

Exergy Analysis of Heat Pump Integration in Solvent-Based CO₂ Capture

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Abstract

CO₂ capture is required for the decarbonisation of hard-to-abate industries and for enabling negative emissions. Solvent-based capture is the most mature technology, but its deployment is limited by the significant heat requirement associated with solvent regeneration. Heat pumping emerges as a potential solution since waste heat is plentiful in the process. This study explores the benefits of heat pump integration through the lens of exergy minimisation. With a capture system integrated to a biomass-fired cogeneration plant as a case study, both vapour-compression and absorption heat pumps are investigated, either to supply the main reboiler or to be integrated as a side heat load in the stripper. Heat pump integration is performed by optimally selecting cold sources identified thanks to pinch analysis. In addition, low-pressure stripper operation is extensively studied because of the conflicting effects of reduced temperature lift and increased reboiler duty. While direct heat integration opportunities are almost non-existent, heat pump integration is found to systematically reduce the exergy requirements of the process by 8% to 25%. Among the investigated options, absorption heat pumps provide the largest exergy reductions. The heat pump integration procedure is also found to be highly sensitive to assumptions.

1. Introduction

CO₂ capture is considered for mitigating hard-to-abate greenhouse gas emissions in relevant industrial sectors (cement, steel, petrochemicals, high-temperature heat...) or to enable carbon dioxide removal (CDR) [1]. In fact, most mid- and long-term emission scenarios require CO₂ capture to some extent in order to remain compatible with current global warming targets [2]. Delays in emission reduction efforts also increase the need for such technologies, given unchanged climate objectives.

Several CO₂ capture technologies exist, relying on different physicochemical principles. To this day, however, solvent-based chemical absorption remains the most prevalent technology [1]. In this process, CO₂ is absorbed by a chemical solvent, which is selected for its affinity for CO₂. Amines are often used as solvents. After absorption, energy is then supplied to separate CO₂ from the solvent, effectively recycling the solvent for further use.

The detailed process is depicted in Figure 1. Consider a gaseous stream from which CO₂ must be separated. Typically, CO₂ concentration ranges from 4-5 vol% (natural gas combustion) to 10-15 vol% (solid fuel combustion), or even higher for specific industrial processes. The gaseous stream is first cooled down and saturated with water in a *direct contact cooler* (DCC) before entering the *absorber*. In this packed column, the gas is brought into contact with a chemical solvent, which captures CO₂ via absorption. Since the absorption reaction is exothermic – and therefore favoured by lower temperatures, cooling the gaseous stream beforehand is justified. Once CO₂ is removed by the solvent, the gas is washed to eliminate entrained solvent droplets and regulate the solvent's water balance. The clean gas is then vented to the atmosphere.

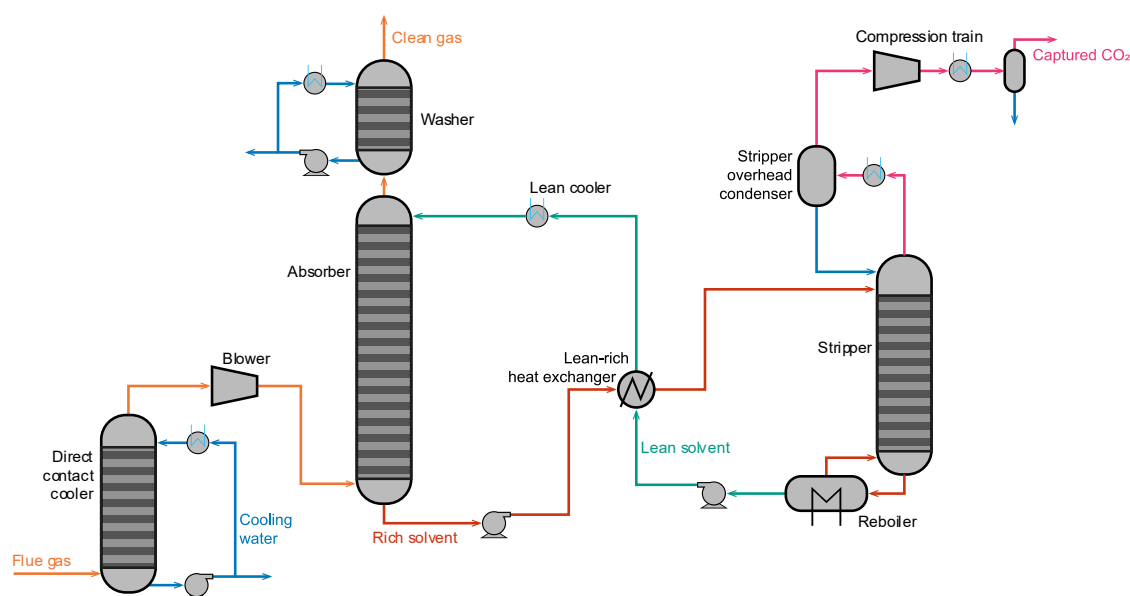


Figure 1. The amine-based CO₂ capture process relies on a chemical solvent that captures CO₂ in an absorption column and that is regenerated in a stripping column.

The chemical solvent (now loaded with CO₂) is collected at the bottom of the absorber and circulated to a second column (called the *stripper*) to be recycled. Heat is applied to break the chemical bond between CO₂ and the solvent, thereby isolating the captured CO₂. Practically

speaking, steam generated in the *reboiler* strips and entrains this captured CO₂. The latter is then washed, cooled down (in the *stripper overhead condenser*) and compressed for transport via pipeline or other means. The depleted solvent is collected and sent back to the first column, effectively closing the loop between absorption and regeneration. Again, since absorption occurs at lower temperatures, the solvent is cooled down: first by preheating the stream that enters the stripper (in a cross-stream heat exchanger called the *lean-rich heat exchanger*), and then by a dedicated cooler (the *lean cooler*) until the target temperature is reached. This capture process requires both cooling (gas cooler, solvent cooler, compression intercoolers...) and heating (reboiler).

Solvent-based CO₂ capture is mature but faces operational hurdles which complicate its large-scale development. In addition, political incentives and social acceptance are required [1]. From a technological point of view, solvent-based capture is mainly hindered by cost, energy use, and solvent degradation. High energy use favours alternatives to CO₂ capture when available and makes decarbonisation costly otherwise. In such cases, minimising the energy use of solvent-based CO₂ capture helps make it feasible, and therefore supports progress in emission reduction efforts. For that purpose, research efforts focus on three main areas: (1) alternative solvents, (2) heat integration, and (3) process modifications.

- 1) Most solvents are amines, and the most common one is monoethanolamine (MEA), more specifically a solution of 30 wt% MEA diluted in water. It is commercially available, has high affinity for CO₂ and limited toxicity, but requires significant regeneration energy. For each tonne of CO₂ that is captured, around 3.7 GJ of thermal energy are required [3]. Many alternatives (other amines, blends, carbonates, ionic liquids, demixing solvents...) with lower specific heat consumption have been proposed over the years. Recently, a new benchmark has been endorsed by the International Energy Agency Greenhouse Gas R&D Programme (IEAGHG). It is a 40 wt% blend of piperazine (PZ) and 2-amino-2-methyl-1-propanol (AMP) in a 2:1 molar ratio, with an corresponding reboiler heat duty of 3.0 GJ/tonne [4].
- 2) Heat integration refers to the optimisation of energy transfers between the capture system and the neighbouring processes. Overall, the aim is to reduce the need for primary energy supply by recycling as much heat as possible in the vicinity of the plant. Many studies report a model in which both the host process (often a power station) and the capture unit are

simulated. However, these studies are case-dependent: utility availability, retrofitting and performance metrics are not standard, and some specificities might facilitate or complicate the integration depending on the context. For example, an adjacent district heating network can be preheated by waste heat generated in the capture system. Systematic guidelines cannot always be identified.

- 3) Process modifications are numerous. From the traditional design depicted in Figure 1, many alternative configurations have been proposed and evaluated, either numerically or even experimentally. Again, most efforts are aimed at the reduction of energy use. Broadly speaking, process modifications can be categorised into absorption enhancement (intercooling, recycling...), internal heat integration (solvent splitting...) and heat pumping (lean vapor compression...). Many studies have analysed individual or combined modifications using techno-economic assessments. Review papers [5,6] then aggregate and compare the results. Unfortunately, most studies are based on different case studies or based on different assumptions. Different performance metrics can also be used. Systematic comparison between these results is therefore difficult.

Within process modifications, heat pumping can be considered as a sensible improvement to the solvent-based CO₂ capture system.

Recent developments in high-temperature heat pump (HTHP) technology report achievable heat sink temperatures of more than 120°C with heat lifts of more than 100 K. Capacities range from tens of kW to tens of MW. Further developments in the coming years may lead to commercially-available HTHPs capable of delivering heat to sinks above 160°C. [7–10].

These technological advancements are beginning to qualify HTHPs for heat supply in solvent-based CO₂ capture systems. In effect, this would simultaneously reduce the need for external energy supply and lower cooling requirements. Heat pump integration can also enable the integration of low-carbon energy carriers, such as renewable electricity, as a replacement for fossil fuel combustion that is usually considered to provide energy to the reboiler. Both energy use and embedded greenhouse gas emissions would therefore decrease.

Several authors have proposed various types of heat pumps to be integrated into the traditional capture system (see Section 2). However, most integration studies are techno-economic analyses.

The conclusions are heavily influenced by the cost estimation methods and the underlying hypotheses. The integration potential is not well outlined for now.

For these reasons, this article analyses the technological potential of heat pump integration into solvent-based capture systems. Both the thermodynamic potential and the expected performance are analysed for different heat pumping systems. This should contribute to the development of heat pump integration by delineating the scope of improvements, helping gauge the worth of integration efforts by knowing the best achievable savings, and suggesting the most promising heat pump technologies.

The scope of the study is limited to the CO₂ capture system itself. The boundaries are set from the hot flue gas (that would otherwise be vented through the stack) to the compressed CO₂ that is ready to be transported to a storage or reuse facility. Integration with the host process is not considered. This allows for the identification of the potential of the capture system itself, regardless of where it is implemented. This is especially relevant for mobile units or in retrofit cases, and for all configurations in which heat integration with the host process is not possible. For instance, all industrial applications that are already heavily integrated have little to no room for additional integration. In cases where the host process allows, broader heat integration could nevertheless enhance heat pump potential.

The rest of the paper is structured as follows. Section 2 outlines the scope of the studied systems and previous works on the matter. Section 3 describes the methodology, the assumptions and the selected case study. Results are then reported in Section 4 and discussed in Section 5.

2. State of the art in heat pump integration

Generally speaking, heat pumps are devices that transfer heat against a temperature difference. They concurrently achieve cooling and heating by drawing low-temperature heat (from the so-called *cold source*) and upgrading it to a higher temperature level (for injection into the *hot sink*). From a high-level perspective, heat pump cycles can be broadly classified as closed-loop cycles (using a dedicated working fluid) and open-loop cycles (using a process stream as the working fluid). This paper focuses on closed-loop cycles. The dedicated working fluid (called *refrigerant*) is either a pure substance or a mixture. These heat pumps interface with heat sources and sinks thanks to heat

exchangers. The refrigerant circulates through a low-pressure *evaporator* where it draws heat from the source, then is compressed to reach a high-pressure *condenser* where it releases heat to the sink. The refrigerant is finally expanded to close the loop.

Heat pump integration can represent a significant increase in capital expenditures. However, economic analysis is not the focus of this paper. While plant designers typically aim for an economic optimum, this paper focuses on the exergy optimum to provide expected ranges for system integration. Both optima may coincide if the operational expenditures (and energy use in particular) are the largest contributors to the cost.

Closed-loop heat pumps can be categorised according to their driving energy source as either mechanically driven or thermally driven. This paper focuses on the principal representative of each category: vapour-compression heat pumps (VCHPs) and absorption heat pumps (AHPs), respectively [7].

In vapour-compression heat pumps (VCHPs), the compression is carried out by a mechanically driven gas compressor, typically powered by an electrical drive. Their performance is represented by the coefficient of performance (COP) which, when studied as heating devices, is given by:

$$COP = \frac{\dot{Q}_H}{\dot{W}} \quad (\text{Eq. 1})$$

where $\dot{Q}_H$ is the heat supplied to the hot sink and $\dot{W}$ the required mechanical power. In high-temperature applications, the COP typically lies between 2 and 5 [7].

Contrarily to VCHPs, absorption heat pumps (AHPs) are thermally driven devices. Indeed, the compression is carried out by a contained absorption-desorption cycle that replaces the gas compressor. In this sub-process, a second fluid (called the *absorbent*) absorbs the refrigerant in a liquid phase once it exits the evaporator by the means of an exothermic reaction. The rich solution is then pumped to a generator where high-grade heat is used to drive the opposite reaction and separate both fluids. The absorbent then cycles back to the absorber while the refrigerant continues towards the condenser. Conceptually, this compression step relies on the same principles as the studied chemical absorption CO₂ capture process. The most common refrigerant/absorbent pairs are NH₃/H₂O and H₂O/LiBr but alternatives are investigated in literature [11].

Two types of AHPs can be defined:

- 1) Type 1 AHPs follow the steps described previously. As illustrated in Figure 2, they draw heat at two temperature levels (at a low temperature T_L in the evaporator and at a higher temperature T_H in the generator) and release it at an intermediate level T_M (both at the absorber and the condenser). They are mainly used for refrigeration applications and are sometimes called *absorption chillers*. Similarly to VCHPs, their performance is represented by the coefficient of performance (COP):

$$COP = \frac{\dot{Q}_M}{\dot{Q}_H + \dot{W}_P} \approx \frac{\dot{Q}_M}{\dot{Q}_H} \quad (\text{Eq. 2})$$

where $\dot{Q}_M$ is the heat supplied by the heat pump, $\dot{Q}_H$ is the high-grade heat consumed at the generator, and $\dot{W}_P$ is the mechanical work that is required within the absorption-desorption loop to pump the loaded solvent. This work is typically orders of magnitude lower than the heat consumption and is consequently neglected in the performance calculation [12]. Typical COP values range from 1.3 to 2.2 [7].

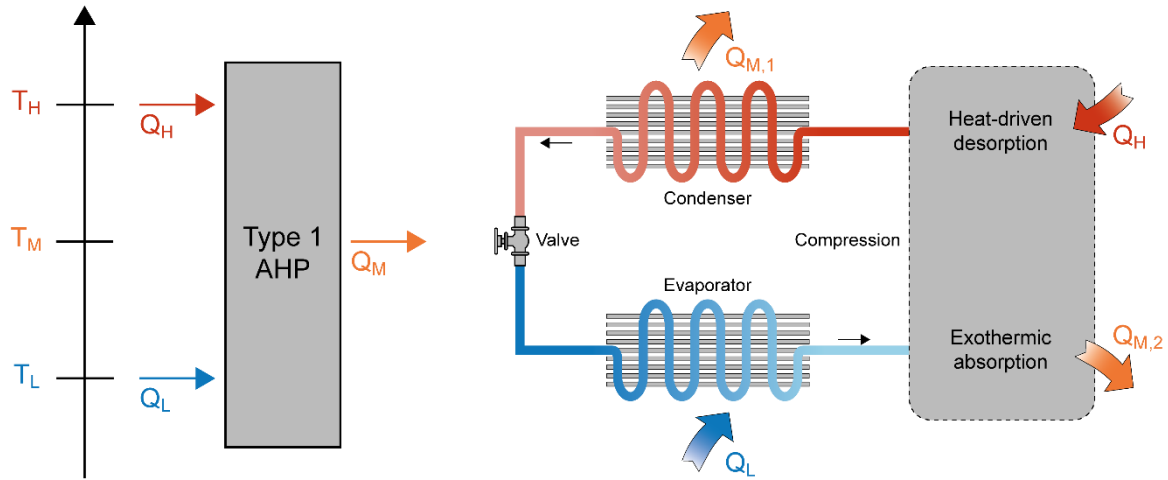


Figure 2. Type 1 absorption heat pumps (AHP) draw heat from two temperature levels and supply it back at an intermediate temperature.

- 2) Type 2 AHPs rely on the same cycle but the high- and low-pressure sides are swapped: the generator and condenser operate at a lower pressure than the evaporator and absorber. Therefore, these AHPs draw heat from a single temperature level T_M (at the evaporator and the generator), upgrade part of it to a higher temperature T_H (at the absorber) and reject the remainder at a lower temperature T_L (at the condenser). This is illustrated in Figure 3.

They are used as heating devices and are sometimes referred as absorption heat transformers (AHT) [11,13]. Their coefficient of performance (COP) is calculated as follows:

$$COP = \frac{\dot{Q}_H}{\dot{Q}_M + \dot{W}_P} \approx \frac{\dot{Q}_H}{\dot{Q}_M} \quad (\text{Eq. 3})$$

Where $\dot{Q}_H$ is the high-grade heat supplied by the heat pump, $\dot{Q}_M$ is the heat consumed, and $\dot{W}_P$ is the mechanical work that is required within the absorption-desorption loop to pump the loaded solvent. Again, this is most often neglected. Because only part of the driving heat is converted into useful output, the COP is generally lower than that of type 1 AHPs, with typical values ranging from 0.2 to 0.5 [7].

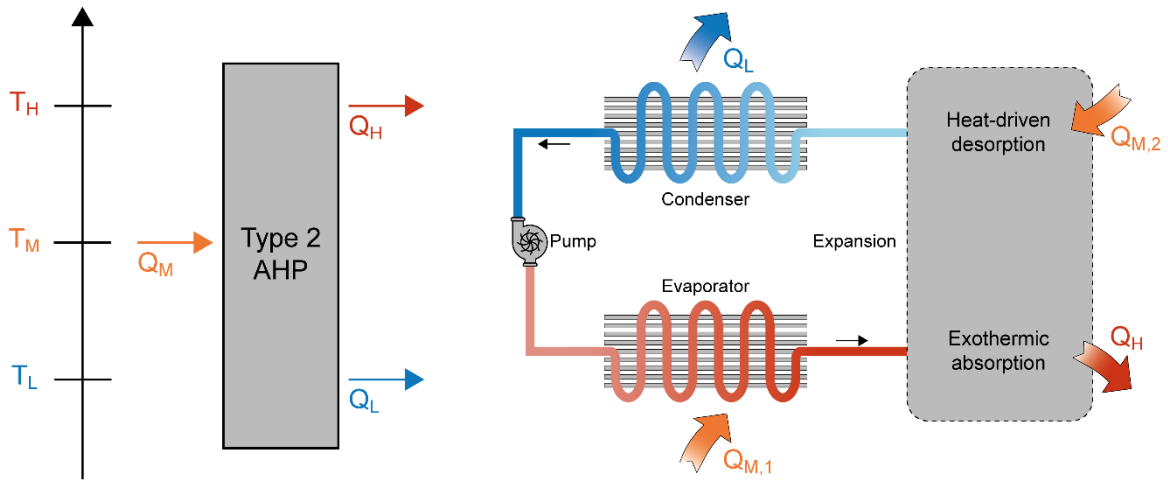


Figure 3. Type 2 absorption heat pumps (AHP) draw heat from a single temperature but releases it at two other temperature levels.

Integration of closed-loop heat pumps in solvent-based CO₂ capture systems has been addressed by several authors.

In their extensive review of process modifications, Le Moullec et al. [6] mention VCHPs (which they name *integrated heat pump* or *IHP*). Based on the review of several patents, they suggest that it should be used to preheat the rich solvent before it enters the stripper. Low quality heat can be obtained from most cooling demands within the process. An estimated benefit is reported as a reduction of 6.5% of the parasitic load that a CO₂ system imposes on a coal-fired plant. In a referenced patent [14], the possibility of completely replacing the traditional steam supply with a heat pump using internal streams is demonstrated.

Guo et al. [15] investigated the integration of a two-stage VCHP drawing heat from the exhaust of the steam turbine in a 600 MW coal-fired power station. The heat pump supplied heat to the CO₂ capture reboiler. While heat pump integration already improved the net electric efficiency of the power plant (from 29% to 30%), several parameters were optimised to further enhance system performance. In particular, operating at a lower reboiler temperature increased the net electric efficiency to 32% and the fuel utilisation efficiency from 29% to 52%.

Wilk et al. [16] studied the integration of a VCHP in an amine-based CO₂ capture system in steel processing. The heat pump was modelled based on the Carnot efficiency and the second-law efficiency. Four heat recovery options were evaluated: from the stripper overhead condenser (SOC), from the lean solvent cooler, from both, and from a plant-wide cooling water loop. The authors concluded that it is best to combine heat recovery from both the SOC and the lean solvent cooler. Reported reductions in both hot and cold demands amount to 50%. The study was also complemented with a cost analysis that identified the cost of steam as the main influence.

Alabdulkarem et al. [17] studied a multifunctional VCHP integrated into the CO₂ capture unit of a natural gas combined cycle (NGCC) plant. The heat pump condenser was split into three segments to allow heat release over different temperatures, effectively providing heating to the reboiler but also process heating at lower temperatures. Seven VCHP configurations were modelled and tested with a total of forty-one refrigerants. Similarly to Wilk et al. [16], they notably studied the stripper overhead condenser (SOC) as the only cold source. The authors found that it was insufficient to replace the reboiler and steam had to be extracted from the NGCC's turbine. They noted, however, that this option remains interesting for energy reduction (16% reduction compared to the base case), while adding limited complexity.

Jensen et al. [18] proposed a two-stage VCHP to replace the natural gas-fired reboiler of the CO₂ capture unit in the biogas upgrading process. Heat was drawn concurrently from the SOC, from the lean solvent cooler, and from intercoolers installed in the absorber. Of particular interest was the study on the reboiler temperature. They highlighted the potential of vacuum-operated strippers (VOS) to increase the COP since it reduces the temperature lift. This study is deepened in the present paper. The conclusion from Jensen et al.'s analysis was that electrification of the amine-based CO₂ capture system was relevant and even robust to electricity prices. In the best scenario,

electrification reduced the absolute magnitude of energy use in the capture process was reduced by 68%.

Absorption heat pumps were also considered by several authors.

Poulidis et al. [19] compared the integration of a VCHP with that of a type 2 absorption heat pump, also known as absorption heat transformer (AHT). Low-quality heat was recovered from the flue gas cooling system at the inlet of the CO₂ capture unit. The authors formulated an optimisation problem to identify the optimal working fluid and operating conditions. They also developed a cascade VCHP-AHT system, effectively combining the two technologies. The AHT outperformed the other configurations, with 53% of the reboiler heat duty supplied by recovered heat, compared with 50% for the cascade configuration. However, the operating conditions of the capture unit were not optimised alongside the heat pump, as they were assumed to be constant.

Zhang et al. [20] investigated the use of a type 2 AHP (AHT) in a CO₂ capture system in a coal-fired plant. Energy savings (in terms of specific energy use per tonne of captured CO₂) amounted to 2.62%. The authors further refined their analysis in the subsequent publication [21]. They compared all three types of closed-loop heat pumps (VCHP, AHP type 1, and AHP type 2) and identified the type 1 AHP as the most relevant but highlighted the large temperature lift.

As part of their review of process modifications, Higgins and Liu [22] investigated a type 1 absorption heat pump (AHP) that uses a lithium bromide (LiBr) solution. Steam served as the high-temperature driving source, while low-grade waste heat was recovered from the outlets of both the absorber and the stripper. The reported heating savings reached 41.2%.

Overall, this review of heat pump integration shows that both vapour-compression heat pumps (VCHPs) and absorption heat pumps (AHPs) have been considered for integration into CO₂ capture units. In all cases, their integration improved the energy performance of the studied systems. While many studies focus on integration with a host process (e.g. power plants or industrial sites), it has also been shown that sufficient waste heat may be available directly within the capture unit. Among the investigated parameters, the reboiler temperature was identified as a key factor for optimising heat pump integration. Finally, performance metrics vary across studies, highlighting the need for a systematic evaluation. This paper proposes such an approach.

3. Methodology

3.1. Case study

Heat pump integration is studied in the context of a case study. The analysis is conducted using a simulation model of an MEA-based CO₂ capture unit developed in Aspen Plus V14. The model is scaled to match the emissions of a biomass-fired combined heat and power (CHP) plant that operates in Liège, Belgium.

The CHP plant caters for the thermal demand of a district heating network (DHN) that is located on a campus of the University of Liège. The CHP plant operates based on a Rankine cycle with two steam turbines. Steam is generated in the biomass boiler at 4200 kPa/420°C and is expanded down to 550 kPa/225°C. Part of it is further expanded to produce electricity in a second turbine while the remainder is routed to the heat exchanger of the DHN. The biomass boiler has a nominal power of 12 MW and is assisted by additional gas boilers to deal with demand peaks [23].

A capture unit is modelled and designed to capture 90% of the CO₂ emitted by the biomass boiler. The gas boilers are left outside of the scope of this study. Since the main boiler is used as a base load unit, its output is constant, and the flue gas stream also is: 54 303 Nm³/h at 159°C with a CO₂ concentration of 9.44 vol%. Over a day, this corresponds to the capture of 135 tonnes of CO₂. Because these emissions originate from biomass combustion, part of these 135 tonnes could be considered as negative CO₂ emissions if captured and stored permanently, but this is not the focus of the current study. Some modelling choices are however relevant:

- The model is adapted from [24] with additional validation from [25]. The selected property method is the Electrolyte Non-Random Two Liquid model with Redlich-Kwong equation of state (ENRTL-RK). The absorber is modelled with six equilibrium stages and the stripper with eight stages. These values are selected because adding additional stages yields only marginal improvements (less than 1% in solvent loading and in energy use).
- To model the lean-rich heat exchanger (LRHX), a temperature difference of 10 K is assumed between the hot stream inlet and the cold stream outlet.
- The operating pressure of the stripper is set at 200 kPa. This choice is motivated by the decrease in specific energy use that comes with higher-pressure operation [26]. However,

marginal gains tend to decrease, and a compromise must be made. This choice is assessed and reviewed later in the paper.

- Captured CO₂ is compressed to 2000 kPa, which corresponds to the operating pressure of the future Belgian CO₂ network [27]. Deviations from this outlet pressure are also studied in later sections. The compression train is modelled as a three-stage compression unit with intercooling, which is done traditionally for compression to up to 30 bar [28]. The overall compression ratio is equally distributed over the compressors. Each compressor has an isentropic efficiency of 80%.
- For operating point investigated in the study, a parametric optimisation is conducted. For a given capture rate (90% in this paper), the operating conditions of the solvent stream can be arbitrarily chosen. Indeed, multiple combinations of lean loadings and solvent flow rates can achieve the capture rate. Moreover, the reboiler heat duty is divided into three components: heating the solvent to the reboiler temperature, generating vapour, and desorbing CO₂ from MEA. These components affect energy use differently and depend on both loading and flow rate. Thus, a minimum energy requirement is expected. To find this minimum, the flowsheet is simulated for different values of these parameters, and the stripper heat duty is recorded. For each operating point, this minimisation is performed but omitted from the discussion to avoid redundancy.

3.2. Pinch analysis

3.2.1. Principles and assumptions

Heat pump integration requires knowledge of both the heating and cooling requirements of the process. A map of these requirements is made by applying pinch analysis to the simulation model.

Pinch analysis is a heat integration method introduced by Bodo Linnhoff in a series of papers published from the late 1970s onwards [29]. Conceptually, it is an application of the Second Law of Thermodynamics to a process. Based on the initial and target temperatures of individual streams, the methodology identifies which streams can exchange heat with one another and to what amount. Indeed, heat transfer is limited since heat can only flow from high to low temperatures. At the end of the heat integration process, the remaining heating and cooling requirements (i.e. that cannot be covered by another stream) correspond to the minimal external utilities that are required by the

process. In effect, this materialises the upper limit on internal heat recovery, or equivalently the lower limit on actual energy use. Pinch analysis is therefore used to optimise heat exchanger networks and heat recovery in industrial processes.

Since pinch analysis is widely used in both academic and industrial settings, the reader is simply referred to Kemp [30] for a step-by-step description of the methodology with worked examples. In this section, only two assumptions are highlighted:

1. The temperature approach in heat exchangers (ΔT_{min}), which defines the energy target of the integration, is a parameter that must be set. Common values are often within the 5°C – 20°C range. In this paper, this parameter is set to 10°C.
2. For a thermodynamically sound operation, it is essential that heat is only supplied in the hot region (above the pinch point) and only removed in the cold region (below the pinch point). There must be no heat transfer across the pinch point. For a heat pump to operate properly under this guiding principle, it must draw heat below the pinch point and release it above. In other words, any heat pump must work across the pinch point. This is called the *Appropriate Placement Principle*, and it is illustrated in Figure 4.

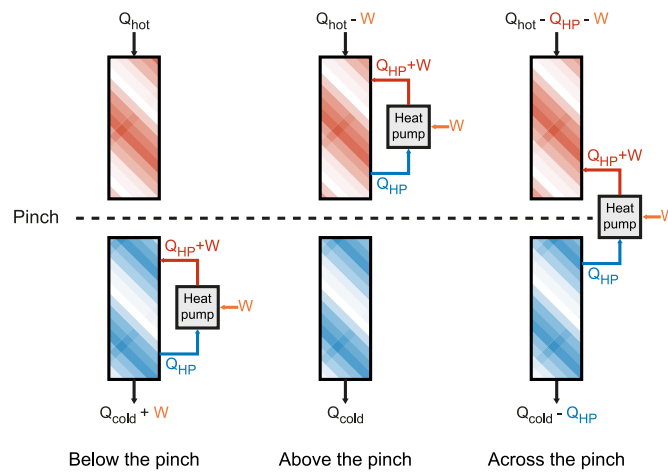


Figure 4. According to the Appropriate Placement Principle, heat pumps must operate across the pinch point to reduce both the heating and cooling requirements. Adapted from [31].

3.2.2. Useful visualisations

The most convenient result from pinch analysis to identify heat pumping opportunities is the Grand Composite Curve (GCC). Indeed, heat requirements (above the pinch) and waste heat availability (below the pinch) can be graphically identified, both in temperature and amount.

The studied system has two columns: an absorber and a stripper. The absorber has neither a cooling demand nor a heating demand and is therefore excluded from the analysis. Implementing some designs aimed at absorption enhancement, such as absorber intercooling, would require the inclusion of the absorber and its utilities into the pinch analysis framework. However, these modifications are out of the scope of this paper. The stripper has both heating (in the reboiler) and cooling (in the condenser) demands. In a first abstraction, both can be modelled in pinch analysis as single, constant-temperature streams.

While this was done in other papers [21], this representation has limitations since it obscures the heat distribution profile and related behaviours. In this paper, three improvements and alternative representations are used:

- 1) *Split Grand Composite Curve (split GCC)*: the reboiler and condenser duties can be represented separately the Grand Composite Curve (GCC) as a standalone box. This is shown in Figure 5, where a mock GCC is generated for illustrative purposes. In most distillation applications, reboiler and condenser loads are similar. In the amine-based CO₂ capture system, the column has no rectifying section, and the condenser duty is significantly smaller than the reboiler duty. The box can therefore resemble a trapezium. This split GCC is useful to understand the potential heat exchange between the column itself and the rest of the process. The overlap between both curves indicates the extent of direct heat integration. It becomes also more convenient to identify which streams could be considered for such integration.

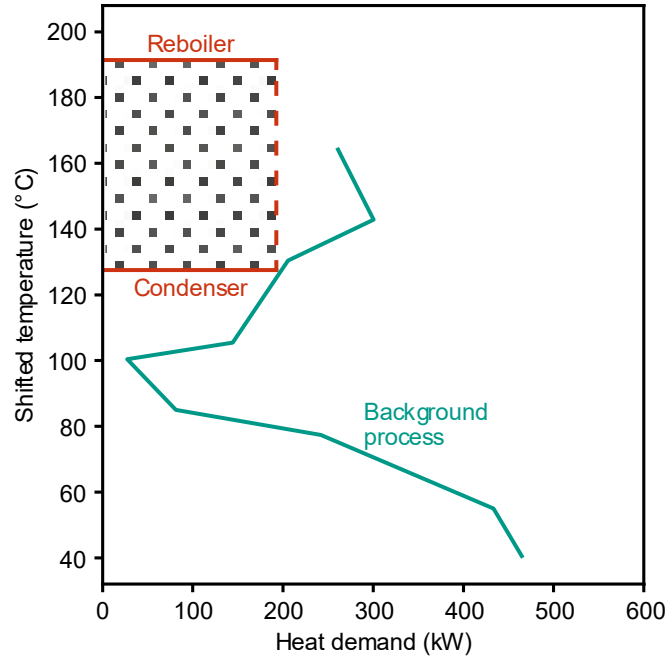


Figure 5. In this mock example of a split GCC, the distillation column (red) can provide heat to the background process (green) since they overlap.

- 2) *Convexity of partial condensation*: to build the GCC, heat capacities are most often assumed constant over each temperature interval. Heat transfers are therefore materialised by straight lines. However, in the solvent-based capture system, the stripper overhead condenser (SOC) is a partial condenser, meaning that only a fraction of the inlet vapour is condensed. More specifically, water is condensed while CO_2 remains gaseous. Because of the progressive reduction in water partial pressure, the saturation temperature progressively decreases. Heat transfer is therefore not linear but follows a convex curve. To account for this phenomenon while performing pinch analysis, the actual heat transfer is linearised into ten segments in the GCC.
- 3) *Column Grand Composite Curve (CGCC)*: a CGCC is an ideal temperature-enthalpy profile of a column. It represents the ideal heat distribution along the temperature profile of the column to achieve Minimum Thermodynamic Condition (MTC). At MTC, the operating line coincides with the equilibrium line, and the energy use is minimal. CGCCs were proposed by Dhole and Linnhoff [32] as an extension of pinch analysis. Applying the profile would require an infinite amount of side reboilers/condensers but CGCCs remain useful to identify promising side loads. This is illustrated in Figure 6.

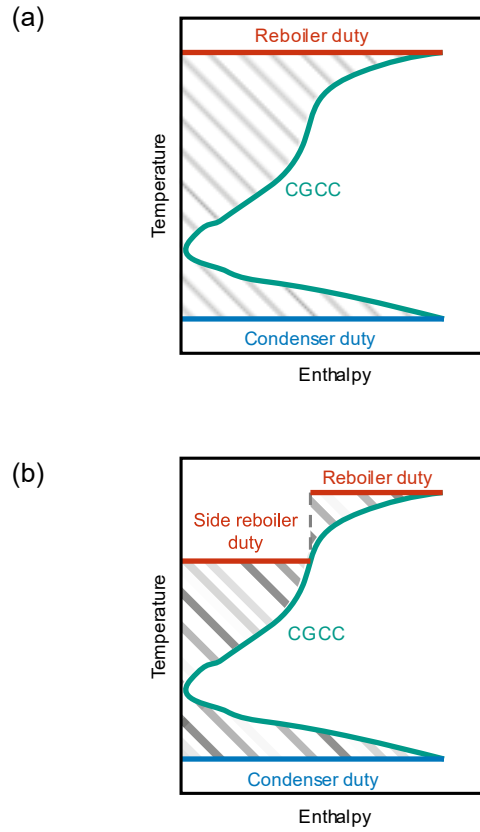


Figure 6. The heat distribution profile suggested by the Column Grand Composite Curve (a) can give insight into potential side loads (b).

3.3. Direct heat integration

Before delving into heat pump integration, pinch analysis results are used to identify direct heat integration possibilities. This helps reduce the required capacity of any heat pump by decreasing the energy demand of the process.

First, the process is analysed for heat integration by direct heat exchange, indicated by *pockets* on the Grand Composite Curve (GCC). The total heating requirement of the process is then given by the largest horizontal distance above the pinch point between the y-axis and the GCC, excluding any area covered by these pockets. Similarly, the total cooling requirement is determined by the largest horizontal distance below the pinch point, excluding heat integration pockets.

Then, modifications to the operating parameters of the CO₂ compression train are investigated. Since CO₂ is compressed on site after capture, intercoolers reject high-grade waste heat that could be used in the process. The waste heat temperature depends on the compression ratio, which

depends on the target outlet pressure. The influence of this target pressure is therefore discussed. In Belgium, where the case study is located, the expected range of operation for the future CO₂ network lies between 2000 kPa and 3300 kPa [27]. The flowsheet is therefore adapted for several pressure levels in this range and pinch analysis is systematically conducted to identify additional reductions in the total energy requirements.

3.4. Heat pump integration

After implementing direct integration opportunities, heat pump integration into the capture process is studied. Because the study is restricted to the capture process itself, the thermal reservoirs that are available for a heat pump are totally defined by the Grand Composite Curve (GCC) introduced previously.

3.4.1. Comparison metrics

Since any heat pump must operate across the pinch point, the hot region of the GCC defines the hot sinks, while the cold region defines the cold sources. Energy savings can therefore be directly assessed by measuring the reductions in total heating and cooling requirements. To compare the performance of different heat pump systems, pinch analysis can be systematically applied both before and after integration, using total heating and cooling requirements as comparison metrics.

However, integrating a mechanically driven heat pump introduces work as a second energy carrier in addition to heat. Therefore, a common basis is necessary for a fair comparison. Exergy is chosen for this purpose. The total exergy requirement of the process is broken down into four components: heating, cooling, heat pumping, and CO₂ compression. They are calculated as follows:

- 1) Heating: the exergy requirement associated with heat consumption is calculated from the following equation:

$$\dot{X} = \dot{Q} \left(1 - \frac{T_0}{T} \right) \quad (\text{Eq. 4})$$

where $\dot{X}$ (kW) is the exergy flow associated with the heat flow $\dot{Q}$ (kW), T_0 is the reference temperature (298.15 K) and T (K) is the temperature at which heat flows. The term $\left(1 - \frac{T_0}{T} \right)$ is called the Carnot factor.

This exergy requirement can be understood as a loss of potential for work production: since heat is consumed at the supply temperature to match the needs of the system, it is not used to power a heat engine between that supply temperature and the environment temperature (T_0). In practice, heat is usually supplied by fuel combustion (e.g. natural gas boilers). Consequently, the temperature at which heat is produced (T_{prod}) is significantly higher than the temperature at which it is consumed (T_{cons}). The degradation of heat from T_{prod} to T_{cons} represents an exergy loss, as the temperature difference could have powered a heat engine to produce work. While this exergy loss can be significant in real systems, it has been disregarded in this paper to maintain a fuel-agnostic approach. In other words, the exergy requirement associated with heat consumption is underestimated in this theoretical framework compared to actual processes.

- 2) Cooling: the exergy requirement associated with process cooling is also calculated based on Equation 4. However, a distinction must be made depending on the temperature at which cooling occurs (T_{cooling}). If cooling occurs above the environment temperature ($T_{\text{cooling}} > T_0$), it would do so at no expense since heat flows naturally from high temperature to low temperatures. A heat engine could even extract work from this transfer. In a way, exergy is made available by such cooling. In practice, cooling still requires actual power (and therefore exergy) to circulate the cooling medium (usually cooling water) but this study is primarily focusing on theoretical principles, so this exergy is neglected. By contrast, if cooling occurs below the environment temperature ($T_{\text{cooling}} < T_0$), work would be needed to upgrade the heat flow from its temperature to T_0 . This corresponds to an exergy requirement for the process. In these cases, the heat flow ($\dot{Q}$ in Equation 4) is treated as negative to yield a positive exergy requirement figure. Jansen and Woudstra [33] provided an extensive discussion on the exergy of cold with worked examples.
- 3) Heat pumping: since work is a high-grade energy carrier that can be readily converted, exergy requirement for mechanically driven heat pumps is equal to work use. For thermally driven devices, exergy requirement is calculated from Equation 4.
- 4) CO₂ compression: similarly, the exergy requirement of CO₂ compressors is equal to their work consumption.

Both heat and exergy share the same units. To help differentiate them, and to facilitate comparison with other case studies, exergy requirement is normalised by the captured CO₂ flow rate, giving a figure in MJ/kgCO₂. This must not be conflated with specific energy use or specific reboiler heat duty, which are metrics that are often used in the field. In this paper, heat is thus reported in joules or watts, with the disadvantage of having case-specific figures.

In addition to the use of a common comparison metric, system performance is benchmarked against the base case process. Since heat pumping is expected to reduce total exergy requirements, the initial process can be considered as an upper bound for the comparison. To evaluate the total exergy requirement of this base case, the methodology presented in this section is applied to the pinch analysis results of the simulation model.

3.4.2. General approach

The integration of different closed-loop heat pumps is limited to single-stage cycles. This choice leads to exergy destruction in heat exchangers since heat sources span over several temperature intervals. However, it ensures simplicity.

Heat pumps are first modelled as ideal cycles, described in subsequent sections. Then, additional assumptions are made to add realism to the models and analyse the eventually changing conclusions. This two-fold approach enables comparisons between:

1. the thermodynamic potential of single-stage heat pumps (ideal model),
2. the improvement potential resulting from the use of actual single-stage heat pumps, with realistic performances (realistic model), and
3. the actual performance of the heat-pump-free process (defined in the previous section as the base case).

Heat pump integration is done by minimising the total exergy requirements, whose components were introduced in the previous section.

3.4.3. Optimisation procedure

First, pinch analysis is used to build the Grand Composite Curve (GCC) which defines the heating requirements (above the pinch point, hot sink) and the waste heat availability (below the pinch point, cold sources).

Then, since heat can be drawn from anywhere below the pinch point, the cold region of the GCC becomes the search space for the optimal cold source. Any point P in this area, defined by a temperature T_L and an amount of heat $\dot{Q}_L$, defines a unique cold source for the heat pump. The corresponding performance is calculated by summing the four components of the total exergy requirements. In other words, the optimal heat pump is identified by minimising the total exergy requirements over all possible cold sources. The objective function is described by Equation 5 and the subsequent constraints (Equation 6):

$$\min_{T_L, \dot{Q}_L} \dot{X}_{tot} = \dot{X}_{heating}(T_L, \dot{Q}_L) + \dot{X}_{cooling}(\dot{Q}_L) + \dot{X}_{HP}(T_L, \dot{Q}_L) + \dot{X}_{compression} \quad (\text{Eq. 5})$$

$$s. t. \quad T_L \in [T_{GCC,min}, T_{pinch}] \quad \text{and} \quad \dot{Q}_L(T_L) \leq GCC(T_L) \quad (\text{Eq. 6})$$

$\dot{X}$ stands for exergy flow (kW), with the subscript *tot* standing for total. The four contributions to the total exergy requirements were detailed in Section 3.4.1. T_L (°C) and $\dot{Q}_L$ (kW) are respectively the cold source temperature and heat. These are the optimisation variables.

Figure 7 illustrates the methodology. The exergy of heating ($\dot{X}_{heating}$) and the exergy of heat pumping ($\dot{X}_{HP}$) depend on the heat supply that the heat pump can manage. Any heating requirement that is not covered by the heat pump must still be supplied by an external utility ($\dot{Q}_{H,utility}$ in the figure). The exergy of cooling ($\dot{X}_{cooling}$) only depends on the amount of waste heat (i.e. below the pinch point) that is consumed by the heat pump ($\dot{Q}_L$). The remaining cooling requirements ($\dot{Q}_{L,utility}$) are dealt with by an external cooling utility operating at the lowest temperature in the GCC ($T_{GCC,min}$), regardless of the cold source temperature of the heat pump (T_L). The exergy associated with CO₂ compression does not depend on the heat pump. A similar method was followed by Knorr et al. [34] for the minimisation of cost and emissions in the context of milk spray drying.

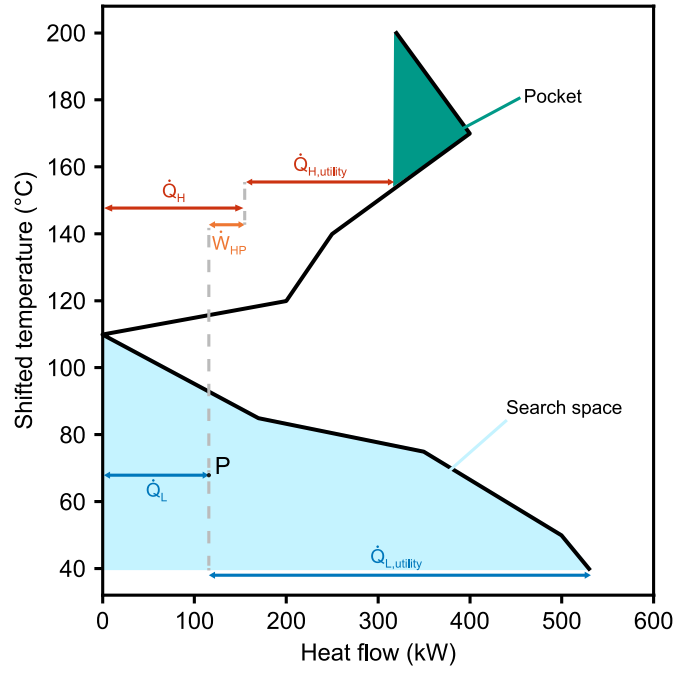


Figure 7. Breakdown of the heat pump integration methodology on a mock Grand Composite Curve (GCC).

The constraints (Equation 6) are set to define the search domain (light blue area in Figure 7). The constraint on the temperature makes sure that heat can only be drawn from under the pinch point. The constraint on the amount of heat is given by the function $GCC(T_L)$ which sets a maximum value of available waste heat for a given temperature level.

The optimisation problem is implemented in Python 3.12 and solved using the Sequential Least Squares Programming (SLSQP) method from the SciPy (v1.15.1) library. Two corrections are applied:

- 1) For any point, if the heat supply to the hot sink is larger than required (i.e. it exceeds the minimum energy requirements identified by pinch analysis), the point is discarded. Any heat pump supplying more heat than required would involve some useless work and stray from the optimum location.
- 2) From the shifted temperatures exhibited by the GCC, one must go back to actual temperatures by considering the chosen temperature approach (ΔT_{min} , defined in Section 3.2):

$$T_{hot,actual} = T_{hot,shifted} + \frac{\Delta T_{min}}{2} \quad (\text{Eq. 7})$$

$$T_{cold,actual} = T_{cold,shifted} - \frac{\Delta T_{min}}{2} \quad (\text{Eq. 8})$$

3.4.4. Vapour-compression heat pump modelling

Vapour-compression heat pumps (VCHPs) are ideally modelled by the reversed Carnot cycle, illustrated in Figure 8. The coefficient of performance (COP), which is used to determine the heat supply based on the waste heat consumption, is calculated as described on the first row of Table 1. In this table, $\dot{Q}$ represents heat flow (kW), T denotes temperature (K) and $\dot{W}$ denotes mechanical power (kW). The subscripts 'H' and 'L' stand respectively for *high* and *low* and refer to the hot and cold reservoirs.

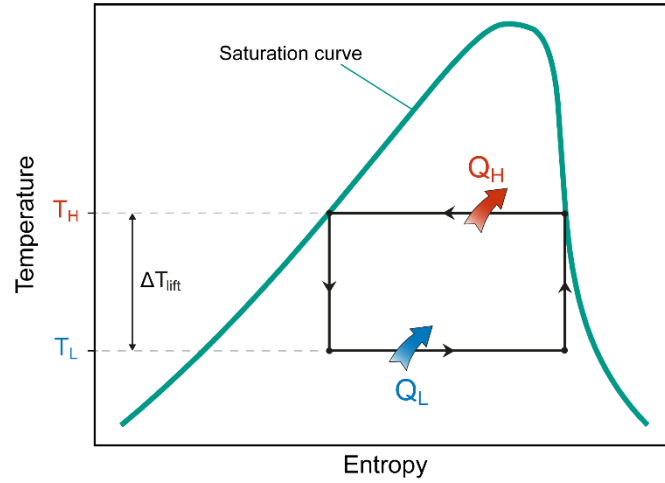


Figure 8. Temperature-entropy (T-s) diagram of the Carnot heat pump.

As a first estimate of the performance of an actual VCHP without considering more advanced topics such as refrigerant selection and cycle optimisation, a realistic COP can be broadly estimated using the second-law efficiency (η_{II}). This parameter is defined as the ratio of the actual COP to the ideal COP:

$$\eta_{II} = \frac{COP}{COP_{ideal}} \quad (\text{Eq. 9})$$

3.4.5. Absorption heat pump modelling

The ideal cycle for AHPs is modelled as a combination of an ideal heat engine with an ideal heat pump [12,36]. In this model, the virtual work generated by the heat engine powers the heat pump.

For type 1 AHPs, as illustrated in Figure 9, the heat engine operates between the generator temperature (T_H) and the intermediate temperature (T_M), while the heat pump operates between the evaporator temperature (T_L) and the intermediate temperature (T_M). For type 2 AHPs, illustrated in Figure 10, the heat engine operates between the intermediate temperature (T_M) and the evaporator temperature (T_L), while the heat pump operates between the intermediate temperature (T_M) and the absorber temperature (T_H).

The corresponding coefficients of performance (COPs) are reported in Table 1. The subscript ‘M’ refers to the *intermediate* temperature level. The ideal COP is derived from the general COP expression using the First and Second Laws of Thermodynamics.

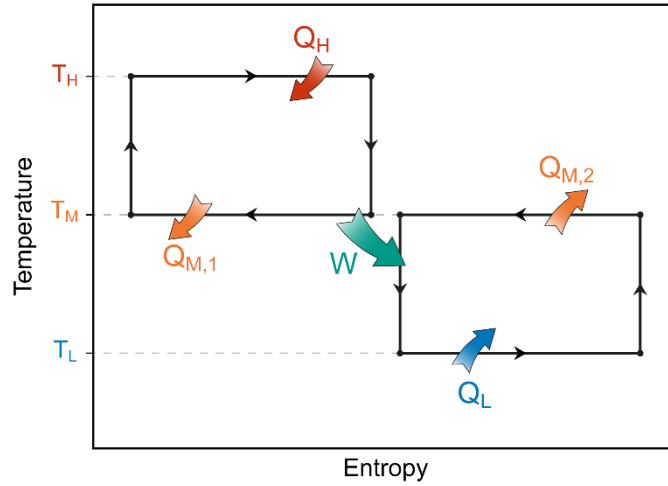


Figure 9. T-s diagram of an ideal type 1 AHP. Adapted from [12].

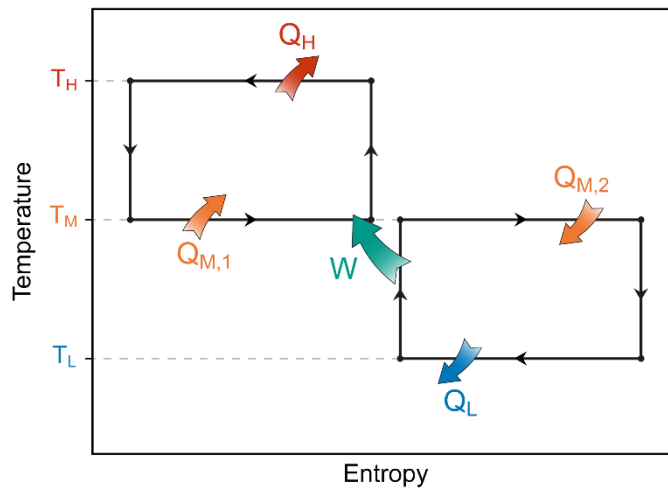


Figure 10. T-s diagram of an ideal type 2 AHP. Adapted from [12].

In literature, absorption heat pumps are described as more expensive, bulkier, less efficient and more complex than VCHPs. They are mainly considered when the unit cost of heat is relatively low compared to electricity [36]. Because these limitations are not related to thermodynamic potential and because some authors have proposed using such systems in CO₂ capture applications, AHPs are still considered in this study.

3.4.6. Influence of the stripper pressure

The operating pressure of the stripper is a crucial parameter for integrating either VCHPs or AHPs. Since the specific heat consumption of the CO₂ capture process decreases with increasing stripper pressure [26,37], stripping is often performed above atmospheric pressure. However, as pressure increases, the reboiler temperature rises accordingly, necessitating heat supply at a higher temperature. Since heat pumping is being considered for reboiler heat supply, it is advantageous to target the lowest possible temperature in order to minimise the temperature lift the heat pump must overcome. The trade-off between higher heat consumption and a smaller temperature lift may favour low-pressure operation, potentially even under vacuum (vacuum-operated stripper, VOS). This was suggested and demonstrated by Jensen et al. [18] and further demonstrated by Beguin et al. [38]. Le Moullec et al. [6] previously discussed the benefits of low-pressure, low-temperature reboilers, though their focus was on the potential use of low-pressure steam rather than reducing the heat pump's temperature lift. To investigate the influence of the stripper pressure, the entire heat pump integration analysis is performed for several, decreasing pressure levels, ranging from 200 kPa (base case) to 50 kPa.

3.4.7. Effect of a side reboiler

Up to this point, heat pump integration has been limited to the main reboiler and therefore constrained to the reboiler temperature. Since CO₂ desorption occurs along the entire stripper height, heat can be supplied at intermediate stages operating at lower temperatures. This motivates the investigation of using a heat pump as a side reboiler in the capture system. First, the side reboiler placement is optimised in the process flowsheet. Then, a heat pump is integrated to supply the side reboiler heat demand.

As explained in Section 3.2.2, for a column to operate at thermodynamic optimum, the ideal heat supply profile requires an infinite amount of side loads, represented by the Column Grand

Composite Curve (CGCC). To limit the scope of this study, only a single side reboiler is introduced. However, the CGCC is used as a basis to optimise its location.

Since exergy is the performance metric used in this work, the optimal side reboiler is identified by exergy minimisation. The CGCC is modified by replacing the temperature (y-axis) with its equivalent Carnot Factor ($1 - \frac{T_0}{T}$), using a reference temperature T_0 equal to 298.15 K. Graphically, heat supplied to a given stage can be represented by a horizontal line at the corresponding temperature (or Carnot Factor). As a result of this variable change, the area under such a line corresponds to the exergy flow associated with the heat supply (see Equation 4). When a side reboiler is added, the total exergy flow associated with heat delivery to the column is represented by the area bounded by the main reboiler line, the side reboiler line, and the x-axis (red, hashed area in Figure 11). By contrast, heat released by the condenser (blue, hashed area in Figure 11) represents exergy made available to the environment and must be subtracted from the previous area. The optimal side reboiler is then determined by minimising the resulting area. An optimum is expected because of the apparent trade-off between supplying heat at high temperature (close to the reboiler) and supplying more heat with higher thermodynamic efficiency.

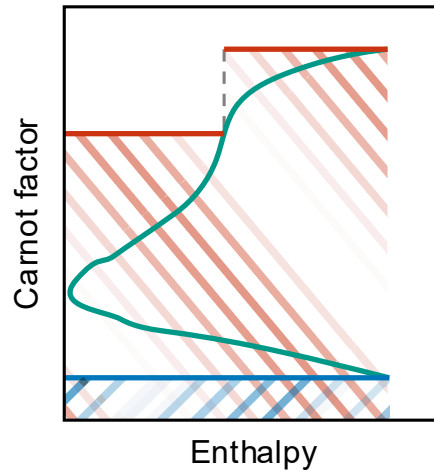


Figure 11. Example of side reboiler optimisation based on the CGCC (green curve). The hashed area represents the exergy requirement of the reboilers and must be minimised.

Because the stripper is modelled with a finite number of equilibrium stages, side reboiler placement is investigated at the corresponding discrete locations, with heat supply defined by the CGCC at each stage temperature. Once the minimal area is identified, the corresponding side reboiler is implemented in the pinch analysis, and a new Grand Composite Curve (GCC) is generated.

From this new GCC, the optimisation procedure introduced in Section 3.4.3 is applied to integrate a heat pump. The algorithm searches for the optimal cold source below the pinch point that satisfies the heat demand of the side reboiler while minimising the total exergy requirements of the process.

Alternatively, the optimal side reboiler could be identified without relying on idealised column models such as the CGCC. For instance, Soares Pinto et al. [39] proposed a column-targeting method based on a minimum driving force derived from simulation results. However, this methodology involves the introduction of additional equilibrium stages, substantially modifying the model as well as the CO₂ capture performances. It has therefore not been applied.

4. Results

4.1. Direct heat integration

The MEA-based capture model is simulated in its base case configuration: no heat pump is integrated, the stripper operates at 200 kPa, and CO₂ is compressed to 2000 kPa. Pinch analysis identifies a pinch point at a shifted temperature of 121°C. The total heating requirement (i.e. the minimum hot utility, above the pinch point) is equal to 6080 kW. The total cooling requirement (i.e. the minimum cold utility, below the pinch point) amounts to 8643 kW. There is no heat integration pocket, meaning that heat recovery by direct heat exchange is not possible. The Grand Composite Curve (GCC) is depicted in Figure 13 (blue). Note the distinct convex shape of the SOC's cooling duty. While this is not shown on the GCC, the CO₂ compression train, which conditions CO₂ for injection into the network, requires 304 kW of mechanical power.

4.1.1. Influence of the CO₂ network pressure

In the selected case study, CO₂ is intended for injection into the future Belgian CO₂ network, which is expected to operate with gaseous CO₂ at pressures ranging from 2000 to 3300 kPa [27]. Depending on this operational pressure, the compression train could present an opportunity for direct heat integration. Indeed, intercoolers are placed between compression stages to improve efficiency and reduce the water content. Higher compression ratios increase the heat of compression, which means that intercoolers may operate from temperatures above the pinch point. Consequently, a higher compression ratio leads to greater potential for direct heat integration. Two main limitations apply:

- 1) It is generally preferable to distribute the compression across multiple compressors, effectively reducing the temperature rise at each step [28].
- 2) The fixed flow rate of captured CO₂, along with the pinch point temperature, constrains the potential for heat recovery. Indeed, valuable heat transfer only occurs in the temperature interval between the initial and pinch temperatures. In other words, the final CO₂ temperature does not affect the direct heat integration potential, although better intercooling can reduce the work required by subsequent compression steps.

In the base case configuration, the outlet temperature of each compressor remains below the pinch point temperature. On the GCC, the cooling duty of the intercoolers is entirely located in the cold region. Higher network pressures could enable direct heat recovery. For the purpose of the analysis, the investigation is extended beyond the planned operational range to explore the boundary condition of compressed gas transportation. CO₂ has a critical pressure of around 7380 kPa [40]. At this outlet pressure, and assuming that the overall pressure ratio is evenly distributed across the three compressors ($\pi_i = 3.33$), a new iteration of pinch analysis identifies a direct heat recovery potential of 122 kW above the pinch point, which represents less than 2% of the total heat demand.

This suggests that direct heat recovery is barely influenced by the CO₂ network pressure. However, some countries operate their CO₂ networks with supercritical or liquefied CO₂ [41]. Depending on the architecture of the compression train, case studies located in such countries could benefit from direct heat recovery from compression intercooling. For the remainder of this paper, the CO₂ outlet pressure is kept at 2000 kPa.

4.1.2. Influence of the stripper pressure

Similarly to the (limited) direct heat integration potential observed at higher CO₂ network pressures, a comparable effect stems from reduced stripper pressure. Indeed, lower stripper pressures require the compression train to achieve a higher compression ratio, resulting in greater heat of compression at the expense of higher compression work.

Stripper pressure is an essential parameter in this heat pump integration study. Lower pressures lead to lower reboiler temperatures and, consequently, lower pinch point temperatures. As a result, any heat pump achieves a smaller temperature lift. Moreover, the valuable temperature interval for internal heat recovery widens accordingly, since it depends on the pinch point temperature.

However, lower stripper pressures also increase the specific heat consumption. These conflicting effects are illustrated in Figure 12.

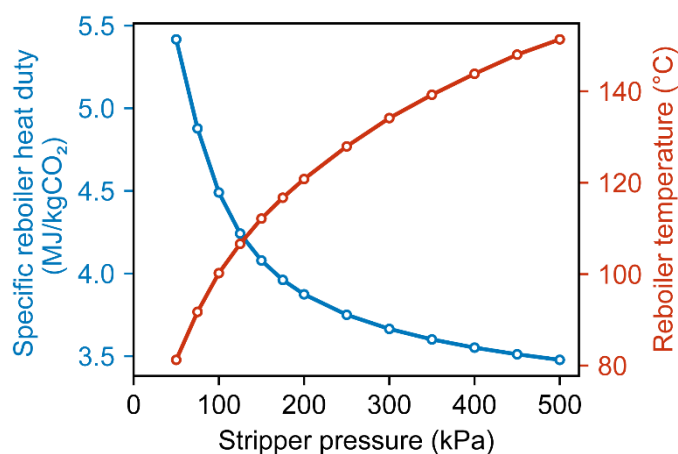


Figure 12. Lower stripper pressures reduce the reboiler temperature (red) but increase the specific heat demand of the process (blue). A trade-off between these effects must be attained.

To evaluate the trade-off, the CO₂ capture system is simulated at various stripper pressures, ranging from the base case of 200 kPa down to 25 kPa. The CO₂ outlet pressure is maintained at 2000 kPa. The resulting Grand Composite Curves (GCCs), shown in Figure 13, reflect the total heating and cooling requirements of the capture process at various pressure levels.

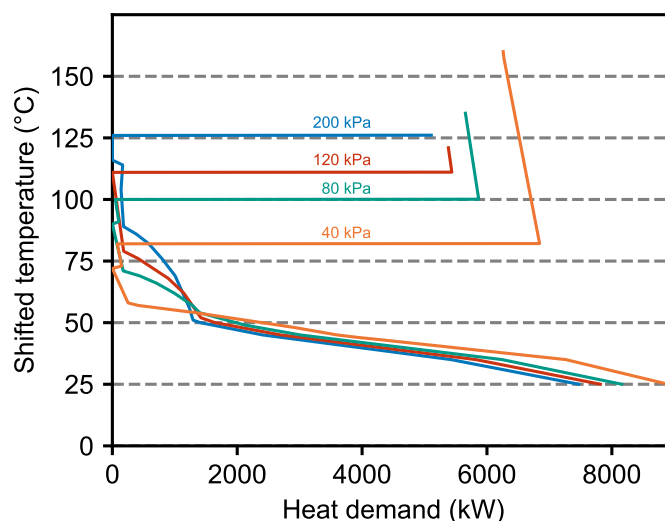


Figure 13. The Grand Composite Curves (GCC) for decreasing stripper pressures show the gradual decrease in reboiler temperature and the increase in specific heat duty.

As mentioned before, the base case (200 kPa) has no direct heat integration pocket. As stripper pressure decreases, the compression ratio for each compressor increases, leading to higher outlet

temperatures. A pocket appears above the pinch point, with an increasingly larger starting temperature due to the fixed number of compression stages. The internal heat integration potential (measured by the projection of the pocket onto the x-axis) increases with decreasing pressure.

However, this gain is offset by the rising reboiler heat duty. The total heating requirement increases from 6080 kW at 200 kPa to 10057 kW at 25 kPa (+65%). At this point, operating at a lower pressure offers no net benefit.

Nonetheless, in the perspective of heat pump integration, two benefits emerge:

- 1) Lower stripper pressures result in reduced pinch point temperatures, as the reboiler operates at a lower boiling point. This decreases the required temperature lift, thereby improving heat pump performance.
- 2) The stripper overhead condenser's (SOC) duty, which is recognisable by its convex shape in the GCC, grows as stripper pressure decreases. As a result, a greater amount of waste heat becomes available for recovery via the heat pump, particularly at temperature levels close to the pinch point.

For these reasons, heat pump integration is studied for several stripper pressures ranging from 200 kPa (base case) down to 50 kPa.

4.2. Heat pump integration

4.2.1. Reference cases

To have an initial comparison basis, the base case process is simulated at the extrema of the studied stripper pressure range (200 and 50 kPa) with no heat pump integration. Heating and cooling requirements are identified from the Grand Composite Curve (GCC), while CO₂ compression power consumption is extracted from simulation results. These values are then converted to exergy requirements and are reported in Table 2. Since heat pumping is expected to reduce total exergy requirements, these reference cases are to be considered as upper bounds.

The heating requirement accounts for the largest share. In practice, this contribution could be even greater, as the calculation assumes that heat is available at the reboiler temperature, which is an

assumption that neglects the exergy destruction from heat transfer with a temperature difference. This assumption is challenged in Section 4.2.2.4.

4.2.2. Vapour-compression heat pumps

4.2.2.1. Ideal cycles

First, vapour-compression heat pumps (VCHPs) are integrated based on ideal models (i.e. with a second-law efficiency of 1) and constraints given by the GCC. The cold region of the GCC is searched for an optimal cold source according to the methodology presented in Section 3.4.

In that context, for a stripper operating at 200 kPa, the optimal VCHP (i.e. the one that minimises the total exergy requirements of the process) is depicted in Figure 14. It draws 4438 kW of heat at a shifted temperature of 40°C and, with a COP of 4.22, supplies 6080 kW to the hot sink (reboiler), thereby fully replacing the conventional reboiler. This outcome is not a consequence of the methodology. The optimiser is free to identify an optimum in which the heat pump only partially replaces the reboiler, with the remainder covered by an external hot utility. In this case, however, heat pumping entirely replaces the external hot utility, as the associated exergy reduction outweighs the exergy cost of heat. It should also be reminded that the exergy cost associated with this external hot utility was previously identified as an underestimation, which further validates this result. More importantly, this also shows that the CO₂ capture process can be entirely electrified by heat pumping under ideal conditions.

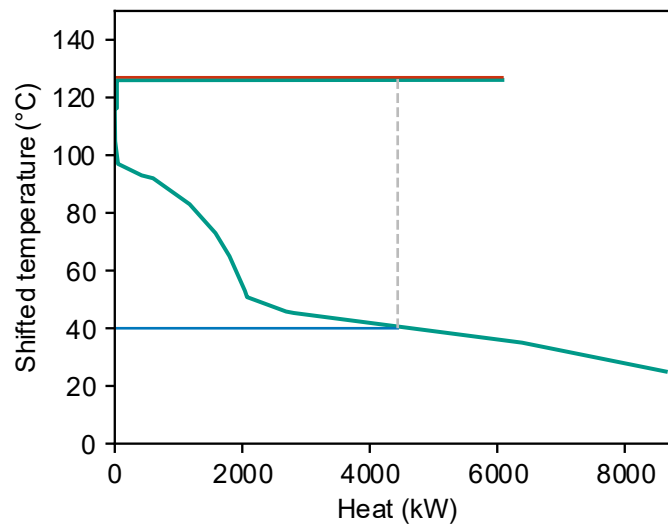


Figure 14. The optimal, ideal VCHP replaces the conventional reboiler. Blue and red lines represent respectively the cold source and the hot sink. $P_{stripper} = 200 \text{ kPa}$.

The total exergy requirement of this optimal VCHP amounts to 1.16 MJ/kgCO₂, which is a 11% reduction compared to the base case at the same stripper pressure. Most of the exergy is used to drive the heat pump (79%) while cooling (4%) and CO₂ compression (17%) complete the picture.

The optimality of this result can be verified by generating a contour plot of the total exergy requirement across the search space. This is done in Figure 15. The black envelope corresponds to the bottom half of the GCC, which is the domain from where heat can be drawn. The colours represent the normalised total exergy requirement of the CO₂ capture process associated with a VCHP. Each point describes a couple formed by the waste heat available to the heat pump and the corresponding cold source temperature. For some data points, too much waste heat is available, and the resulting heat pump overdelivers compared to the actual heat requirements above the pinch point. The subdomain formed by these points is excluded (hashed area).

The figure confirms the location of the optimum at a shifted temperature of 40°C. In addition, the figure shows that, besides the identified minimum, the lowest values are located along the GCC, where the cold source is constrained by heat availability. This implies that, for a given cold source temperature, the total exergy requirements of the process are obtained by integrating an ideal heat pump that extracts all available heat.

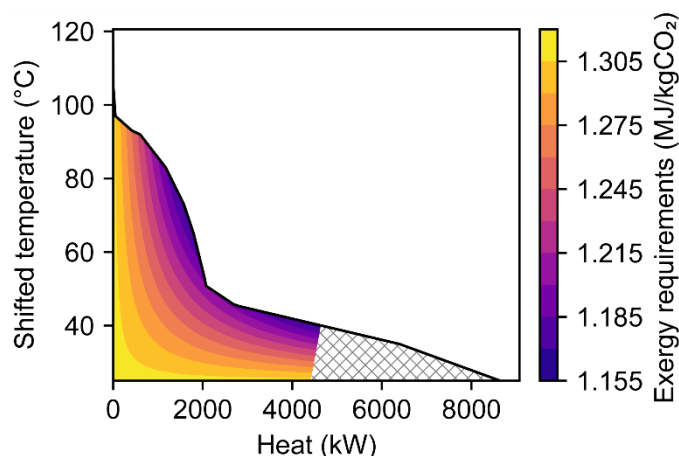


Figure 15. Contour plot of the exergy requirements across the entire search space for ideal VCHP integration.

Interestingly, the identified optimum corresponds to the highest temperature at which sufficient waste heat is available for the heat pump to supply the entirety of the heat demand. At lower temperatures, waste heat is also sufficient but supplying the same amount of heat with a greater temperature lift results in a larger exergy requirement. The boundary between the excluded points

(too much heat produced) and the accepted points is slightly tilted. For a given hot sink temperature, an identical amount of heat from the cold source gives an increasingly larger heat supply as the cold source temperature decreases. Indeed, the work that is required increases with decreasing cold source temperatures due to the decreasing performance of the heat pump.

It is also important to notice that local minima exist at upper temperature levels, between 60°C and 80°C (shifted). If the optimisation is constrained to this region, the optimum reaches an exergy requirement of 1.17 MJ/kgCO₂. This is very close to the global minimum identified earlier, making the optimisation very sensitive. This also indicates the potential of a smaller capacity heat pump, which would bring comparable benefits at a lower cost. However, a second heat source would then be required, at the expense of additional capital.

4.2.2.2. *Influence of the stripper pressure*

The optimisation is replicated for several stripper pressures that range from the initial 200 kPa down to 50 kPa. Figure 16 shows the total exergy requirements for all investigated pressures.

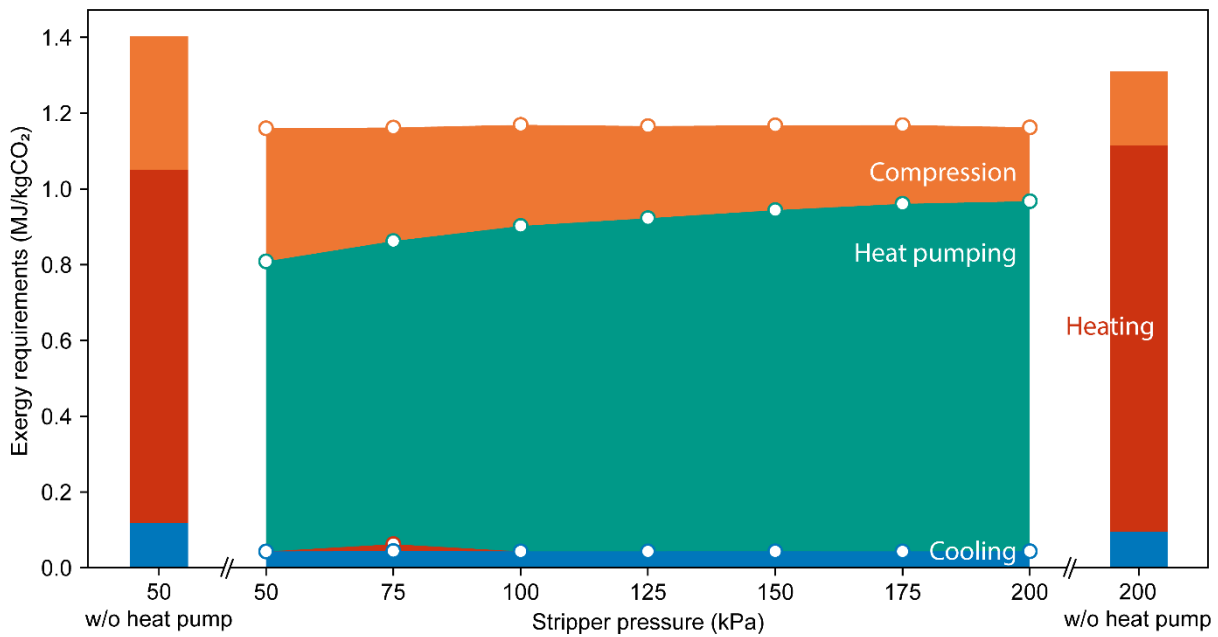


Figure 16. Total exergy requirements are reduced with an ideal VCHP at any pressure level. Reducing the stripper pressure does not further reduce the requirements because of the increase in compression exergy.

For all stripper pressures, VCHP integration reduces the exergy requirement of the process by at least 11% compared to the base case. Moreover, most optimal configurations involve reboiler replacement, meaning that the heat pump entirely substitutes external, thermal energy use with

heat pumping. The only exception appears at 75 kPa, where the minimum still involves a heating utility. This highlights the sensitivity of the optimisation, which was previously identified. Full reboiler replacement is still achievable, but at a slightly higher exergy requirement.

As expected, the improvement in heat pump performance at lower stripper pressures, resulting from a reduced temperature lift, outweighs the increase in specific reboiler heat duty. As also expected, the power consumption associated with CO₂ compression increases accordingly. However, the novel finding is the near-complete balance between these two effects. In other words, under ideal VCHP conditions, there is little benefit in operating the stripper at low pressures.

This highlights the relevance of accounting for CO₂ compression. In a previous work from the authors [38], it was reported that operating a CO₂ capture pilot at low stripper pressures is beneficial, but this was because CO₂ compression was not included in the pilot plant configuration.

However, these results alone are not sufficient to validate or challenge the claim that vacuum-operated strippers should be preferred in real-world applications [18]. The current result shows that thermodynamic limitations remain almost identical across the studied range of stripper pressures in the case of ideal VCHP integration. However, actual VCHP integration may yield a different balance as studied in the next section.

4.2.2.3. *Realistic cycles*

The temperature lift of the optimal, ideal VCHP is significant ($\Delta T_{lift} = 91$ K). While some actual, high-temperature heat pumps are capable of such lifts, their COPs typically fall in the range of 2 – 2.5 [7–9]. The second-law efficiency (defined by Equation 9 in Section 3.4.4) is used to calculate a more realistic COP and therefore expand the study to actual systems. In this paper, the second-law efficiency of a VCHP is assumed to be equal to 50%. This value corresponds to a common techno-economic optimum for high-temperature heat pumps [7–9].

With this adjusted COP, the methodology applied for ideal VCHPs is replicated. An optimal, realistic VCHP is searched for from all possible cold sources in the cold region of the GCC.

When the stripper operates at 200 kPa, the optimisation identifies an optimal VCHP that draws 692 kW at a shifted temperature of 91°C. It supplies 892 kW to the reboiler with a COP of 4.46. An external hot utility is therefore required for a large share of the reboiler duty (85%). The

corresponding exergy requirement amounts to 1.28 MJ/kgCO₂. Compared to the base case, exergy savings drop from 11% (ideal VCHP) down to 2% (realistic VCHP). This is quite logical since only a small fraction of the heat demand is covered by the heat pump, the remainder being still supplied by the external hot utility.

Figure 17 shows total exergy requirement across the search space for this realistic VCHP. There is a stark contrast between ideal VCHPs (Figure 15) and realistic VCHPs (Figure 17). In the latter case, the lowest exergy requirements are found for heat pumps with a limited lift. At the corresponding temperatures, waste heat availability is poor, and the fraction of heat supply that is covered by the VCHP is consequentially low. In addition, the lowest exergy values are also found for systems without heat pumps (along the vertical axis). This indicates that the competition between heat pumping and external utility heating is in favour of the external utility. Moreover, in the contour plot for realistic VCHPs (Figure 17), the domain of excluded points is larger since more work is required for any temperature lift. Therefore, the heat supply from a given waste heat temperature is dominated by the contribution of work and the cold source consumption is smaller.

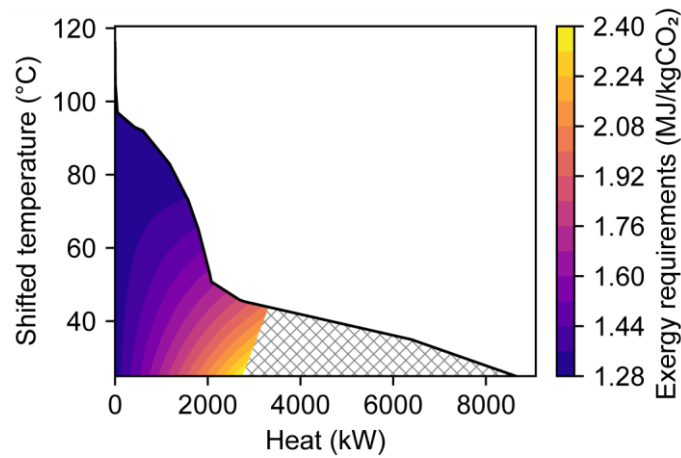


Figure 17. Contour plot of the exergy requirements across the entire search space for realistic VCHP integration ($\eta_{II} = 0.5$).

It must be noted that full electrification (i.e. a heat pump that fully replaces the reboiler) is still achievable, but this no longer constitutes the optimum in terms of exergy requirements.

The influence of the stripper pressure is investigated by repeating the optimisation at each pressure level. The results are shown in Figure 18. The large share of exergy associated with the hot utility

(red) indicates that the optimiser selects partial replacement across the entire range. As a result, the exergy savings vary from 2% at 200 kPa to less than 1% at 50 kPa.

As the stripper pressure decreases, exergy requirements associated with CO₂ compression grow. On a per-unit-of-heat basis, the exergy of heat pumping decreases thanks to the lower temperature lift. However, the relative share of the heating demand covered by the heat pump does not really grow because of the competition with the exergy of heating, which becomes cheaper (exergy-wise) since the reboiler temperature decreases. Since heat pumping does not become gradually a larger contributor to the heat supply, the effect of increasing specific reboiler duty (see Figure 12) degrades the performance of the system at lower stripper pressures.

With ideal VCHPs, the total exergy requirements were nearly constant across the studied pressure range. However, with realistic VCHPs, operating at lower pressures becomes less advantageous.

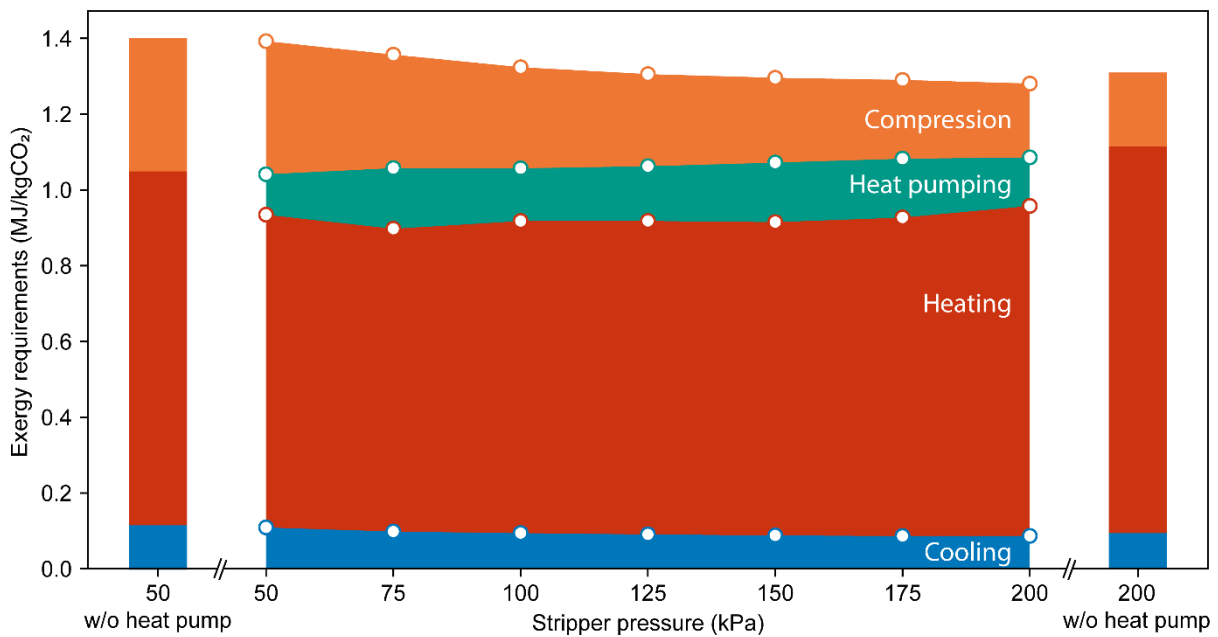


Figure 18. With a more realistic VCHP model, heat pumping becomes less predominant in the optimal exergy distribution. Operating at lower pressures also becomes less interesting.

4.2.2.4. Sensitivity of the exergy cost of heat

The previous conclusions must be nuanced by noting that heat was always assumed available at the reboiler temperature. Consequently, the exergy cost of heat decreased with a lower reboiler temperature. In reality, however, the external hot utility is unlikely to be available at the consumption temperature, but rather sourced from another process operating at a fixed

temperature. For instance, it can be steam withdrawn from a turbine or steam generated from a central boiler. In some cases, particularly when the capture system is integrated from the design stage, steam withdrawal can be arranged so that its temperature matches the reboiler operating point. In many applications, however, and especially in retrofits, the pressure level of available steam is not adjustable and must be accepted as a constraint.

To optimise the placement of a heat pump in that context, the optimisation is repeated for various hot utility temperature levels. For consistency with realistic assumptions, the stripper is operated at 200 kPa and a heat pump second-law efficiency of 50% is applied.

Figure 19 shows the increase in total exergy requirements for the entire capture system with an increasing hot utility temperature. The blue curve shows the gradual increase of the baseline exergy requirements, i.e. before heat pump integration. In this case, the exergy costs of cooling and compression remain unchanged, while the higher hot utility temperature increases the exergy cost of heat according to Equation 4. The red curve corresponds to the total exergy requirements with an optimally integrated, realistic VCHP. When the hot utility temperature matches the reboiler temperature (leftmost point), the results from the previous section indicate that the heat pump covers only a small part of the reboiler demand. As the hot utility temperature increases, total exergy requirements rise, but the heat pump progressively supplies a larger share of the demand, as shown by the growing gap between the two curves. At a certain point, the red curve reaches a plateau, indicating that the heat pump covers the entire reboiler duty. Beyond this point, a further increase in the exergy cost of heat does not affect the total exergy requirements, since the hot utility is no longer utilised.

The figure also shows the temperatures of the medium- and high-pressure steam relevant to the case study. If steam is extracted at medium pressure, the exergy requirements fall from 1.85 MJ/kgCO₂ without a heat pump to 1.70 MJ/kgCO₂ (-8%) with an optimal, realistic VCHP supplying 30% of the reboiler demand. By contrast, if steam is extracted from the main boiler, the heat pump fully replaces the reboiler, reducing the total exergy requirements from 2.51 MJ/kgCO₂ down to 2.03 MJ/kgCO₂ (-19%). This outcome implies that supplying additional steam by burning extra fuel would lead to a configuration in which the optimal heat pump entirely displaces the reboiler due to the high exergy value of this extra fuel.

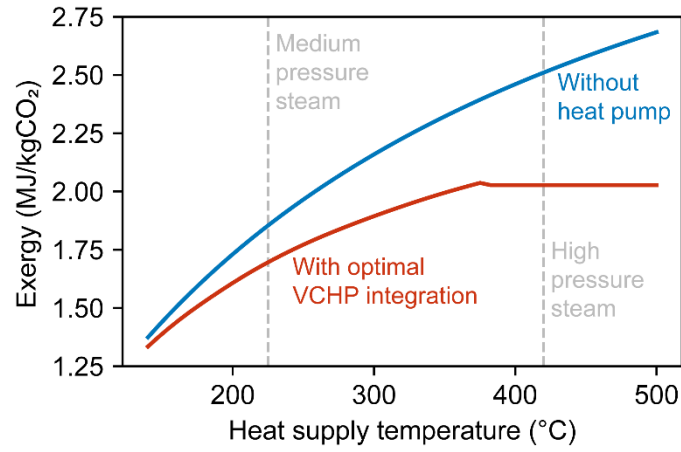


Figure 19. As the hot utility temperature increases, the optimisation attributes a larger fraction of heat supply to the heat pump. Mechanical work becomes progressively more interesting than high-temperature heat.

To better understand the emergence of the plateau, it is useful to examine the evolution of the optimal cold source temperature as the hot utility temperature increases (Figure 20).

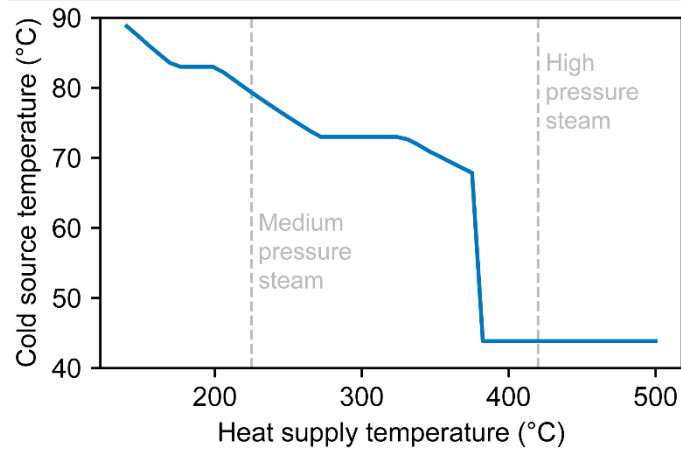


Figure 20. As the exergy cost of heat increases, the optimiser selects a gradually decreasing optimal cold source temperature. At a given point, the optimiser jumps to the second zone.

As the hot utility temperature rises, the heat pump supplies a larger share of the reboiler duty, and the cold source is gradually chosen at lower temperatures to access sufficient waste heat. At the point corresponding to the plateau, the optimal cold source temperature shifts from the upper region (around 70°C) of the search space towards the lower region (around 40°C). This shift reflects the presence of two distinct local minima in the optimisation landscape (see Section 4.2.2.1). Once the exergy cost of heat becomes sufficiently high, the second minimum offers a more favourable solution.

For the remainder of the paper, the hot utility temperature is fixed at 225°C, which corresponds to the medium-pressure steam in the case study.

4.2.2.5. *Effect of a side reboiler*

To investigate the potential of using a heat pump as a side reboiler, the optimal side reboiler is first identified based on the Column Grand Composite Curve (CGCC, Figure 21), which is obtained from simulation results. Side reboiler placement is independently studied for each of the 7 stages (stage No. 8 represents the main reboiler). For each of them, the side load suggested by the CGCC of the base process is implemented, while the remainder of the heating demand is supplied at the main reboiler. Using the Carnot factor, the column-specific exergy requirements are computed by adding the exergy requirements of both the current side and main reboilers and by subtracting the exergy released in the stripper overhead condenser (SOC, labelled as stage No. 0). The side reboiler is finally placed on the stage where the column-specific exergy requirements are minimal.

The CGCC for the stripper operating at 200 kPa is depicted in Figure 21. The optimal side reboiler and the remaining heat supply at the main reboiler are represented in the same figure.

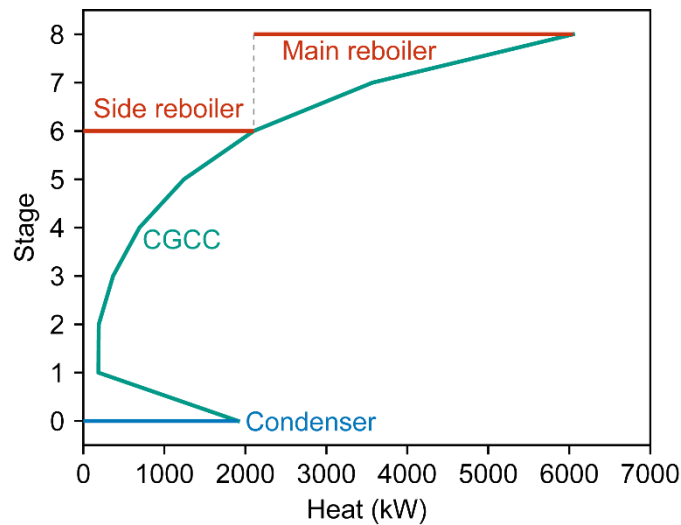


Figure 21. Column Grand Composite Curve (CGCC) of the base case stripper ($P = 200$ kPa). The eight equilibrium stages operate at temperatures ranging from 121°C (reboiler) to 102°C (topmost stage).

This optimal side reboiler supplies 2105 kW to the column, leaving the remaining 3942 kW to be supplied by the main reboiler. This distribution of heat supply across two temperature levels is reflected in the Grand Composite Curve (GCC) generated from these results (Figure 22), where the multi-temperature heat supply is clearly visible.

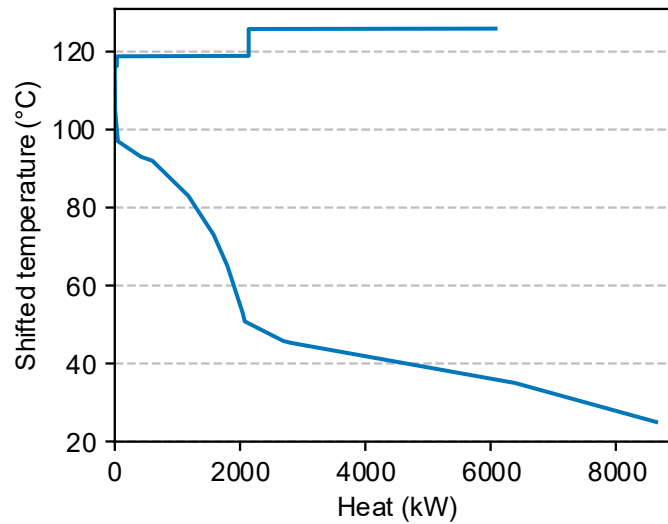


Figure 22. Grand Composite Curve of the capture process with the integration of the optimal side reboiler. The heat pump integration aims at fulfilling this secondary load.

From this GCC, a realistic VCHP is then integrated based on the optimisation procedure introduced earlier. Instead of supplying an unconstrained amount of heat to the main reboiler, the heat pump only deals with the side reboiler demand. In practice, this means identifying the highest cold source temperature at which there is sufficient waste heat to supply the entirety of the side reboiler duty.

At 200 kPa, the corresponding VCHP draws 1545 kW of heat at a shifted temperature of 74°C and, with a COP of 3.61, meeting the side reboiler duty. The corresponding total exergy requirement of the entire system is 1.67 MJ/kgCO₂. This represents a 10% reduction compared with the base process. Figure 23 shows the breakdown of this total exergy requirement when it is assumed that the heat pump fully addresses the side reboiler, as well as the results for other investigated stripper pressures.

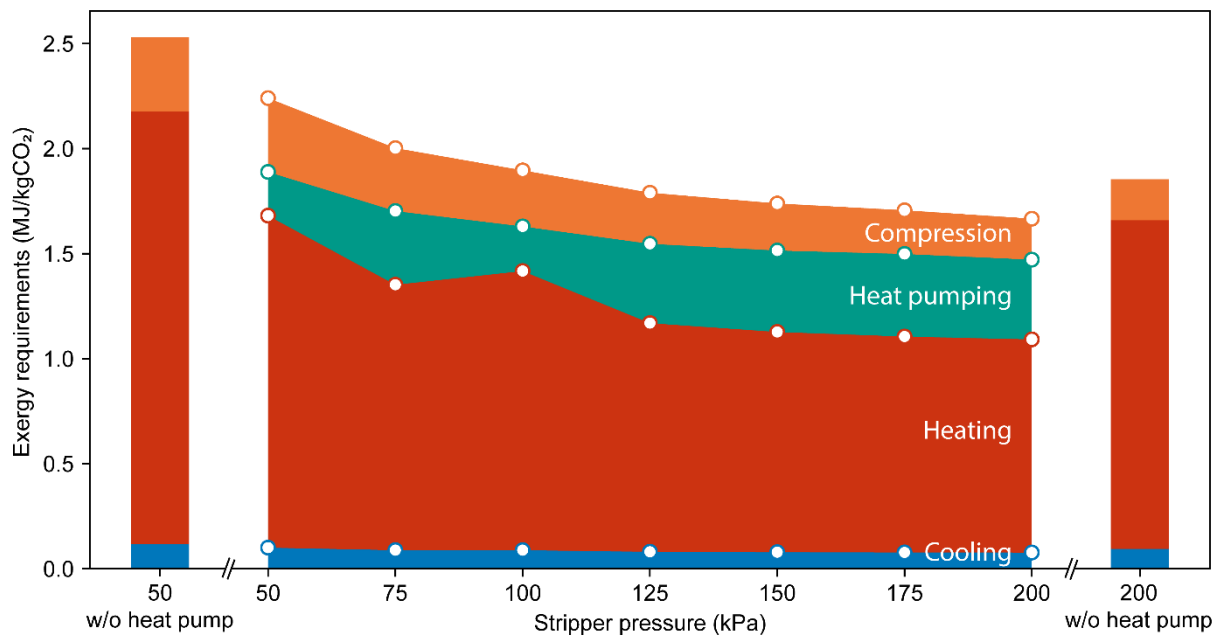


Figure 23. As the realistic VCHP is only integrated as a side reboiler, a hot utility is still required. Operating the stripper at lower pressures becomes even less interesting.

Note that side reboiler placement is performed independently at each stripper pressure. Consequently, the definition of the heat pump hot sink varies from one pressure level to another. This explains the bump that appears at 100 kPa and it highlights the sensitivity of the side reboiler placement: because stages operate at similar temperatures, placing the side reboiler at different stages yields very similar total exergy values. However, it can noticeably shift the distribution of exergy supply between heat pumping and conventional heat while maintaining a similar total exergy requirement.

Overall, achievable exergy savings reach approximately 10% across the entire range. Using a VCHP as a side reboiler alone results in slightly higher savings (10%) compared with VCHPs integrated in the main reboiler (9%), as reported in Section 4.2.2.4. Exergy performance at low pressures, however, is more sensitive. Indeed, there is a gradual increase in total exergy requirements as pressure decreases. Since the heat pump's share of the total heat supply is limited by the capacity of the optimal side reboiler, there is only a constrained opportunity to offset the increasing specific reboiler heat duty.

4.2.3. Absorption heat pumps

4.2.3.1. Assumptions

Since vapour-compression heat pumps are powered by an external source of mechanical work (usually an electric drive), they are easily integrated into any process. By contrast, absorption heat pumps (AHPs) are thermally driven and heavily depend on the availability of thermal energy. The key assumption for AHP integration is therefore the availability of driving heat.

Type 1 AHPs require high-grade heat to drive the heat extraction from the cold source. They are well suited for processes with a significant heat demand above the pinch point (Figure 24a), which is the case for the amine-based capture system (Figure 13). To remain consistent with the additional assumptions from the previous sections, it is assumed that heat is available as medium-pressure steam at 225°C. This steam can either drive the AHP or serve as a hot utility for the reboiler if the AHP does not supply the entire reboiler duty.

Because this realistic assumption is made regarding driving heat availability, it is preferable to avoid using ideal models to perform the AHP integration analysis as the results would be a mix of both. Therefore, the coefficients of performance (COP) introduced in Table 1 are also adjusted with a second-law efficiency, as it was done for realistic VCHPs. An average value of 40% is selected based on a review of modelling [36,42] and experimental studies [43–45].

Type 2 AHPs use heat at an intermediate temperature. They are integrated into a GCC by drawing a large amount of heat below the pinch point to meet a smaller demand above it (Figure 24b). No assumption is required regarding the availability of external heat. However, the temperature level at which waste heat is rejected by the type 2 AHP must be selected. This temperature is fixed at 20°C, matching the cold utility temperature. Moreover, for consistency, the integration is performed for realistic cycles, with a second-law efficiency of 70%. Again, this value is selected based on modelling [42] and experimental studies [13,46].

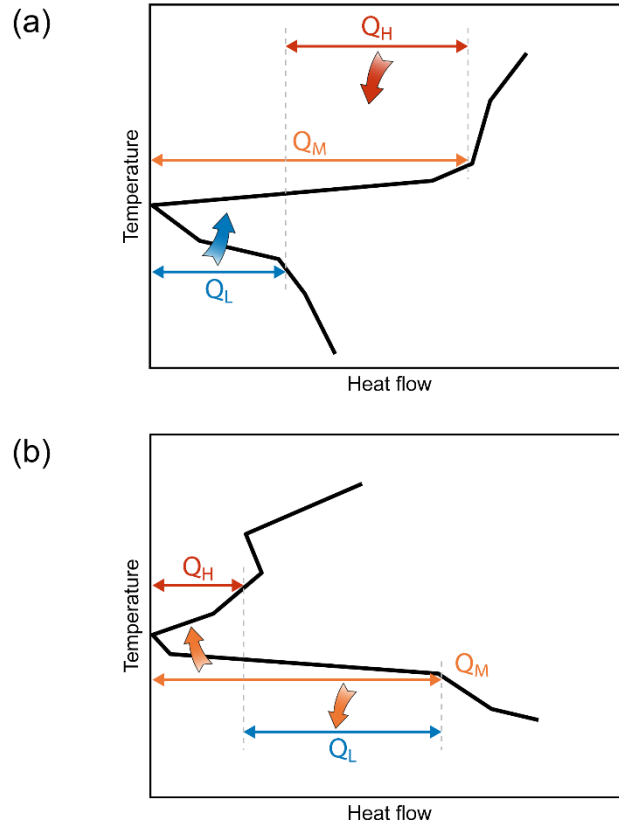


Figure 24. Typical integration of a type 1 AHP (a) and a type 2 AHP (b). The GCCs are generated for illustrative purposes and do not correspond to the studied system.

4.2.3.2. Type 1 absorption heat pumps

The optimisation procedure applied for VCHPs is repeated with the AHP model. In this case, the bottom half of the GCC is searched for the pair $(T_L, \dot{Q}_L)$ (see Figure 24) that minimises the total exergy requirements of the process. For a stripper operating at 200 kPa, the contour plot that results from the integration procedure is shown in Figure 25.

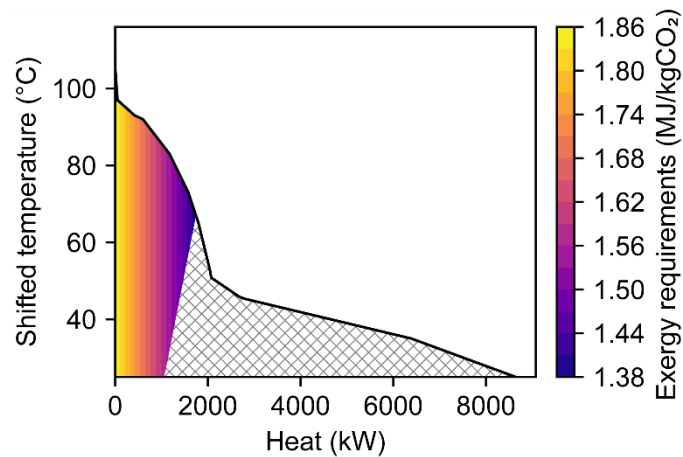


Figure 25. The contour plot of the type 1 AHP integration no longer follows a hyperbolic shape, but a vertical shape instead.

Compared with VCHP integration, more points are excluded (hashed area in Figure 25). For a given heat demand at the reboiler, a larger share of it is provided by the driving heat in an AHP than by work in a VCHP. Therefore, less waste heat is required. Consequently, in the search space, more cold sources deliver excess heat with AHPs, which leads to more points being excluded in the contour plot.

A striking difference is the change in contour shape. For VCHPs, the contours were hyperbolic, whereas with type 1 AHPs, they are vertical. This can be explained by the competing uses of heat at 225°C: it can either be used to drive an AHP or directly as heat supply to the reboiler. The former use is more efficient since waste heat also contributes to the energy balance, effectively reducing the amount of high-temperature heat that is required. In other words, it is always more favourable to use the driving heat to power an AHP. For a given cold source temperature, the optimiser therefore selects the largest possible amount of waste heat, leading to vertical contours. The overall optimum is found where waste heat is available in the largest quantities (besides the excluded zone), corresponding to the minimum shown in Figure 25.

At this minimum, the total exergy requirement is 1.39 MJ/kgCO₂, representing a 25% reduction compared with the base case (1.85 MJ/kgCO₂). As expected, the entire reboiler duty is covered by the AHP.

As with VCHPs, the influence of stripper pressure on AHP performance is also investigated. The methodology is applied for pressures ranging from 200 kPa down to 50 kPa, with results shown in Figure 26. Here, ‘heat pumping’ corresponds to the exergy associated with the high-grade heat driving the generator. Exergy savings range from 25% (200 kPa) to 39% (at 50 kPa). However, as with VCHPs, exergy savings associated with the reduced temperature lift at lower stripper pressures are balanced by the increase in compression power and in the specific reboiler heat duty. Thus, operating at low pressures is not advantageous.

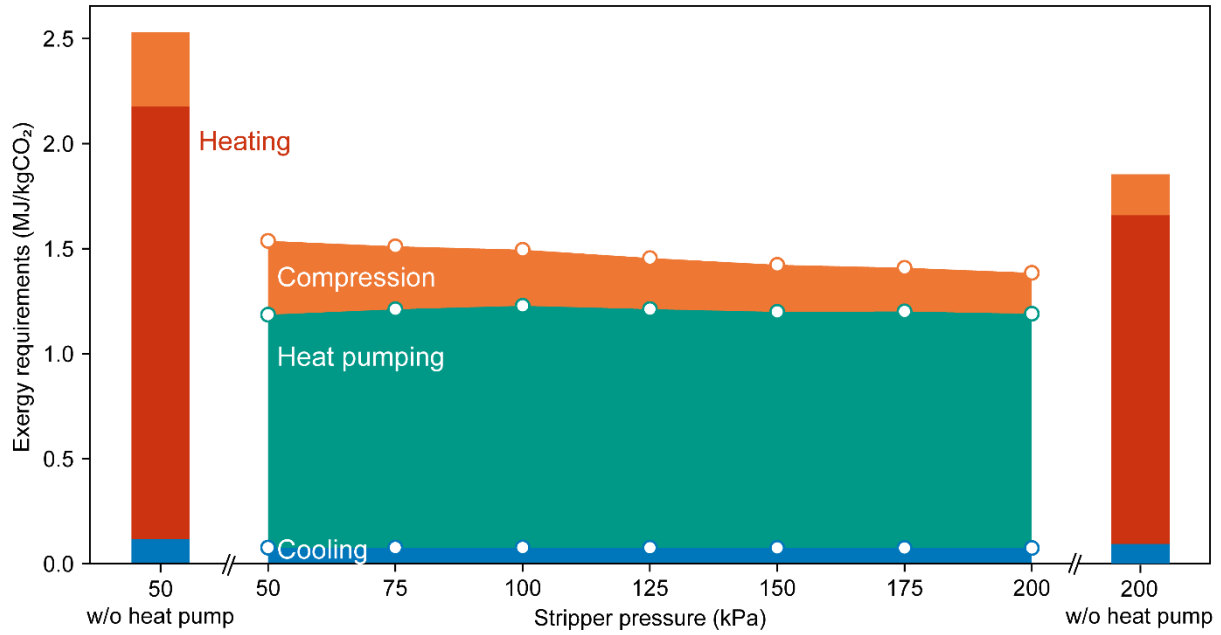


Figure 26. While the exergy savings associated with type 1 AHPs are significant, operating at lower pressures is still not advantageous.

4.2.3.3. Type 2 absorption heat pumps

The optimisation procedure is replicated for type 2 AHPs. In this case, the bottom half of the GCC is searched for the pair $(T_M, \dot{Q}_M)$ (see Figure 24) that minimises the total exergy requirements of the process.

The optimisation yields an optimal type 2 AHP with a waste heat source at 36°C (shifted)0D. The corresponding minimum total exergy requirement is 1.64 MJ/kgCO₂, which corresponds to savings of 11%.

Among all studied heat pump types, this is the lowest cold source temperature. This is explained by the nature of type 2 AHPs: they must extract much more heat at the intermediate temperature than they deliver above the pinch point. When the trade-off between external utility and heat pumping favours heat pumping, the intermediate source must be as large as possible. The optimiser therefore selects a source with abundant waste heat, as confirmed by the contour plot in Figure 27.

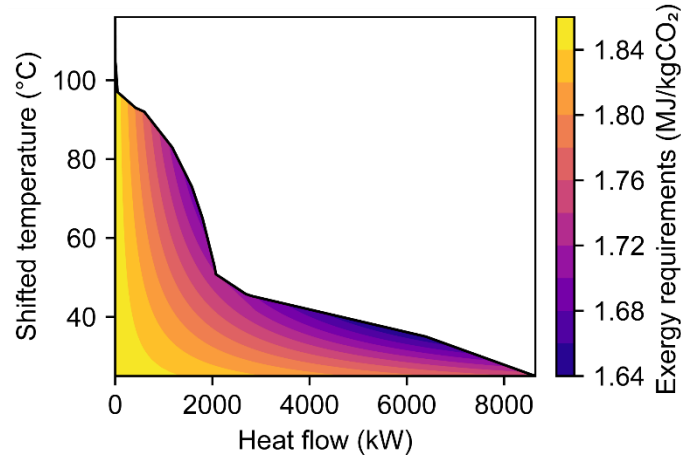


Figure 27. The optimal type 2 AHP placement selects a source with significant waste heat availability, since only part of this heat can be converted to useful heat supply above the pinch point.

At the optimal temperature, the AHP consumes all available waste heat (5961 kW), but most of this is rejected as cooling demand (5162 kW). Only about 13% is effectively upgraded to supply heat to the process.

Given the geometry of the GCC, type 2 AHPs are not sufficient to fully replace the reboiler in this CO₂ capture application. This is also noticeable in the contour plot, where no heat source is excluded for overdelivering.

The influence of stripper pressure is again investigated. Figure 28 shows the total exergy requirements for the investigated stripper pressures. Exergy savings range between 11% and 18%. Similarly to the initial stripper pressure ($P_{stripper} = 200$ kPa), none of the optimal type 2 AHP configurations can replace the reboiler. Furthermore, consistent with other heat pump types, operating at low pressures or under vacuum increases overall exergy requirement.

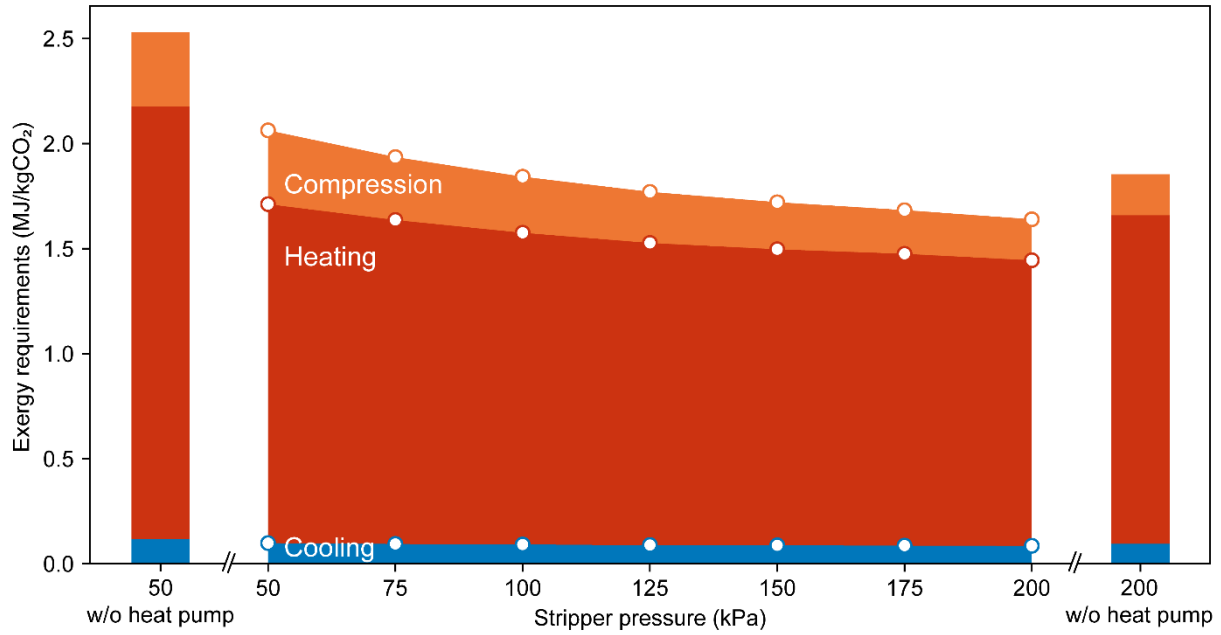


Figure 28. A significant contribution of the exergy cost of heat remains for type 2 AHPs. Operating at low stripper pressure is still not advantageous.

5. Conclusion

Leaving aside the results for ideal heat pumps and assuming that medium-pressure steam (550 kPa/225°C) is used as the hot utility, the different types of closed-loop heat pumps are compared in Table 3.

As expected, all heat pump integrations help reduce the total exergy requirements. Even the least effective case achieves an 8% reduction. Integrating heat pumps into the process therefore consistently lowers the exergy requirement of the entire CO₂ capture process while maintaining an identical capture rate.

Under realistic assumptions, the optimal heat pump placement leads to hybrid integration scenarios in which external heat is still required. The optimal vapour-compression heat pump (VCHP), whether integrated into the main reboiler or operated as a side reboiler, is implemented as a small-capacity heat pump supplying approximately 30% and 35% of the thermal energy demand, respectively. The remaining demand is provided by medium-pressure steam. While the optimal type 1 absorption heat pump (AHP) is able to cover the entire reboiler heat duty, it still requires high-grade driving heat. The optimal type 2 AHP only supplies only about 13% of the thermal energy

demand. This is a consequence of its nature, as only a fraction of the consumed waste heat is upgraded.

The influence of the exergy cost of heat is significant, with optimal integrations shifting towards full replacement when high-pressure steam is used instead of medium-pressure steam. Accounting for the true exergy cost of fuel would therefore favour complete substitution of hot utilities by heat pumps.

Across all closed-loop heat pump options, type 1 AHPs exhibit the largest potential, since their reported exergy savings exceed those of VCHPs and type 2 AHPs. Consequently, implementing type 1 AHPs in CO₂ capture systems appears to be the most promising option. However, detailed modelling of real type 1 AHPs is still required to validate this conclusion in practice.

While economic aspects were not addressed in this work, exergy can serve as a proxy for operational expenditures (OPEX). The reported reductions may therefore translate, at least partially, into OPEX savings. However, the integration of heat pumps is likely to remain constrained by the significant capital expenditures (CAPEX) associated with their installation. Comprehensive economic assessments are therefore required to evaluate their practical feasibility.

Using a VCHP as a side reboiler appears equally effective as using it for the main reboiler. Combining both configurations may further improve the performance. Indeed, in this study, heat pump integration was limited to single-stage machines. Heat pump performance could be increased with advanced cycle designs, such as multi-stage architectures. For example, multiple evaporators could distribute the thermal load across different side reboilers, while multi-stage machines could exploit multiple cold sources at different temperatures. An advanced optimisation procedure should therefore be applied to identify the optimal combination of system architecture and refrigerant to maximise process performance.

In the present study, a specific case study was required to obtain realistic heat loads. Although the results were normalised, local operating conditions still influence the calculated values. For instance, the flue gas temperature influences the heat rejected in the direct contact cooler (DCC), which contributes to waste heat availability for heat pump integration. In this case, this amounts to 2194 kW below the pinch point. This contribution is not critical for VCHPs and type 1 AHPs, since the selected cold source often exclude heat streams from the DCC. By contrast, type 2 AHPs

take full advantage of this additional heat, making their performance more case specific. Nevertheless, the proposed methodology can be applied to other case studies to calibrate expectations prior to detailed engineering.

More importantly, the CO₂ concentration in the treated flue gases is a key parameter. In this study, a concentration of 9.44 vol% was considered (biomass combustion). Different flue gas compositions would result in different energy demands, and heat pump integration would need to be reassessed.

Throughout the paper, the sensitivity of the optimal solutions was highlighted. Contour plots, which were introduced to map the exergy savings associated with any cold source, showed that local minima exist at very close values. In practice, techno-economic optima may shift towards these currently suboptimal solutions. Additionally, selecting lower-capacity heat pumps may reduce equipment costs associated with their integration.

The influence of stripper pressure was analysed for all heat pump types. In each case, the reduction in temperature lift was almost entirely offset by increased work consumption in the CO₂ compression train and by higher specific reboiler heat duty. In contrast to techno-economic optima, such as those identified by Jensen et al. [18], exergy-based optima are largely insensitive to stripper pressure. Furthermore, for VCHPs integrated as side reboilers and for type 2 AHPs, low-pressure operation is largely suboptimal, approaching the performance of a heat-pump-free process at the lowest pressures.

Limited heat availability at the highest temperatures below the pinch point is the main barrier to efficient heat pump integration. In this region, most heat is rejected from the stripper overhead condenser (SOC), whose duty is significantly smaller than that of the reboiler. In distillation applications, where columns also have a rectifying section, reboiler and condenser loads are more balanced. This explains the larger potential for heat-pump-assisted distillation, as reported in the literature, compared with the energy integration of solvent-based CO₂ capture.

Although amine degradation was not explicitly included in the model, thermal degradation is known to increase with temperature [26,47]. Operating at lower stripper pressures could therefore mitigate solvent degradation, reducing the need for solvent reclaiming and the associated costs.

CO₂ capture units may also include other types of mechanical compression systems, such as lean vapour compression or overhead vapour compression, to improve energy efficiency. Several examples are described in [6]. Because such compressors act directly on process streams without the use of a refrigerant, they can be described as “open-loop heat pumps”. A comparison between open-loop and closed-loop heat pumps would be valuable to determine which integration strategy delivers the greatest reduction in exergy requirements.

Finally, accounting for the characteristics of the host process is essential whenever possible. Extending the scope of pinch analysis to include the full heat-exchange network of the host facility can uncover additional synergies and opportunities for heat recovery. In particular, new cold sources may become available, effectively increasing the potential for heat pump integration and thereby improving the overall performance of the capture process.

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Nomenclature

Variables

$\dot{Q}$	Heat transfer rate (kW)
$\dot{W}$	Mechanical power (kW)
T	Temperature (K or °C)
ΔT_{min}	Minimum approach temperature in heat exchangers (K)
$\dot{X}$	Exergy transfer rate (kW)
ΔT_{lift}	Temperature lift (K)
π	Compression ratio
η	Efficiency

Subscripts

H	Hot
C	Cold
in	Consumed by the device
M	Intermediate
P	Pumping
0	Reference or environment
prod	Produced
cons	Consumed
tot	Total
min	Minimum
i	Individual, for each compressor
II	Second Law
rev	Reversible

Acronyms

HTHP	High-temperature heat pump
VCHP	Vapour-compression heat pump

AHP	Absorption heat pump
COP	Coefficient of performance
SOC	Stripper overhead condenser
VOS	Vacuum-operated stripper
AHT	Absorption heat transformer (type 2 absorption heat pump)
DHN	District heating network
HP	Heat pump
GCC	Grand composite curve
CGCC	Column grand composite curve
MTC	Minimum Thermodynamic Condition
LRHX	Lean-rich heat exchanger
DCC	Direct contact cooler
CC	Composite curve

Tables

Heat pump type	COP definition	Ideal COP
VCHP	$COP = \frac{\dot{Q}_H}{\dot{W}}$	$COP = \frac{T_H}{T_H - T_L}$
Type 1 AHP	$COP = \frac{\dot{Q}_M}{\dot{Q}_H}$	$COP = \left(1 - \frac{T_L}{T_H}\right) \frac{T_M}{T_M - T_L}$
Type 2 AHP	$COP = \frac{\dot{Q}_H}{\dot{Q}_M}$	$COP = \left(1 - \frac{T_L}{T_M}\right) \frac{T_H}{T_H - T_L}$

Table 1. Coefficients of performance (COP) for different types of closed-loop heat pumps [7,12,35].

Stripper pressure (kPa)	200	50
Heating requirement (kW)	6080	8005
Associated exergy requirement (MJ/kgCO ₂)	1.02	0.93
Cooling requirement (kW)	8643	10755
Associated exergy requirement (MJ/kgCO ₂)	0.09	0.12
CO ₂ compression consumption (kW)	304	548
Associated exergy requirement (MJ/kgCO ₂)	0.19	0.35
Total exergy requirement (MJ/kgCO ₂)	1.31	1.40

Table 2. Breakdown of the total exergy requirements for the reference cases.

Heat pump type	Exergy requirements (MJ/kgCO ₂)					Relative reduction
	Heating	Cooling	Heat pumping	Compression	Total	
Base case	1.56	0.09	0	0.19	1.85	-
VCHP	1.09	0.08	0.33	0.19	1.70	8%
VCHP as a side reboiler	1.01	0.08	0.38	0.19	1.67	10%
Type 1 AHP	0	0.08	1.12	0.19	1.39	25%
Type 2 AHP	1.36	0.09	0	0.19	1.64	11%

Table 3. Summary of total exergy requirements for different heat pump types with realistic assumptions for a stripper operating at 200 kPa and with an external hot utility at 225°C.