] evaluates a Kalina cycle for WHR in cement plants. The reference cement plant has a daily capacity of 2100 tons clinker. A power generation of 2439 kW is achieved with an exergetic and energetic efficiency of 48.8 and 24.0%, respectively. They concluded that a rise in turbine inlet pressure does not significantly increase the power generation but decreases the specific cost of electricity generation.

Kizilkan [35] compared a tCO_2 BC and a steam RC in a cement plant thermodynamically. He found that the tCO_2 BC is about 3.5% more energy and 7% more exergy efficient. CO_2 also showed highest performance compared to other supercritical working fluids. A dynamic model and the control of a single recuperator recompression supercritical CO_2 power cycle for waste heat of 380℃ is also presented in Olumayegun and Wang [51]. The case study and preliminary design of the process is carried out for exhaust gas from the cement industry with a flow rate of 100 kg/s.

Nami and Anvari-Moghaddam [53] studied combined cooling, heating, and power systems. A LiBr-H20 absorption chiller is used together with either a steam RC or a recuperative ORC. The study concluded that the CCHP system operating with an ORC using hexamethyldisloxane as a working fluid has the highest performance while the steam Rankine-based cycle has the faster payback period.
Ishaq, et al. [57] examines the performance of heat recovery from furnace cement slag in combination with a thermochemical copper-chlorine cycle for hydrogen production. The blast furnace waste heat from the cement's slag is used to heat water, which enters the Cu-Cl cycle at 550℃. The heat from the high temperature oxygen gas stream is used in a triple stage reheat Rankine cycle.

3.8 Other heat to power solutions

Other heat to power technologies like membrane technologies, Stirling engines, thermoelectric generators or phase change materials are excluded from this report since they are impractical in a cement plant either due to the temperature level or due to the size of the process.

Nevertheless, an interesting analyse of a thermoelectric generator for cement plants is presented in Mirhosseini, et al. [58]

An interesting absorption cycle is the carbon carrier cycle. Instead of a condenser, a carbon dioxide chemical absorption process is used to create an efficient pressure reduction downstream of the turbine. The efficiency in the cycle can get close to an ideal Carnot cycle. With a heat source of 90℃ the cycle can produce three times more electricity compared to current low temperature conversion technologies [59]. However, the patent states it is for waste heat of 150℃ or lower, which excludes this cycle also for the applications in cement plants.

3.9 Identification of the most relevant cycles

The most promising waste heat to power cycles are the simple steam RC and the simple ORC. The steam Rankine cycle has a high efficiency, low pump power consumption and a low-cost working fluid. The disadvantages of the steam RC are the additional water-treatment system to deionize the water and a deaerator to remove oxygen from the water and reduce corrosion of metallic parts. An additional disadvantage due to the medium temperature waste heat in a cement plant is the superheating constraint and the possible formation of droplets during the expansion. Nevertheless, turbines can operate with expansion into the two-phase region, Moreover, the waste heat temperature in the cement plant is on the lower boundary where Rankine cycle can be considered.

The advantages of the ORC are that superheating can be avoided, the simplicity of the cycle, which does not require any additional equipment, its compactness due to higher fluid densities and the lower evaporating pressure. The disadvantages are the lower efficiency compared to the Rankine cycle and the working fluid characteristics, which are often flammable and possibly toxic and non-environmentally friendly. Cyclohexane, cyclopentane, isopentane, toluene, and R142b are working fluids used for waste heat to power cycles in cement plants (Table 2). In the direct comparison between RC and ORC by Sanaye, et al. [43] and Karellas, et al. [8] the RC was superior to the ORC with higher efficiency and lower pay-back times.

More complex configurations of the RC or ORC should be avoided. They result in slightly higher efficiencies, but the costs of additional equipment usually make these configurations inferior to the simple configurations. The only configuration that could be considered for the cement plant is recuperated cycle. Fierro, et al. [46] found a considerably shorter payback time including a recuperator into their ORC. On the other hand, Lecompte, et al. [16] recommends recuperated cycles only if a lower cooling limit of the flue gas exist, which is not the case in the cement plant.

The Kalina cycle and the trilateral cycle can be excluded from the considerations. The Kalina cycle is complex and the initial costs are high. Economically it will be inferior to the simple RC and ORC. The trilateral cycle has a large increase in volume flow over the expander. Moreover, it is unclear if efficient two-phase expanders in the size needed are available.

Interesting to investigate are zeotropic fluids for ORCs. They may increase efficiency of the cycle without additional costs. A benzene/toluene mixture is recommended for considered temperature range in the cement plant [15]. However, limited practical experience exists. Therefore, unexpected challenges may occur, e.g. pressure increase in the condenser due to blockage effects because only one fluid condense.

Recently, several authors proposed for cements plant supercritical CO_2 BCs [35, 51]. We will call it tCO_2 Rankine cycle. They report higher efficiencies. However, an economic analyses and comparison to a steam RC and ORC was not performed. The tCO_2 RC must be operated at high pressures of about 200-250 bar. This makes the cycles very compact. However, thicker pipes and other material are required. Therefore, long pipes should be avoided otherwise they might become a considerable cost factor.

In the following thermodynamic analyses of a RC, ORC with a single working fluid and a zeotropic mixture and a tCO_2 RC are performed.

4 Heat exchangers in cement plant – Examples from case studies

The heat exchanger design is an important part of the waste heat recovery. However, the optimal heat exchanger design will be individual for each cement plant. The primary heat exchangers for waste heat recovery might be arranged in parallel with existing air-to-air heat exchangers, or spray towers. They can also be installed as stand-alone systems and replace the existing air-to-air heat exchangers or spray towers. It is, however, more common to install a bypass to avoid jeopardizing the primary process in case of failures of the turbine or generator in the heat to power cycle. Moreover, unstable and varying process conditions that can lead to overheating make a good control strategy essential for the operations of the heat exchangers. The control system is out of the scope of this deliverable.

Different heat exchanger design is necessary for the two waste heat sources at the preheater and the clinker cooler. The dust load and dust quality at both sources are different. Usually, the dust load in the exhaust gas from the preheater is relatively high. The dust can be very fine and might be sticky. The compounds that condense at the heat exchanger pipes can, for example, be chlorine salts of heavy metals. The dust concentration from the clinker cooler exhaust gas is relatively low if a cyclone is installed upstream of the heat exchanger. But the flow rate and temperature vary continuously. Nevertheless, a heat exchanger for the clinker cooler exhaust gas is smaller and simpler to design than the heat exchanger after the preheater. The main concern for the clinker cooler heat exchange is to control the air flow distribution, which can be investigated by aerodynamic simulations.

The following techniques can be applied in the HX to handle the dust loads [60]:

- Air blowers
- Impact dust removal techniques (knocking etc.) to remove dust from the heat exchanger walls.
- Soot blowing equipment to remove dust from the heat exchanger walls.
- If the clinker cooler is installed upstream of the dust removal equipment a steel ball "soot cleaning" system can be installed to remove dust.

A careful design of the heat exchanger to avoid dust deposition is important. Moreover, accurate welding of pipes in the heat exchanger is of importance and must be verified for each pipe. Otherwise, loss of liquid (diathermal oil) or false air in-leakage can happen. Moreover, detection of these holes is after installation impossible without removing the insulating material of the heat exchanger.

4.1 Examples from real installations

There are a few examples of heat recovery from cement flue gas in real plants. An overview over examples described in the literature is given in Table 3. In these examples tube bundles without fins are usually used for the air-to-liquid heat exchangers.

Table 3. Overview of installed heat exchangers in cement plants.

Reference	Heat to power cycle	Preheater heat exchanger	Clinker cooler heat exchanger
Mirolli [60]	Kalina cycle	- Retractable sootblowing equipment to handle dust. - 80-200 g/Nm^3 dust load - 350-400°C exhaust gas temperature	- Retractable sootblowing equipment to handle dust. - 5-10 g/Nm^3 dust load (0.03-0.05 g/Nm^3 after dust removal) - 200-300°C exhaust gas temperature

Rizzi, et al. [61], Rizzi, et al. [62]	ORC	- Dust removal by cyclones prior to HX. - Crossflow tube bundles with dust evacuation - 320°C exhaust gas temperature	
Ramshaw, et al. [55]	ORC	- Crossflow tube bundles with horizontal gas flow and hammer rapping system. - 80 g/Nm^3 dust load - 370°C exhaust gas temperature	- Crossflow tube bundles - Prior to HX dust removal with electrostatic precipitator - 290°C exhaust gas temperature
Börrnert [56]	ORC	- Crossflow bare tube bundle with horizontal gas flow and knocking technology cleaning system. - 50-100 g/Nm^3 dust load - 370°C exhaust gas temperature	- dust settling chamber (cyclon de-duster) at entry of clinker cooler HX.
Hunter and Ray [41]	Steam RC	- Boiler with sootblowing passages	- Boiler with sootblowing passages

In Mirolli [60], a case study for installing a Kalina cycle, it is reported that the dust quantity in the preheater gas is about 80-200 g/Nm^3 at a temperature of 350-400°C At the clinker cooler the dust loads are lower with 5-10 g/Nm^3 at a temperature of 200-300°C, but the dust is abrasive. After the dust removal the dust content of the clinker cooler exhaust gas is 0.03-0.05g/Nm^3. As dust removal techniques cyclones, electrostatic precipitator or baghouse are mentioned as possibilities. Without dust removal it is recommended to maintain the hot air velocity at less than 6m/s to prevent erosion in the clinker cooler heat exchanger. A conventional retractable sootblowing equipment is suggested to remove dust from the heat exchanger walls. Sootblowing equipment was implemented in another cement plant WHR project. In a sootblower usually steam is used as a blowing medium to remove soot from the heat exchanger tubes. It is mentioned that experience with Kalina cycle power plants with heat source temperatures up to 900°C with dust laden gases exist.

Rizzi, et al. [61], Rizzi, et al. [62] describe the installation of an ORC for a cement plant in Ait Baha, Morocco, which uses waste heat from the kiln. The exhaust gas is at a temperature of about 320°C and is cooled down to about 230°C in the heat exchanger. The heat is transferred to a thermal oil loop using a diathermic oil. The dust is removed by cyclones prior to the heat exchanger. The exchangers themselves are tube bundles with a dust evacuation (Figure 16 and Figure 17).

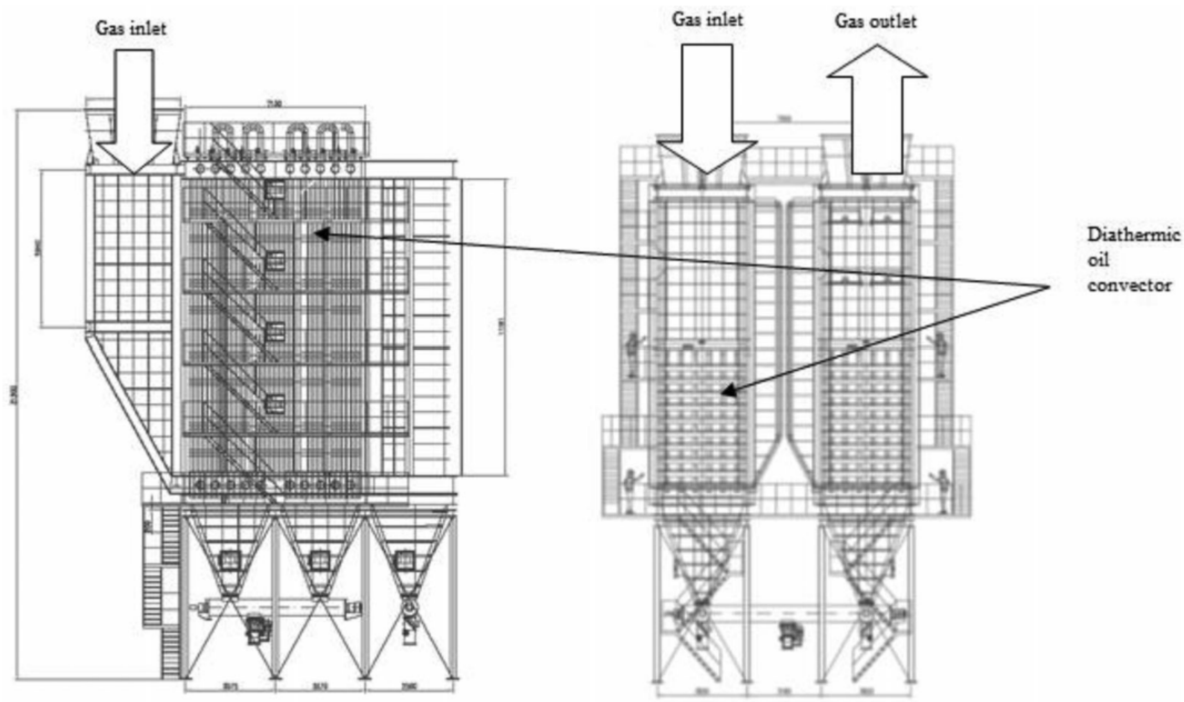


Figure 16. Heat exchanger and convectors to recuperate heat from the kiln gases [61].

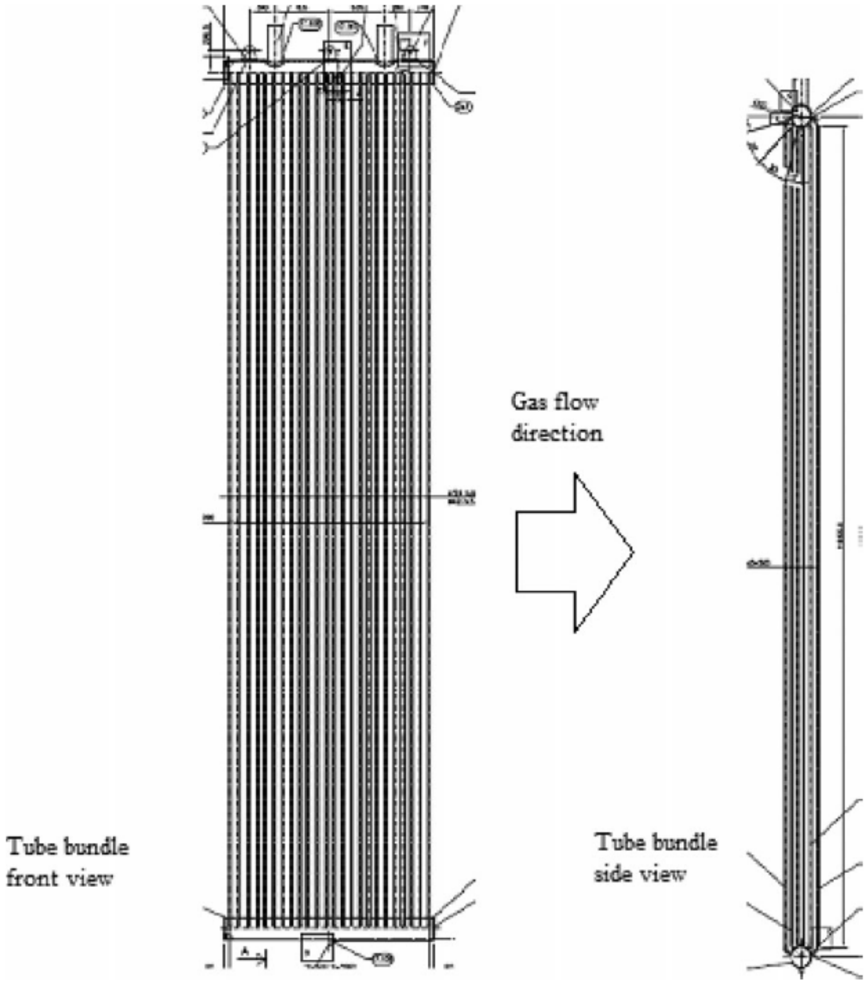


Figure 17. Heat exchanger in Ait Baha, Moroco [62].

A case study for the experience with an ORC in a cement plant is presented by Ramshaw, et al. [55]. Heat transfer tubes are used to transfer the heat from the clinker cooler (290°C) and the kiln preheater (370°C) to a thermal oil circulating in the tubes (Figure 18). The kiln preheater consists of two parallel heat exchangers with horizontal gas flow. Horizontal gas flow design allows closer tube stacking and a smaller volume compared to vertical flow heat exchangers. However, it was chosen because it reduces the risk of fouling. The gas crosses the tube bundles, and a hammer rapping system is adopted to keep the tubes clean on the gas side, where the dust load is about 80g/Nm3.

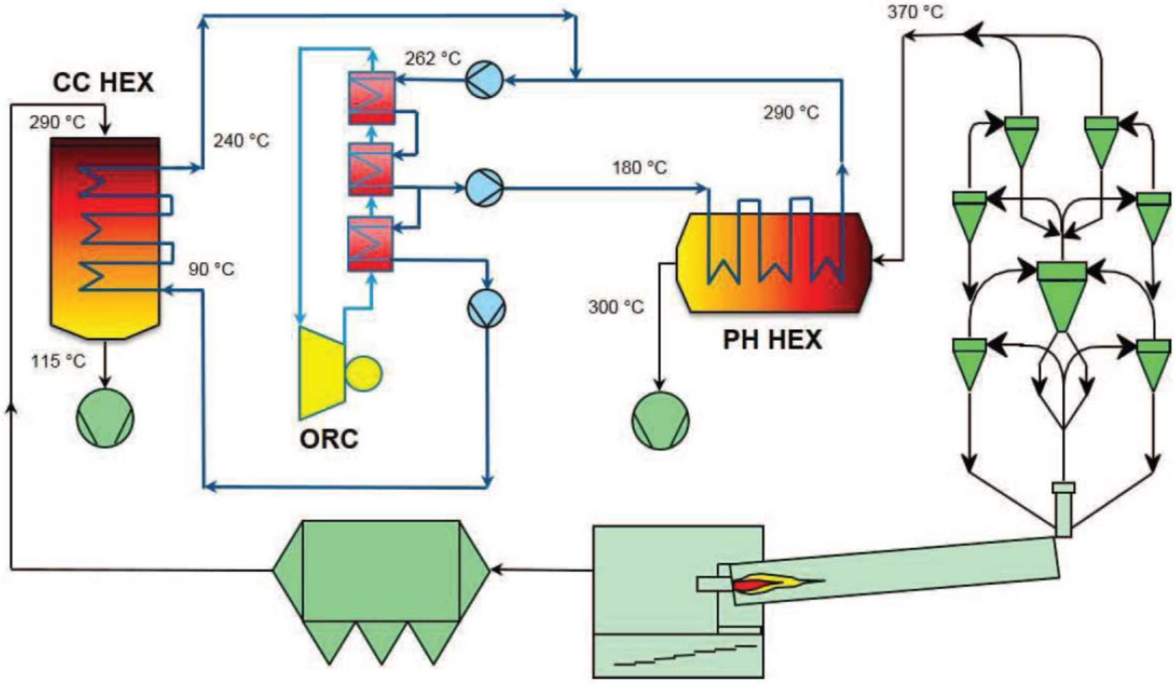


Figure 18. ORC waste heat recover system [55].

The heat exchangers are installed in a very narrow space between existing preheater tower and cooling tower (Figure 19).

Figure 19. Preheater heat exchanger (left) and clinker cooler heat exchanger (right) [55].

The clinker cooler heat exchanger is installed downstream the clinker cooler electrostatic precipitator and is also a gas/thermal oil heat exchanger with tube bank arranged in cross flow with the gas. The thermal oil is not only used to transfer heat to the ORC but also to supply heat to a catalytic emission reduction section to reduce NOx.

Börrnert [56] describes the installation of an ABB ORC power plant at a cement plan in Untervaz, Switzerland. The waste heat covers about 20% if the plant's entire power consumption.

The heat is transferred to the power plant by an intermediate cycle using pressurized water. Two heat exchangers are installed, which recover heat from the gas flow after the preheater (370°C) and after the clinker cooler. The dust load in the preheater exhaust gas is about 50-100g/Nm^3. The horizontal gas flow heat exchanger from the preheater is a bare tube type with the geometry design according to the dust levels. Especially important is sufficient tube inter-space to avoid fouling. However, large tube inter-space reduces the heat exchange The integrated cleaning system based on knocking technology is included in the heat exchanger. The gas from the clinker cooler is totally dry but the dust is abrasive. Therefore, a dust settling chamber (cyclon de-duster) is installed at the entry of the clinker cooler vertical gas flow heat exchanger.

Hunter and Ray [41] describe their waste heat recovery system for the cement industry. It is not clear if their system was implemented in a cement plant. The heat exchanger (a boiler) is a tube bundle with water in the tubes Several sootblowing passages are installed to handle dust accumulation (Figure 20). In fact, it is mentioned that the boiler design used here is more accommodating of dust loading than heat exchangers typically found in clinker cooler applications.

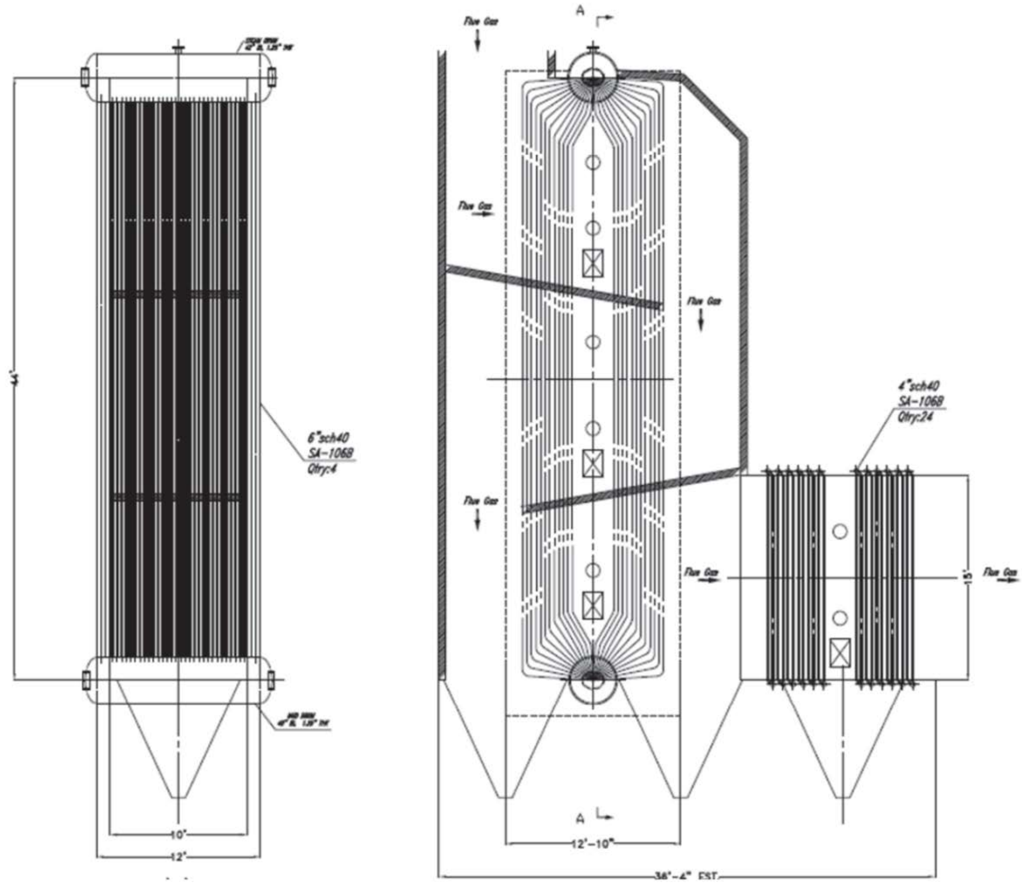


Figure 20. Vertical boiler arrangement illustrating downflow design, water in tubes, bottom hopper and sootblowing passages [41].

4.2 Research and development

A system to recover waste heat and handle dust in the exhaust gas from cement plants is described in the patent by Kalina [63]. The system consists of cyclones to remove large particle, a scrubber to reduce the dust load further, a pump to increase the pressure in the gas flow existing the scrubber, a filter (e.g. knitted mesh filter) to remove the rest of the dust before entering the waste heat recovery heat exchanger. A heat exchanger design to handle dust loads in cements plants is presented in a patent by Chawla [64]. In the heat exchanger the gas passes narrow stream which cross each other through s structured packing. The transferred moment from the streams induce rotation. The droplets and particles in the gas stream are thrown to the strips of the packing and separated from the gas. Another heat exchanger design patent for gas with high dust loads is presented in Berkestad, et al. [65]. The design is like the tube bundles used in the previously presented case studies. The heat exchanger has a U-shape with tube bundles inside. The U-shape helps to remove dust from the gas. It is, however, unclear if these three patents were ever installed in a cement plant.

Brandt, et al. [66] shows the development of a heat exchanger for an electric arc furnace at steel mills. The exhaust gas has also high dust loads for which the heat exchanger design is optimized. Important parameters to reduce fouling are the tube diameter and tube arrangement in the heat exchanger. The heat exchanger is used to transfer heat to a thermal oil. Another study of designing heat exchangers for metallurgical off-gas is presented in Skjervold, et al. [67]. An in-house modelling framework developed by SINTEF Energy Research [68] is used in the design of the heat exchanger. While there are similarities with the cement case, it must be pointed out that the dust characteristics and composition is different which also can cause very different problems and solutions for the heat transfer compared to a cement plant. Nevertheless, it is expected that the heat exchanger design and dust handling in a cement plant might be easier than in the metallurgical industry. Moreover, none of the challenges with the dust concentration in a cement plant cannot be addressed by a well-designed heat exchanger.

5 Thermodynamic analysis

In this section thermodynamic analyses of the most promising waste heat to power cycles are presented.

In this section it is assumed that a hot oil loop transfers the heat to the waste heat to power cycles. The hot oil loop operates in a temperature range of 314℃ to 40℃ and has a mass flow of 19.10 kg/s. The temperature difference between hot oil loop and the exhaust gas stream of the cement plant is assumed to be 80K, which is believed to be conservative. The hot oil loop can provide 11.85 MW of heat to the heat-to-power cycle, which is in the same magnitude as the heat exchanger duty of the ORC designed in the CEMCAP project. In the CEMCAP project 18.2 MW of heat was extracted from the exhaust gas from which 6.35 MW was used to heat up the air to the raw mill. The purpose of the thermodynamic analysis is, however, not to replicate the results in CEMCAP, but compare different bottoming cycles. It is, therefore, expected that differences in the power production of this work and the CEMCAP project exist. The sensitivity of the heat to power cycle regarding temperature changes in the hot oil will also be studied. These temperature changes would be equivalent to a smaller ΔT and a less conservative assumption about the heat transfer to the hot oil loop. In all studied cases the ambient temperature and cooling water temperature in the inlet of the condenser are set to 15℃. The minimum temperature difference in all heat exchangers in the cycle is limited to 10K and a minimum vapor fraction of 0.85 is allowed in the expansion.

The heat to power cycle is optimized using a detailed SINTEF in-house model for heat exchangers and optimizing of bottoming cycles build upon the NLPQL subroutine from Shittkowski [69, 70]. The fluid libraries thermopack [71] and REFPROP 10.0 [72] are used to represent the organic fluids and water.

5.1 Steam Ranking cycle

A simple steam Ranking cycle is studied (Figure 9). The steam RC is optimized to produce maximum power for the BAT cement plant. The variables and their constraints for the model are displayed in Table 3.

Table 4. Constraints of the steam Ranking cycle.

	Minimum	**Maximum**
Inlet pressure to turbine [bar]	8	220
Condensing pressure [bar]	0.035	3.8
Inlet temperature to turbine [℃]	150	490
Working fluid mass flow [kg/s]	0.25	75

Flowrate of cooling fluid in condenser [kg/s]	50	5000

In the optimal case the steam RC recovers 7.2 MW from the hot oil stream and can produce a net power of 1.96 MW. A hot oil cycle is used for a better comparison with the ORC. The waste heat utilisation is 27.06% and a first law efficiency of 15.54% is achieved. A disadvantage of the steam RC is the requirement to superheat. 'This allows only a small mass flow of the working fluid of 2.5 kg/s. As a result, the heat from the hot oil stream cannot be recovered efficiently. Only 7.2 MW of the 11.85 MW available heat is transferred to the working fluid. Moreover, the hot oil stream has still a temperature of about 166℃ when it leaves the evaporator. The problem are two opposing effects. The mass flow of the working fluid should be increased to recover more heat, but on the other hand, superheating is necessary for the power production and the vapour fraction after the turbine must be larger than 0.85, which limits the mass flow. The consequence of the limited mass flow is a relatively low first law efficiency of the steam RC.

The temperature and pressure profile of the cycle is given in Table 3 and the T-s and energy diagram is displayed in Figure 16.

Table 5. Temperatures and pressures in the optimised steam Rankine cycle.

Variable	Temperature	Pressure
Inlet turbine	285	13
Inlet condenser	31.3	0.046
Outlet condenser	25.0	0.036
Inlet evaporator	25.2	16.25

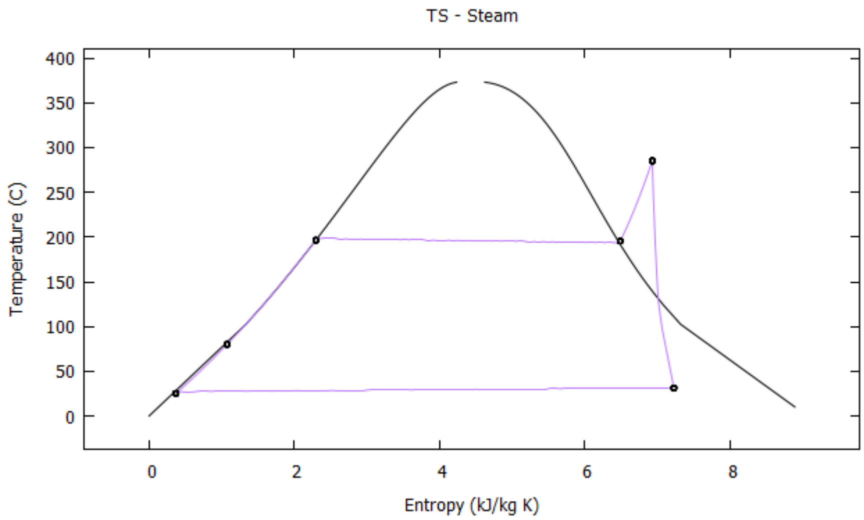


Figure 21. T-s diagram of the Rankine cycle.

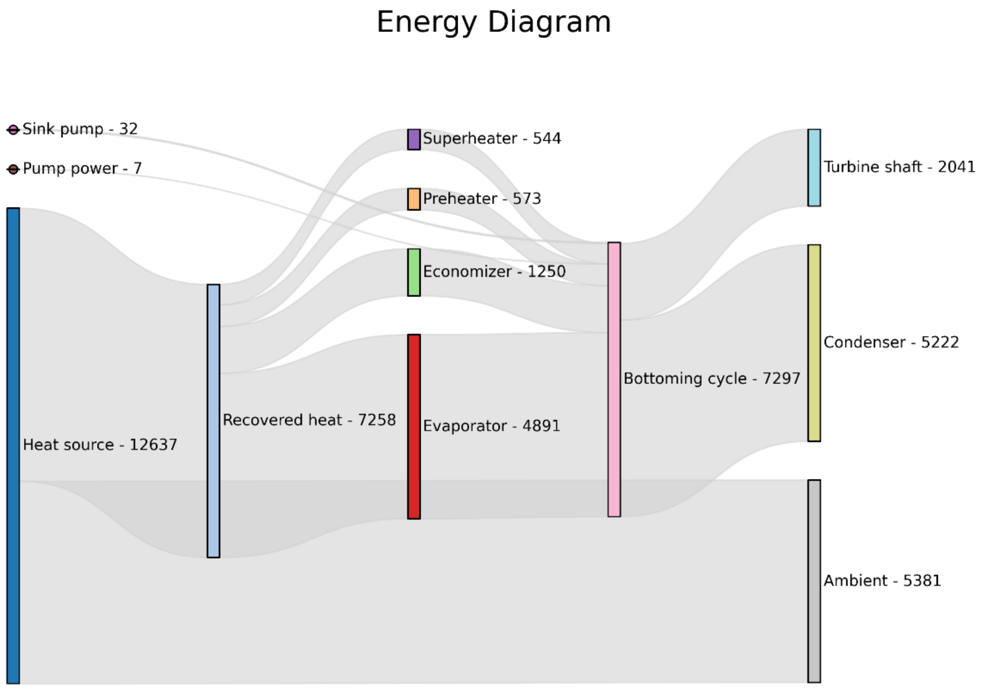


Figure 22. Energy diagram of steam RC.

Interestingly, the pressure is quite low at the optimum of the cycle. Moreover, it is even at the lower boundary of the constraints. However, the power production increases only negligibly at a lower pressure. A large part of low temperature energy is lost in the condenser. This is low temperature energy, which cannot be utilised elsewhere in the process.

In the following a sensitivity study is performed. It is tested how the steam RC is affected by higher temperatures of the hot oil source (Table 4). The hot oil can only be used up to a temperature of 350℃. For higher temperatures, a direct heat recovery is assumed.

Table 6. Sensitivity of the steam Rankine cycle to temperature changes in the hot oil source.

	Increased temperature but same available heat				**Increased temperature but same heat exchanger duty**		
T [℃]	Available heat [MW]	Net produced power [MW]	Utilisation in evaporator [%]	1st law efficiency	Heat exchanger duty [MW]	Net produced power [MW]	Heat utilisation [%]
314	12,64	1,96	57,42	15,5	7,26	1,96	27,0
344	12,64	2,28	65,67	18.0	7,26	1,99	27,4
374	12,64	2,15	62,63	17,0	7,26	1,97	27,14
404	12,64	2,38	60,23	18,8	7,26	2,27	31,3
434	12,64	2,55	64,10	20,2	7,26	2,29	31,5

Higher temperatures can increase the net produced power and the efficiency of the steam Rankine cycle. While the heat utilisation within the steam RC is high, the first law efficiency is considerably smaller because of the already mentioned problem of transferring the heat to the cycle.

5.2 Organic Rankine cycle

In this section a thermodynamic analyse of the ORC is performed. Important for the performance of the cycle is the working fluid. Cyclohexane, toluene, isopentane, R142b, benzene and methylcyclopentane were tested. Benzene resulted in a high power production and relatively consistent convergence of the optimization algorithm. Therefore, benzene was chosen as one of the working fluid. In addition, methylcyclopentane and butane are tested.

5.2.1 ORC with benzene

The variables and constraint for this case are displayed in Table 6. The main difference to the steam RC is that the maximum allowed expander pressure was reduced. It was noted that the optimization with some working fluids resulted in high cycle pressures. The constraint limited these pressures.

Table 7. Variables and constrained for the optimization of the ORC.

	Minimum	**Maximum**
Turbine inlet pressure [bar]	8	60
Condensing pressure [bar]	0.035	3.8
Inlet temperature to turbine [℃]	80	490
Working fluid mass flow [kg/s]	0.25	75
Flowrate of secondary fluid in condenser [kg/s]	50	5000

The temperature and pressure profile of the optimized ORC with benzene as the working fluid is presented in Table 7. The T-s diagram is displayed in Figure 18.

Table 8. Temperature and pressure changes in the ORC with benzene.

Variable	Temperature	Pressure
Inlet turbine	216.8	18.6
Inlet condenser	71.0	0.15
Outlet condenser	25.0	0.14
Inlet evaporator	26.0	21.9

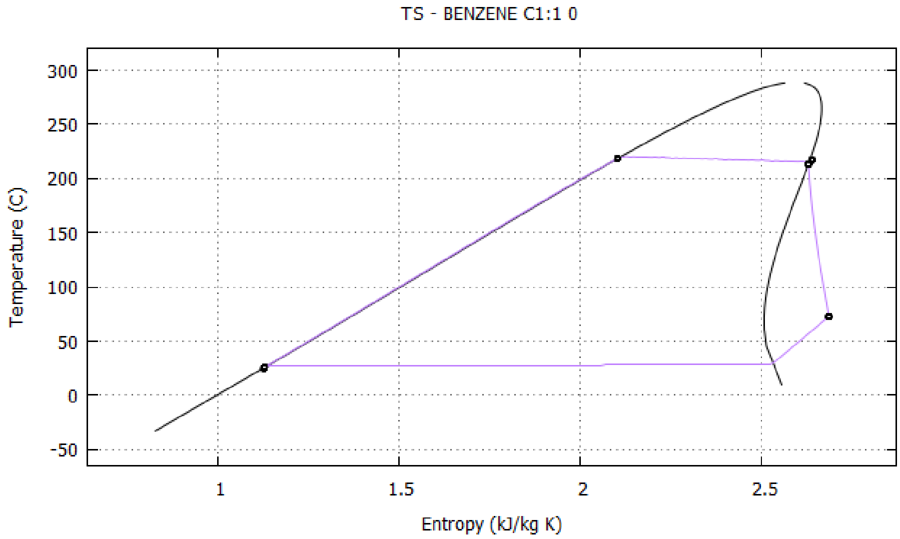


Figure 23. T-s diagram of the ORC with working fluid benzene.

Interestingly, the ORC can recover considerably more heat from the hot oil cycle. The duty in the evaporator is about 11.0 MW, which is about 4 MW more recovered heat than in the steam RC. This is also the reason for the considerably larger net produced power production of the ORC, which is 2.8 MW. The ORC has a recovered heat utilisation of 25.0%, which is approximately the same as the steam RC. However, the first law efficiency is 22.5%, which is considerably larger than for the steam RC. Superheating is not necessarily required for the ORC using benzene, which allows greater variability in the working fluid mass flow. Consequently, the ORC can recover more heat from the heat source.

It can be observed that most of the energy (about 7.9 MW) is lost in the condenser. The inlet temperature to the condenser is about 71℃ in the considered case using benzene. Consequently, there is useful energy available that can potentially be used for heating up the air for the raw mill. A desuperheater could be installed between turbine and condenser. The option must be evaluated economically. Nevertheless, the heat demand to heat up the air stream to the raw mill is about 6.35 MW, which can be partly taken from the superheated working fluid in the ORC. In the current ORC with benzene as a working fluid about 0.9 MW of heat is removed from the working fluid in a desuperheater. Using this heat in the cement plant can result in a higher WHR rate and larger net produced power in the ORC.

5.2.2 ORC with methylcyclopentane

If the heat after the turbine can be utilised elsewhere in the cement plant choosing a working fluid with a higher temperature behind the turbine might be favourable. Methylcyclopentane can be operated transcritically with a turbine inlet pressure of 40 bar. It recovers 11.9 MW from the hot oil stream. 24.4% of the recovered heat is utilised in the cycle, which results in a first law efficiency of 22.7% and a net produced power of 2.9 MW. The working fluid mass flow is 16.6 kg/s and the inlet temperature to the condenser is 130℃. Moreover, about 2.7 MW of heat is removed between turbine and condenser. The pressure for the ORC with methylcyclopentane was chosen manually, but it was observed that higher pressures increase the power production.
If the ORC with methylcyclopentane is operated with a turbine inlet pressure of 30 bar the ORC is operated subcritical. Then it produces 2,8 MW net power (1st law efficiency 22.3%) and between turbine and condenser heat of about 2,6 MW is removed from the working fluid (Figure 18). These low temperature heat could be utilized to heat up the air stream to the raw mill.

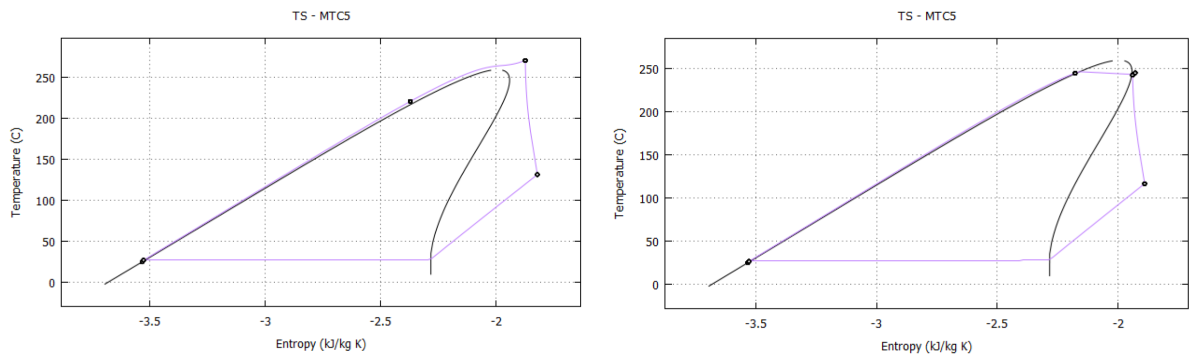


Figure 24. T-s diagram of ORC using methylcyclopentane. Left transcritical; Right subcritical

5.2.3 ORC with butane

Butane is another working fluid that can be used in the temperature range considered. Butane is more environmental friendly and easier to handle than, for example, benzene. The ORC with butane is operated transcritically at a high pressure (Figure 20). The maximum pressure of the cycle was limited to 103 bar. The positive consequence of high pressures is the compactness of the resulting cycle. The disadvantages are high pumping power and possible safety concerns. The ORC with butane can produce 2,4 MW of net power with a first law efficiency of 19.3%. The expander produces 2,8 MW, but about 0,4 MW pumping power is required due to the high pressures in the cycle. The advantage of the ORC using butane is the high condensation pressure of 2.6 bar. The other ORCs considered in this section operated at subatmospheric pressure, which might cause leakage into the cycle and pollution of the working fluid. The inlet temperature to the condenser is 167℃ and the working fluid mass flow about 14.2 kg/s. It is possible to remove heat of 4.1 MW between turbine and condenser. The heat could be used to heat up a source for heating up the air stream to the raw mill. A hot oil stream, for example, of 34,8 kg/s could be heated up from 15 to 159℃ in the condenser. This hot oil stream carries heat of 9,3 MW, which is sufficient to heat to heat air up to about 130℃ assuming a pinch temperature in the hot oil/air heat exchanger of 29℃. Moreover, in the current set-up it is assumed that the heat for the hot air is extracted before the ORC. Consequently, more power could be produced by using instead the heat extracted from the ORC in the desuperheater and condenser.

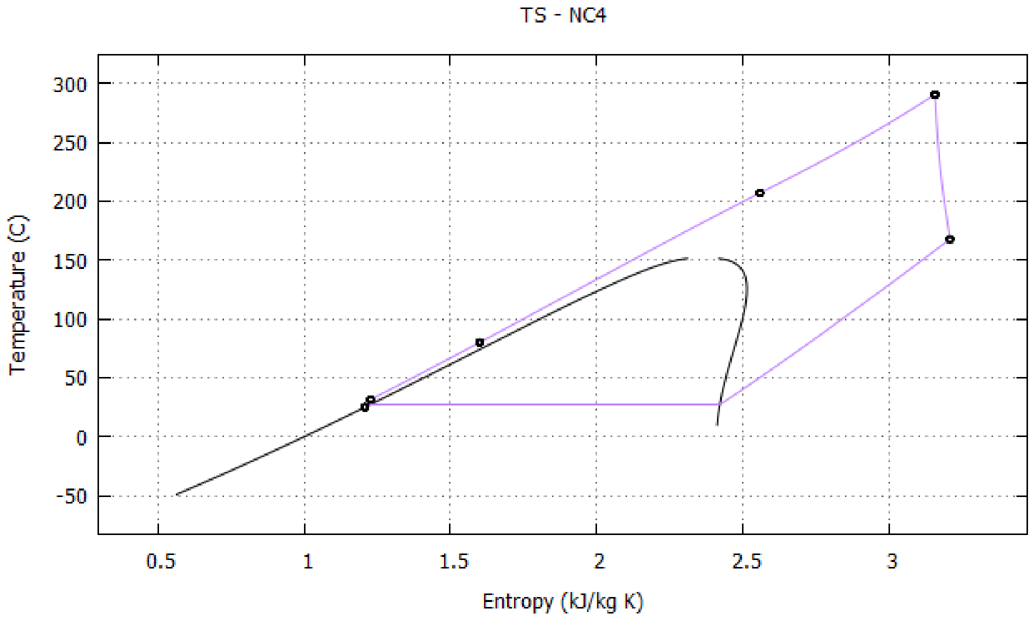


Figure 25. T-s diagram of ORC using butane.

5.2.4 Sensitivity analysis for ORC with benzene

In this section the ORC is tested for higher temperatures (Table 8). The hot oil Therminol 66 can be used until 352℃, thereafter Therminol 72 can be used. Therminol 72 has an operation range of 80℃ to 380℃. Therefore, the highest temperature investigated for the ORC is 374℃. The produced power increases with increasing temperatures. At a temperature of 374℃ the ORC is operated transcritically. In comparison with the steam Rankine cycle the ORC still produces more power even at higher temperatures. The first law efficiency at 374℃ is 29%, while the steam Rankine cycle reaches 20.2% at 434℃. The heat utilisation in the cycle is, however, similar for the ORC at 374℃ and the steam Rankine cycle operated at 434℃. The challenge in the steam RC is, however, to transfer available heat to the bottoming cycle.

Table 9. Sensitivity of ORC.

	Increased temperature and heat exchanger duty				Increased temperature but same heat exchanger duty		
T [℃]	Available heat [MW]	Net produced power [MW]	Utilisation in evaporator [%]	1st law efficiency	Heat exchanger duty [MW]	Net produced power [MW]	Heat utilisation [%]
314	12,64	2,84	89,74	22,50	11,34	2,84	25,1
344	12,64	3,29	94,86	26,00	11,34	3,11	27,40
374	12,64	3,67	99,15	29,00	11,34	3,38	29,80

5.2.5 Zeotropic mixtures

The ORC with zeotropic mixtures that outcompeted the simple ORC with a pure working fluid was not found. However, only a limited time was spent to test a few mixtures including a toluene/benzene mixture. Therefore, it might be possible that mixtures can be found that result in larger power production than with a pure working fluid.

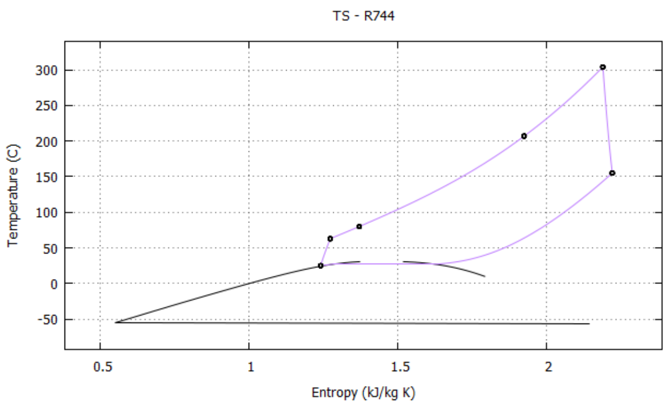
5.3 Transcritical CO_2 Rankine cycle

In this section the transcritical CO_2 RC (ttCO_2 RC) is thermodynamically analysed. A transcritical cycle needs considerably higher cycle pressures than an ORC or steam RC. Therefore, the constraints on the pressure at the turbine and condenser are increased (Table 9). In Kizilkan [35] and Olumayegun and Wang [51] the cycle is called supercritical Brayton cycle. We choose to call it transcritical Rankine cycle.

The tCO_2 RC recovers 10,8 MW heat of the 12,6 MW heat from the hot oil cycle (Figure 20).

Table 10. Constraints for the tCO_2 Rankine cycle.

	Minimum	Maximum
Turbine inlet pressure [bar]	8	320
Condensing pressure [bar]	0.1	100
Inlet temperature to turbine [℃]	80	490
Working fluid mass flow [kg/s]	0.25	75
Flowrate of secondary fluid in condenser [kg/s]	50	5000



Variable	Temperature	Pressure
Inlet turbine	304	320
Inlet gas cooler	155	67
Outlet gas cooler	25	67
Inlet evaporator	63	323

Figure 26. T-s diagram and process data of tCO_2 Rankine cycle.

At the turbine it produces 3,2 MW power, which is considerably larger than for the ORC and the steam RC. However, the net produced power is only 1,9 MW, since 1,3 MW is used to operate the pumps of the cycle. Consequently, the larger power production as reported in Kizilkan [35] is observed here as well, but only if the required pumping power is neglected. Included this loss the tCO_2 RC has a lower cycle efficiency. The first law efficiency of the tCO_2 RC is just 14.8%, which is lower than the first law efficiency of the steam RC and the ORC. Alternatively, the use of R125 was proposed to reduce the pumping power [73], but in our analysis it was not superior to the tCO_2 RC.

The advantage of the tCO_2 RC might be the heat recovery at the condenser. The gas cooler inlet temperature is 155℃. A large amount of heat could be used here to heat up the air stream to the raw mill. The heat available from the gas cooler is 8,74 MW, which is sufficient to heat up the required air stream. Consequently, only a small amount of additional heat must be removed from the cement plant to heat up the stream to 190℃. However, even if about 5-6 MW can be saved for heating up air, the tCO_2 RC with an efficiency of about 15% will produce just about 0.9 MW more power, which is still less than the ORC. The advantage of the tCO_2 RC can be the hight operating pressure and the small volume flows. Consequently, the components are smaller than for the other cycles.

6 Discussion

The thermodynamic analyses showed that the simple ORC is superior to the other heat to power cycles considered in this report. Therefore, the recommendation is to study ORC for the oxyfuel cement plant.

The steam RC was not able to recover the same amount of heat from the hot oil cycle since the available working fluid mass flow is constrained to allow superheating in the evaporator. Karellas, et al. [8] assumed full heat recovery in the evaporator and the power production is based on the efficiency of the cycles. Mohammadi, et al. [42] had a lower temperature constrained of 180℃ on the exhaust gas, which might favour the steam RC. Unclear is, however, why Sanaye, et al. [43] was able to recovery more heat with the steam RC than with the ORC and why they did not have the same problem between increasing working fluid mass flow to recovery more heat in the evaporator and decreasing the mass flow to allow more superheating. In our case the ORC was superior to the steam RC even at source temperature of 434℃. At these source temperatures the ORC is operated at 374℃ since maximum temperature of Therminol 72 is 380℃ while the steam RC can be operated at higher temperatures using a direct heat exchange between exhaust gas and bottoming cycle. It is expected that the ORC is still superior to a steam RC even if the steam RC is operated with a direct heat exchange and the ORC with an indirect heat exchange using a hot oil loop. In addition, the often smaller kilns and higher raw material moisture in northern Europe might be additional reasons to choose the often compacter, simpler, and fully automated ORC.

The tCO2 RC was in our thermodynamic analysis also clearly inferior to the ORC. The difference between our analysis and the one by Kizilkan [35] is the efficiency of the cycle. Kizilkan [35] found a utilisation of the recovered heat of about 36% while our heat utilisation in the tCO_2 RC was just about 29%. The considerably larger degree of utilisation allowed a larger efficiency of the tCO_2 RC even if the pump and compression power consumption was deducted from the produced power. In our case the larger pump and compression power could not be recovered by the slightly higher heat utilisation in the tCO_2 RC.

The ORC allows low heat recovery of the working fluid before the condenser. This can be an interesting heat source for the hot air stream, which is supplied to the raw mill. The first law efficiency of the ORC is about 22%. Consequently, about 1/5 of the energy supplied to the hot oil loop can be transformed to power. If 4-5 MW of low temperature heat can be utilised to heat up the hot air steam to the raw mill, the power production could be increased by 0.8-1 MW. This is an interesting option that should be considered in the further design of the heat to power cycle of the oxyfuel cement plant.

7 Conclusion

The document investigates waste heat recovery options the oxyfuel cement plant. In the AC²OCem project three different oxyfuel cement plant configurations are studied. The BAT plant is a theoretical case while the Lägerdorf and Slite plants are actual cement plants. For the initial investigations of waste heat recovery solutions, the temperature levels are of importance, which are in the range of 340 - 460℃. In the Lägerdorf plant V1 oxyfuel the temperature level is just 220℃, which is considerably lower than the primary temperature range studied in this document.

Several waste heat recovery solutions are thoroughly reviewed, and the most promising solutions are identified. These options are thermodynamically analysed and compared. The ORC is the best options for the oxyfuel cement plant with the current specifications. The ORC is thermodynamically superior to the steam RC and tCO_2 RC. Moreover, the potential installation of a desuperheater, which allows low heat recovery for potential low temperature heat demands in the plant, increases the advantage of the ORC compared to the steam RC.

These results are in line with the result from the CEMCAP project. It is expected that other heat to power cycles, like the Kalina cycle, are economically inferior to the ORC because of the more complex system design which requires more components. For the trilateral cycle it is questionable if a two-phase expander of the size needed in a cement plant is available. Other technologies, like thermoelectric generators, are clearly inferior with respect to the power production. It is, therefore, recommended to investigate in the next tasks ORCs or CO2 RC. While it is expected that the latter is inferior in power production it has advantages in component sizing. Moreover, CO_2 might be easier to handle than organic working fluids.

The heat exchanger design for the preheater exhaust gas and clinker cooler exhaust gas will be similar for each of the heat to power cycles. A common design are tube bundle heat exchangers with knocking technology to remove accumulating dust on the heat exchanger walls of the preheater HX. The clinker cooler HX is usually also a tube bundle with dust removal at the entry to the heat exchanger. The heat exchanger design should not be a limiting factor for the implementation of heat to power systems in cement plants.

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