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Evaluation of different heat pump concepts for production of process steam from surplus heat at Glencore Nikkelverk

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Abstract
This memo summarizes the main findings of a case study targeting surplus heat upgrading for steam production in a nickel refinery plant. The available surplus heat source is water at 90°C and the steam demand is 4.5 MW at 9 bar. The objective is to perform an initial evaluation of several heat pump concepts, with focus on energy efficiency, to form a basis for further work and discussions with vendors. Four environmentally friendly concepts are proposed, differing mainly in the refrigerant used and compressor technology applied. Potential savings in electricity demand and related CO_2 emissions, compared to the existing electric boiler, ranges between 50% and 75%. A concept using water as refrigerant and blowers as compression technology showed the highest saving. The lowest savings was found for a gas refrigeration cycle (reversed Brayton) with CO_2 as refrigerant, but this concept offers instead the most compact and less complex system. For all evaluated heat pump solutions, the study clearly shows a trade-off between energy efficiency, compactness, complexity and technological readiness level (TRL).

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1 Introduction

This memo describes the evaluation of four different heat pump solutions for surplus heat upgrading and steam production at Glencore Nikkelverk.

1.1 Background

The focus on efficient and low-emission energy systems requires improved solutions to cover heat demands in industrial processes. Such heat demands are often at a high temperature and covered by electrically heated or fossil fuelled steam boilers. At the same time, many industrial processes produce large amounts of surplus heat which is, due to its low temperature level, usually released into the environment and in many cases with an additional cooling effort. By installing a high-temperature heat pump (HTHP) this low-temperature waste heat can be recovered by lifting its temperature and supplying high-temperature heat demands.

The nickel refinery plant at Glencore Nikkelverk uses 9 bar steam with an average consumption of 4.5 MW (+/- 30%), which today is supplied from an old electrical boiler of 5 MW. A waste heat source at 90°C is available for the local district heating plant but not utilized. The scope of work for this case study is a preliminary investigation of different heat pumping concepts to enable recovery and up-grading of this surplus heat to produce the required steam demand (see Figure 1-1). Focus is put on energy-efficiency, but other criteria such as system complexity, compactness and technology readiness will also be briefly discussed. The aim is to provide a basis for future work and discussions with vendors.

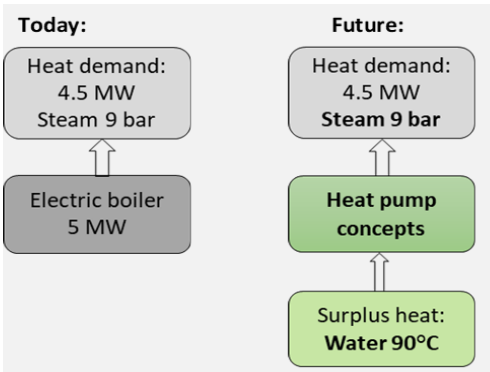

Figure 1-1: Case study definition.

1.2 Case description

The available waste heat source is 90°C water in a closed loop with a pressure of typically 2.5-3.5 barg. By implementing a HTHP this waste heat could supply the required steam demand at 9 bar. The feedwater to the current electrical boiler is preheated up to 80°C with another waste heat source. There is little/no return of condensate as most of the steam goes into the nickel production process.

The waste heat flowrate is adjustable, typical values are 350-400 m^3/h. An outlet heat source temperature of minimum 80°C was assumed to ensure that enough waste heat is available for producing the average steam demand (4.5 MW). However, the flowrate and outlet temperature can be adjusted to optimize heat pump performance. Waste heat in excess is cooled by the existing seawater cooling system.

Due to the large variation in steam consumption during start-up and shut-down a back-up is needed, which could be either a new electric boiler or a heat pump solution. The steam supply must be reliable and 100% available, 8760 h/year (back-up included). Table 1-1 lists the key heat source and heat sink specifications.

Table 1-1 Key heat source and heat sink specifications.

	Heat source - surplus heat	Heat sink – steam demand
Heat duty	**Calculated**	**4.5 MW**
Inlet / outlet	**90°C / 80°C**	**70°C[1] water / 9 bar steam (175°C)**

[1] The feed water is assumed to be pre-heated to 70°C

1.3 High temperature heat pumps

Implementing heat pumps is in general an energy efficient way to use a heat source at a low temperature level to provide heat to a heat sink (heat demand) at a higher temperature level. The basic design of a heat pump, shown schematically in Figure 1-2, is a closed refrigerant circuit consisting of two heat exchangers (evaporator and condenser), a compressor and an expansion valve.

The energy efficiency of a heat pump in a specific heating operation point is normally expressed as its coefficient of performance (COP), defined as

$$COP = \frac{\dot{Q}_{\mathrm{Heating}}}{P_{\mathrm{el}}}$$

where $\dot{Q}_{\mathrm{Heating}}$ [MW] is the heat supplied to the heat sink and P_{el} [MW] is the power consumption of the compressor.

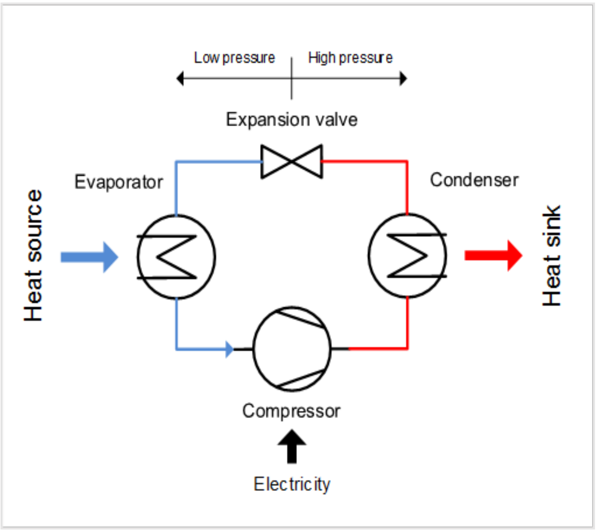

Figure 1-2: Simple heat pump scheme.

High-temperature heat pumps (HTHPs) are normally defined as heat pumps delivering heat at a temperature above 90-100°C. They have the potential to cover a wide range of industrial heating demands using only a fraction of the electric or fossil energy compared to direct heating. Examples of applications include reheating of district heating water, production of hot water, steam production or thermal upgrade of excess heat for industrial processes [1].

The number of industrial HTHPs available on the market has grown steadily in recent years. There are more than 20 heat pump models, supplying heat in the range of 90°C - 150°C, with heating capacities of 20 kW up to 20 MW. Different types of refrigerants are used, but for delivery above 90°C mostly synthetic refrigerants are applied, having either high global warming potential (GWP) and/or potential toxicity [2]. In this case study different HTHP cycles using natural refrigerants are evaluated, having a negligible GWP-value.

The performance of a heat pump, as well as the design of components and thus investment cost, are strongly dependent on the choice of refrigerant. Differences in thermodynamic properties will for the same amount of delivered heat imply differences in, for example, energy efficiency (COP), operating pressures, required amount of waste heat, additional components needed, material requirements, compressor size/type as well as system compactness and complexity.

The compressor is a key component, both related to energy efficiency and investment cost. For HTHPs there is generally a large temperature difference between heat source and heat sink, and thereby a large pressure ratio over the compressor, implying the need for several compressor stages. The number of stages, together with the total compressor suction volume flow, gives an indication of both the space requirements and complexity of the heat pump system. Another important operation parameter related to the compressor is the discharge temperature, which is limited by the material used. Water injection (de-superheating) between compressor stages is applied to reduce the discharge temperature and also to reduce the suction volume flow.

2 Methodology

2.1 Choice of heat pump concepts

Generally, promising HTHP solutions include open or closed steam cycles, cascade cycles with steam and hydrocarbons, as well as the so-called reversed Brayton cycle with carbon dioxide (CO_2) as refrigerant. The choice of which solutions to be evaluated in this case study was based on previous studies within HighEFF and other related projects such as HeatUp[2] and DryF[3]. The proposed HTHP solutions are developed at a lab or concept stadium and all use natural working fluids.

2.2 Modelling of heat pump concepts

To evaluate operation and performance parameters, the heat pumps concepts were modelled in Excel with input and output parameters as shown in Table 3-1. An example is shown in Figure 2-1 for one of the evaluated concepts. The values with grey background represent input parameters while the values with white background are calculated.

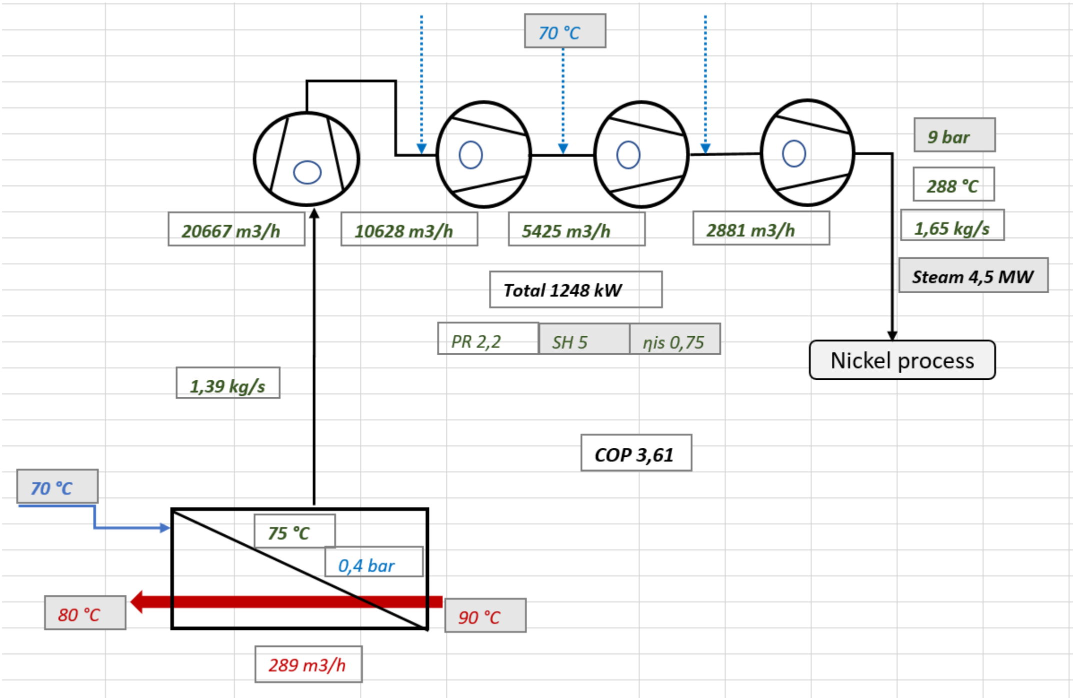

Figure 2-1: Modelling example for one of the evaluated heat pump concepts.

2.3 Calculation of energy- and emission savings

Possible savings in elctricity (energy) consupmtion and CO_2 emissions were calculated to demonstrate the potential of the HTHP concepts to meet HighEFF goals, which are 20-30% reduction in energy consumption and 10 % reduction in greenhouse gas emissions for the centre as a whole.

The baseline scenario, to which the calculated savings are referred to, is here defined as the currently installed electrical boiler (5 MW). For the calculation of emission savings, the Norwegian energy mix for electricity production was used; 18.9 grams of CO_2 emitted per kWh[4].

[2] https://www.sintef.no/projectweb/heatup/
[3] http://dry-f.eu/
[4] Calculated by NVE for 2018

3 Heat pump concepts

In this chapter the four evaluated heat pump concepts are described, while the evaluation results are presented and discussed in chapter 4.

3.1 Steam heat pump cycles

Heat pump cycles using steam/water (R718) as refrigerant have shown a large potential for applications in the temperature range of 100-180°C. Water is a natural working media with good thermodynamic properties but has a drawback in its low volumetric capacity leading to large volumes flows. Thus, the compressor must be designed for volumetric capacities multiple times higher than for other refrigerants [1]. To facilitate the market introduction of R718 heat pumps, different research projects and pilot tests have focused on development of cost-effective compressors [3]. Steam compression heat pumps for use in industrial processes are described in more detail in the deliverable "D3.2_2017.05 Possibilities for energy recovery by steam compression cycles".

Two different steam heat pump concept are evaluated, shown in Figure 3-1 and Figure 3-2. The essential difference between them is the compression technology applied for raising the steam pressure. The first concept (Figure 3-1) uses turbo compressors, while the second instead applies a larger number of blowers/fans operating at a lower pressure ratio but at a higher efficiency.

3.1.1 Open steam cycle with turbo-compressors

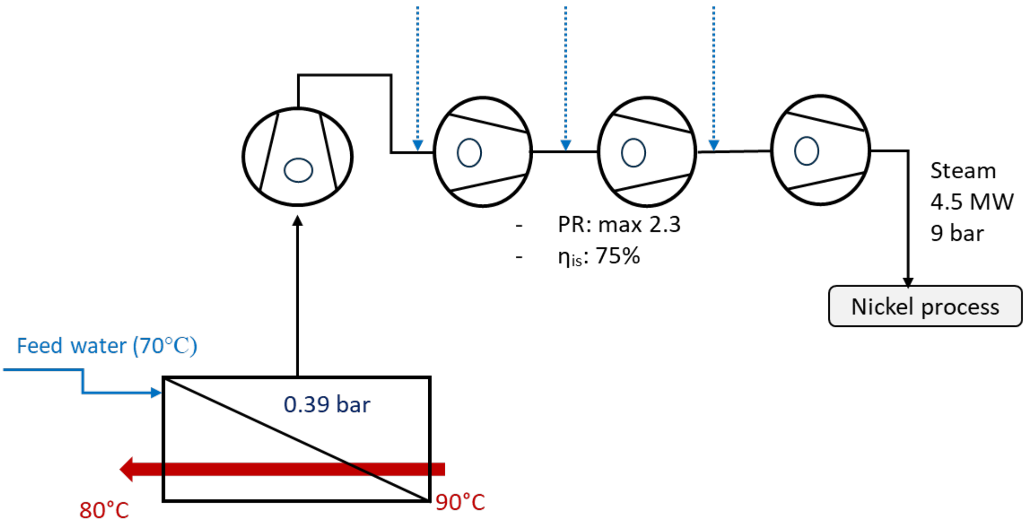

Figure 3-1: Sketch of the open steam (R718) heat pump cycle with turbo-compressor (4 stages).

With the assumed values on outlet heat source temperature and minimum temperature differences in heat exchangers (see Table 3-1), the maximum steam pressure in the evaporator becomes 0.37 bar. Such a low pressure implies a large pressure ratio and large volume flows. To keep the pressure ratio below 2.3, which is the maximum pressure ratio recommended by compressor manufacturers, to achieve a reasonable isentropic efficiency (75%) and not too high discharge temperatures, the compression process must be divided in at least 4 stages. De-superheating of the steam between each compressor stage is enabled by injection of feed water to a superheating of 5 K [3].

3.1.2 Open steam cycle with centrifugal blowers

For the concept with steam blowers (Figure 3-2) instead of compressors the pressure ratio is limited to 1.4 while the isentropic efficiency is stated by the manufacturer to be above 80%. To address the drawbacks of the low steam pressure, a double stage evaporator can be implemented (which of course also could be done in the concept shown in Figure 3-1). It includes two evaporators coupled in series on the heat source side and in parallel on the feed water side. The two evaporators operate with the same duty but at different temperature/pressure levels. To raise the steam pressure to 9 bar requires in total 11 blower stages. De-superheating of the steam between each stage is made by injection of feed water to a superheating of 0 K, i.e. saturated steam.

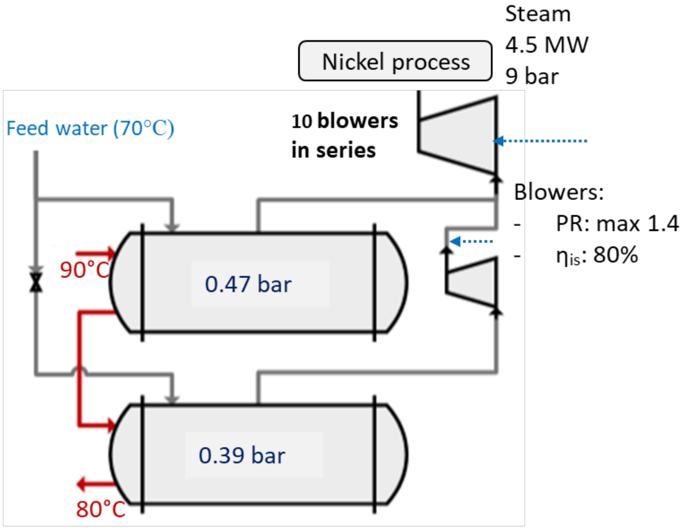

Figure 3-2: Sketch of the open steam (R718) heat pump cycle with blowers (11 stages).

3.2 Cascade cycle with a hydrocarbon bottoming cycle

Cascade cycles, as principally shown in Figure 3-3, consist of two heat pump cycles; a bottom cycle and a top cycle, operating with different refrigerants. Compared to the steam cycles presented in section 3.1, the bottom cycle enables a reduced pressure ratio, as well as volume flow, in the open steam top cycle. Suitable refrigerants for the bottom cycle, i.e. enabling the steam to evaporate in the cascade HEX at a higher pressure/temperature, includes ammonia, pentane and butane. A more detailed evaluation of cascade heat pumps with different refrigerants is found in the deliverable "D6.2_2017.03 Heat pump alternatives at HLNG".

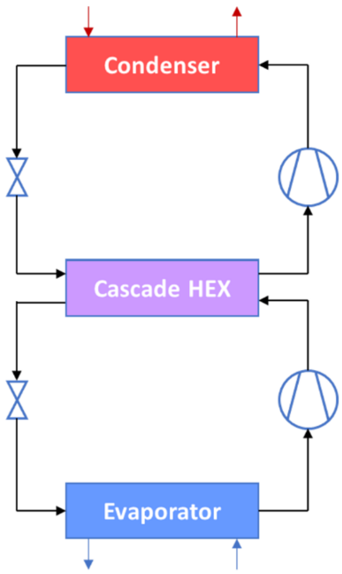

Figure 3-3: Simplified scheme of a cascade heat pump cycle.

Based on results from previous studies, performed in HighEFF and HeatUp, butane (R600) was chosen as refrigerant in the bottom cycle. According to a recent state-of-the art review regarding HTHPs [2] there exists two experimental prototypes of butane heat pumps delivering heat at a temperature of 100°C or above; one delivering water at 100°C, the other delivering steam at 2.4 bar (125°C), both from a heat source of 60°C. Within HighEFF a test-rig of a butane heat pump was installed at the SINTEF/NTNU laboratory, delivering heat at a temperature of 100-130°C using a heat source at 50°C or 60°C. As described in the deliverable "D3.2_2018.03 Performance of a high-temperature heat pump using butane as refrigerant", stable operation with COP values between 2.0 and 3.7 was achieved.

The evaluated cascade HTHP concept is principally shown in Figure 3-4. The compressor in the bottom cycle is a "standard" hydrocarbon scroll compressor, with an assumed isentropic efficiency of 75%. To ensure enough superheat of the butane refrigerant at the compressor inlet, and thereby avoiding wet compression, a suction gas heat exchanger (SGHX) is included [4]. Based on previous studies, to include a SGHX in a butane cycle also increases the COP. As seen in Figure 3-4, including a butane bottoming cycle enable a steam evaporation pressure of 2 bar, instead of 0.4 bar in the open steam heat pump cycle Figure 3-1.

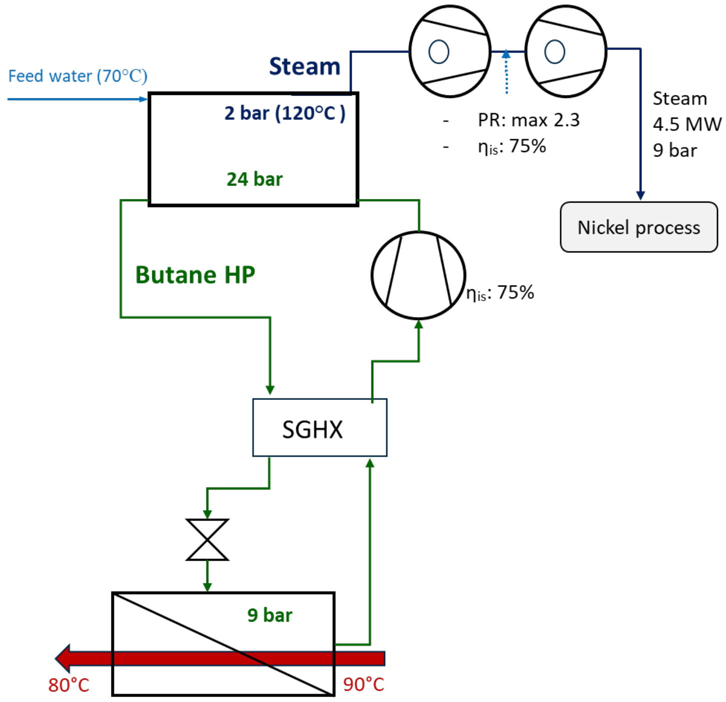

Figure 3-4: Sketch of a cascade heat pump cycle, consisting of a bottoming cycle using butane (R600) and a steam (R718) top cycle with two turbo-compressor stages.

3.3 Reversed Brayton cycle using carbon dioxide

The last concept is based on the so-called reversed Brayton cycle (or gas refrigeration cycle), principally shown in Figure 3-5. As seen, it is similar in design to the conventional heat pump cycle, but since the refrigerant operates solely in its gas phase, thus without the liquid/vapor phase change, the cycle does not have any evaporator and condenser. Instead, it includes a gas heater, operating at the lower pressure and temperature level, and a gas cooler operating at the higher pressure and temperature level. Another important difference is that the energy of expansion is partly recovered in the expander, thus reducing the power consumption of the compressor [5].

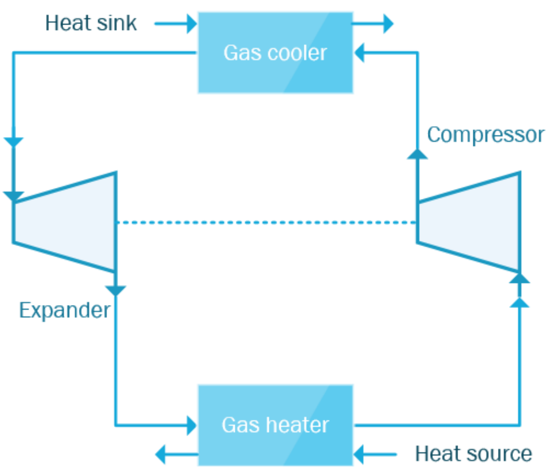

Figure 3-5: Principal sketch of a reversed Brayton heat pump cycle.

The reversed Brayton cycle is well known on a feasibility study-level, especially for applications requiring heat sink temperatures up to 150-200°C, and has a high potential for large scale (10-100 MW) applications where process heat is required at temperatures up to 500°C [6]. This type of cycle is widely used in jet aircrafts, for air conditioning systems using air from the engine compressors, and in the LNG industry [7]. CO_2 (R744) can be used as refrigerant in a reversed Brayton cycle, at conditions where it operates above its critical point (73 bar, 31°C).

In Figure 3-6, the evaluated concept with a reversed Brayton cycle using CO_2 (R744) as refrigerant is shown. As seen, it enables the steam to be directly produced from the CO_2 heat pump cycle at a pressure of 9 bar, i.e. no steam compressor is required. Based on previous case studies, the efficiency is increased by installing a SGHX. The turbo-compressor is mounted on the same shaft as the turbo expander, enabling a partial recovery of the expansion work generated by the expansion of CO_2 from a high-pressure level in the gas cooler (steam generator) to a lower pressure level in the gas heater.

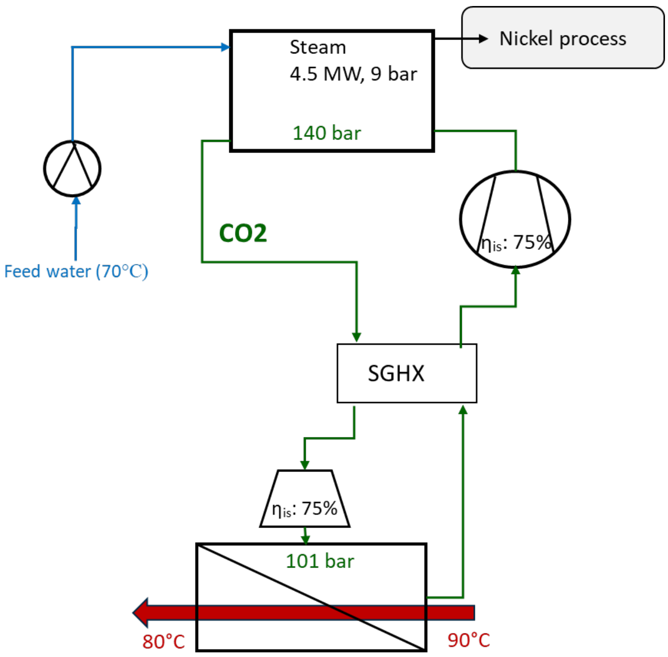

Figure 3-6: Sketch of the reverse Brayton heat pump cycle operating with CO_2 (R744) as refrigerant.

Since the whole process occurs in the super-critical region, CO_2 will be in gas phase in all points in the cycle, meaning that the temperature and pressure in the gas cooler and heater are not connected as in a conventional evaporator and condenser. This feature allows to control the pressure and temperature independently to optimize the system performance [8].

Based on previous studies the performance is increased with a high gas cooler pressure. The maximum feasible gas cooler pressure today is expected to 140 bar [6]. The gas heater pressure was varied to find the maximum COP, without pinch violations (see Table 3-1) and heat balance deviations.

3.4 Input parameters

Table 3-1 presents the input parameters for the performance evaluation of the different HTHP concepts. These input parameters imply several assumptions / simplifications, such as

- Each compressor (expander) efficiency is fixed (i.e. does not vary with pressure ratio)
- Pressure drop and heat transfer characteristics are neglected

Table 3-1: Input parameters for the different HTHP concept. The parameters in italic differ between the different concepts, while the other parameters are the same for all evaluated concepts.

INPUT PARAMTER	VALUE
Heat source inlet / outlet temperature	90°C / 80°C
Heat sink inlet pressure, temperature	1 bar, 70 °C
Delivered steam	4.5 MW
Delivered steam pressure	
Steam cycles	9 bar
Butane bottom cycle	2 bar
CO_2 reversed Brayton Cycle	9 bar
Desuperheating between pressure stages	
Steam cycle with blowers	0 K
Steam cycle with compressors	5 K
Isentropic efficiency / maximum pressure ratio	
Steam compressors	2.5 / 75%
Steam blowers	1.4 / 80%
Butane compressor	3 / 75%
CO_2 compressor and expander	2 / 75%
Minimum pinch point / mean temperature difference	3K /10K

For some operating/design parameter a sensitivity analysis was performed, which is presented in chapter 5.

3.5 Evaluation parameters

In Table 3-2 the key parameters included in the evaluation are presented. The focus of the investigation is on energy efficiency, to show the potential savings in electricity consumptions and CO_2 emissions. However, since heat pump concepts based on water/steam as refrigerant generally achieve high efficiencies but may have challenges related to high compressor discharge temperatures and high specific volume flows, some operating parameters were also highlighted.

Table 3-2: Evaluation parameters.

Energy efficiency parameters	Operating/design parameters
COP [-]	Total compressor suction flow [m^3/h]
Power demand [MW]	Discharge temperature [°C]
Electricity savings [MW], [%]	
CO_2 emission reduction [t/year], [%]	
Waste heat used [MW]	

4 Performance evaluation results

Here the results from the evaluation of the four different heat pump concepts are presented and discussed.

4.1 Energy and emission savings

First, the COP and power consumption for the different concepts are presented, followed by the corresponding savings in electricity consumption and CO_2 emissions. The utilised waste heat flows for producing 4.5 MW steam are also shown, to indicate any potential for a larger steam production.

4.1.1 COP and power consumption

In Figure 4-1, the different HTHP concepts are compared for a heat delivery of 9 bar, in terms of energy efficiency (COP) and power consumption. As seen the COP ranges between 1.9 to 4.1 meaning that the heat pump solutions can produce between 2-4 times more steam per kW electricity input compared to the electric boiler. As seen, to produce 4.5 MW steam the required electricity consumption reduces from 5 MW (current electric boiler) to 1.3 – 2.6 MW (with an assumed electricity efficiency of 95%).

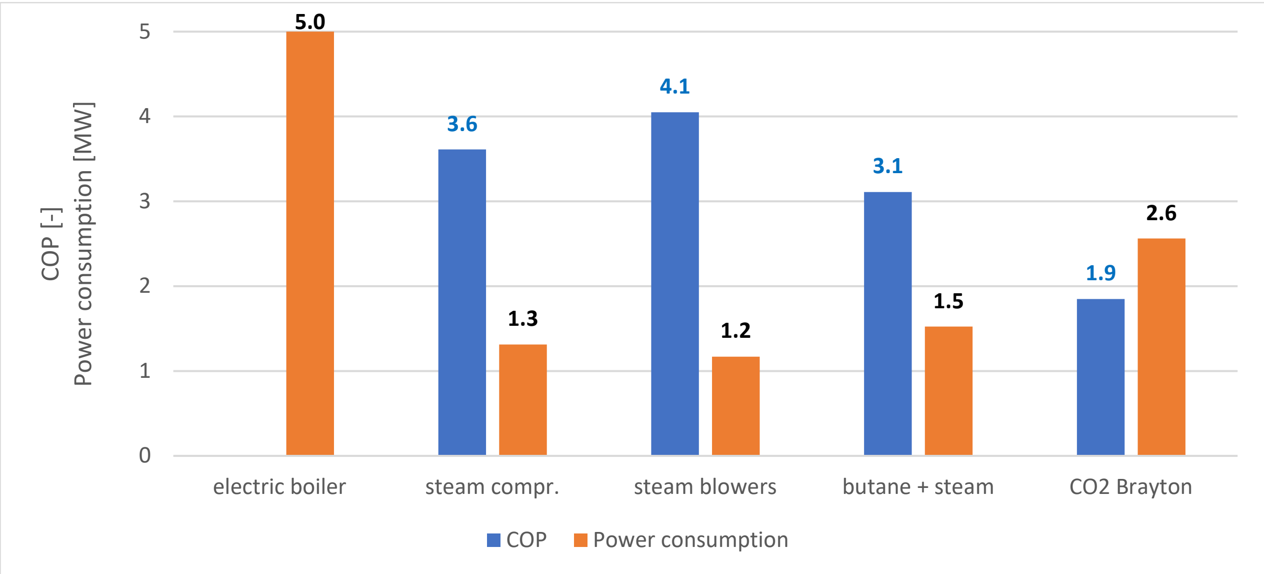

Figure 4-1. COP and power demand for the different HTHP concepts.

Comparing the two steam cycles ("steam compr." and "steam blowers") the concept applying steam blowers offers a higher COP, explained by the lower pressure ratio for each compression stage and the corresponding higher isentropic efficiencies. In addition, the "steam blower" concept applies a double-stage evaporation which is thermodynamically favourable (see section 5.4.1). The cascade cycle ("butane + steam") shows a lower COP, and a corresponding higher electricity consumption of 15-30% compared to the steam cycles.

The reversed Brayton cycle ("CO_2 Brayton") requires about double the amount of electricity compared to the steam cycles. One explanation is that steam heat pumps generally show a high efficiency for cases in which both the heat source and sink have relatively constant temperatures, while the reversed CO_2 Brayton cycle reaches higher efficiencies for cases with large temperature changes [5]. Thus, the CO_2 cycle would perform better for an application were hot water is produced (i.e. a heat sink with a large temperature difference), rather than steam (for which the heat demand is mostly at the constant evaporation temperature).

Concludingly, from an *energetic point of view*, there is no gain in including a bottom cycle or in producing the steam directly in a reversed Brayton CO_2 cycle.

4.1.2 Savings in electricity consumption and CO_2 emissions

If assuming steam is required 8760h/year the annual power consumption for the existing electric boiler is 44 GWh corresponding to yearly CO_2-emissions of 828 ton. Table 4-1 shows the ranges in annual power consumption and CO_2 emissions for the evaluated heat pumps concepts, as well as the resulting reductions in consumption and emissions compared to the electrical boiler.

Table 4-1 Energy consumption and CO_2 emissions in reference scenarios and in heat pump scenario.

Scenario	Power consumption [MW]	Annual power consumption [GWh]	Annual CO_2 emissions [tons CO_2]
Electric boiler	5	43.8	828
HTHP concepts	1.2 - 2.6	10.2 - 22.4	194 - 424
Reduction with HTHP	2.4 - 3.8	21.4 - 32.3	404 - 634
	49 -77 % reduction in electricity consumption and CO_2 emissions		

In Figure 4-2 the potential power savings when implementation the different HTHP concepts are shown. Note that since the savings is compared to an electrical boiler, the percentage reduction in CO_2 emission are the same as the electric power savings.

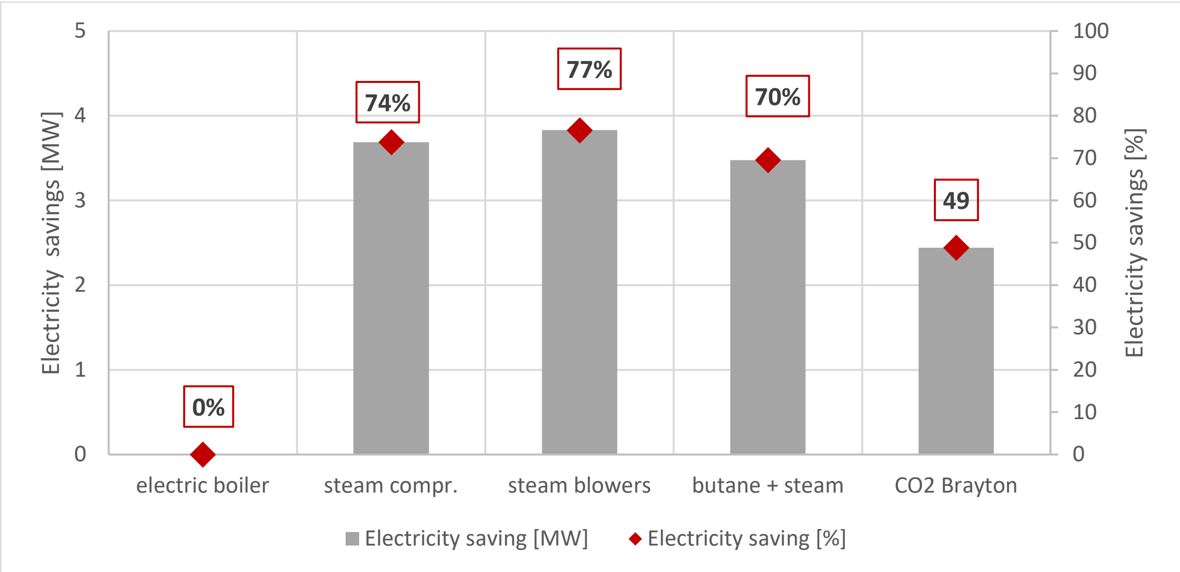

Figure 4-2: Potential power savings of implementing HTHP concepts, compared to the existing electrical boiler.

4.1.3 Utilised waste heat

Typical values of available waste heat are 350-400 m^3/h. With the assumed heat source outlet temperature of 80°C, the required waste heat flow for producing 4.5 MW is smaller than this. As shown in Table 4-2, it ranges between 183 and 289 m^3/h. This implies that either a larger amount of steam can be supplied (using all the available waste heat flow) or the 4.5 MW can be supplied with a slightly higher efficiency by keeping the heat source outlet temperature on a higher level. This is further evaluated in section 5.3.

Table 4-2: Required waste heat flow, with an outlet temperature of 80°C, for producing 4.5 MW steam.

Utilised waste heat flow [m^3/h]	Steam compr	Steam blower	Cascade	CO_2 Brayton
	289	300	271	183

4.2 Operation / design parameters

In this section some of the key parameters related to operation and design are presented.

4.2.1 Volume flows

Figure 4-3 shows the total volume flow, defined as the sum of the suction volume flow to all compressors and compression stages.

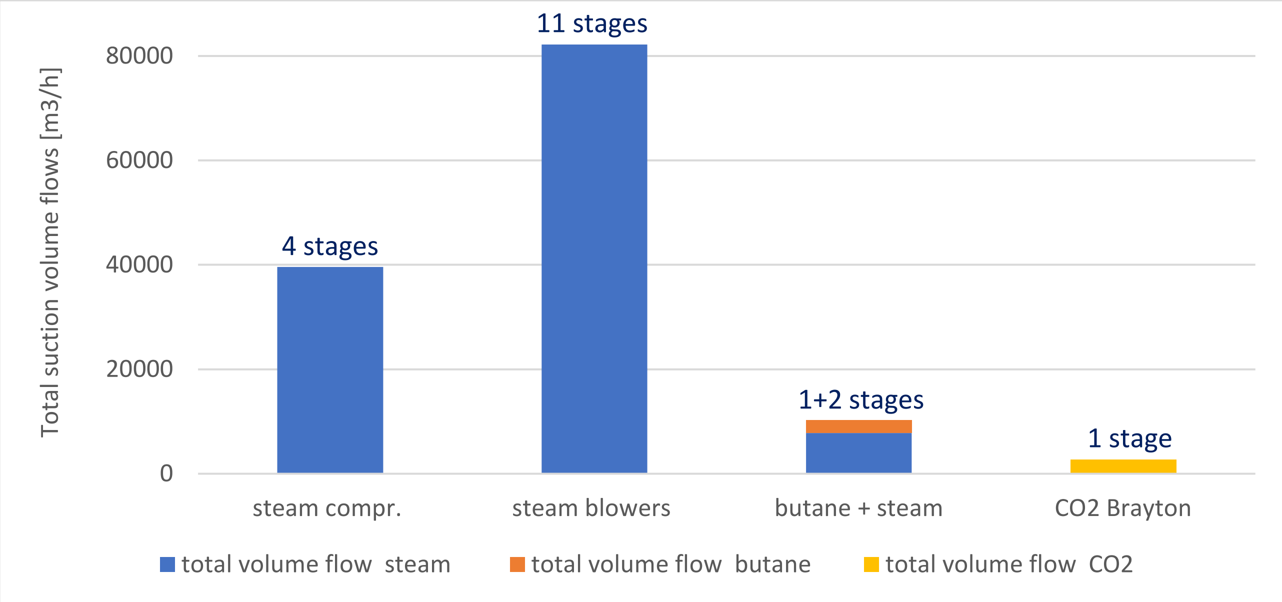

Figure 4-3: Total volume flow of the different refrigerants (working media) for the different HTHP concepts.

As discussed before, a challenge with using water/steam as refrigerant is the large volume flows, which is clearly shown in Figure 4-3. The two concepts using only steam compressors require 15 - 30 times more volume than the CO_2 Brayton cycle and 5 - 10 times more than the cascade cycle. Comparing the two steam cycles, the concepts using blower requires twice as much due to the larger number of stages.

It is also clearly shown that a high COP (large power saving) corresponds to larger volume flows (lower volumetric heating capacity). Thus, even if the cascade cycle and CO_2 Brayton cycle are less favourable in relation to energy efficiency, they show advantages in much more compact systems (and possibly lower investment costs).

4.2.2 Discharge temperature

In Figure 4-4, the discharge steam temperature for the different concepts are shown, and also the discharge temperature from the butane and CO_2 compressor. For the steam cycles, the high discharge (superheated) steam temperature presents a challenge. So far, the maximum reported discharge temperature for centrifugal compressors dedicated to heat pumps reported is 225°C for a steam turbo-compressor. However, in a SINTEF test rig a turbo compressor is operated up to 300°C.

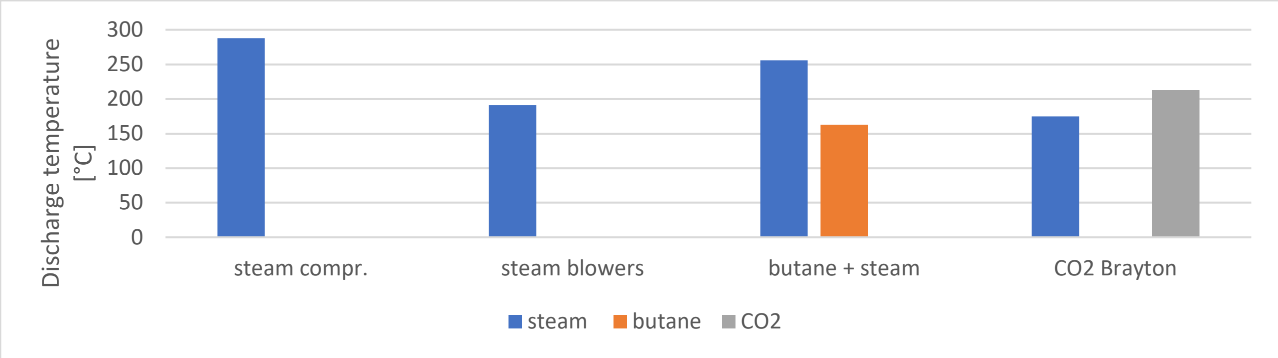

Figure 4-4: Compressor discharge temperatures for the different HTHP concepts. Note that the delivered steam saturation temperature is 175°C (9 bar).

4.2.3 Summary of performance and operating parameters

In Table 4-3 some other operating parameters for the evaluated HTHP concepts are shown, such as operational pressure levels and compressor pressure ratios.

Table 4-3: Operating parameters for the different steam cycles.

	R718-compr	R718-blower		R600+R718		CO_2
		Evap 1	*Evap2*	*R600*	*R718*	
COP [-]	3.61	4.05		3.11		1.85
Evaporator pressure [bar]	0.39	0.39	0.47	9.1	2.0	101
Condenser /gas cooler pressure [bar]	n.a.	n.a.		24.2	n.a.	140
Total pressure ratio steam [-]	23.1	20.9		4.5		n.a.
Pressure ratio each stage [-]	2.2	1.4		2.7	2.1	1.4
Volume flow, 1st stage [m^3/h]	20655	10740	17481	2452	5250	2717
Volume flow, total [m^3/h]	39577	82185		10263		2717
Discharge temperature [°C]	288	191		163	256	213
Waste heat flow [m^3/h]	289	300		271		183
Waste heat [MW]	3.3	3.4		3.1		2.1

5 Sensitivity analysis

A minor evaluation of the influence of certain operating/design parameter has been performed in this section.

5.1 Scope of analysis

Table 5-1 shows for which parameters the analysis is performed, together with the change in parameter value and the considered HTHP concept(s). The results are presented in the following sections.

Table 5-1: Scope of study for the sensitivity analysis.

Operational /design parameter:	Concept evaluated	Initial	Changed to
Produced steam – pressure level	All	9 bar	7 bar
Utilised waste heat flow	All	calculated	400 m^3/h
Outlet heat source temperature	Steam cycle with compressors	80	85
Evaporation – single or double stage	Steam cycle with blowers	double	single
Steam evaporation pressure	Cascade cycle (butane/steam)	1 bar	2 bar
Gas cooler pressure	Reversed Brayton cycle	140 bar	200 bar

5.2 Reduced steam pressure

All results presented in the previous chapters are related to a heat delivery of 9 bar steam, as the process is today. In Figure 5-1, the savings in electricity consumption is shown for a steam supply of 9 and 7 bar respectively.

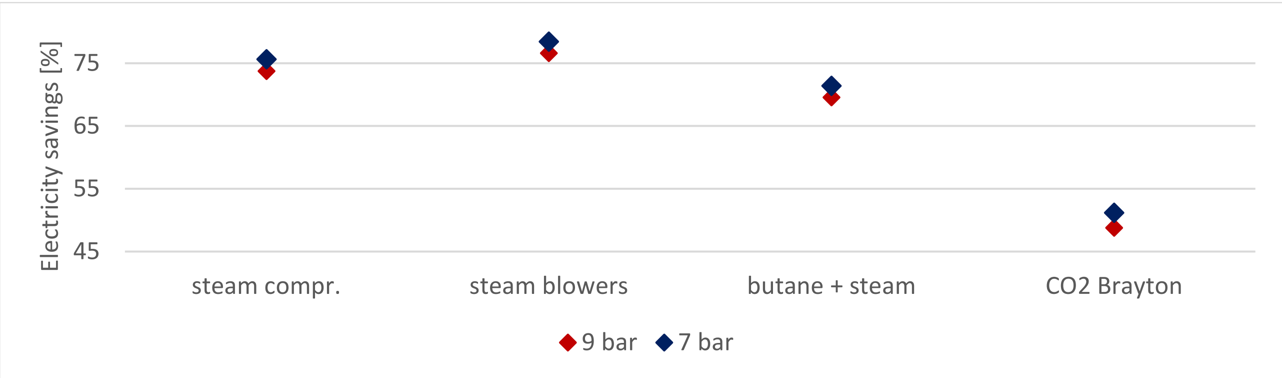

Figure 5-1: Comparison of electricity savings between steam supply at 9 and 7 bar.

As seen in Figure 5-1, only a marginal increase in power savings is achieved by decreasing the pressure level of the supplied steam. Reasons for this, maybe unexpected, low influence includes;

- A larger steam mass flow is required to still supply 4.5 MW
- The gain related to reduced pressure ratio is not fully considered
- The decrease in pressure ratio is not enough to reduce the number of steam compression stages

This can be further explained when looking at the influence on key evaluation parameters, shown in Figure 5-2.

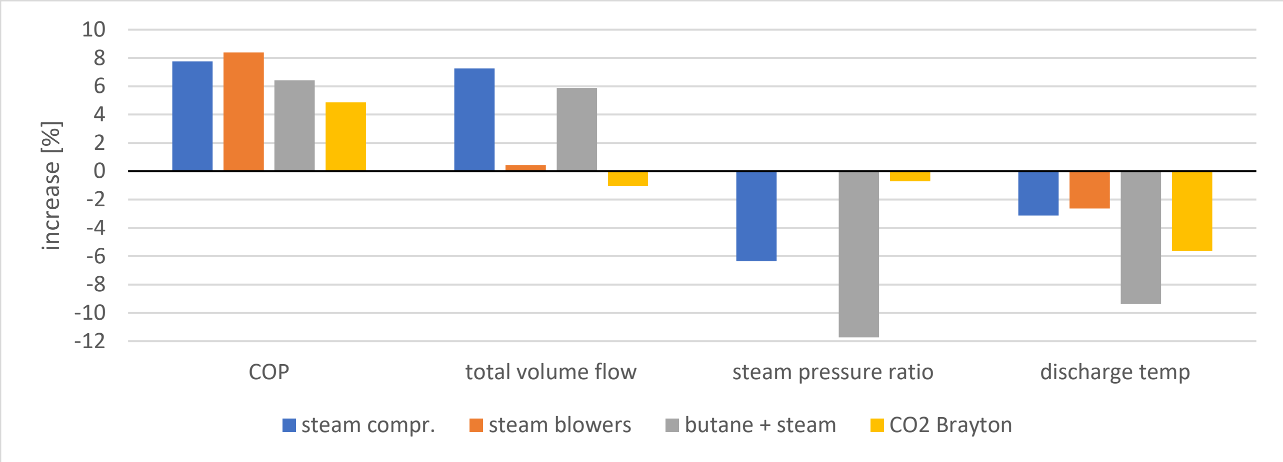

Figure 5-2: Change in key parameters when decreasing the steam supply pressure from 9 bar to 7 bar.

As seen in Figure 5-2, the increase in COP is largest for the steam cycles. For the steam cycle with compressors, the pressure ratio is decreased, but still four stages are required. However, the reduced pressure ratio might imply a higher isentropic efficiency and thus a larger increase in COP than is shown here. For the steam blower cycle the pressure ratio is kept the same (1.4) but the number of stages is only marginally reduced (from 11 to 10 stages). The lower steam discharge temperature might also facilitate a heat pump implementation, since todays solutions are limited in temperature.

When delivering the steam at 7 bar instead of 9 bar, a somewhat larger amount of steam has to be produced to still deliver 4.5 MW. This explains the increase in total volume flow for the steam compressor concept and for the cascade cycle. For the steam blower concept, there is only a negligible increase in total volume flow, since operating at a lower pressure level enabled one blower stage less. For the Brayton cycle the volume flow is actually slightly decreased, explained by a higher suction density of CO_2 (due to a slightly larger pressure and lower temperature when operating at 7 bar steam).

5.3 Waste heat flow and temperature

As discussed in section 4.1.3, all available waste heat flow is not required for producing 4.5 MW. This implies that either a larger amount of steam can be produced (increasing the waste heat flow) or that the same amount (4.5 MW) can be produced at a slightly higher efficiency, by keeping the heat source outlet temperate higher than 80°C.

Heat source flow. Typical available waste heat flows were stated to 350-400 m^3/h. Figure 5-3 shows the possible steam production using 400 m^3/h, still cooling it down to 80°C. For the steam-cycles and the cascade cycle around 6 MW steam can be supplied and almost 10 MW with the CO_2 Brayton cycle.

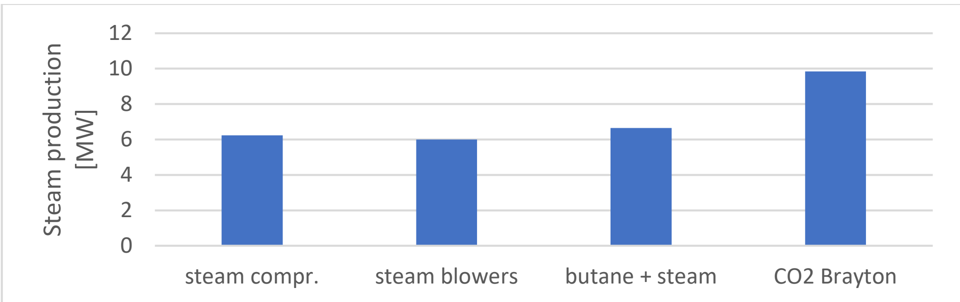

Figure 5-3: Maximal steam production for a waste heat flow of 400 m^3/h and outlet temperature of 80°C.

Heat source outlet temperature. The possibility to achieve a higher COP by operating at a higher outlet temperature of the heat source is exemplified for the open steam cycle with compressors. Figure 5-4 shows the influence on key parameters when keeping the outlet temperature on 85°C instead of 80°C. The higher outlet heat source temperature enables the system to operate with a higher pressure in the steam evaporator leading to all the positive effects seen in Figure 5-4; increase in COP and at the same time reduced volume flow, pressure ratio and discharge temperature. However, the extra saving in electricity consumption is only 70 kW, corresponding to 1.5% additional saving in electricity consumption. Note also that this requires a waste heat flow of almost 600m^3/h, which is typically not available.

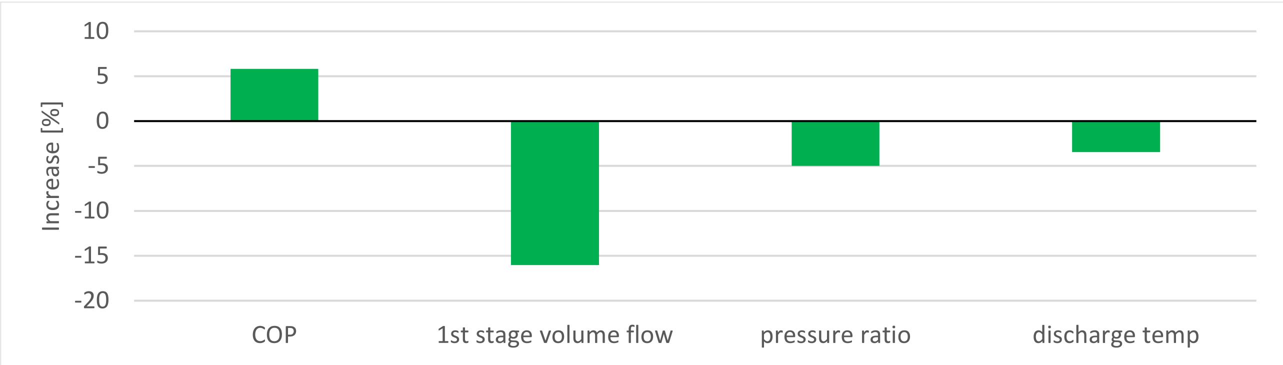

Figure 5-4: Influence on key parameters of keeping the heat source outlet temperature on 85°C instead of 80°C.

For the cascade cycle a similar influence was seen, while for the CO_2 concept there is no gain, since the temperature and pressures are not related in the same way and since this cycle is favoured by a large temperature difference of the heat source.

5.4 Some design/operating choices

5.4.1 Double-/single-stage evaporation

The concept with steam blowers is equipped with a double-stage evaporator (Figure 3-2) to increase the performance. Since this is a more expensive and complex design, even though more flexible, an evaluation of the gain in performance, compared to standard single-stage evaporation was performed.

In Figure 5-5, the change in COP and volume flow is shown when applying single-stage evaporation instead of double-stage. Even though the COP decreases and total volume flow increases, the difference in electricity savings is less than 1%.

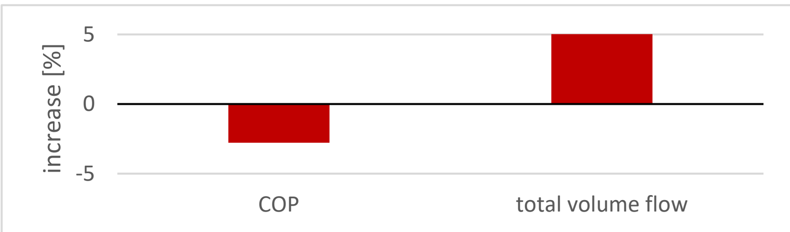

Figure 5-5: Change in COP and volume flow when operating with single-stage evaporation instead of double-stage.

5.4.2 Temperature level in cascade cycle

In a cascade cycle the temperature level for steam generation (i.e. the bottom cycle condenser) can be optimised in relation to COP and steam volume flow. The effect of this was evaluated by decreasing the steam generation temperature (by decreasing the temperature lift in the butane cycle), from 120°C to 110°C.

As seen in Figure 5-6, reducing the temperature level is, as expected, negative in relation to volume flow, pressure ratio and discharge temperature, implying a potential need for one extra compressor stage. The increase in COP is not as obvious; COP for the bottom cycle will be increased due to a lower pressure ratio, while the top cycle COP will be reduced due to a larger PR. As shown in previous case studies, and confirmed here, the COP of the bottom cycle has a large influence of the COP for the total system.

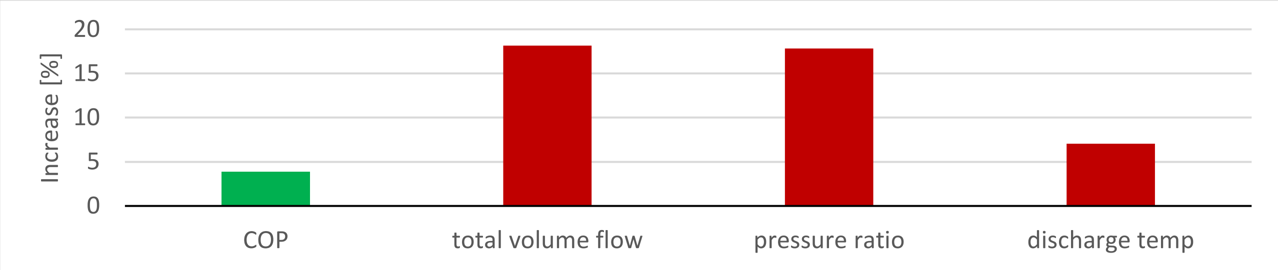

Figure 5-6: Change in key parameters when reducing the temperature of the steam, generated by the butane bottom cycle, from 120°C to 110°C.

5.4.3 Gas cooler pressure in the reversed Brayton cycle

The change in performance of operating the CO_2 cycle at higher gas pressure, 200 bar instead of 140 bar, was evaluated. As mentioned in section 3.3, for the CO_2 Brayton cycle, the low and high pressure are not given by the heat sink and heat source directly, since CO_2 is in gas phase at each state point in the cycle. Thus, the temperature and pressure are not connected as in a conventional heat pump including an evaporator and a condenser. In conventional cycles COP tends to decrease with an increase in the high-pressure level, but for cycles operating as trans-critical or super-critical processes, as the reversed Brayton cycle, the behaviour is quite different. This is explained mainly by the fact that in the critical region the enthalpy and entropy are greatly influenced by the pressure [8]. For example, normally the enthalpy at condenser outlet mainly depends on temperature, but the enthalpy at the gas cooler outlet is also influenced by the pressure.

Figure 5-7 shows the influence of gas cooler pressure ("high pressure") on COP, suction volume flow, pressure ratio and discharge temperatures.

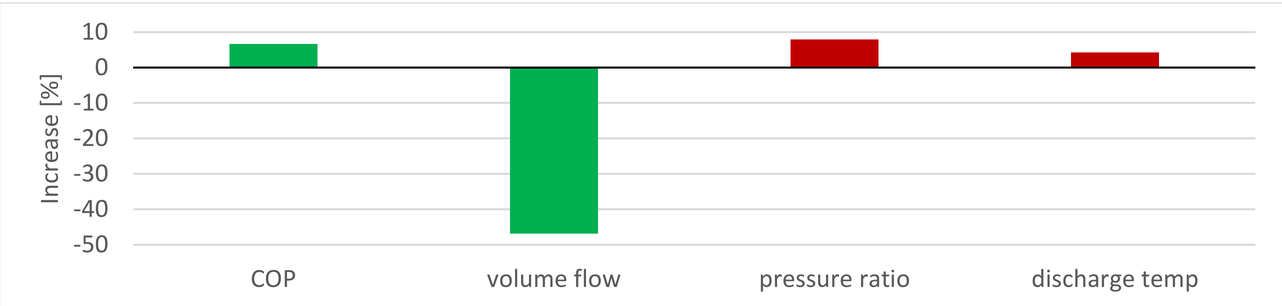

Figure 5-7: Change in key parameters when increasing the high pressure level (CO_2 gas cooler pressure) from 140 bar to 200 bar in the reverse Brayton cycle.

As seen, operating at a higher gas cooler pressure gives a 7% increase in COP, and at the same time also the effect on volume flow is positive. Again, this is due to the non-dependence of temperature – pressure. The gas heater pressure increases also (from 100 bar to 133 bar). Thus, the increase in pressure ratio is only minor (still a pressure ratio of only 1.5).

5.5 Summary

Table 5-2 presents a principal summary of the heat pump evaluation by a rough grading of some important aspects. As seen, to choose the most suitable concept includes a trade-off between energy efficiency, system compactness, complexity and technological readiness.

Table 5-2: Trade-off between different aspect for the evaluated heat pump concepts.

Heat pump system	Complexity	Energy efficiency	Heating capacity (compactness)	Tecnological Readiness Level (TRL)
Steam compressors	🟡	🟢	🔴	7-8
Steam blowers, double-stage evaporation	🟡	🟢	🔴	7-9
Cascade: butane + steam	🔴	🟡	🟡	5-6
Gas cycle: CO2 reversed Brayton	🟢	🔴	🟢	5

Technological readiness level (TRL): The steam cycles have a high TRL, but existing solutions are limited in steam temperature/pressures. There are suppliers offering heat pump concepts with steam turbo compressors or blowers delivering steam up to 5 bar /150°C, but to reach 9 bar further development is required. Another option is to consider adding a top-stage cycle with another compressor technology, raising the pressure from 5 bar to 9 bar.

For the cascade cycle, the lower TRL is mainly due to the low capacity of existing butane compressors, while for the CO_2 reversed Brayton there are suppliers ready to design and build but no such system exists today.

6 Conclusions and further work

The following high-temperature heat pump concepts were evaluated for upgrading 90°C waste heat (water) to deliver 4.5 MW of 9 bar steam to a nickel refining process, thereby replacing the existing electrical boiler:

- Open steam cycle with turbo compressors (4 stages)
- Open steam cycle with steam blowers (11 stages) and double-stage evaporation
- Cascade cycle with the bottom cycle using butane, delivering steam at 2 bar to the steam top cycle
- Reversed Brayton cycle using CO_2, producing 9 bar steam directly (no steam compressor)

From a performance evaluation of the different concepts, the following indicative conclusions were drawn:

- ➢ The highest energy efficiency (COP) is achieved for the steam cycles, with a COP of around 4, while COPs of 3 and 2 were achieved with the cascade cycle and the reversed Brayton cycle, respectively.

- ➢ The corresponding savings in power consumption and CO_2 emissions are around 75% for the steam cycles and 50% for the reversed Brayton cycle, compared to the existing electrical boiler.

- ➢ Even though the steam cycles have the highest TRL level of the evaluated concepts, there are challenges relates to the high steam delivery temperature, which today is limited to around 150°C, (5 bar) and also to the large volume flows required.

- ➢ In terms of total compressor suction volume flows, the open steam cycles require 15-30 times higher volume flow than the revered Brayton cycle and 5-10 more than the cascade cycle.

- ➢ Reducing the steam delivery pressure from 9 to 7 bar only resulted in 2% additional energy savings.

- ➢ All available waste heat flow is not utilised to produce 4.5 MW, enabling either a larger steam production or a production at a slightly higher efficiency. With 400 m^3/h waste heat flow, a steam production of 6 - 10 MW can be achieved, and the Brayton cycle enables the largest steam production.

- ➢ The choice of the most appropriate concept includes a clear trade-off between energy efficiency, system compactness, complexity and technological readiness.

Suggestions for future work:

- Discussions of technical feasibility, together with potential suppliers of compressors, heat exchanger etc, aiming at a first concept design.
- Economic evaluation of CAPEX, OPEX and payback period with respect to different concepts and implementation scenarios.

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