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INFLUENCE OF HEAT SOURCE COOLING LIMITATION ON ORC SYSTEM LAYOUT AND WORKING FLUID SELECTION: THE CASE OG ALUMINIUM INDUSTRY

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Abstract

Energy recovery from the raw gas in aluminium production plant is investigated in this paper. This hot gas contains acid compounds and it should not be cooled below its dew point in order to avoid corrosion problems. This limitation is generally not considered in the published literature Common working fluids were screened, with two heat recovery configurations:

Direct and indirect (thermal oil loop) systems. The power production loss for an indirect heat recovery system compared to a direct system was relatively small, about 5%. The acid dew point temperature restriction had less influence on the indirect system, which could utilize fully the low heat sink temperature available on the site. This factor almost fully compensates for the temperature loss induced by a second heat exchanger.
Indirect systems can use standard ORC modules which could reduce the cost of the plant

Introduction

The metallurgical production industry is a major consumer of energy; as such relatively moderate improvements in energy efficiency can have a substantial impact on society. Hydro aluminium is a major supplier of aluminium and aluminium products. The production of aluminium from alumina is achieved through an energy intensive electrolysis process, where alumina is reduced consuming carbon electrodes (Hall-Heroult process). The company early realized the necessity to improve the specific energy consumption of the production process (kWh/kg aluminium produced).
In a typical Norwegian smelter about 40% of the energy consumed in the electrolysis process ends up chemically bound in the aluminium product, while the rest is lost as heat to the cell exhaust gases (raw gas) and the electrolysis hall ventilation air [1]. As most smelters are located remotely from communities that could have benefitted from this heat e.g. district heating, converting the heat energy to electricity is an attractive alternative.
About 50% of the surplus heat is contained in the raw gas sucked off the cells. Typical raw gas temperature is close to 100 °C. To produce electricity from a heat source of such a relatively low temperature, the classical steam power cycle is inefficient and other working fluids have to be considered.
The Organic Rankine Cycle (ORC) is now a well-established technology for power production from low temperature heat sources. Working fluids are generally organic compounds used in the refrigeration industry, hydrocarbons or a few other natural fluids (e.g. ammonia). Common applications for ORCs are electricity production from geothermal fields [2, 3], biomass plants [4] and gas turbine bottoming cycles [5, 6]. Other applications include solar application [7-9], combustion engines [10, 11] and energy recovery from waste heat in industry *[12]*.
Technology providers exist for these applications, among them are ORMAT (geothermal, [13, 14] and TURBODEN (biomass, [4]). Research in ORC technology is still very active, focusing both on component development [15-17] and working fluid selection [18-23]. Working fluid selection studies generally do not consider possible limitation on how much the heat source can be cooled. In the case of the raw gas, acid compounds might cause corrosion problems if the gas is cooled below its dew point. This paper investigates how this important limitation affects the working fluid selection.

Boundary conditions for the power production unit

The energy in the raw gas has to be recovered by use of a heat exchanger. The heat recovery heat exchanger (HRHE) can be installed either before or between the FTP stages. In the dry scrubber the raw gas is mixed with alumina particles which trap the fluoride. A substantial amount of air at ambient temperature is entrained with the alumina particles and, as a consequence, the off gas temperature downstream the dry scrubber is on average reduced by 20 K. A raw gas HE in position installed upstream the FTP is therefore thermodynamically favourable in order to maximize the available heat for power production and the thermal efficiency of the power production unit (PPU). Heat recovery from particle laden gas is challenging as the risk of heat exchanger surface fouling and even blocking is high. Heat exchanger surface fouling issues were addressed by Næss et al. [24], and are not considered in the present work.

The temperature of the raw gas throughout the year follows closely the outside temperature. It was measured to vary from an average of 98 °C in winter to an average of 111 °C in summer.
Fresh water from the nearby river is available for heat rejection, average summer temperature is 12C, average winter temperature is 2 °C

The raw gas contains SO_x gases that originate from the carbon anodes used in the electrolysis process to produce aluminium. The majority of the SO_x gas is in the form of SO_2, which remains gaseous when the raw gas is cooled down in a heat exchanger. However, the gas also contains small amounts of SO_3 and H_2SO_4, which will condense if the gas reaches its dew point temperature. The dew point temperature is mainly dependent on the SO_3, H_2SO_4 and water vapour concentration, but is difficult to estimate because of the complex chemical composition of the raw gas.
In 2001, an acid dew point probe was used in the raw gas at Hydro Sunndalsøra [25]. The results indicated that the dew point was below 40 °C. As no exact value was found, it was assumed that as long as the temperature of the raw gas and the heat

exchanger surface are above 40 °C, SO_3 and H_2SO_4 condensation is avoided.

Two main systems for heat recovery were considered in this study:

- Direct heat recovery: Heat from the raw gas was directly recovered by the working fluid of the ORC.
- Indirect heat recovery: Heat from the raw gas was first recovered by a heat transfer fluid (thermal oil or water in this case) and transported to the ORC boiler where it was transferred to the working fluid.

By direct heat exchange between the raw gas and the ORC working fluid, installation and operation costs are minimized. In addition, the irreversibility provided by an additional temperature difference is avoided. As such, the efficiency should be higher than for an indirect system.
On the other hand, the indirect system avoids limiting the condensation pressure of the ORC to be directly limited by the acid dew point temperature limitation. As already stated, the raw gas dew point limits the (cold) inflow temperature of the heat exchanging fluid (water for an indirect system and ORC working fluid for a direct system) to 40 °C. As a result, the condensation temperature in a direct system is limited to 40 °C if no internal heat exchanger (IHE) is used in the ORC cycle. If an IHE is used, the condensation pressure can be reduced, but less heat can be recovered from the heat source. For an indirect system the temperature limitation is on the intermediate heat carrier fluid and not on the ORC working fluid. Therefore it is possible to take full thermodynamic advantage of the cooling medium (fresh water) and condense the working fluid at lowest pressure (and temperature) possible, increasing thermal efficiency.

Calculation procedure

A thermodynamic model (illustrated in Figure 1) was implemented in excel using Refprop v8 [27], for thermodynamic properties calculations. Calculations were performed based on simple assumptions:

- Constant isentropic efficiencies for the pump and expander
- Constant minimum temperature differences in the heat exchangers
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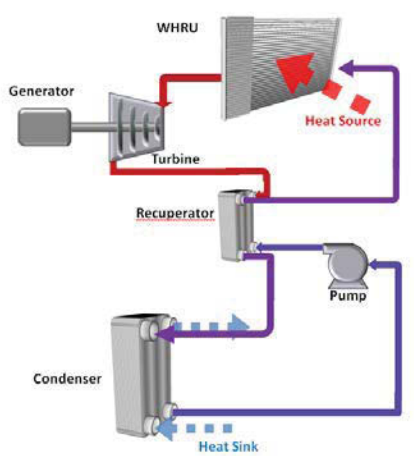

Figure 1 Layout of the ORC cycle

The following parameters were selected as dimensioning for the ORC unit:

Table 1: Dimensioning parameters

Parameter	Value	Unit
Raw gas temperature, summer	110	°C
Raw gas temperature, winter	100	°C
Cooling water temperature, summer	12	°C
Cooling water temperature, winter	2	°C
Min. DT in raw gas heat exchanger	15	°C
Min. DT in pure fluid heat exchangers	5	°C
Min. temperature of working fluid in HRHE	40	°C
Raw gas mass flow rate	522.54	kg/s
Cooling water mass flow rate	∞	kg/s
Turbine efficiency	80	%
Pump efficiency	70	%

Pump and compressor efficiencies as well as temperature pinches for the heat exchangers are typical for this kind of system and are commonly used in the literature.
Working fluid selection is an important part of the system optimization. General rules for the selection can be found in [26]. Our particular application imposes additional criteria to be fulfilled by the working fluids:
Condenser saturation pressure should be at least 50 kPa above atmospheric pressure (in order to provide sufficient suction head to avoid cavitation occurring in the pump and to prevent atmospheric air from entering the system).
Only standard sub-critical operation was considered, i.e. the critical temperature of the ORC working fluid should be above the maximum heat source temperature of ca. 120 °C.
Working fluids restricted by the Montreal Protocol (1987) were not considered. In the Copenhagen Amendment (1992) it was decided to phase out all CFCs by 1995. The Beijing Amendment (1999) instructs the developed countries to freeze production of HCFCs by 2003. Neither CFCs nor HCFCs were therefore considered in the present study.
When there are several isomers within the same functional group only one of the isomers was considered (e.g. butane and isobutane).
Highly oxidizing or toxic fluids, like nitrous oxide (N_2O) and hydrogen sulphide (H_2S), were not considered.

Results for direct heat recovery

Figure 1 presents the results for both summer and winter conditions

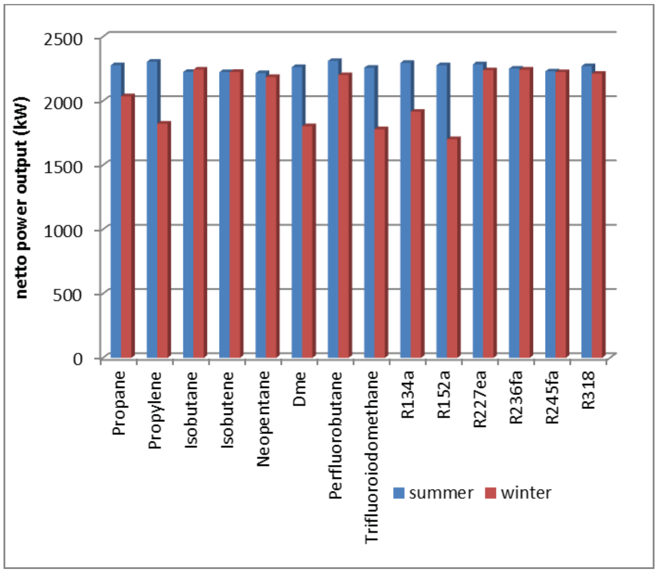

Figure 2 Results for direct heat recovery.

Based on the simulation results, a slightly higher power output was obtained at the summer conditions than in the winter even though the Carnot efficiency was higher for the winter scenario. The reason for this was that the summer scenario had a higher condenser temperature, and as a consequence less heat can be extracted from the raw gas than in the winter scenario.

The working fluid that obtained the highest power output overall was R227ea, having a power output of 2285 kW for the summer scenario and 2239 kW for the winter scenario. However it was observed that all considered working fluids obtained approximately the same net power output for the given conditions. While the pressure levels and mass flow rates varied depending on working fluid, the net power output remained approximately the same. For the winter scenario, the results were more diverse. All fluids having a negative sloped saturation vapour line, i.e. propane, propylene, Dme, trifluoroiodomethane, R134a and R152a, were unable to supply enough heat, from the recuperator, to operate at 7 °C condensation temperature in the winter scenario still avoiding working fluid temperature below 40 °C in the heat recovery heat exchanger. The condensation pressure had to be raised which resulted in a lower thermal efficiency and thus a lower power output. Figure 2 illustrates the observation. The fluids with negative vapour saturation line had on average 18% lower power output than isobutane, which had the highest power output in the winter scenario (2244 kW).

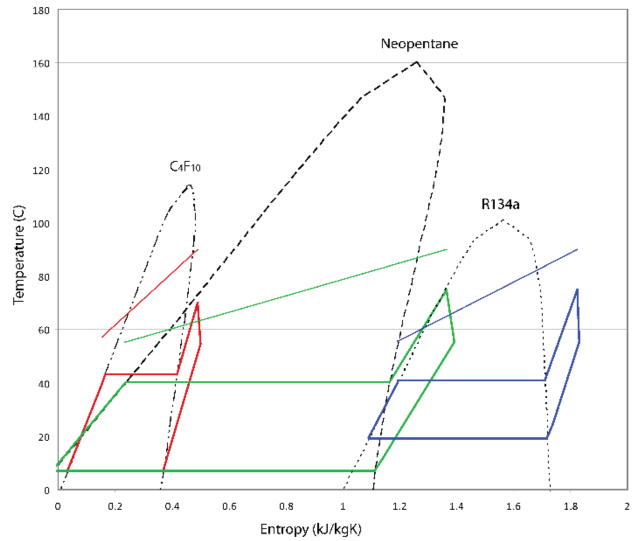

Figure 3 Optimized cycles for C_4F_{10}, neopentane and R134a for a direct heat recovery system in winter scenario

Results for the indirect heat recovery

The indirect heat recovery system model did not have any temperature restrictions directly connected to the ORC as temperature limitations in the HRHE is put on the heat transfer fluid loop. The primary system design did therefore not include an IHE as the direct system design. The heat transfer fluid loop had a 40 °C minimum temperature restriction at the inlet of the HRHE, because of the risk of sulphuric acid formation in the raw gas heat exchanger. For simplicity, the temperature drop of the raw gas was assumed to be equal to the temperature rise of the water in the raw gas heat exchanger. The water mass flow rate would thereby be constant and could be calculated from the raw gas mass flow and the specific heat capacities of the fluids. Inserting the following values

- $c_{p\_rawgas} \approx c_{p\_nitrogen}$ (100 °C) = 1.043 kJ/kgK
- c_{p_water} (80 °C) = 4.197 kJ/kgK
- mass flow rawgas = 522.54 kg/s

one gets a water mass flow rate of 129.86 kg/s.

Figure 3 presents the results for both summer and winter conditions

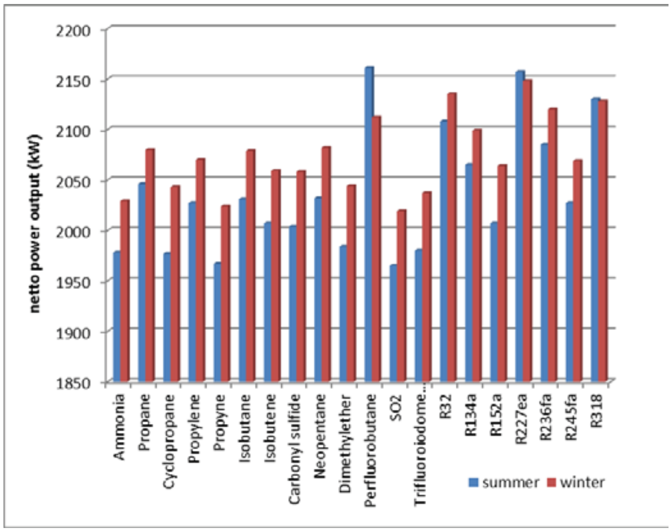

Figure 4 Results for indirect heat recovery.

As a consequence of the lenient system requirements (compared to those for the direct system), all working fluids managed to fulfil the system requirements for both scenarios. As a result of the more numerous potential cycle configurations the net power output (P_{net}) of the different working fluids varied to a greater extent than for the direct system. For both scenarios, the net power output for the different working fluids stayed within a 10% range. All fluids experienced a higher thermal efficiency during the winter scenario. This coincided with the power output, which was higher in the winter scenario than in the summer scenario for almost all working. This can be explained by the lower heat sink temperature in winter which allows for lower condenser pressure. The working fluid that obtained the highest power output overall was R227ea with a net power output of 2157 kW for the summer scenario and 2148 kW for the winter scenario.

Conclusion

The heat source for the ORC unit considered is raw gas from the production of aluminium. The raw gas had a mass flow rate of 1 456 000 Nm^3/h and an average temperature of 110 °C during summer and 100 °C during the winter. Acid compounds might cause corrosion problems if the gas is cooled below dew point. As a result the cooling of the raw gas is limited to 70C and the working fluid temperature in the heat recovery heat exchanger is maintained higher than 40C.

The heat sink was fresh water, estimated to have an average temperature of 12 °C in the summer and 2 °C in the winter. The heat sink was considered to be an unlimited source as no data concerning this issue was available.

Restriction on the cooling of the heat source was found to have a significant effect on working fluid selection.

For a direct heat recovery system it was necessary to install an IHE for optimal cycle performance. R227ea came out as the best performer for the direct system with an average power output of 2262 kW. However, the differences between the power outputs with the different ORC working fluids were small.

The indirect system solution had a broader selection of relevant working fluids than the direct design because of the absence of the IHE. The simulations showed again that the net power output did not vary much for the different working fluids, even though they displayed a wide range of thermophysical properties. This can be interesting as it enables use of standard working fluid with available components like R134a in an efficient energy recovery system. Again, R227ea achieved the highest performance with an average power output of 2153 kW.

The main finding of this study was that the power production loss for an indirect heat recovery system compared to a direct system was relatively small, about 5%. The acid dew point temperature restriction had less influence on the indirect system, which could utilize fully the low heat sink temperature available on the site. This factor almost fully compensates for the temperature loss induced by a second heat exchanger. Indirect heat recovery system with a thermal oil loop is today the usual system for commercial ORC available on the market. On the other hand, direct system could provide investment saving in addition to slightly better performance.

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