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Performance evaluation of high-temperature heat pump systems for hot water and steam generation in food processing

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ABSTRACT

A large amount of fossil fuels is consumed for hot water and steam generation in food processes such as smoking, scalding, drying, pasteurization, sterilization, cleaning, and cooking, which currently rises both economic- and environmental concerns. At the same time, there is a considerable number of low-grade waste heat available, often from associated cooling processes. High-temperature heat pumps (HTHPs) are considered as a promising solution for steam boilers replacement and waste heat recovery owing to their high energy efficiency and sustainability. In the present study, the performance of three heat pump systems — a trans-critical CO_2 heat pump system, a cascade propane-butane heat pump system and an ammonia-water absorption-compression heat pump (ACHP) system, for hot water and steam production in food processing is evaluated based on different application scenarios. In all the scenarios and temperature lift levels, the ACHP exhibits the best thermal performance with moderate pressure levels and low-pressure ratios. When applying the ACHP for both the cleaning and cooking processes, the achievable energy saving rate can reach 79%. Further, the optimal HTHP system architectures for different application scenarios are discussed.

Keywords: High-temperature heat pump, steam generation, ammonia-water mixture, food industry, GHG-emissions reduction, natural refrigerants.

1. INTRODUCTION

In the food supply chain, a large amount of fossil fuels is consumed and applied for hot water and steam generation. Phasing out the fossil-fuel based boilers is vital to reduce greenhouse gas (GHG) emissions and achieve climate neutrality by 2050 in this sector. Steam boilers are usually applied to enable heating processes such as scalding, smoking, cooking, sterilization and pasteurization in food production plants. At the same time, there is a considerable number of low-grade waste heat available from associated cooling processes such as chilling, freezing and air conditioning. With a further development of heat pump technology, supply temperatures of more than 100 °C and up to 150 °C are achievable with existing industrial high-temperature heat pumps (HTHPs) (Arpagaus et al., 2018; Jensen et al., 2015). This means that the HTHPs are considered as a promising solution for steam boilers replacement and surplus heat recovery owing to their high energy efficiency, compactness, and sustainability.

The HTHPs using natural refrigerants as working fluid have gained particular interest in recent years due to their low global warming potential (GWP) and well known environmental impacts (Mateu-Royo et al., 2021). The major limitations of employing HTHPs for high-temperature heat supply are the critical point of the refrigerants and the compressor operating constraints at high temperature levels and pressures (Ommen et al., 2015). Nekså et al. (1998) pointed out that the safety limit for avoiding oil degeneration is 180 °C for most of the commonly used lubrication oils in compressors.

CO_2 is considered a promising natural refrigerant for HTHPs since it is neither toxic nor flammable and widely available on a global scale. The trans-critical CO_2 heat pump is capable of produce hot water from cold water directly up to 90 °C without operational problems (Nekså et al., 1998). Some commercial trans-critical CO_2 heat pumps on the market can even achieve supply temperatures up to 120 °C for hot air production (Arpagaus et al., 2018). The modelling results of White et al. (2002) revealed that the pressurized water can also be heated up to 120 °C with a trans-critical CO_2 heat pump and the reduction of heating capacity and COP is relatively small when increasing the hot water temperature from 65 °C to 120 °C, however, the CO_2 HTHPs have to be considered as high temperature lift units able to heat cold water to high temperature levels. The lower the cold-water inlet temperature to be heated, the higher the energy efficiency of these kind of HTHPs.

Hydrocarbons are another favourable natural working fluid for the use in HTHPs, due to their similar thermodynamic properties and performance parameters to refrigerants applied during the last decades. From the evaluation results of Bamigbetan et al. (2018), butane (R600) exhibits the highest potential for future application in HTHPs for up to 125 °C supply temperatures in the category of natural refrigerants, owing to its high operating flexibility and the commercially available compressors. Pachai et al. (2021a, 2021b) pointed out that butane (R600), iso-butane(R600a), pentane(R601) and heptane(R603) are good environmentally friendly solutions for HTHP applications with supply temperature below 230 °C, based on their simulations. Water vapor is the only option for supply temperatures from 230 °C to 250 °C. Wemmers et al. (2017) developed a butane HTHP employing commercially available components which can produce low pressure steam with up to 125 °C (2.4 bar) from a heat source temperature of 60 °C at a COP of 1.9. While the butane based HTHP developed by Moisi and Rieberer (2017) achieved a heat sink outlet temperature of 110°C at a COP of 3.5 utilizing a heat source at the same (60 °C) temperature level. Bamigbetan et al. (2019) studied the performance of a propane-butane (R290 - R600) two-stage cascade HTHP which employs propane as the working fluid in the low temperature cycle (LTC) and butane as the working fluid in the high temperature cycle (HTC). The heat sink temperatures of up to 115 °C were achieved by recovering waste heat at 25-35 °C. Bergamini et al. (2019) evaluated the performance of single-stage, two-stage and cascade HTHP systems with four different natural refrigerants (propane, butane, ammonia and water) for boiler substitution. Compared with natural gas boilers, the HTHPs exhibit good performance in supplying heat up to a temperature level of 180 °C, especially for the cycles applying ammonia and water as refrigerants.

Absorption-compression heat pumps (ACHP) which are based on the principle of the Osenbrück cycle (Osenbrück, 1895) have been of high interest in the past years because of the low discharge pressure levels and the favourable temperature glides in the heat exchangers on both sides. The heat supply temperatures of up to 150 °C and temperature lifts of up to 60 K are achievable with a HACHP employing commercially available components. These units are more economical, compared when applying gas boilers (Jensen et al., 2015). The fully integrated energy system of an investigated dairy, which utilizes an ACHP as HTHP, demonstrated that this heat pump provides all required temperature levels for the various heating and cooling demands. It achieved a waste heat recovery rate of up to 95% and GHG emissions reductions between 23.2% and 91.7% compared to a conventional energy system (Ahrens, Foslie, et al., 2021).

An integrated energy system combining colling- and heating processes with the HTHPs can directly utilize the surplus heat from refrigeration systems as well as medium temperature processes and provide high temperature heat for hot water and steam consumers in a food plant. In the present study for hot water and steam production, the performance of three different heat pump systems are analysed: — a trans-critical CO_2 heat pump system, a cascade type propane-butane system and an ammonia-water ACHP system. The achievable energy savings and GHG-emissions reduction with the different HTHP systems are investigated and compared. Further, the optimal HTHP system architecture for different application scenarios are discussed.

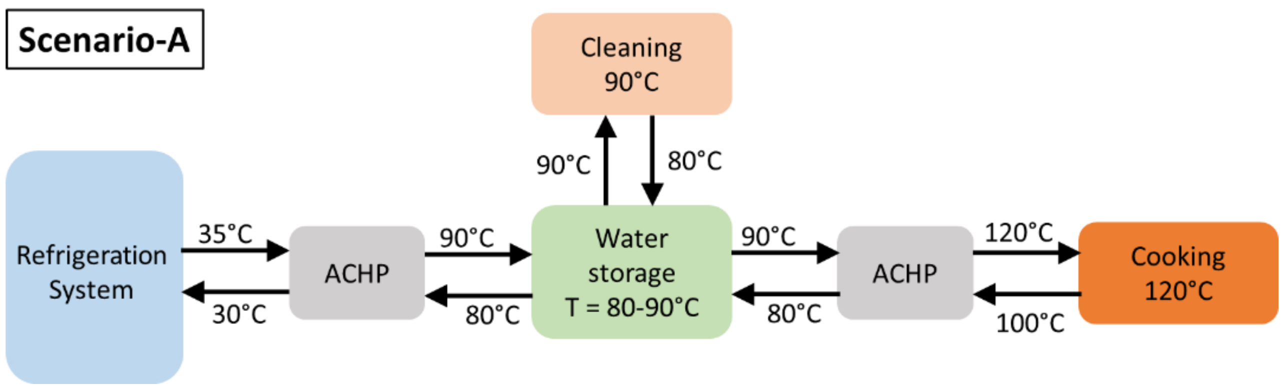
2. METHODS

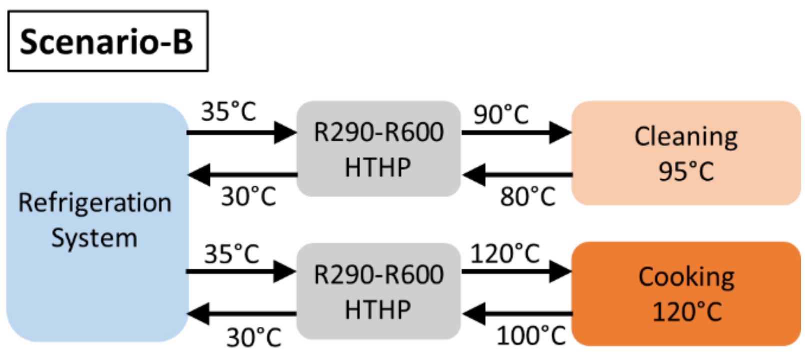
The performance evaluation of three high-temperature heat pump systems for hot water and steam production is conducted based on defined demand cases. The assessment concentrates on the HTHP behaviors on power demand and the GHG emission reduction potential.

2.1. Demand cases

Usually in food plants like meat production plants and dairies, the thermal demands for cooling processes such as chilling, freezing and storage-room cooling are provided by refrigeration systems which are major consumers of electric energy (Iten et al., 2021). The surplus heat from the refrigeration systems is at relatively low temperature levels and often rejected to the ambient. Steam boilers are used in the processes with heating demands, for instance; sterilization, steaming & cooking, cleaning, smoking, scalding, drying, pasteurization and ultra-high temperature (UHT) treatment (Ramirez et al., 2006a). The temperature requirements of these processes are different and varying with external conditions including fluctuations of product demands and seasonal variations. Ahrens, Selvnes, et al. (2021) discussed different system integration strategies for surplus heat recovery in food processing plants. The condensing heat from the refrigeration system can be recovered and used as the heat source for heating processes applying heat pumps. For the processes demanding high temperature levels, such as cleaning and cooking, the heating capacity can be supplied by HTHPs instead of steam boilers.

The present study aims to explore the suitable HTHP solutions for hot water and steam supply in integrated food processing systems. According to the heat demands of common thermal processes in industrial food production (Ramirez et al., 2006a; Ramirez et al., 2006b), several heating processes with specific temperature requirements are defined, as listed in Table 1. Three HTHP systems are evaluated to supply heat for the defined processes, including a trans-critical CO_2 HTHP system, a cascade propane-butane HTHP system and an ACHP system. In addition, based on the different system integration strategies, five application scenarios of the studied HTHPs are determined, as shown in Figure 1.





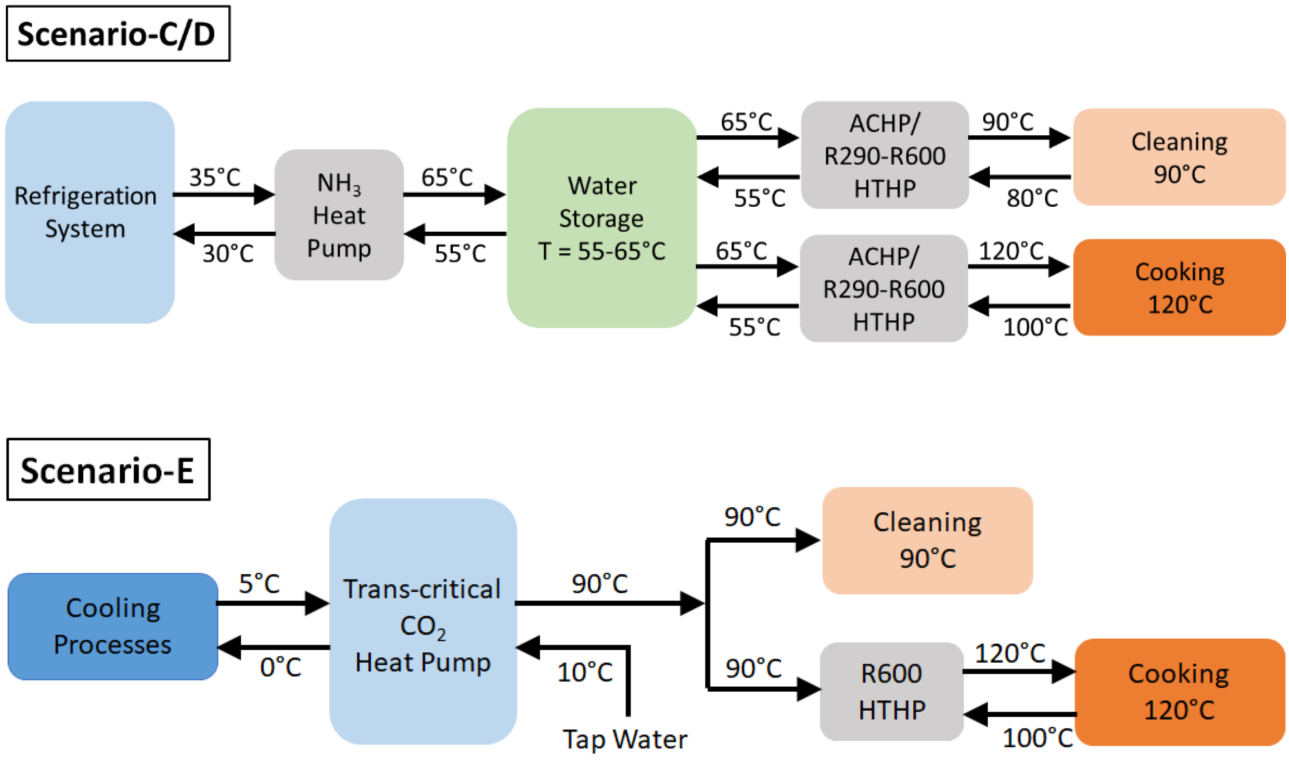


Figure 1: Application scenarios of the studied HTHPs

Table 1. Heat requirements of process fluids

Process	Supply temperature	Return temperature
Cooking	120°C	110°C
Cleaning	90°C	80°C
Water heating	90°C	10°C

In Scenario-A, the condensing heat from the refrigeration system at approximately 35 °C is used as the heat source for an ACHP to provide heat (process water) at 90°C for the cleaning process. The rest of the condensing heat from the ACHP is stored in an integrated water tank and utilized as the heat source for another ACHP which supplies heat at 120 °C for the cooking process.

In Scenario-B, two propane-butane cascade HTHPs utilize the 35 °C condensing heat from the refrigeration system as the heat source to produce high temperature heat of 90°C and 120°C for the cleaning and cooking processes respectively.

In Scenario-C/D, the waste heat from the refrigeration system is firstly heated up to 55-65 °C by a NH_3 heat pump and stored in a water tank which is used as the heat source by two HTHPs supplying heat for cleaning and cooking processes. Two ACHPs and two propane-butane cascade HTHPs are applied in Scenario-C and Scenario-D, respectively.

In Scenario-E, a trans-critical CO_2 heat pump is applied to provide heat for both the cooling and heating processes. On the cold side, cold process water of 0°C is supplied to the consumers with cooling demands and represents the corresponding heat source. On the warm side, hot process water of 90°C is produced in the gas cooler by heating up water from 10°C. The hot process water is used for multiple purposes in the food plant including cleaning and serving as the make-up water for a butane HTHP circuit, supplying heat for the cooking process.

The performance of the three HTHP systems in different application scenarios are examined. The assessment focuses on their behaviors on energy consumption and CO_2 emissions reduction. Due to the lack of practical operational data of food plants, the same exemplary demand and capacity of 5000 kW is used for both the cleaning and cooking processes to provide a comprehensive assessment of the energy systems in the studied scenarios.

2.2. HTHP system description

2.2.1. Trans-critical CO_2 heat pump cycle

Due to the low critical temperature of CO_2 (31 °C, 71 bar), a trans-critical cycle is required for high temperature lift applications. A simplified sketch of a typical trans-critical CO_2 heat pump and the corresponding T-S diagram are illustrated in Figure 2. The heat absorption in the evaporator occurs at constant temperature and in sub-critical condition. While the heat rejection in the gas cooler takes place in the supercritical region with a large temperature glide and the minimum temperature difference between refrigerant and water to be heated. Therefore, the trans-critical CO_2 cycle is suitable for applications which requires a large heat sink temperature glide and low heat source temperature levels providing useful cooling capacities. Moreover, the high volumetric efficiency and good heat transfer characteristics of CO_2 results in compact and efficient heat exchangers in trans-critical CO_2 heat pumps.

2.2.2. Propane/butane cascade heat pump cycle

A simplified sketch of a propane-butane cascade heat pump and the corresponding T-S diagram are illustrated in Figure 3. Propane is applied as the working fluid in the low temperature cycle (LTC) and butane is utilized as the working fluid in the high temperature cycle (HTC). Heat is transferred through a cascade heat exchanger between the LTC and HTC. The heat transfer of heat source and sink sides occurs mostly at constant temperature levels. For cases requiring a low temperature lift and heat source temperature levels close to the critical temperature of propane (96 °C), a single butane cycle can be used as HTHP for the high temperature supply.

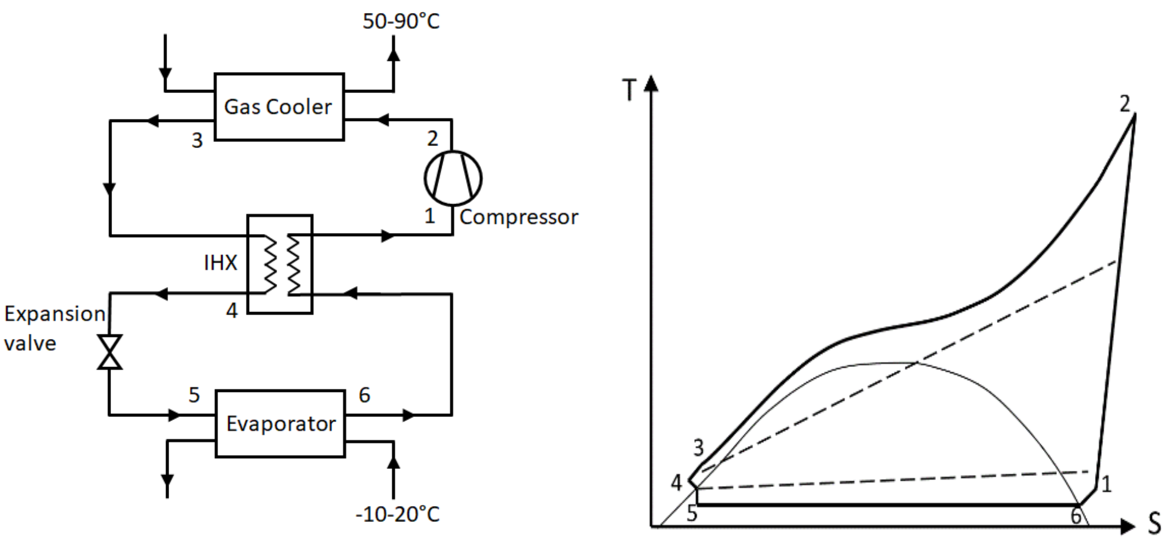


Figure 2: Simplified sketch of a typical trans-critical CO_2 heat pump and the corresponding T-S diagram

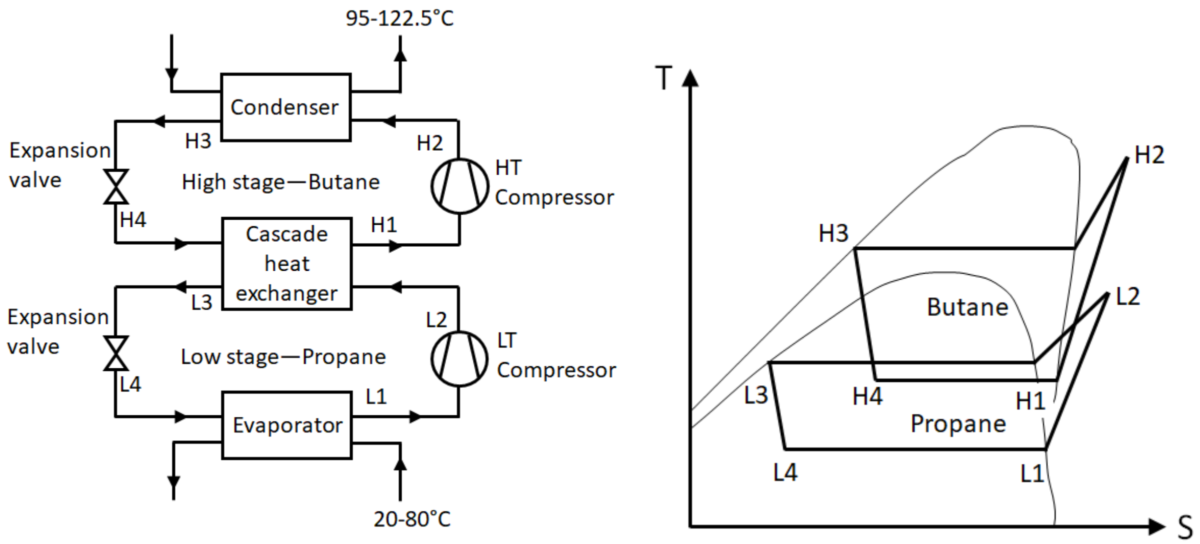


Figure 3: Simplified sketch of a propane-butane cascade heat pump and the corresponding T-S diagram

2.2.3. Ammonia-water absorption-compression heat pump cycle

The ammonia-water ACHP is designed based on the Osenbrück cycle as vapor compression cycle with an additional liquid circuit. A simplified sketch of a ACHP system and the corresponding P-T diagram are illustrated in Figure 4. It combines the technologies of an absorption and vapor compression heat pump with a mixture of ammonia and water as the working fluid. Heat is extracted and released at non-constant / temperature glides in the absorber and desorber. The necessary compression ratio is lower compared to conventional vapor compression heat pumps. High heat sink temperatures of up to 130 °C can be achieved with comparatively high COPs. In addition, the capacity can be controlled by changing the overall composition of the working fluid mixture, which ensures high system flexibility and adaptability.

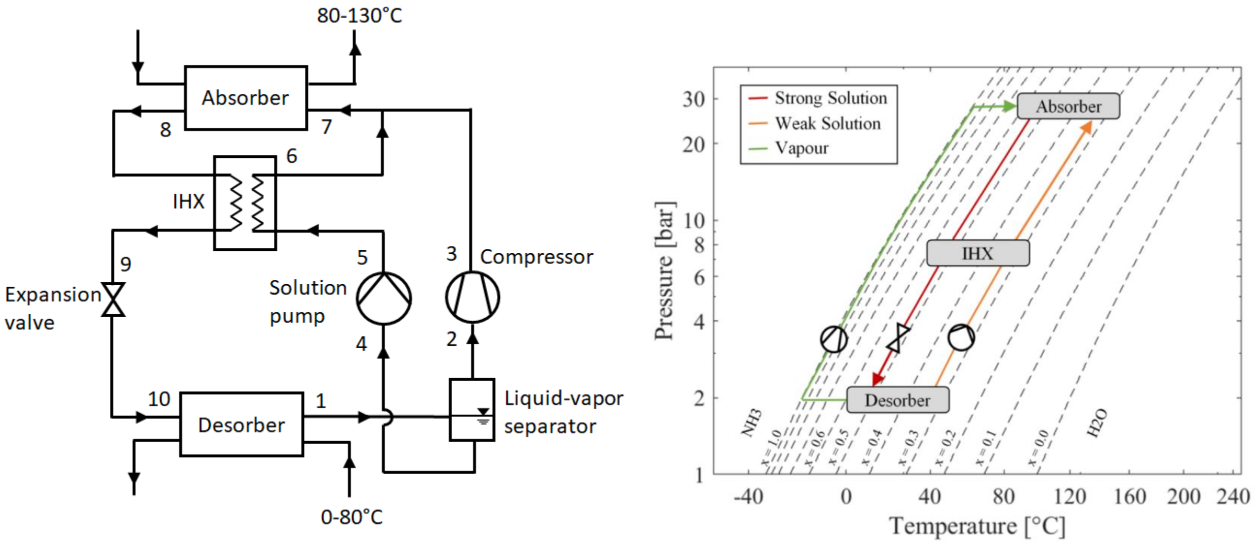


Figure 4: Simplified sketch of a ACHP system and the corresponding P-T diagram

2.3. Performance analysis

The thermodynamic performance of the different HTHP systems is evaluated using Coefficient of Performance (COP) which is defined as the ratio of useful heating capacity provided by the heat pump to the required power input, as shown in Eq.(1).

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

where $\dot{Q}_h$ is the useful heating capacity supplied by the heat pump system and $\dot{W}$ is the power consumed by the heat pump system. The Carnot COP is the theoretical maximum COP achieved by the heat pump in the given temperature range, as shown in Eq.(2).

$$COP_{Carnot} = \frac{T_{sink}}{T_{sink} - T_{source}} \qquad \text{Eq. (2)}$$

The ACHP system employs the ammonia-water zeotropic mixture as working fluid. Therefore, it rather follows the Lorenz process than the Carnot process due to the occurring temperature glides in the heat source and the heat sink side. The COP of a Lorenz cycle is defined as follows:

$$COP_{Lorenz} = \frac{T_{m,sink}}{T_{m,sink} - T_{m,source}} = \frac{T_{m,sink}}{\Delta T_{m,lift}} \qquad \text{Eq. (3)}$$

where $T_{m,source}$ and $T_{m,sink}$ are the logarithmic mean values of the heat source temperature and heat sink temperature, respectively, and $\Delta T_{m,lift}$ is the mean temperature lift, as seen in Eq.(4).

$$T_{m,sink} = \frac{T_{sink,out} - T_{sink,in}}{Ln(\frac{T_{sink,out}}{T_{sink,in}})} \qquad T_{m,source} = \frac{T_{source,in} - T_{source,out}}{Ln(\frac{T_{source,in}}{T_{source,out}})} \qquad \text{Eq. (4)}$$

2.4. System modelling

The HTHP systems are modelled based on the application scenarios and system demands presented in Figure 1 and Table 1. The COPs of the trans-critical CO_2 heat pump and NH_3 heat pump are calculated based on specific operation temperature levels using CoolPack software (Version 1.50). For the propane-butane cascade HTHP and ACHP, the COPs are determined based on the 50% of Carnot COPs and Lorenz COPs respectively, which are the achievable values for the industrial cases of ENOUGH. The lowest temperature of the heat source is used as the heat source temperature in the modelling. In terms of the condensation temperature, a minimum difference of 7 K to the maximum heat sink temperature is adopted. The overall performance of the energy system in every scenario is evaluated by combining the HTHP results on the heat supply for the cleaning and cooking processes. The operation conditions of the studied HTHP systems in different scenarios are listed in Table 2.

To simplify the comparison, similar assumptions are applied to all the HTHP systems and working fluids. A fixed isentropic efficiency of 0.7 is employed for the compressors and occurring temperature- and pressure losses are neglected. The defined loads and defined COPs are employed to compute the power demand. In each individual process, 90% of the power provided through the compressor is added to the heat on the source side. The thermal efficiency of electric- and gas boilers is assumed at 100% for the performance comparison so that the power demand relates to the heating capacity.

To investigate the performance of the studied HTHP systems on GHG emissions reduction, the CO_2 equivalent values for electricity are calculated based on Norwegian (NO) and European (EU) cases, respectively. In the NO case, a CO_2 emission factor for electricity of 50 g $CO_{2,eq.}$ kWh^{-1} is used according to the range presented by Clauß et al. (2019). In the EU case, the CO_2 emission factor for electricity is 230 g $CO_{2,eq.}$ kWh^{-1} on the basis of the EU energy consumption reported by European Environment Agency (EEA, 2021).

Table 2. Operation conditions of the working fluids of the studied HTHPs in different scenarios

	Process	Applied HTHPs	Temperature	
			Source	Sink
Scenario-A	Cleaning	ACHP	30-35°C	87-97°C
	Cooking	ACHP	80-90°C	107-127°C
Scenario-B	Cleaning	R290-R600 HTHP	30°C	97°C
	Cooking	R290-R600 HTHP	30°C	127°C
Scenario-C	Cleaning	ACHP	55-65°C	87-97°C
	Cooking	ACHP	55-65°C	107-127°C
Scenario-D	Cleaning	R290-R600 HTHP	55°C	97°C
	Cooking	R290-R600 HTHP	55°C	127°C
Scenario-E	Cleaning	Trans-critical CO_2 HTHP	0°C	13-120°C
	Cooking	Butane HTHP	90°C	127°C

3. RESULTS AND DISCUSSION

The source data for all the presented results can be accessed via the link shown at the end of the refence list.

The comparison results of the calculated COPs, energy savings and the utilized surplus heat rate for the studied HTHP systems in the different application scenarios are presented in Figure 5-7. The temperature lift discussed in this study is defined as the difference between the heat sink outlet temperature and heat source outlet temperature.

In both Scenario-A and Scenario-B, the HTHPs upgrade the heat from 30-35°C to the temperature level of 80-90°C for the cleaning process, while in Scenario-C and Scenario-D, the heat required for the cleaning is supplied by the HTHPs from the heat source at 55-65°C. Therefore, the same HTHPs shows better thermal performance, i.e. a higher COP, in Scenario-C and Scenario-D than in Scenario-A and Scenario-B due to the lower temperature lifts, as shown in Figure 5-6. Moreover, in Scenario-A and Scenario-B, the ACHP and propane-butane cascade HTHP have the same COP values. While in Scenario-C and Scenario-D, when the heat source temperature increases and temperature lift decreases, the ACHP system has the higher COP than the propane-butane cascade HTHP. This is due to the differences in heat transfer, i.e. compared with a vapor compression heat pump using azeotropic working fluid, the thermal performance of the ACHP which employs ammonia-water zeotropic mixture as the working fluid which improves the condensation in the absorber, especially when the temperature glides of the heat source and heat sink are larger. However, as the temperature lift increases, the performance of the Lorenz cycle approaches the Carnot cycle and the advantage of the ACHP over the vapor compression heat pump declines. Therefore, in Scenario-A and Scenario-B, the two HTHPs exhibit the same thermal performance in the heat supply for the cleaning process.

In Scenario-E, a trans-critical CO_2 HTHP is employed to supply both the cooling and heating load and its COP is slightly higher than the COPs of ACHP and propane-butane cascade HTHP in Scenario-A and Scenario-B, which means the trans-critical CO_2 heat pump has better thermal performance at higher temperature lift compared with the other two HTHPs when producing hot water by heating it from 13 °C to 120 °C. To supply the heat sink temperature of 90°C, the compressor discharge pressure of the trans-critical CO_2 HTHP is around 125°C and the pressure ratio is 3.8. In Scenario-E, the trans-critical CO_2 HTHP cannot be directly used to supply heat for the cooking process due to the high temperature requirement. To achieve the high heat sink temperature of 120°C, the compressor discharge temperature is about 155°C which is at the upper limit of the compressor operating temperature. In addition, the trans-critical CO_2 HTHP is not applicable to Scenario-A-D because of the low critical temperature of CO_2, i.e. it fits perfect for hot water production and low inlet temperatures of the cold water.

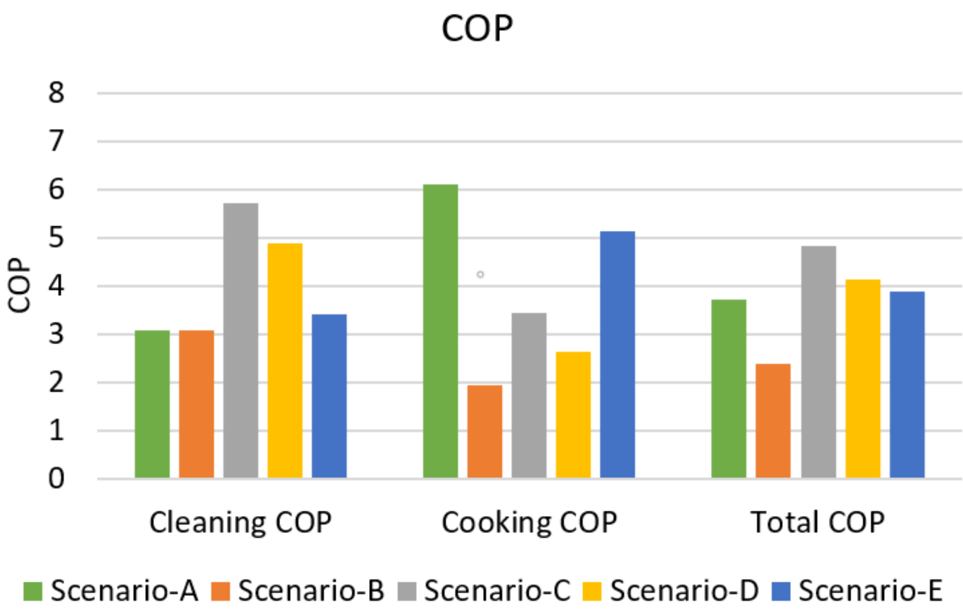


Figure 5: The results of COPs of studied HTHP systems in different scenarios

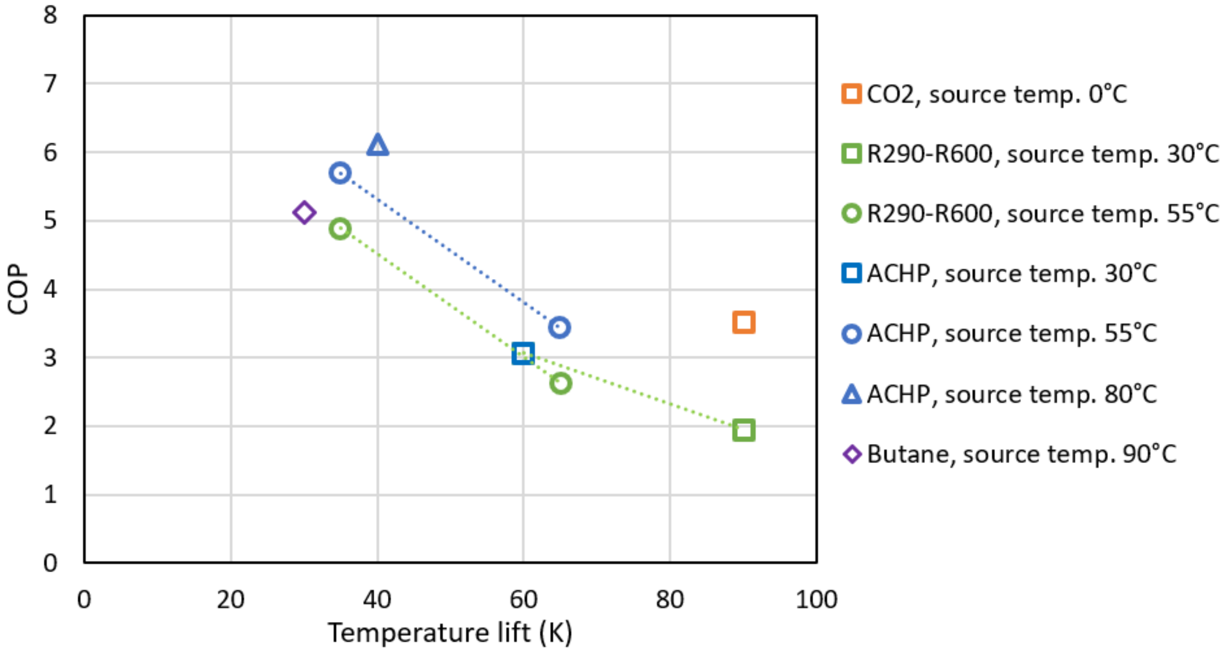


Figure 6: The COPs of the studied HTHP systems with temperature lift at different heat source temperatures (only the lowest heat source temperature is presented for the ACHP system)

For the cooking heat supply, the ACHP obtains the highest heating COP of 6.1 in Scenario-A because of the low temperature lift required and high heat sink temperature glide, as illustrated in Figure 6. The COP of the ACHP in Scenario-C is lower than it in Scenario-A and even though the performance of the ACHP in Scenario-C is better than the propane-butane cascade HTHP in Scenario-D, the advantage is not very remarkable, which further proves that the decrease of temperature lift can significantly improve the thermal performance of ACHP. When the temperature lift decreases to 25 K, the COPs increase 59 % and 78 % for the heat sink outlet temperature of 90 °C and 120 °C respectively. Furthermore, to achieve the 120°C heat sink outlet temperature, a moderate compressor discharge pressure of 25 bar is required for the ACHP with a relatively low pressure ratio of 3.6.

Compared with the performance in Scenario-B, the COP of the propane-butane cascade HTHP is slightly higher in Scenario-D when the temperature lift is lower. The heat sink temperature of 120 °C can be achieved with compressor discharge temperature of 136 °C and a discharge pressure of 24.5 bar. The evaporation and condensation processes in the propane-butane cascade HTHP are isothermal so that it is appropriate for the applications with the requirement for a constant heat sink temperature. However, owing to the high flammability level of propane and butane, it is necessary to optimize the charge of the working fluids and extra safety assessments should be conducted for specific circumstances, which can be mitigated by following national and international standards. Due to the high heat source temperature as well as the constant heat source/sink temperatures, a single butane cycle is used for the cooking heat supply in Scenario-E and exhibits good performance for a 30 K temperature lift.

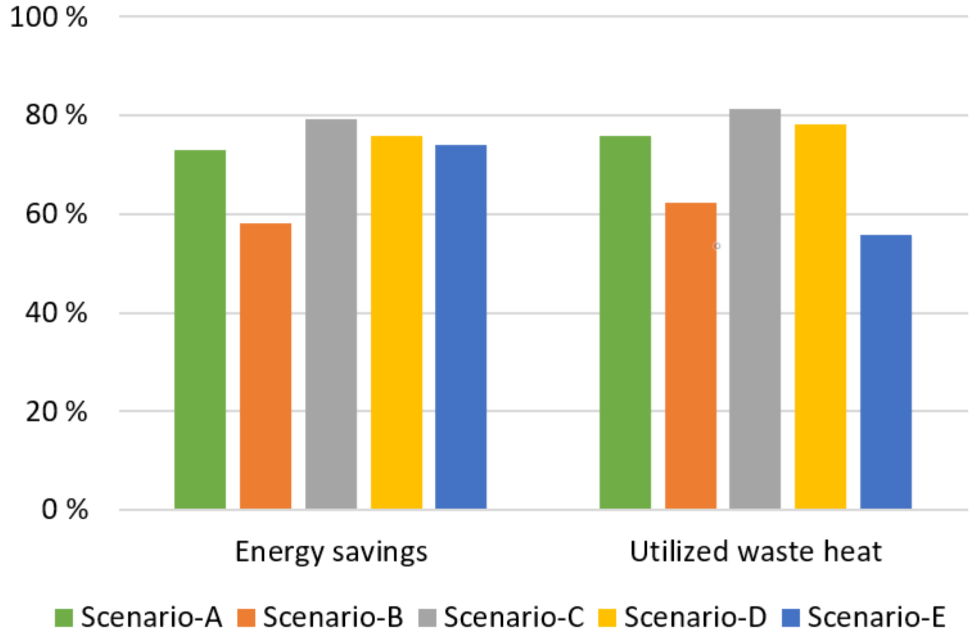


Figure 7: The energy savings compared with using a gas/electric boiler and the proportion of the utilized waste heat in the total heating load of the studied HTHP systems in different scenarios

From the calculation results, the ACHP exhibits the best performance in both the cleaning and cooking processes and is the most promising solution for the steam boiler replacement among all the studied HTHP systems. Even though the toxicity of ammonia should be taken into the consideration especially for the applications in the food processing, ammonia is widely used in industrial cooling processes and the toxic issue can be addressed by trained operators and when strictly following national and international standards. The compressor discharge temperature is the major constrain for the application of the ACHP for the heat sink temperature supply higher than 130 °C. As this circumstance, the steam production with the ACHP is difficult to be achieved using commercially available components. The system optimizations such as using multi-stage compression or liquid injection could be effective manners to further increase the temperature lifts and heat sink discharge temperature (Ahrens, Loth, et al., 2021).

On the energy system level, the total performance of Scenario-C is the most efficient and has the highest emission reduction potential, because the use of the NH_3 heat pump increases the system integration as well as decreases the required temperature lifts, as shown in Figure 5. The total COP of Scenarios-D is slightly lower than Scenario-C due to the better performance of the ACHP than the propane-butane cascade HTHP in the same operation condition. Scenario-A has the better performance than Scenario-B because of the higher system integration level. The heat at the temperature level of the cleaning process is stored in a water tank and used as the heat source of the second ACHP to produce heat for cooking. As shown in Figure 7, the heating system in Scenario-C can recover more waste heat than the systems in the other scenarios when consuming the same amount of electric power. Compared with using gas or electric boilers to provide heat for the cleaning and cooking processes, a significant amount of energy can be saved by employing the studied HTHPs. In Scenario-C, using the ACHP for both the cleaning and cooking processes, the achieved energy saving rate is 79 % while a energy saving rate of 58 % is achieved by utilizing the trans-critical CO_2 HTHP for hot water production in Scenario-E. The higher COP results in the lower power demand and higher waste heat recovery. The GHG emissions reduction is evaluated for the studied HTHP systems based on the Norwegian (NO) and EU cases. Table 3 illustrates the CO_2 emissions reduction for one-hour period in different application scenarios compared with the CO_2 emissions generated by gas/electric boilers. The CO_2 emissions reduction in Scenario-C is more than twice the values in Scenario-B.

Table 3. CO_2 emissions reduction achieved in different scenarios

	$NO_{,el}$ (kg $CO_{2,eq.}$ kWh^{-1})	$EU_{,el}$ (kg $CO_{2,eq.}$ kWh^{-1})
Scenario-A	521	2399
Scenario-B	290	1336
Scenario-C	710	3265
Scenario-D	658	3028
Scenario-E	380	1740

4. CONCLUSIONS

In the present study, the thermal performance of three heat pump systems — a trans-critical CO_2 heat pump system, a cascade propane-butane system and an ammonia-water ACHP system, for hot water and steam production is evaluated based on the defined demand cases in different application scenarios. In all the scenarios and temperature lift levels, the ACHP exhibits the best thermal performance with moderate pressure levels and low-pressure ratios. The highest heating COP achieved by the ACHP is 6.1 with the heat sink outlet temperature of 120 °C and temperature lift of 40 K. The application of the trans-critical CO_2 HTHP is constrained by its low critical temperature, but its performance is still good when heat source temperature is low and there is a large heat sink temperature difference, i.e. low cold water inlet temperature. The propane-butane cascade HTHP exhibits good performance in the scenarios with low temperature lifts and it is appropriate for the applications with demands for constant heat source and sink temperatures.

On the energy system level, the total performance of Scenario-C is better than in the other scenarios because the use of the NH_3 heat pump increases the system integration and decreases the required temperature lifts. Therefore, the achieved energy savings and GHG-emission reduction of Scenario-C are also higher than in the other scenarios. In Scenario-C, using the ACHP supplying heat for both the cleaning and cooking processes, the achieved energy saving rate can reach up to 79 % compared to traditional boiler systems.

From the results of the present study, the HTHPs exhibit excellent thermal performance on hot water and steam production and are a competitive technology for steam boiler replacement in the food industry. More specific studies and demonstrators should be conducted to optimize the HTHP systems for specific application requirements and further improve the performance as well as the energy efficiency.

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