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Heat Pump and Steam Compressor Cascades: Strategic Decision-Making for a Successful Market Entry

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Abstract

Steam is a crucial energy carrier in industrial processes, integral to various heating and process applications. Mechanical vapor compressor (MVC) and vapor compression heat pump (VCHP) cascades have emerged as highly efficient technologies for steam generation, outperforming standalone VCHP systems for steam generation in terms of energy efficiency. This increase in efficiency motivated first industrial implementations of MVC-VCHP cascades that are either in imminent realization or already operational.
The three primary MVC technologies—piston, screw, and turbo compressors—exhibit unique performance characteristics and operational limitations, but also show some overlap in their operational margins, complicating the selection process for optimal use cases. This challenge is further intensified under partial load conditions, where performance deviations become increasingly pronounced and technology specific.
We present a methodology designed to evaluate industrial site specifications, energy costs and individual equipment capital expenditure as well as performance data. The outcomes of this calculation are then subjected to a sensitivity analysis, allowing for the assessment of variations in input parameters. The levelized cost of heat (LCoH) holds as important metric for long term investment decisions and is therefore chosen as key metric to compare the technologies.
This methodology results in a realistic evaluation of optimal operational windows for each technology, allowing for informed decision-making for both VCHP and MVC suppliers. By identifying the machine configurations tailored to the equipment operating windows, these results contribute to minimizing primary energy consumption and optimal LCoH for the steam supply.

Keywords: heat pump; MVC; steam compressor; LCoH; techno-economic assessment

1. Introduction

Steam is the primary energy carrier in industrial processes due to its excellent heat transfer properties during condensation. With the highest latent heat content of any known fluid, steam offers a high energy density and delivers optimal heat transfer during the condensation process. These characteristics allow for a significant reduction in the size of heat exchangers and piping, particularly in high-pressure steam applications, compared to other heat transfer media.

In many modern industrial plants, steam is primarily produced by fossil fuel-fired boilers. These boilers are favoured for their relatively low investment costs. However, these fossil fuel-based systems emit substantial amounts of CO_2, which has a severe impact on the global climate. In the European Union, industrial processes account for 20.3% of CO_2 emissions [1]. In these industrial processes, heat below 200°C accounts for 39% of

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the total heat demand, which is currently met primarily by fossil fuels, most commonly through natural gas boilers [2].

Direct steam production via electric steam generators offers a potential solution for reducing CO_2 emissions. However, in high electricity-to-gas price scenarios, the associated higher energy costs make them less competitive compared to the fossil fuel fired equipment. This is particularly problematic in energy-intensive industries, where the increased energy costs for electrification result in significant competitive disadvantages in a globalized economy.

State-of-the-art heat pumping technologies offer a promising alternative, providing process steam with lower primary energy input, particularly for temperature ranges around 200°C and below [3–5]. By investing electrical energy, these systems utilize low-grade waste heat and upgrade it to high-temperature process heat at the required temperature level. The pressure, and thereby the temperature, of the saturated steam can be further increased through direct steam compression. This is achieved using mechanical vapor compressors (MVC), a technology known from mechanical vapor recompression (MVR) applications. These technologies offer additional benefits, as they can reach higher saturated steam temperatures than current VCHP technologies [5]. However, they come with certain drawbacks, particularly in applications with low waste heat temperatures, due to the low density of steam in sub-atmospheric conditions.

To leverage the strengths of both VCHP and MVC technologies, cascades of these systems, as illustrated in Figure 1, have been investigated theoretically [6–8] and first devices are already operational [9, 10] or close to full operation [11], demonstrating increased efficiency and operational benefits for each technology when used in combination.

Given the distinct performance characteristics of each technology, the optimal operating window for the cascade varies across different industrial processes. Determining the steam pressure and temperature for transitioning an optimal handover from the heat pump to the steam compressor is not straightforward.

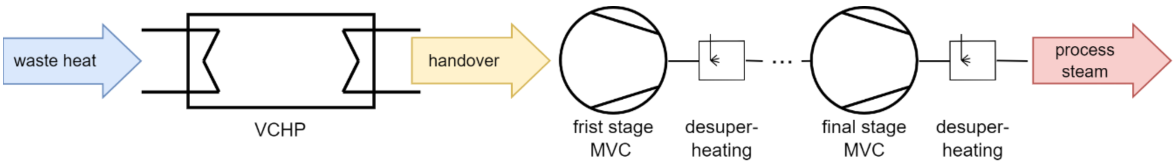


Figure 1. Sketch of the VCHP-MVC cascade under investigation with injection after each MVC stage.

This work aims to address this question from a techno-economic perspective. We take a high-level approach, avoiding detailed modelling of the vapor compression cycle, to make our results applicable across a wide range of process parameters. However, the approach allows for user inputs to adjust process and system parameters, such as the efficiencies of each used technology. Consequently, we identify the point at which operational and investment costs are minimized, determining the optimal handover temperature from the VCHP to the first MVC stage. Since many industrial processes operate off their design point for a significant portion of the year, we also examine how part-load behaviour affects the cascade's performance, which may lead to a different handover temperature than what is observed in full-load conditions.

To illustrate the results of the presented calculation we picked a showcase example that is partially based on specific input data from openly available literature sources. Especially when it comes to equipment investment cost, there is a large uncertainty in the available data.

2. Heat Pump Technologies for Steam Generation

2.1. Vapor Compression Cycle

2.1.1. Technology overview

The vapor compression cycle is the most widely used refrigerant cycle in industrial heat pumps. It relies on heat exchange mainly during the condensation and evaporation of a refrigerant at two distinct states, each defined by fixed pressure and temperature conditions. This characteristic makes the cycle well suited for comparison with the Carnot COP (Coefficient of Performance), which represents the maximum achievable COP for any heat pump and is defined by

$$COP_C = \frac{T_h}{T_h - T_c} \tag{1}$$

This fundamental description of the heat pump performance is based on the two characteristic temperatures of the vapor compression cycle: the condensation temperature T_h and the evaporation temperature T_c. Since a real heat pump operates below the theoretical maximum efficiency, its performance is typically expressed as a fraction of the Carnot limit, quantified by the second-law efficiency η_{2nd}. In general, the COP of a heat pump relates the delivered heat output $\dot{Q}_H$ to the electrical power input P_{el} and thus indicates how much more heat can be supplied compared to the electrical energy invested as can be seen by

$$COP = \frac{\dot{Q}_H}{P_{el}} = \eta_{2nd} COP_C \, . \tag{2}$$

Since we do not delve into the specifics of the refrigeration cycle and instead focus on the heat pump's performance based on external parameters, the temperatures T_h and T_c are referred to external parameters. In the case of steam-generating heat pumps, we set T_h as the steam temperature and T_c as the source outlet temperature, as these temperatures best correspond to the condensation and evaporation temperatures in the refrigeration cycle.

In Arpagaus et al. [3] different approaches of modelling the heat pump performance based on source and sink temperature are presented. We have compared these approaches based on the resulting 2nd law efficiency for different temperature lifts $T_h - T_c$ in Figure 2. Besides the second law efficiency model they present two other approaches that show divergent behaviour at low lifts and therefore fail in describing the heat pump performance in these conditions. Since we want to describe the heat pump performance for arbitrary handover temperatures, we chose the simple but proven approach of describing the heat pump performance by an efficiency model where $\eta_{2nd} = 0.45$ is applied to the Carnot COP, which offers the advantage of being applicable over a wider temperature range.

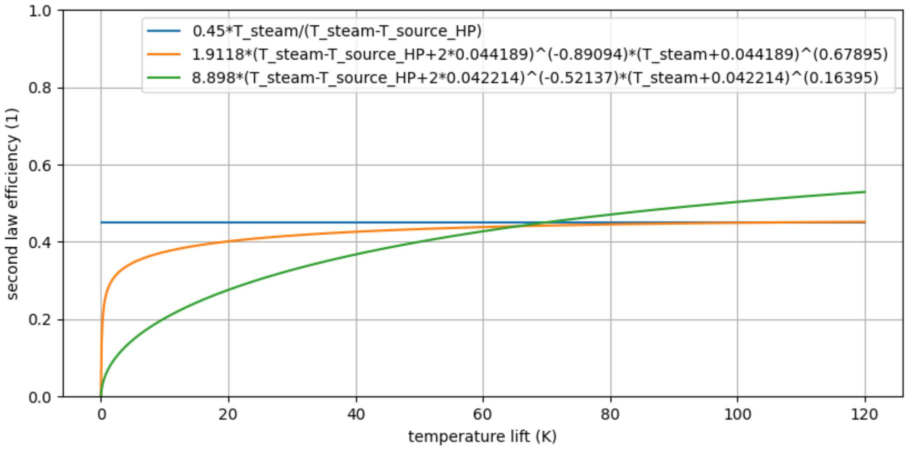


Figure 2. Comparison of different heat pump performance models based on 60°C heat source temperature and variable steam (heat sink) temperature based on equations of Ref. [3].

2.1.2. Off-design-performance

Depending on the compressor technology used in the refrigeration cycle, the off-design performance of VCHPs can vary significantly. While piston compressors maintain a relatively constant performance over a wide range of steam mass flow rates [12] , screw compressors often experience a drop in efficiency when the mass flow—and thus the heat output—deviates from the design point. This will be demonstrated based on the part-load performance of a screw steam compressor in Section 2.2.2. In the showcase example, we assume that the VCHP employs a piston compressor and that the heat pump experiences no efficiency loss during part-load operation as motivated by equipment manufacturer information.

2.2. Mechanical Vapor Compression

2.2.1. Technology overview

Steam compressors are a well-established technology. In this study, we employ them to raise the steam pressure beyond what is achievable with state-of-the-art VCHP systems. For certain applications, we even observe improved performance for lower handover temperatures in the cascade.

By investing electrical power, steam is mechanically compressed, resulting in an increase in both pressure and temperature at the output. This process is often referred to as an open-loop heat pump cycle, as the compressed steam is directly used in the process and not reintroduced into the MVC after condensation. This contrasts with the closed-loop cycle of a VCHP, where the same refrigerant is reused after each cycle. However, an MVC system can also be employed as a compressor in a closed-loop heat pump cycle that uses water (R718) as the refrigerant.

The performance of a compressor is often quantified by its isentropic efficiency η_{is}. This metric describes the deviation of the actual final state of the compressor from the theoretically most efficient final state, following from isentropic compression, where no entropy is produced.

In practical MVC applications, a single compression stage is often insufficient to achieve the desired steam pressure. As a result, steam compressor systems typically employ multiple stages, as shown in Figure 3. After each stage water is injected to reduce superheating, which is also referred to as dry compression since the suction conditions for each stage are never in the wet steam regime (steam quality $x < 1$). Excessive superheating can lead to thermal stress on the equipment and reduce efficiency, as compression becomes increasingly inefficient the further it moves from the saturation curve.

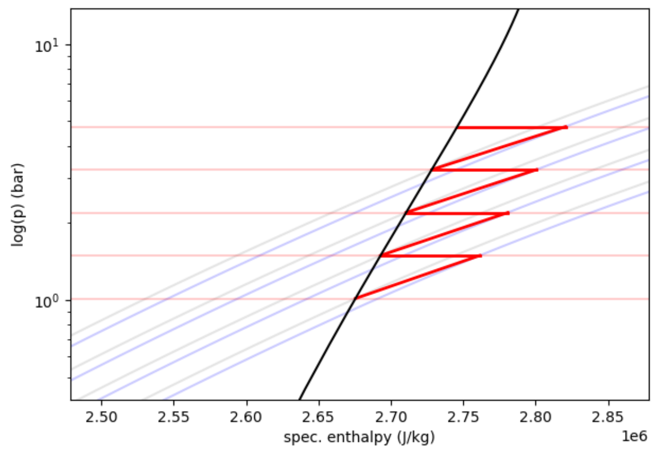


Figure 3. Log(p)-h diagram of a stage wise dry steam compression from 100°C to 150°C saturated steam at a stage pressure ratio PR=1.47. Water is injected after each stage to decrease superheating and reach saturated steam conditions.

The performance of MVC technologies in multistage applications largely depends on both the stage isentropic efficiency and the maximum achievable lift per stage. These two parameters determine the second-law efficiency shown in Figure 4, where its decrease with increasing lift per stage is illustrated.

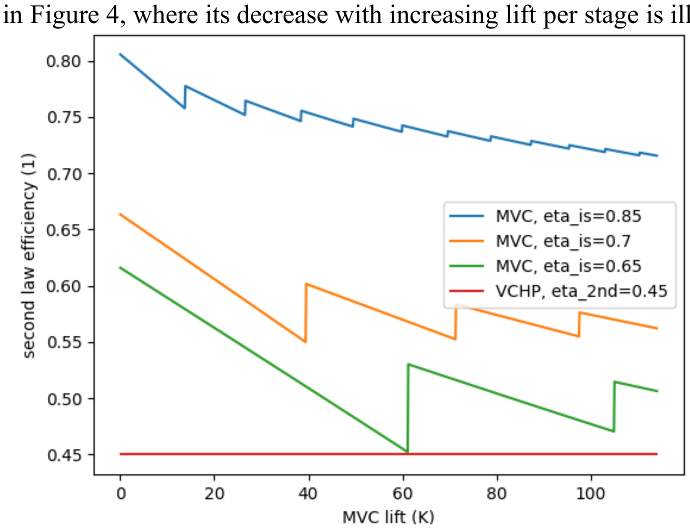


Figure 4. Second law efficiency of MVC technologies with typical isentropic efficiencies and stage pressure ratios compared to the constant efficiency of a VCHP. (Turbo: $\eta_{is} = 0.85$, $PR_{max} = 1.5$, piston: $\eta_{is} = 0.7$, $PR_{max} = 3.5$, screw: $\eta_{is} = 0.65$, $PR_{max} = 8$)

To summarize the approach taken in this work, we describe the performance of the VCHP based on the second-law efficiency, whereas the performance of the MVC technology is characterized by the isentropic efficiency of the compressor. This provides a more detailed description of steam compression. Moreover, this approach remains flexible in describing the heat pump cycle without requiring detailed modelling of its internal dynamics and is applicable across a wide range of operating conditions.

2.2.2. Off-design performance

The off-design performance of an MVC largely depends on the compression technology used. Turbo compressors, for example, have a relatively narrow operating range and experience a significant decrease in performance when deviating from the design point. This behaviour is often illustrated in turbo compressor maps. The efficiency in this context is determined by the pressure ratio and volume flow. Since we are considering fixed pressure ratios for each handover temperature, a slice through the compressor map at a constant pressure ratio can be used to represent the decline in isentropic efficiency as a function of the mass flow ratio relative to the design point. We utilized the compressor map from an experimental steam compressor in Ref. [13] and found a normalized fit function with maximum efficiency $\eta_{\text{is}} = 0.78$ at the design point at 100% mass flow (x) of

$$f(x) = -0{,}728 \cdot x^2 + 1{,}460 \cdot x + 0{,}268 \tag{3}$$

For piston steam compressors we expect the same off-design performance as in heat pumps with piston compressors and therefore assume that the performance doesn't degrade under partial load conditions.

In the work of Deguerece et al. [14] a screw steam compressor was subjected to a performance review, which found an average isentropic efficiency of $\eta_{\text{is}} = 0.65$ across all test points. For part-load behaviour at a constant pressure ratio, we referred to data from manufacturers of screw compressors used in refrigeration applications. The average performance decrease of a screw compressor can be described using the normalized fit function

$$f(x) = 1{,}263 \cdot x^2 - 1{,}316 \cdot x + 1{,}053 \tag{4}$$

2.3. Steam compressor – heat pump cascade performance calculation

An illustration of the equipment arrangement in the cascade is shown in Figure 1. The VCHP at the bottom of the cascade delivers steam at an intermediate temperature to the first stage of the MVC setup. After each stage of compression, water is injected to reduce superheating, and at the final stage, process steam at the required saturated steam temperature is delivered.

Since both technologies produce steam, we chose the latent heat in the outlet steam as the metric for comparison. as it is always required at the same temperature regardless of whether it is generated by the VCHP or the MVC. To calculate the fluid properties of the steam we used the TILMedia python fluid library and determined the heat in the outlet steam based on the enthalpy $h_{steam,saturated}$ at steam temperature T_{steam} and qualities x on hand of

$$\dot{Q}_{\text{steam}} = \left(h_{steam,saturated}(T_{steam}, x = 1) - h_{liquid,saturated}(T_{steam}, x = 0)\right) \cdot \dot{m}_{\text{outlet}} \tag{5}$$

Each MVC technology has a maximum attainable pressure ratio per stage PR_{max} relating the inlet and outlet pressures. The pressure ratio determines the number of stages required for the pressure increase from p_{initial} to p_{final}

$$n_{\text{stages}} = \left\lceil \frac{ln\left(\frac{p_{\text{final}}}{p_{\text{initial}}}\right)}{ln(PR_{max})} \right\rceil \tag{6}$$

The actual compression ratio per stage PR_{actual} is computed on the hand of

$$PR_{\text{actual}} = \left(\frac{p_{\text{final}}}{p_{\text{initial}}}\right)^{1/n_{\text{stages}}} \tag{7}$$

The performance of a cascade can be computed using a stage wise algorithmic approach based on the isentropic efficiency of each steam compressor stage and the second-law efficiency of the heat pump. The performance calculation begins with the desired steam mass flow required for the process heat, which correlates to the heat output. Each preceding stage has a lower mass flow $\dot{m}_{\text{stage}}$ due to the desuperheating water injection $\dot{m}_{\text{inj}}$, which is subtracted. After all MVC stages, the required steam mass flow coming out of the bottom-end steam generating heat pump is determined. The calculation for each MVC stage is presented in the following list.

1. Calculate the inlet steam pressure p_{in} for the current stage based on the outlet pressure p_{out} and the pressure ratio PR_{actual}. First, the inlet pressure of the final stage is computed, where p_{out} is the desired steam output pressure required for the process. For each further step of the calculation p_{out} is equivalent to the inlet pressure of the next higher stage.

$$p_{\text{in}} = \frac{p_{\text{out}}}{PR_{\text{actual}}} \tag{8}$$

2. Compute the thermodynamic state conditions at the inlet of the current stage

$$h_{\text{in}} = h(p_{\text{in}}, x = 1), \quad s_{\text{in}} = s(p_{\text{in}}, x = 1) \tag{9}$$

3. Compute the outlet state according to isentropic compression (ideal compression)

$$h_{\text{is, out}} = h(p_{\text{out}}, s = s_{\text{in}}) \tag{10}$$

4. Compute real output enthalpy according to isentropic stage efficiency

$$h_{\text{out}} = h_{\text{in}} + \frac{h_{\text{is, out}} - h_{\text{in}}}{\eta_{\text{is}}} \tag{11}$$

5. Reduce desuperheating by water injection of an amount depending on the inlet mass flow

$$\dot{m}_{\text{inj}} = \frac{\dot{m}_{\text{stage}} \cdot \dfrac{h_{\text{out}} - h_{\text{sat, vap}}}{h_{\text{sat, vap}} - h_{\text{liq}}}}{1 + \dfrac{h_{\text{out}} - h_{\text{sat, vap}}}{h_{\text{sat, vap}} - h_{\text{liq}}}} \tag{12}$$

6. Electrical power per stage with overall efficiency of 95% depending on the inlet mass flow

$$P_{\text{MVC,stage}} = \frac{(\dot{m}_{\text{stage}} - \dot{m}_{\text{inj}}) \cdot (h_{\text{out}} - h_{\text{in}})}{0{,}95} \tag{13}$$

7. The inlet mass flow is handed over to preceding stage as new outlet mass flow

$$\dot{m}_{\text{stage}} \leftarrow \dot{m}_{\text{stage}} - \dot{m}_{\text{inj}}, \quad p_{\text{out}} \leftarrow p_{\text{in}} \tag{14}$$

The heat pump thermal capacity is then computed based on the output steam mass flow, which is equivalent to the inlet steam mass flow of the first MVC stage.

$$\dot{Q}_{\text{VCHP,required}} = \dot{m}_{\text{evap}} \cdot \left(h_{\text{sink}} - h_{\text{source}} + c_p \Delta T\right) \tag{15}$$

In the sensible heat part of $\dot{Q}_{\text{VCHP,required}}$ we calculated the heat capacity at constant pressure c_p at the handover temperature between the heat pump and the steam compressor, while the temperature difference ΔT is the difference of the handover temperature and a fixed feedwater temperature of 20°C.

3. Cost Model

3.1. Investment cost

3.1.1. VCHP investment cost

Typically, the specific investment cost (e.g., €/kW) of heat pumps decreases with increasing capacity. This trend is represented by specific cost functions found in the literature. The main contributors to heat pump costs are the compressor and the two heat exchangers used for condensation and evaporation. These components generally scale with the heat demand at the sink, which motivates relating the specific investment cost to the heat capacity $\dot{Q}_{\text{h}}$ of the heat pump, as described in Ref. [3]. This relationship is based on a fit of commercially available VCHPs.

$$c_{inv,HP} = 3157 \cdot \dot{Q}_h^{-0.322} \text{ for } \left[c_{inv,HP}\right] = €/\text{kW}_{\text{th}} \tag{16}$$

3.1.2. MVC investment cost

For MVC technology, the specific cost is sometimes reported in relation to the thermal capacity [15]. However, since the steam compressor is not equipped with a heat exchanger, this may not always provide the most accurate representation, as the cost is largely influenced by the number of stages, which increases with the temperature lift. Given that electrical power correlates with the temperature lift as well as the mass flow according to Eq. (13), we chose it as a representation of the equipment cost. Due to the limited data available, we adopted heat specific values to electrical power specific to describe the cost of MVC equipment. Based on realized projects published in Annex 58 – High Temperature Heat Pumps [5] we estimated the specific cost for reciprocating piston MVC technologies to $970€/\text{kW}_{\text{el}}$ derived from a project where the installation costs were not included. For turbo MVC technologies we found a project with unclear details regarding installation costs, therefore we decided to halve the reported investment cost, assuming a specific investment of around $1500€/\text{kW}_{\text{el}}$. As no publicly available price data was found for screw MVC systems, we assumed their investment cost to be equivalent to that of the piston MVC.

3.2. Operating cost

The total annual heat demand Q_{heat} is then reflected in the operating expenditure $OPEX$, calculated using the electricity price EP, the electric power demand $P_{el,tot}$ and the utilization time per year t_{spec}. EP only considers electricity consumption cost, while electricity demand cost has been ignored. These costs are combined with the annual maintenance expenditure $OPEX_M$ to determine the total operating expenditure

$$OPEX = OPEX_M + t_{spec} \cdot P_{el,tot} \cdot EP \tag{17}$$

A literature study suggests that the average annual maintenance cost is about 3.2% of the investment cost [3]. To account for additional unknown factors, we chose to set it to 4% for our further calculations.

3.3. Levelized Cost of Heat

LCoH has been chosen as the key metric to compare the total cost when we compare the share that vapor compression heat pumps and steam compressors cover in the cascade

They consider CAPEX as well as OPEX, which are set in relation to the generated heat

$$Q_{heat} = t_{spec} \cdot \dot{Q}_{heat} \tag{18}$$

The calculation of the levelized cost of heat is based on the calculation of levelized cost of electricity presented by Short et al. [16]. However, due to the assumption of constant energy demand and cost it can be simplified by employing the CRF factor and then reads

$$LCOH = \frac{CRF \cdot CAPEX + OPEX}{Q_{heat}} \tag{19}$$

Here capital recovery factor CRF takes account of the weighted average cost of capital (WACC) in the following form

$$CRF = \frac{WACC(1 + WACC)^t}{(1 + WACC)^t - 1} \tag{20}$$

In our calculation the weighted average capital cost $WACC$ is chosen to 8% by factoring in interest rates and capital risk. In general it depends also on tax credits and rates, depreciation and insurance cost and is often estimated between 5 and 15% [3][16].

4. Results

4.1. Exemplary use cases of cascades

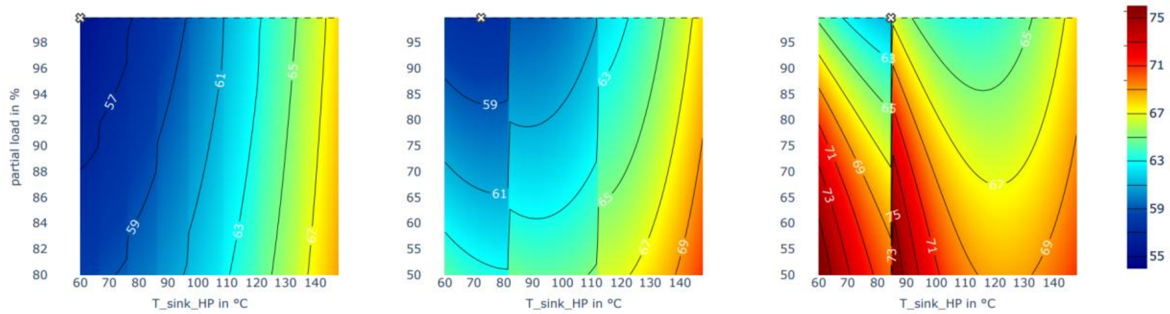
To showcase the performance of heat pump cascades in different scenarios we defined three distinct use cases. All cases operate at the same process steam temperature of 150°C and a heat source temperature of 60°C. The cost of electricity is estimated at 100 €/MWh, and the total operating time for the machinery is set to 10 years. Under full load conditions, all cases deliver 5.4 t/h or 3.17 MW of steam, which were chosen since it is in the operating range of each MVC technology in the given process conditions. Cases A and B compare the effect of heat pump efficiency, with both cases having the same utilization of 6000 hours per year. Case C represents a scenario with low utilization, operating for just 3000 hours per year. Each case is considered with 50% of the time in part-load operation. The case parameters are provided in Table 1, the resulting LCoH heat maps for Cases A, B and C are displayed in Figure 5, Figure 6 and Figure 7.

Table 1. use cases for VCHP-MVC cascades

Use Case	A (Base case)	B (eff. HPs)	C (low utilization)
t_{spec}	6000	6000	3000
η_{2nd}	0,45	0,55	0,45

4.2. Visualization of results

The multitude of input parameters presents challenges when visualizing the results in two dimensions. Out of a user perspective, we chose to plot the handover temperature, which is equivalent to the VCHP sink temperature, and the percentage of part-load operation on individual axes of the outcome diagrams. In each diagram, the LCoH is represented by a colour gradient of a heat map, with the minimum LCoH highlighted by an “x” mark. This representation provides the user with insight into how much heat is delivered by the VCHP and MVC, while also illustrating how the optimal handover temperature changes as the part-load factor increases.

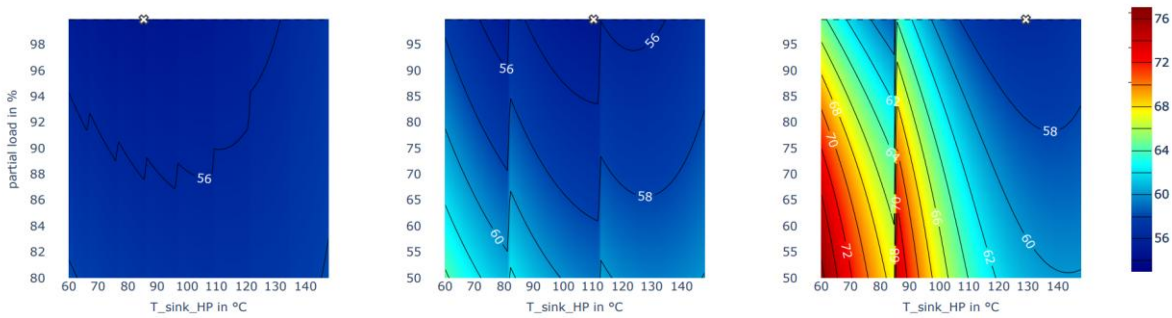


Turbo: $\eta_{is} = 0.78, PR_{max} = 1.5$
Possible part load: 80-100%

Piston: $\eta_{is} = 0.7, PR_{max} = 3$
Possible part load: 50-100%

Screw: $\eta_{is} = 0.65, PR_{max} = 8$
Possible part load: 50-100%

Figure 5. LCoH plotted on a heat map for use case A for three different MVC technologies



Turbo: $\eta_{is} = 0.78, PR_{max} = 1.5$
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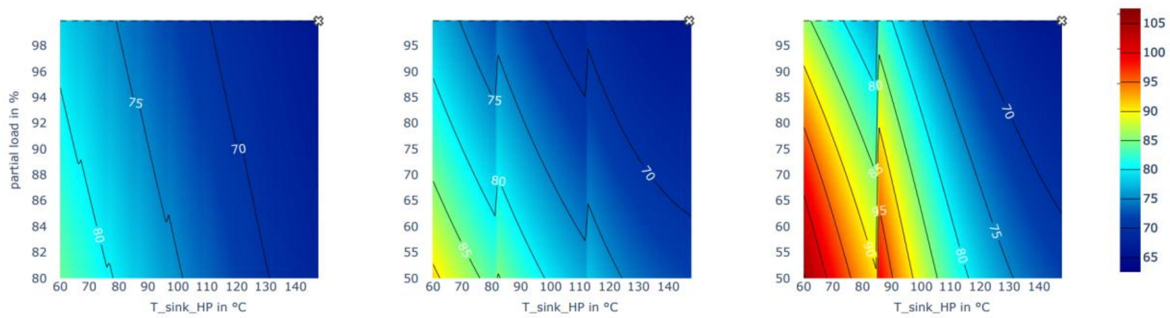
Piston: $\eta_{is} = 0.7, PR_{max} = 3$
Possible part load: 50-100%

Screw: $\eta_{is} = 0.65, PR_{max} = 8$
Possible part load: 50-100%

Figure 6. LCoH plotted on a heat map for use case B for three different MVC technologies



Turbo: $\eta_{is} = 0.78, PR_{max} = 1.5$
Possible part load: 80-100%

Piston: $\eta_{is} = 0.7, PR_{max} = 3$
Possible part load: 50-100%

Screw: $\eta_{is} = 0.65, PR_{max} = 8$
Possible part load: 50-100%

Figure 7. LCoH plotted on a heat map for use case C for three different MVC technologies.

The sharp changes in LCoH in Figure 5 to Figure 7 stem from the introduction of a new stage of MVR to achieve the desired lift under the given maximum pressure ratio.

4.3. Sensitivity analysis

Since all input parameters are subjected to some degree of uncertainty, we performed a sensitivity analysis at the minimum LCoH point for each heat map and varied each input parameter individually from 50 to 150% individually while keeping the others constant.

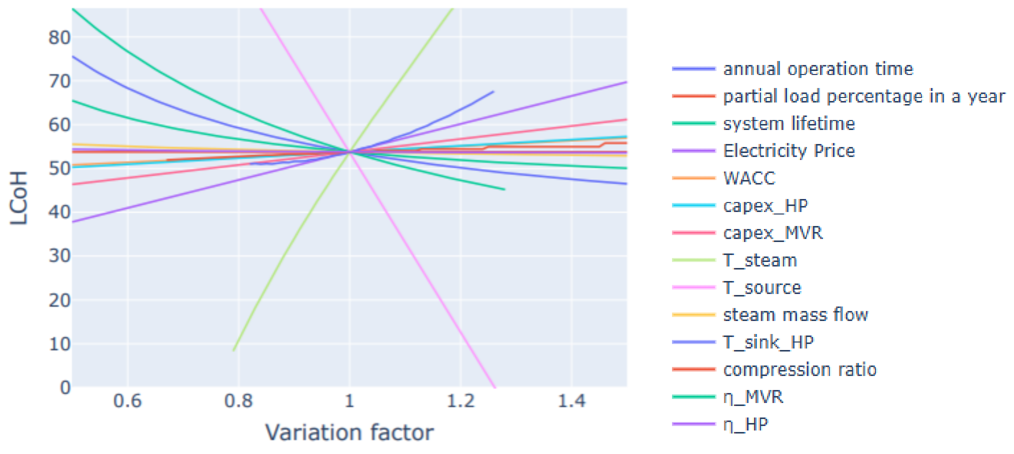


Figure 8. Sensitivity analysis at minimum LCoH of a turbo MVC for Case A (with $\eta_{2nd} = 0.45$ at 6000h/a)

As can be seen in Figure 8, the system is most sensitive to the output steam temperature and the source temperature. This behaviour is expected since these two temperatures directly influence the maximum attainable efficiency COP_C of the total cascade. We also see that the influence of electricity cost shows a linearly increasing behaviour on the LCoH.

The investment cost and cost of capital only have a marginal influence on the LCoH due to the high equipment utilization in this case.

In use case C, with low utilization of the equipment, the influence of CAPEX and WACC becomes more pronounced, as can be seen in Figure 9. The increased gravitas of the heat pump CAPEX is also evident, as the optimal handover temperature is equal to the outlet steam temperature, resulting in no use of MVC.

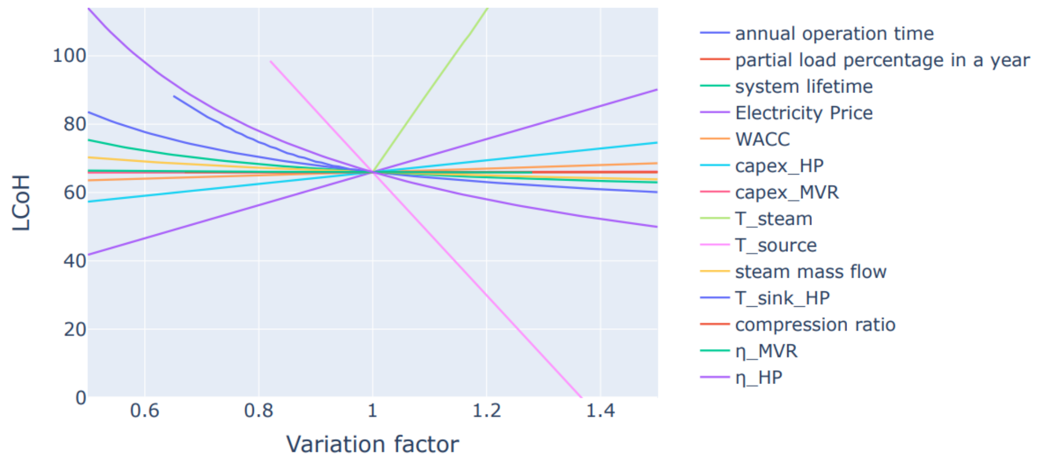


Figure 9. Sensitivity analysis at minimum LCoH of a turbo MVC for Case A (with $\eta_{2nd} = 0.55$ at 3000h/a)

5. Conclusions and Outlook

It is important to note that our analysis relies solely on publicly available data. Price data is difficult to obtain and is based on multiple sources across different years. As a result, the findings should not be interpreted as absolute truths but rather as a directional guide indicating where each technology might be most suitable. Nevertheless, we have developed a methodology that allows users to input values from manufacturer quotes and therefore achieve a more accurate result. Through the sensitivity analysis, it is possible to assess which parameters are more critical and where less precise information might still yield reliable results.

However, general trends can be derived from the results presented in this work. For Case A (Figure 5), a low handover temperature is optimal due to the high utilization of the equipment and the high efficiencies of the MVC technologies. For Case C (Figure 7), the opposite trend is observed, as investment costs carry a higher weight under low-utilization conditions. Case B (Figure 6) further shows that efficient vapor compression heat pumps can be favourable even at high utilization when their efficiency, represented by a second-law efficiency of $\eta_{2nd} = 0.55$, exceeds that of the base case (case A).

5.1. Other criteria that must be considered for investment decisions

Our methodology provides valuable insights into the techno-economic factors that must be considered when making investment decisions regarding operational parameters. These insights can be helpful not only for site managers but also for equipment manufacturers when deciding how much R&D effort to invest in developing machinery with larger operational windows. However, the used LCoH metric does not account for variations in energy demand and energy costs over the entire utilization period. To more accurately represent such behaviours, the more detailed calculation method of Short et al. [16] is required. Additionally, LCoH is just one factor to consider when making an investment decision.

Other important criteria must be considered before installing a heat pump cascade at an industrial site. These include the equipment footprint, operational reliability and redundancy, investment risk, and equipment utilization, among other considerations. For some processes, clean steam, free from contamination by debris from the compressor, is needed, rendering the use of some MVC technologies unfeasible.

As steam pressure decreases, so does its density, requiring an increase in volume flow to deliver the same amount of heat. This increase in volume flow leads to larger condensers of the heat pump and larger piping to the steam compressor. Such increases in equipment size result in higher costs, which are not fully captured by the LCoH calculation in relation to the handover temperature. On the other hand, CAPEX has little effect on LCoH when equipment utilization is high.

5.2. Extensions to our methodology

5.2.1. Dry vs. wet compression in the MVC

We only considered dry compression, where the steam at the inlet of the MVC is already in a saturated state, becomes superheated during compression, and is then returned to a saturated state after compression through water injection, as shown in Figure 3.

In wet compression, water is injected before compression, and the steam enters the compressor in a wet state ($x < 1$). Some MVC suppliers offer this type of compression, which has shown potential for improving efficiency and lowering superheated outlet temperatures, thus reducing thermal stress on the machinery. Hosseinnia et al. [6] demonstrated that wet compression can be 2.1% more efficient than a dry compression in a scenario of 6 injection ports. With an increasing number of injection ports the efficiencies of dry and wet compression converge since both follow the saturated vapor line.

5.2.2. Sensitivity analysis based on handover temperature instead of LCoH

While the sensitivity analysis based on LCoH provided valuable insights into which parameters influence the investment decision, another important criterion is how these parameters affect the handover temperature from the VCHP to the MVC in the cascade. This would require determining the minimum LCoH for each parameter variation using an optimization approach, making it computationally more intensive. However, this approach would offer valuable insights into how the load share between the bottom and top stages changes when uncertainties in certain parameters arise.

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References

[1] European Parliament, *Greenhouse gas emissions by country and sector.* [Online]. Available: https://www.europarl.europa.eu/topics/en/article/20180301STO98928/greenhouse-gas-emissions-by-country-and-sector-infographic (accessed: Nov. 17 2025).

[2] Frauenhofer ISI, *Direct electrification of industrial process heat: An assessment of technologies, potentials and future prospects for the EU.* [Online]. Available: https://www.agora-industry.org/

publications/direct-electrification-of-industrial-process-heat (accessed: Nov. 20 2025).
[3] ARPAGAUS C., BLESS F., and BERTSCH S., "Techno-economic analysis of steam generating heat pumps for integration into distillation processes," 2022.
[4] F. Bless, C. Arpagaus, S. S. Bertsch, and J. Schiffmann, "Theoretical analysis of steam generation methods - Energy, CO 2 emission, and cost analysis," *Energy*, vol. 129, pp. 114–121, 2017, doi: 10.1016/j.energy.2017.04.088.
[5] D. T. Benjamin Zühlsdorf, "Annex 58 High-Temperature Heat Pumps Task 1 - Technologies: Task Report," *IEA Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)*.
[6] S. M. Hosseinnia, L. Amiri, H. Nesreddine, D. Monney, and S. Poncet, "Thermodynamic analysis of high temperature cascade heat pump with R718 (high stage) and six different low-GWP refrigerants (low stage)," *Case Studies in Thermal Engineering*, vol. 53, p. 103812, 2024, doi: 10.1016/j.csite.2023.103812.
[7] V. Wilk, F. Helminger, S. Dusek, M. Lauermann, and T. Fleckl, "Steam generation from waste heat with heat pumps," in *Proceedings of DKV Tagung 2019*.
[8] Kenneth Hoffmann, Michael Bantle, Kjetil Evenmo and Vebjørn Nilsen, "Ammonia - Steam cascade heat pump for +100˚C steam generation," *14th IEA Heat Pump Conference Procedings*, 2023.
[9] Michael Bantle, Christian Schlemminger, Vebjørn Nilsen, "Integration of a 1.6 MW Steam Supplying Heat Pump into the Feed Production Process," *High-Temperature Heat Pump Symposium 2024 Proceedings*, pp. 429–436, 2024.
[10] KOBELCO Compressors Corporation, *Steam Grow Heat Pump / SGH165*.
[11] Johannes Riedl, Felix Hubmann, Reinhard Jentsch, Verena Sulzgruber, Bernd Windholz, Veronika Wilk, "Optimierung eines Wärmepumpensystems für Kälte und Dampf," *DKV-Tagung 2023 - Konferenzband*, 2023.
[12] OMMEN T., JENSEN J., MEESENBURG W., and et al., "Generalized COP estimation of heat pump processes for operation off the design point of equipment," doi: 10.18462/IIR.ICR.2019.0648.
[13] BANTLE M., SCHLEMMINGER C., TOLSTOREBROV I., and et al., "Performance evaluation of two stage mechanical vapour recompression with turbo-compressors," doi: 10.18462/IIR.GL.2018.1157.
[14] B. Degueurce, F. Banquet, J. Densant, and D. Favrat, "Use of a twin screw compressor for steam compression," *Hydraulic Pneumatic Mechanical Power*, vol. 1984, no. 30, pp. 335–337. [Online]. Available: https://core.ac.uk/download/pdf/147936761.pdf
[15] S. Wolf, "Integration von Wärmepumpen in industrielle Produktionssysteme: Potenziale und Instrumente zur Potenzialerschließung," Dissertation, Institut für Energiewirtschaft und Rationelle Energieanwendung, Universität Stuttgart, Stuttgart, 2017. Accessed: Nov. 23 2020. [Online]. Available: http://dx.doi.org/10.18419/opus-9593
[16] W. Short, D. Packey, and T. Holt, "A Manual for the Economic Evaluation of Energy Efficiency and Renewable Energy Technologies," 1995.