COMPARISON OF CONVENTIONAL AND CO_2 POWER GENERATION CYCLES FOR WASTE HEAT RECOVERY

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ABSTRACT

A theoretical analysis of waste heat recovery in a cement plant was performed. Conventional power generation cycles, namely the steam Rankine cycle (SRC) and the organic Rankine cycle (ORC) were compared with novel carbon dioxide (CO_2) cycles. Particularly three cases were investigated, the supercritical CO_2 Brayton-cycle (sCO_2-BC), the transcritical CO_2 Brayton-cycle (tCO_2-BC) and the transcritical Rankine-cycle (tCO_2-RC). Simulations showed that the SRC ($3297kW$) and the ORC ($3915kW$) generate less power than the CO_2 cycles. The tCO_2-RC yielded the maximum net power output ($5445kW$) while the tCO_2-BC generates more power ($4488kW$) than the sCO_2-BC ($4197kW$) because in this case the expansion is limited by the critical pressure of the CO_2.

INTRODUCTION

Heat recovery from industrial processes is of rising interest due to the Energy Efficiency Directive of the EU and the latest United Nations Framework Convention on Climate Change in Paris, France 2015. At the investigated cement plant two major heat sources for heat recovery are available, see Fig. 1:

- the off-gas from the rotary kiln ($\theta = 295°C, \dot{m} = 78\ \frac{kg}{s}$)
- the cooling air from the grate cooler at the exit of the rotary kiln ($\theta = 410°C, \dot{m} = 14\ \frac{kg}{s}$)

The hot gas from the chlorine bypass is not included in this study due to its low mass flow. For these two sources different concepts of process integration for the heat recovery system have been evaluated in the present paper, taking into consideration steam Rankine cycle (SRC), organic Rankine cycle (ORC), see (Karellas, et al., 2016) and (Wang, et al., 2015), and CO_2-cycles (operated as Brayton or Rankine cycle). From a thermodynamic point of view efficiencies of different heat engines are equal, if the thermodynamic temperatures of heat input and output are equal for processes to be compared. Nevertheless the attainment of high temperatures for heat input and low temperatures for heat output is even influenced by the used working medium and furthermore different working media like water, an organic substance (e. g. Isobutane) for ORC or even CO_2 are characterized by certain properties having influence on the overall system performance as well as certain components.

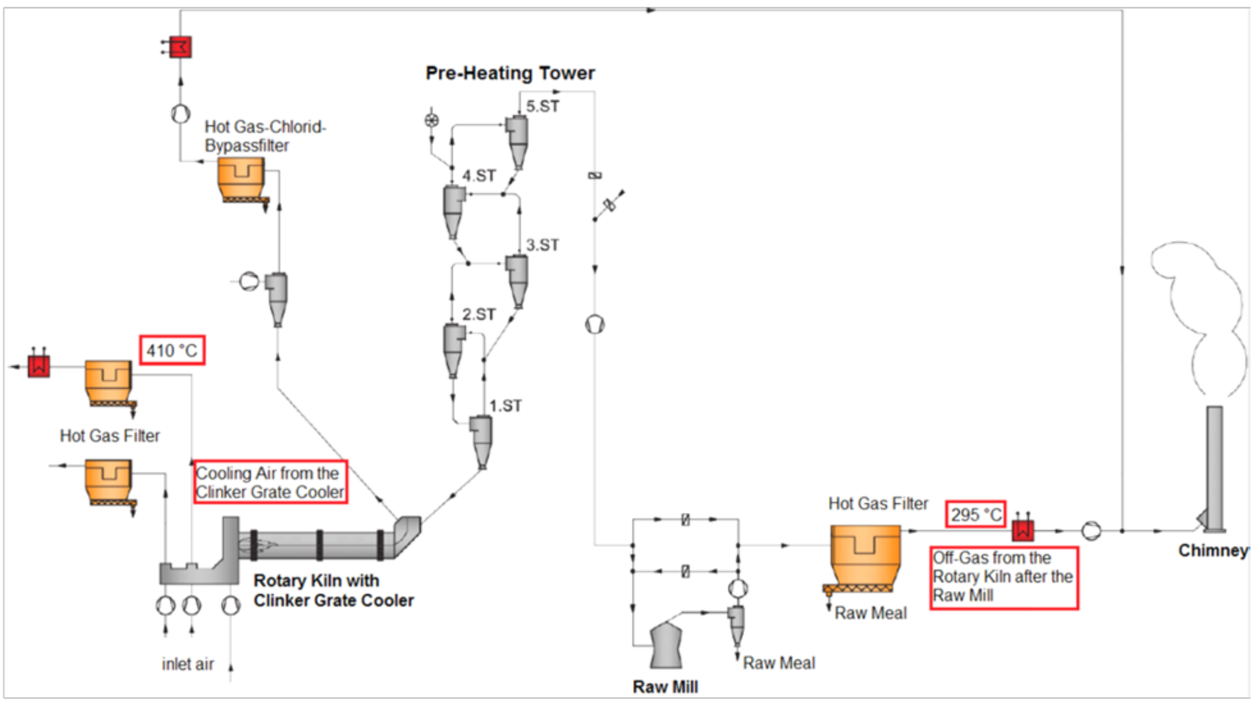

Fig. 1: Schematic of the investigated cement plant

E. g. (water) steam Rankine cycles become expensive for low temperature heat sources in comparison to potential CO_2-systems. Despite this fact SRC is well known and proven and therefore an interesting possibility for power generation from waste heat. ORC-systems offer the possibility to select the best suited working fluid for the temperature level of the heat recovery application. A potential disadvantage of ORC-media is the necessity of a further heat transfer medium between heat source and ORC due to its inflammability. Nevertheless even thermo oil, which is used for heat transfer in many cases, is even combustible and so the problem is rather reduced than solved. Pressurized water cycles as an alternative, on the other hand demand high pressure levels to prevent evaporation, which enhances costs and safety requirements. Besides SRC and ORC the Kalina process (Li & Dai, 2014), has been developed for heat recovery especially for low temperature heat sources. The most important advantage of Kalina cycles is the phase change at variable temperature which reduces the irreversibility of the heat transfer process, due to the fact that the cycle applies a mixture of NH_3/water as working medium. Nevertheless in the mean time, supercritical ORC's have been developed so this major advantage was lost. An interesting alternative are (s)CO_2-cycles for power generation. CO_2 as a working medium is harmless, fluid properties of CO_2 lead to small heat recovery system dimensions and no additional heat transfer cycle is necessary. Additionally critical data of CO_2 (p_c=73,75 bara, ϑ_c=30,98°C) are advantageous for heat transfer to the ambient. Low compressibility values for the real gas in the vicinity of the critical point (CP) create conditions for the development of closed gas turbine cycles for heat recovery, by using CO_2 as working medium.

All simulations were performed in IPSEpro V 6.0 developed by SimTech Simulation Technology (SimTech, 2015). IPSEpro is process-simulation software with modelling capabilities. Libraries contain the thermodynamic properties of the included fluids and all basic models required for process-engineering.

STEAM RANKINE CYCLE (SRC)

The Steam Rankine Cycle is a frequently used and well developed system for electricity generation from exhaust gas. Dependent on the type of application, various different power plant types, such as one- or two-pressure systems with optional feed water preheating systems are common. The choice of the best system for a specific application depends on the input temperature as well as the further usage of the cooled exhaust gas. For this simulation a two-pressure system was chosen because it allows lower exit flue gas temperatures and thus retrieving more heat. Generally multi-pressure systems, enable lower exit gas temperatures and reduce exergy losses (Effenberger, 2000).

Water is advantageous as a working fluid because it is harmless and not environmentally hazardous. However water treatment units and the size of steam turbines required make the configuration highly complex and expensive. Due to its high critical pressure and temperature ($224\ bara$ and $374\ °C$) water is a very good working fluid for high temperature applications but is less suited for low temperature systems.

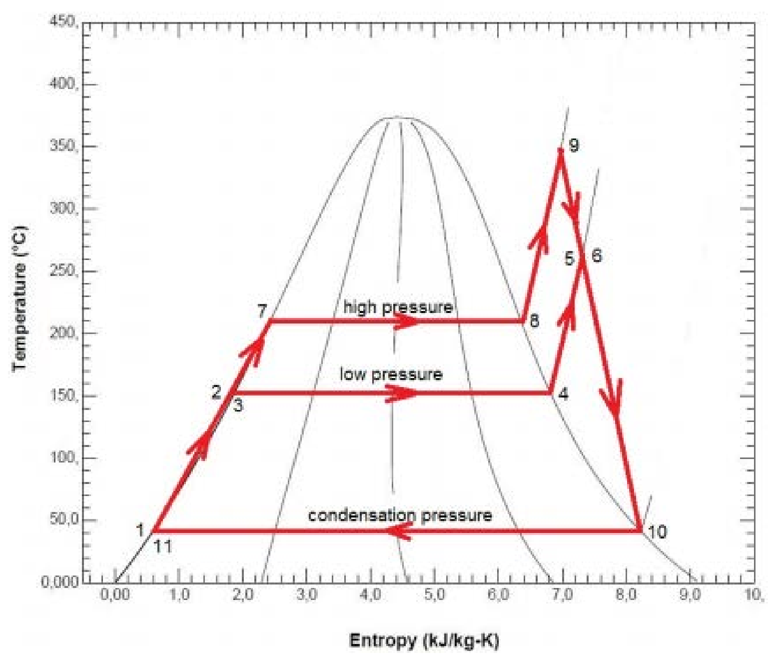

Fig. 2: Temperature vs. entropy diagram of a SRC two-pressure system

A two pressure system with superheating was investigated in order to achieve the maximum power output. The T, s-diagram of the cycle is depicted in Fig. 2. In such a plant, water is pressurized to the high pressure (HP) level (1), flows through the economizer (2) and is heated up to the inlet temperature of the low pressure (LP) evaporator (3), where it is split up into a LP and HP stream. The HP stream is heated further (7), evaporates partly and the separated steam (8) from the HP steam drum is superheated (9). It enters the HP turbine and is expanded to the LP level (6), where it is mixed with the superheated LP steam (6) from the LP steam drum (4). The mixed flow is fed to the LP turbine and is expanded to the condensation pressure (10). The vapor is condensed (11) before entering the feed water pump and closing the cycle.

Simulation

The operating parameters are listed in Table 1.

Steam Rankine Cycle - Operating Parameters	
High pressure level	$20 bara$
Low pressure level	$6.5 bara$
Max. steam temperature	$395°C$
Minimum allowed ΔT in HEX	$15°C$
Turbine isentropic efficiency	0.75

Table 1: Operating parameters of the steam Rankine cycle

The power output of the cycle with the given heat source is 3297 kW while the thermal efficiency of the cycle is calculated as $\eta_{thSRC} = 22.39$ %.

ORGANIC RANKINE CYCLE (ORC)

While the SRC performs well with high grade temperature sources, it is not the best solution for low temperature sources. The ORC is mainly applied for low temperature heat-recovery, it puts to use the same principle as the SRC but uses organic fluids instead of water. The organic compounds are characterized by higher molecular mass and lower boiling/critical temperature than water (Tchanche, et al., 2011). These properties allow the system to recover heat from low-temperature heat sources more effectively.

The selection of the best working fluid depends on the temperature of the heat source, the plant design and on the physical and thermodynamic properties of the fluid. It should be nontoxic, incombustible or inodorous; it also should have low ozone depletion and global warming potential, and a low water hazard class. Organic fluids can be divided into dry, isentropic and wet fluids. Wet fluids show a negative, isentropic a vertical and dry a positive slope at the saturated vapor curve (Chen, et al., 2010). In order to avoid the expansion into the two phase region, wet fluids like water or ammonia must be superheated while dry fluids do not reach the two-phase region after expansion. Furthermore their evaporation enthalpy is low meaning that less energy is required for vaporization.

For dry organic fluids the temperature behind the turbine is higher than the condensing temperature. Therefore it is possible to implement an internal heat exchanger (IHE) to recover heat to preheat the pressurized working fluid. The T, s-diagram of such a system is shown in Fig. 3. Due to safety reasons and in order to avoid temperature peaks, a thermal oil loop is used to transfer the heat from the heat source to the organic medium. The pressurized organic fluid (1) is heated by the IHE (2), further heated (3) and evaporated (4). The saturated vapor is expanded in the turbine (5) and cooled in the IHE (6). Finally it is condensed (7) and pressurized closing the cycle.

Simulation

Due to the high number of working fluids and their different properties it is essential to select the appropriate medium in order to maximize the cycle efficiency. No strict rules about how to select the proper organic medium exist. He et al. (He, et al., 2012) calculated the optimal evaporation temperatures of working fluids for subcritical ORC for the highest net power output and concluded that the critical temperature of the fluid should be as close as possible to the temperature of the heat source. Another indicator is the heat transfer capacity which determines the cost of the heat exchanger.

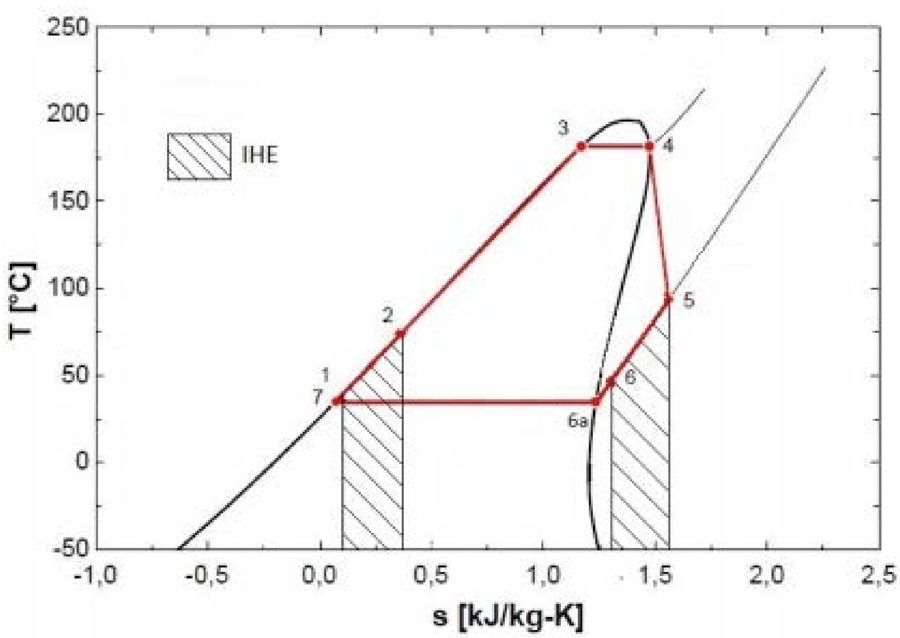

Fig. 3: T, s - diagram of an organic Rankine cycle configuration with an internal heat exchanger

Several organic fluids (namely neopentane, isopentane, cyclepentane, n-butane) were chosen according to these parameters and cyclopentane provided the best results compared to the other considered media.

Organic Rankine Cycle (Cyclopentane) - Operating Parameters	
Operating pressure	$34.6 bara$
Condensation pressure	$0.6 bara$
Max. operating temperature	$217°C$
Cooling water Temperature	$15°C$
Turbine isentropic efficiency	0.75

Table 2: Operating parameters of the ORC (cyclopentane)

The operating parameters of the cyclopentane cycle are summed up in Table 2. The values for the operating and condensation pressure were chosen so that a minimum temperature difference of $5\ K$ arises between the ORC medium and the heat source and the cooling water respectively.

The net power output of this system is $3915\ kW$ and the thermal efficiency is $\eta_{thORC} = 19.56\ \%$. These values correspond to an increase of 3% regarding net power and to an efficiency increase of 5.5 % compared to an ORC without an IHE.

SUPERCRITICAL CO_2 CYCLES

The supercritical power cycle takes advantage of the real gas behavior of the CO_2 in order to achieve high thermal efficiency. The main improvement of the supercritical CO_2 cycle is the reduced compressor work because of property changes in the region of the CP. Another benefit is the low critical temperature of CO_2 (31°C), which makes it possible to use water at ambient temperatures as a coolant. The main advantages of CO_2 can be summed up (Kuhlanek & Dostal, 2011) as:

- the high operating pressure enables smaller size components, Fig. 4
- sCO_2 cycles achieve high efficiencies at low temperatures
- well known thermodynamic properties
- stability
- non-toxicity
- not hazardous to waters
- non-flammable
- low critical temperature

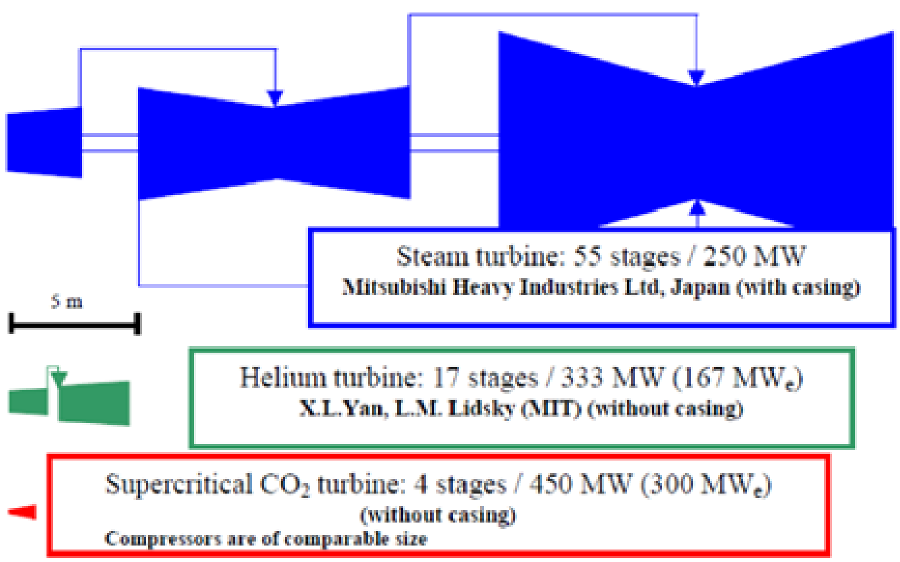

Fig. 4: Size comparison of a steam, a helium and a carbon dioxide turbine (Dostal, 2004)

- abundantly available
- high power density
- low surface tension (reduced effects of cavitation in the machinery)
- low molecular leak due to higher molecular mass
- low costs
- easy handling
- plant personnel accustomed to CO_2

Using CO_2 as working fluid for cycle processes in supercritical state means that pressure and temperature exceed the CP. CO_2-cycle processes, where all thermodynamic changes of state take place above the CP, are called supercritical CO_2-cycles (sCO_2-cycles). If the thermodynamic changes occur above and below the CP, the cycle processes are called transcritical CO_2-cycles (tCO_2-cycles). The T, s-diagramm in Fig. 5 shows the regions in which the described cycles operate in a. The CP of CO_2 is marked by the red sign and is located at $31°C$ and $73.8 bara$.

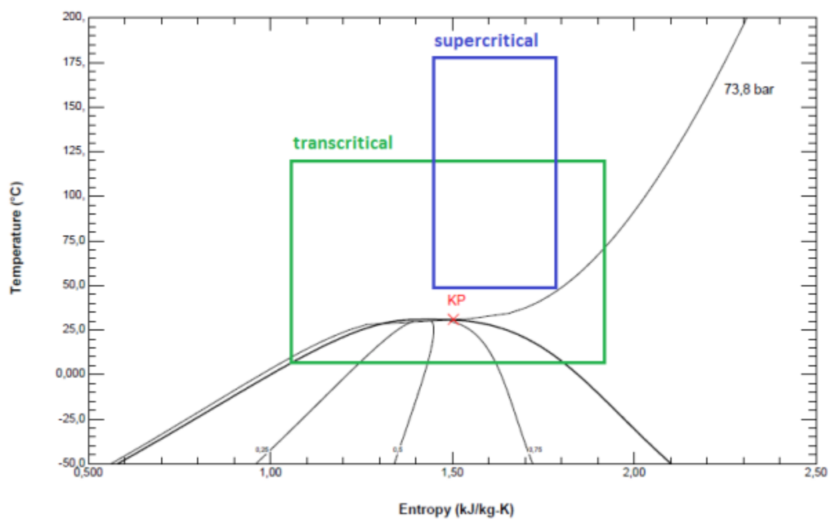

Fig. 5: Difference between sCO_2- and tCO_2-cycles

sCO_2-Brayton-Cycle (sCO_2-BC)

In the supercritical Brayton-Cycle the condition of the working fluid is always above the CP. Fig. 6 shows the schematic diagram and the T, s-diagram of a simple sCO_2-BC with a recuperator. By use of a compressor the working fluid is processed to the upper pressure level of 221.4 $bara$. This equates to the line from point 0 to 1 in the diagram. After this compression the fluid passes the recuperator to heat up the CO_2 (1 to 1*).

Subsequently the working fluid passes the heat exchangers that transfer the waste heat from the two different sources of the cement plant to the CO_2 (1* to 2). Following the heat up, the working fluid is expanded to the low pressure level of 73.8 bara in the turbine, which drives the generator to produce electricity (2 to 3). Finally the CO_2 passes through the recuperator and the cooler to bring the temperature of the CO_2 back to the initial value (3 to 3* and 3* to 0).

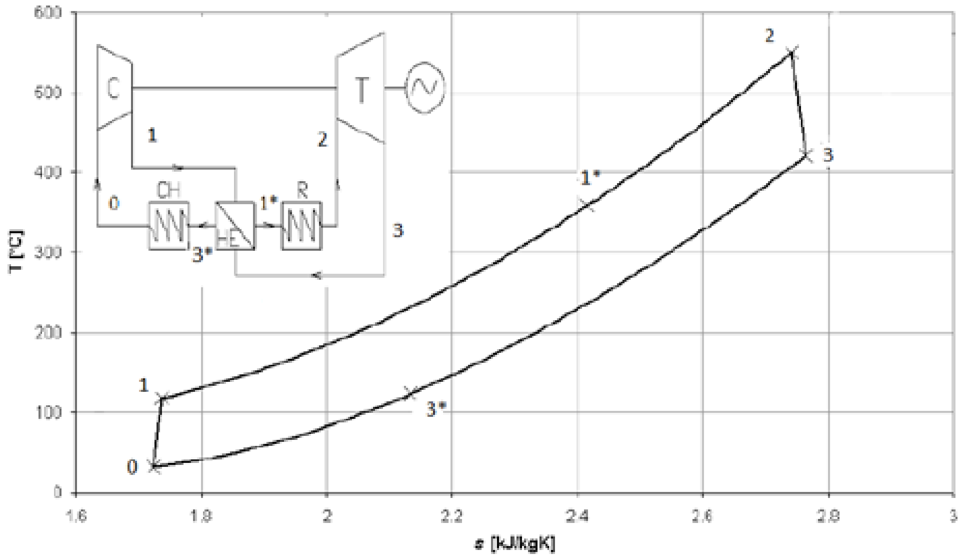

Fig. 6: Schematic diagram and T, s-diagram of a simple sCO_2-BC with a recuperator

Simulation

The operating parameters for the simulation of the supercritical CO_2 Brayton cycle are shown in Table 3.

sCO2-Brayton-Cycle - Operating Parameters[1]	
upper pressure level	$221.4 bara$
lower pressure level	$73.8\ bara$
lowest temperature	34 °C
cooling water temperature	12°C
turbine isentropic efficiency	0.91
compressor isentropic efficiency	0.89

Table 3: Operating parameters of the supercritical CO_2-Brayton cycle

The electrical power output of this supercritical CO_2-Brayton cycle is 4007 kW and the thermal efficiency is $\eta_{th,sCO2\_BC} = 20.8\ \%$.

tCO_2-Brayton-Cycle (tCO_2-BC)

The transcritical CO_2-Brayton cycle is very similar to the supercritical CO_2-Brayton cycle, but the entry to the compressor is below the CP. The CO_2 is still in gaseous state and is located on the saturated vapor line, what is shown inFig. 7. The schematic diagram corresponds to the one of the sCO_2-cycle. Like mentioned before, this cycle runs below and above the CP.

[1] The values for the isentropic efficiency of the sCO_2 machinery were taken over from (Mercangöz, et al., 2012).

The pressure p_0 and the temperature at the entry of the compressor are set to 64.34 bara and 25 °C respectively. If the higher pressure level p_1 is the same as the one of the sCO_2-cycle, then a higher amount of energy can be produced because of a smaller compressibility factor at the lower pressure level p_0. Consequently the power consumption of the compressor is lower.

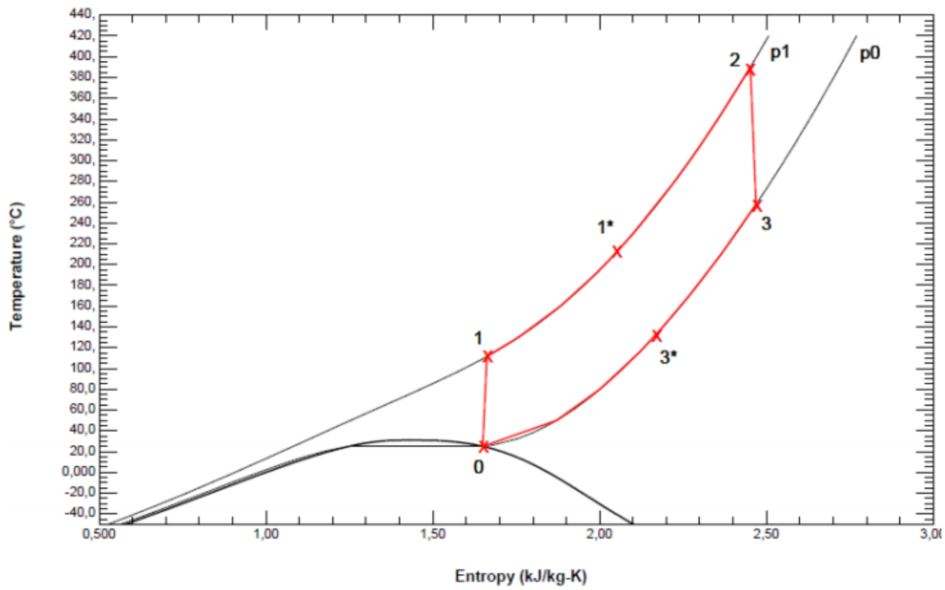

Fig. 7: T, s-diagram of a simple tCO_2-BC

Simulation

The operating parameters for the simulation of the transcritical CO_2-Brayton cycle are listed in Table 4.

tCO_2-Brayton Cycle - Operating Parameters	
upper pressure level	$221.4 bara$
lower pressure level	$64.34\ bara$
lowest temperature	25 °C
cooling water temperature	12°C
turbine isentropic efficiency	0.91
compressor isentropic efficiency	0.89

Table 4: Operating parameters of the transcritical CO_2-Brayton cycle

For this transcritical CO_2-Brayton cycle the electrical power output is 4295 kW and the thermal efficiency is $\eta_{th,tCO2\_BC} = 22.3\%$. Compared to the sCO_2-cycle a higher water mass flow for cooling down the CO_2 to 25°C is needed.

tCO_2-Rankine-Cycle (tCO_2-RC)

The transcritical CO_2-Rankine cycle is like the steam rankine cycle a condensation process. The CO_2 passes through the two-phase region during the cooling process and is completely liquefied.

That is the main difference compared to the two Brayton-cycles. Another difference is that instead of a compressor a pump is used to pressurize the working fluid. Fig. 8 shows a T,s-diagram of a simple transcritical CO_2-Rankine cycle. The entry in the pump (point 0) is located on the saturated liquid line below the CP. Again the schematic diagram corresponds to the one of the supercritical Brayton cycle with recuperator.

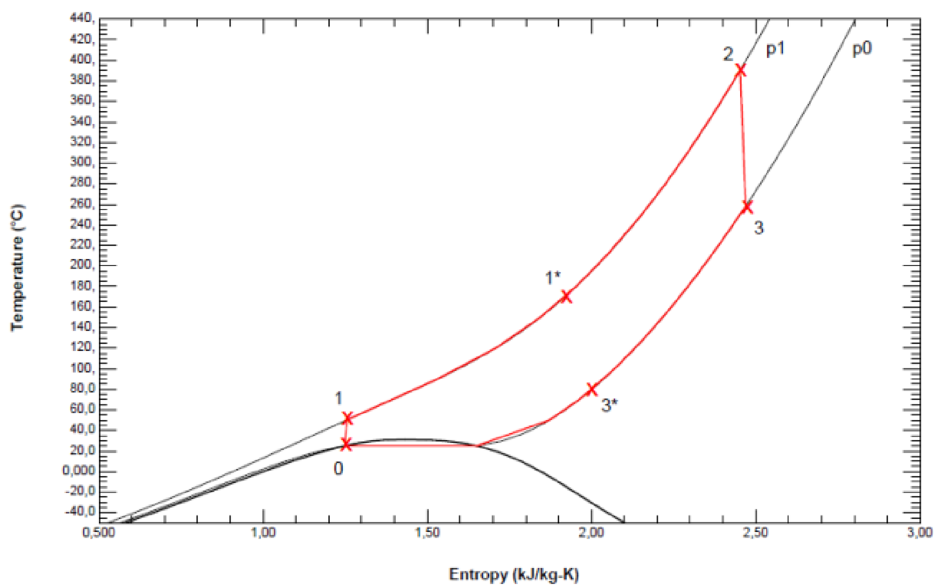

Fig. 8: T,s-diagram of a simple transcritical CO_2 Rankine-cycle

The specific work rate for the pump is notably smaller than the one from the compressor of the two Brayton cycles. This is because of the high density (liquid state) of the CO_2 and the resulting small compressibility factor. Using the same boundary conditions for the tCO_2-RC like at the two Brayton cycles, notable more electric power can be harvested. In terms of the isothermal condensation, a special matter is the pinch point difficulty in the cooling section. If the temperature of the cooling water is given, the water mass flow has to be increased to liquefy the CO_2. Depending on the ambient conditions (e. g. water sources), this can lead to problems in a technical and economic perspective.

Simulation

The operating parameters for the simulation of the transcritical CO_2-Rankine cycle are summed up in Table 5. For this transcritical CO_2-Rankine cycle the electrical power output is 5192 kW and the thermal efficiency is $\eta_{th,tCO2\_RC} = 26.3\%$. For the given water temperature of 12 °C a water mass flow of 420 kg/s is calculated, to reach the required minimum temperature difference in the cooler.

tCO$_2$-Rankine Cycle – Operating Parameters	
upper pressure level	$221.4 bara$
lower pressure level	$64.34\ bara$
lowest temperature	25 °C
cooling water temperature	12°C
turbine isentropic efficiency	0.91
compressor isentropic efficiency	0.86

Table 5: Operating parameters of the transcritical CO$_2$-Rankine cycle

CYCLE COMPARISON

The simulation results for the electrical power output, the heat input and the thermal efficiency of all different cycles are shown in Fig. 9. The tCO$_2$-RC supplies the highest rate of electrical power of all investigated cycles. The reason for this result is the low specific work rate of the pump. The tCO$_2$-RC is producing 5192 kW, which is 897 kW more than the tCO$_2$-BC and 1185 kW more than the sCO$_2$-BC. The ORC (3915 kW) and the SRC (3297 kW) provide 1277 kW and 1895 kW less electrical power respectively.

Like mentioned before the cooling of the tCO$_2$-RC can lead to problems, this is definitely a main disadvantage of this system. For the given water temperature of 12°C a water mass flow of 420 kg/s is required. On the contrary the tCO$_2$-BC and the sCO$_2$-BC require 130 kg/s and 70 kg/s water mass flow respectively. A possible solution to reduce the water mass flow for the tCO2-RC could be an independent cooling cycle for the working fluid, but this would lead to technical and economic penalties.

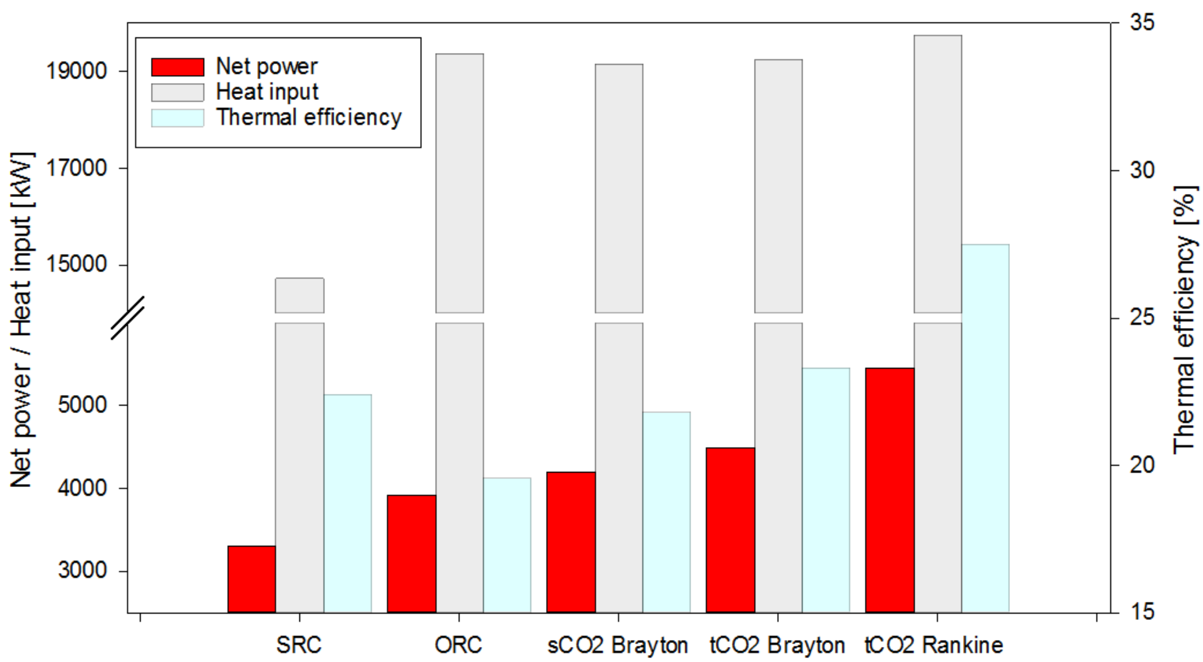

Fig. 9: Simulation results for the electrical power output, the heat input and the thermal efficiency of all investigated cycles

Looking at the results for the heat input leads to the conclusion that the SRC has significantly less heat input (about 5000 kW) than all the other investigated cycles when using the same boundary conditions for the waste heat sources. This is because water is not a suited medium for recovering low temperature heat. The high critical pressure and temperature leads to the conclusion that the SRC produces less power than the other cycles. The chosen two-pressure system enables an extended cooling-down of the heat source-medium. This can be achieved by setting the pressure levels in such a way to match the temperature level of the available heat source.

Fig. 9 also compares the results of the thermal efficiency of the different cycles. The tCO_2-RC reaches the best efficiency value with 26.3 %. The tCO_2-BC (22.3 %), the steam Rankine cycle (22.4 %), the sCO_2-BC (20.8 %) and the ORC (20.6 %) deliver smaller efficiency values respectively. Compared to the ORC and the sCO_2-BC, the steam Rankine cycle has a higher thermal efficiency, despite its lower power output. The reason therefore is the smaller amount of heat which is transferred to the SRC.

For the further evaluation of the results the heat recovery effectiveness, which is the ratio between heat input to the cycle and the heat made available by the external heat source[2] and the total heat-recovery efficiency, which is the product of the thermal efficiency of the cycle and the heat recovery effectiveness were calculated. Fig. 10 shows that due to the lower heat input in the SRC the heat recovery effectiveness of this system is significantly lower in comparison to the other cycles where the heat input is approximately equal. Also it is shown that the tCO_2-RC yields the highest total heat-recovery efficiency since it amounts to the highest net power produced.

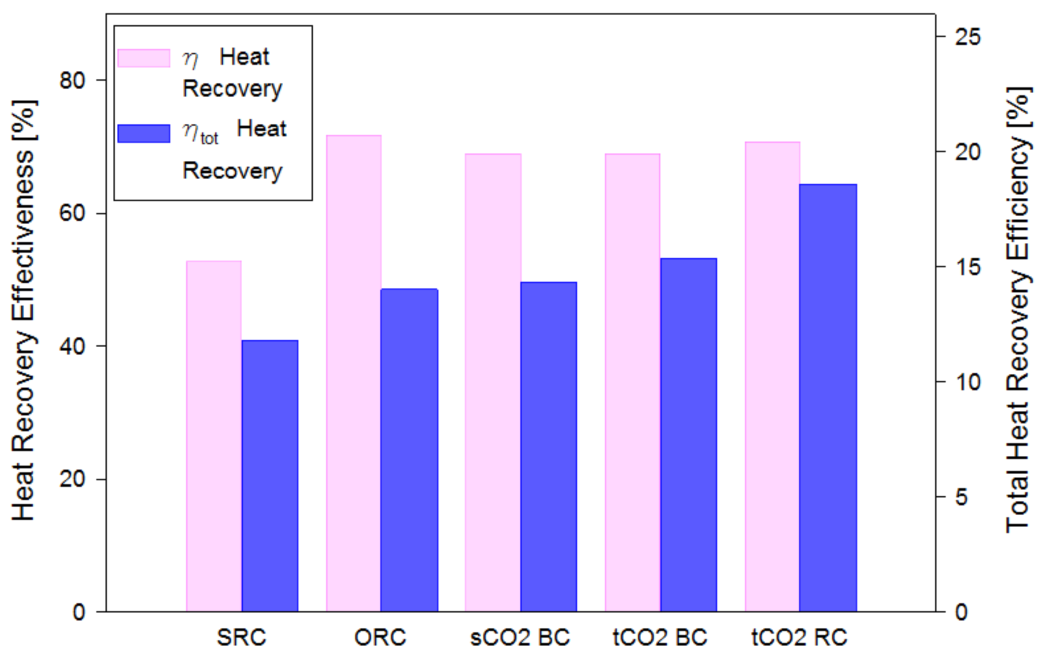

Fig. 10: Heat recovery effectiveness and total heat-recovery efficiency of all simulated cycles.

[2] The maximum available heat is referred to cooling down the exhaust air to ambient temperature (25°C).

CONCLUSION

The potential application of a waste heat recovery system at the cement plant in Gmunden, Austria was analysed. Different power cycle configurations were simulated and discussed. The aim was to compare the currently applied cycles for waste heat recovery namely the SRC and the ORC with novel sCO_2 concepts.

The tCO_2-RC proved to be the system with both the highest efficiency (26.3 %) and net power output (5192 *kW*). A disadvantage of this configuration is the vast amount of cooling water required. The tCO_2-BC that operates between exactly the same pressure levels as the tCO_2-RC, results in lower thermal efficiency (22.3%) and net power (4195 *kW*). This is attributable to the increased power consumption by the compressor in comparison to the pump used in the tCO_2-RC. On the other hand the boundary conditions of the sCO_2-BC were changed since this cycle operates only in the supercritical region. Particularly the low pressure level and the compressor inlet temperature had to be adjusted. Due to the smaller pressure difference and the limitation to the supercritical region this cycle resulted in even lower values for the thermal efficiency (20.8%) and net power (4007 *kW*).

The simulated results of all CO_2 cycles proved to be superior to the yield of the SRC and the ORC. Although water is the medium of choice for high temperature applications it is not suited for the considered temperature range. Simulations showed that the net power produced (3297 kW), is the lowest between the compared cycles and also that it is not possible to cool the exhaust gas as much as in the other cases. Although this leads to an increased thermal efficiency (22,4%), the SRC ends up producing the lowest amount of net power without being able to cool the source as much as the other systems.

From the currently applied systems the ORC is the only one that approaches the results of CO_2 cycles. Generally organic working fluids exhibit several disadvantages (flammable, expensive among others) in contrast to CO_2. Finding the best fit between working fluid and heat source is not an easy task since there is abundance of organic working fluids available. Cyclopentane, which proved to be the optimal medium yielded the highest thermal efficiency (19.6%) and net power (3915 kW) among all considered organic working fluids.

The bottom line is that supercritical or transcritical CO_2 power cycles offer both better thermal efficiency values and harvest more net power than the current state of the art systems. Therefore this technology should be further pursued and developed.

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