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Analysis of possibilities to utilize excess heat of supermarkets as heat source for district heating

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Abstract

The paper analyses the possibilities to recover excess heat from CO_2 supermarket refrigeration plants for supply to district heating networks. It was analyzed to operate the refrigeration system at an increased gascooler pressure to directly supply heat and to install a cascade heat pump to recover the heat from lower temperatures. Increasing the gascooler pressure appeared promising during summer, while the cascade heat pump showed higher COPs during colder periods. The investment for the cascade heat pump could be compensated within 4 years at district heating prices above 37 €/MWh.

Keywords: CO₂; District Heating; Excess Heat Recovery; Supermarket Refrigeration; Sector Coupling

1. Introduction

The advancements in district heating (DH) technologies are characterized by decreasing supply temperatures. The 4th generation of DH is described by supply temperatures between 40 °C to 70 °C, [1,2]. The 4th generation of DH

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comprises low-temperature district heating (LTDH) and ultra-low-temperature district heating (ULTDH) networks. LTDH networks are designed with a supply temperature around 60 °C, which is just high enough for direct heat supply for domestic hot water. ULTDH networks operate at around 40 °C and are designed for direct supply of space heating, while the domestic hot water production requires a booster heat pump at the customer site.

The boundary conditions determine, which of the solutions constitutes the thermodynamic and economic optimum. Several studies have therefore compared various DH technologies under consideration of different conditions, such as consideration of existing network components, availability of heat sources, network layouts and other aspects [2–5].

Since the availability of heat sources for heat pump based district heating networks represents a key aspect with respect to their performance, different works in this field were carried out. Bühler et al. [6–8] have conducted economic analyses of the possibilities to use industrial excess heat from thermal processes for district heating. They considered spatiotemporal aspects and ensured the economic feasibility by comparisons to alternative heat sources for the DH network and to energy-efficiency measures, which would decrease the availability of industrial excess heat. Their studies outlined the general potential of excess heat recovery for DH purposes but also emphasized the importance of beneficial boundary conditions, such as short distances between heat source and customers and low integration cost.

Lund et al. [9] indicate, that the refrigeration systems of supermarkets represent another potential heat source for DH systems. These are rejecting excess heat to the ambience throughout the year and are typically located in close vicinity to areas with increased heating demands. While conventional refrigeration systems with e.g. R-404a, reject heat during a subcritical condensation just above the ambient temperatures, CO_2 systems reject the heat to a relatively large extent significantly above the ambient temperatures during desuperheating or transcritical gascooling processes. The increased temperature differences represent irreversibilities of CO_2 systems and thereby a potential for improvements.

CO_2-based systems constitute the state of the art technologies in cold and moderate climates and are emerging as well in warmer climates [10–12]. Especially fully integrated systems, which integrate the air conditioning and heating requirements into the system while exploiting the peculiarities offered by the refrigerant properties of CO_2, show convincing performances and an increased utilization of existing equipment. Most of the studies focus on the heat recovery to cover demands for space heating and hot water preparation on-site or in close vicinity [10–16].

The studies demonstrated the possibilities for integrating on-site heat demands in the refrigeration system. The examples indicated high performances for cases, in which the heat demand and its temperatures matched well with the available heat. On the other hand, highly integrated systems often require storage tanks for balancing the temporal shift between availability and demand. The amount of available excess heat can furthermore exceed the heat demand.

LTDH networks represent an alternative potential customer of low-temperature excess heat. It was therefore suggested by e.g. [9,17,18] to couple the supermarket refrigeration units to LTDH networks. LTDH networks constitute a flexible customer of the available heat and the supermarkets are representing a decentral heat source for LTDH networks, often implying small distances and in many cases an existing connection to the DH networks, [7].

This study therefore focuses on an analysis of the possibilities for the recovery of excess heat from CO_2 supermarket refrigeration units and evaluates the boundary conditions, for which the sector coupling appears economically feasible.

2. Methods

2.1. Possible system layouts

The following subchapters present the different possibilities to recover excess heat from supermarkets for supplying heat to a LTDH network while heating the return stream from 45 °C to 70 °C. The approaches are compared on an annual basis assuming average ambient temperatures of 20 °C, 9 °C and 1 °C for summer, spring/autumn and winter, respectively, and for an average cooling demand of 100 kW throughout the year.

2.1.1. Supply of excess heat during operation at increased gascooler pressure
Figure 1 a) shows the layout of a simple refrigeration plant with one evaporator, an internal heat exchanger, an ejector and the possibility for direct heat recovery. The gascooler pressure and thereby the temperature profile of CO_2 during heat rejection is during normal operation determined by the ambient temperature, e.g. [12,19]. Consequently, the temperature profile and thereby the amount of excess heat that is directly recoverable, varies as well.

The heat that is directly recoverable during normal operation is often rather limited. The gascooler pressure can alternatively be operated at an increased pressure to not only cover the cooling demand of the supermarket but also to actively supply heat to the DH network. Both the amount of heat that is recovered as well as the COP for supplying the heat are dependent on the additional pressure increase [15]. The contributions increase nonlinearly and suggest that there is an optimal marginal pressure increase, yielding a maximum COP.
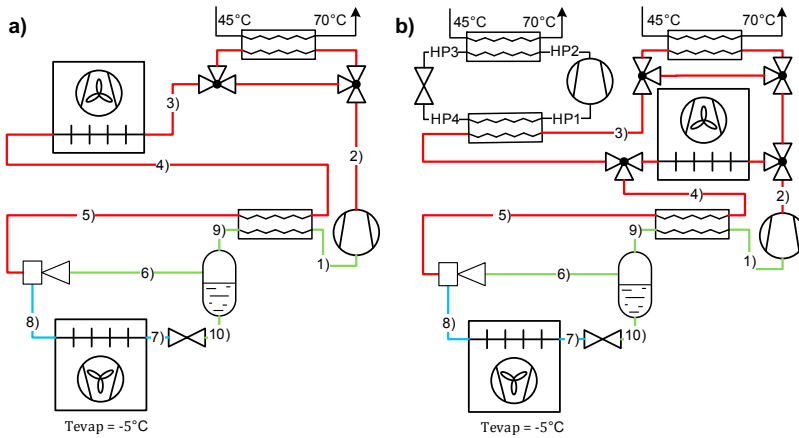

Figure 1: Flow sheets for a standard CO_2 refrigeration system with an ejector and one evaporation temperature with a) direct heat exchange at operation with increased gascooler pressure, b) direct heat exchange and a cascade heat pump in serial connection

Table 1: Assumptions for modelling the CO_2 refrigeration system and the cascade heat pump

	CO_2 System	Cascade HP
Isentropic efficiency of compressor:	80 %	75 %
Heat loss from the compressor:	5 %	5 %
Efficiency of motor:	95 %	95 %
Lumped isentropic efficiency of ejector:	25 %	-
Effectiveness of internal heat exchanger:	70 %	-
Pinch point temperature differences in HXs:	5 K	5 K
Minimum gascooler outlet temperature:	10 °C	-
Evaporation temperature:	-5 °C	-

In order to analyze the trade-off among additionally supplied heat and the performance, a numerical model was implemented in EES, [20]. The model consists of mass and energy balances for all components, as well as equations modeling their performance. Table 1 summarizes the assumptions for the model of the refrigeration system. The gascooler outlet temperature is determined by the ambient temperature and a pinch point temperature difference of 5 K while the gascooler pressure was set to optimal conditions with respect to cooling COP_c. The coefficient of performance for cooling $COP_c = \dot{Q}_c / \dot{W}_{T_{amb},opt}$ is defined as the ratio of the cooling rate $\dot{Q}_c$ to the power utilized to operate system optimally according to the ambient temperature $\dot{W}_{T_{amb},opt}$. The compressor was modelled with an isentropic efficiency for the compression process and an additional efficiency for the motor. Heat losses from the compressor were considered as well.

The model assumes that all heat is rejected by the gascooler to the ambient during standard operation. In order to supply heat to the DH network, the compressor discharge temperature must be high enough to enable cooling of the CO_2 while heating the DH stream from 45 °C to 70 °C and respecting a minimum pinch point temperature difference of 5 K. After the direct heat exchange, the CO_2 is further cooled to the gascooler outlet temperature that is defined by the ambient temperature and the pinch point temperature difference. If the compressor outlet temperature is too low for heating the stream up to 70 °C, the entire heat is rejected through the gascooler.

The performance of supplying the additional heat rate was described by a coefficient of performance $COP_{HS,CO2}$, which is defined by the ratio of the heat supplied in the direct heat exchanger $\dot{Q}_{dHX}$ to the additional power that is used to operate the refrigeration system at an increased gascooler pressure. The additional power is defined by the difference between the power during operation at the optimal pressure according to the ambient conditions $\dot{W}_{Tamb,opt}$ and the power during operation at an increased gascooler pressure $\dot{W}_{comp}$ while maintaining a constant cooling load.

$$COP_{HS,CO2} = \frac{\dot{Q}_{dHX}}{\dot{W}_{comp} - \dot{W}_{T_{amb},opt}} \tag{1}$$

The trends for the additional power, the additional heat rate and the COP will be studied, and a compromise will be chosen as the reference mode for the comparisons.

2.1.2. Supply of excess heat using a cascade heat pump

The heat that is rejected to the environment because it is below the minimum supply temperature can alternatively be used as a heat source for a cascade heat pump, which supplies the district heating network, too. Figure 1 b) shows the system layout and Table 1 summarizes the assumptions for modelling the cascade heat pump.

Utilizing a cascade heat pump enables choosing the gascooler pressure of refrigeration system independently of the ambient conditions. A parameter variation of the gascooler pressure showed that the system performance is optimal for subcritical heat rejection at a condensation pressure of 70 bar, corresponding to a temperature of 28.7 °C. At this operation, the compressor discharge temperature was just high enough to enable direct heat supply.

The coefficient of performance for heat supply in the cascade heat pump scenario $COP_{HS,HP}$ was defined by the ratio of the accumulated supplied heat rate over the power that is additionally invested compared to standard operation for supply of cooling. The supplied heat includes both the heat flow supplied by the cascade heat pump $\dot{Q}_{HP}$ and the direct heat exchange $\dot{Q}_{dHX}$. The total power is composed by the compressor power of the heat pump $\dot{W}_{HP}$ and the additional power spent in the refrigeration system to operate at a condensation pressure of 70 bar instead of the optimal condensation pressure corresponding to the ambient temperature ($\dot{W}_{p_{cond}=70\ bar} - \dot{W}_{T_{amb,opt}}$).

$$COP_{HS,HP} = \frac{\dot{Q}_{HP} + \dot{Q}_{dHX}}{\dot{W}_{HP} + \left(\dot{W}_{p_{cond}=70\ bar} - \dot{W}_{T_{amb,opt}}\right)} \tag{2}$$

2.2. Optimization of the cascade heat pump working fluid

This study assumed that the heat supply is realized in a heat exchanger heating the DH stream from 45 ° to 70 °C while respecting a pinch difference of 5 K. This means that the source inlet stream to the evaporator of the cascade heat pump enters at 50 °C, is cooled down to the condensation temperature of approximately 29 °C and condenses at a constant temperature. This temperature profile is nonlinear and it is expected, that a certain mismatch compared to the temperature profile of the cascade heat pump working fluid that evaporates at a constant temperature occurs.

The authors have shown in previous studies that utilizing mixtures constitutes a promising approach to match the temperature profile of the working fluid during evaporation and condensation to the temperature profile of heat sink and source, [5,21,22]. This approach enabled reducing the irreversibilities during heat transfer and thereby improving the performance. The previous studies focused on selecting the working fluid for boundary conditions in which the heat source and sink had a linear temperature glide, while the temperature profile of CO_2 is expected to be nonlinear.

A working fluid screening was conducted as described in [5,21,22]. The model for the cascade heat pump was therefore evaluated for several mixtures, considering all possible binary mixtures of a list of fluids including 14 natural fluids and 4 hydrofluoroolefins (HFOs). The results were analyzed with respect to the thermodynamic performance and to indicators of the investment, such as the volumetric heating capacity VHC or the pressure levels.

2.3. Economic Evaluation

The economic performance is evaluated according to the Danish boundary conditions. The production of district heating with a heat pump includes two key taxes in Denmark. Firstly, there is the excess heat tax which concerns the utilization of any excess heat related to process equipment. It is implemented to counteract tax exemptions that are given to the production of goods by companies. For heat pumps, only the amount of heat produced above a COP of 3 is taxed, and the taxation is 25 % of the heat sale, up to a maximum of 7.40 €/GJ.

$$\text{Excess Heat Tax} = 25\ \% \cdot \left(\frac{COP_h - 3}{COP_h}\right) \cdot \text{Heat Sales} \tag{3}$$

The second taxation of interest is the electricity to heat tax. Whenever electricity is used to produce DH there is a surplus tax compared to when electricity is used for process equipment. This will however be reduced in 2020 [23], and the Public Service Obligations (PSO) which are currently imposed on all electricity will also be removed. Together these contributions will drop the current electricity price from ~134 €/MWh to ~94 €/MWh. Heat prices vary a lot

both with regards to geography and season, from the low end which is 27 €/MWh to the gas substitution price of up to about 60 €/MWh, at specific places even higher.

In order to estimate the investment of the cascade heat pump, a specific investment cost per unit of supplied heat of 671 €/kW is assumed [24]. The value of 671 €/kW was related to the construction of industrial heat pumps, which imply a relatively high cost due to case specific engineering and manufacturing. It is assumed that the development of a standard unit could reduce the cost for manufacturing and on-site integration to 400 €/kW, as it can be observed for domestic heat pumps [25]. Estimating the investment cost when using mixtures appears to be more difficult due to the lack of practical experiences with the construction and operation of heat pumps using zeotropic mixtures. Previous studies, [5,22], suggest initial investment costs of competitive solutions using mixtures, comparable to the investment costs for heat pumps using pure fluids or up to 100 % higher. The studies outline the strong dependency on the specific cases. It may furthermore be noted, that the above-mentioned costs included both the cost for acquisition and integration of the heat pump, while the heat pump using mixtures requires the same cost for the integration. For a simplified assessment of the economic performance of a modular cascade heat pump unit, specific investment costs per unit of supplied heat of 600 €/kW were assumed. The cost for connecting the refrigeration system directly to the DH network were estimated with 17,000 €, based on a realized project and neglecting the influence of the capacity. The connection cost for the cascade heat pump solutions are assumed to be included in the above-mentioned estimates. The analysis included furthermore a maintenance cost per unit of supplied heat of 1.60 €/MWh for all scenarios using a cascade heat pump.

The solutions were compared based on the total annual cash flow, which includes the income from the heat sales, all costs for electricity and taxes as well as the levelized investment cost. The comparison was based on a capital recovery factor of 0.08, assuming a lifetime of 20 years and an effective interest rate of 5 %. Lastly, a simple payback time for the investment was calculated by relating the annual net income of each solution to its initial investment cost.

3. Results

3.1. Thermodynamic comparison of heat supply technologies

The first option for supplying heat to a DH network was to operate the supermarket refrigeration system with an increased gascooler pressure to use part of the heat from desuperheating for heating the DH stream from 45 °C to 70 °C, while the remaining part of the heat is rejected to the environment. The results from e.g. [15] have shown the nonlinear dependency between the pressure increase and the additionally recoverable heat and indicated the existence of an optimal COP with respect to heat supply performance.

The first part of the analysis consisted therefore of an analysis of the performance for supplying heat for different gascooler pressures while taking different ambient conditions as reference scenario. Figure 2 shows the trends for the different energy flows and the performance for supplying heat COP_{HS} for the ambient conditions for a) summer, b) autumn/spring and c) winter.

The optimal pressure during normal operation for supply of cooling is defined by the saturation pressures of the gascooler outlet temperature that is determined by the ambient temperature and the assumed pinch point temperature difference. The optimal gascooler pressure for refrigeration operation is 45 bar during winter, 50 bar during autumn/spring and 64 bar during summer. The amount of power required to lift the pressure increases steadily, while the heat flow rates show discontinuities. The heat flows show that the pressure has to be increased up to 80 bar during winter, up to 78 bar during autumn/spring and up to 68 bar during summer to obtain sufficiently high compressor discharge temperatures, enabling direct heat transfer to the network. The absolute pressures that enable heat supply varied depending on the season, due to the different superheating from the internal heat exchanger.

The amount of power that needs to be additionally invested to reach the point at which the direct heat supply is possible, when assuming the optimal operation for supply of cooling as a reference point, is significantly larger during winter and autumn/spring than during summer. The amount of heat that is supplied in the case of just high enough temperatures is in all cases around 33 kW for an average cooling capacity of 100 kW. This yields high COP_{HS} during summer, mainly due to the small amount of additionally required power for the relatively small pressure increase. For the cases of autumn/spring and winter, it may be noted that the COP_{HS} are lower, due to the lower reference pressure of optimal operation for cooling supply.

Increasing the gascooler pressure further, yields an increase of both the amount of supplied heat and the COP before the COP_{HS} shows an optimum around 120 bar for autumn/spring and winter conditions. The amount of supplied heat can be increased even further with higher pressures, while the performance decreases. Under summer conditions, the trend looks different. Above the point at which heat recovery is possible, the performance is high and drops significantly with an increasing pressure, while reaching a plateau between 90 bar and 110 bar, while the amount of supplied heat increases.

a) Summer: $T_{ambient} = 20\ °C$ b) Autumn/Spring: $T_{ambient} = 9\ °C$ c) Winter: $T_{ambient} = 1\ °C$

Supplied heat in direct HX — Rejected heat in gascooler — – – Total rejected heat
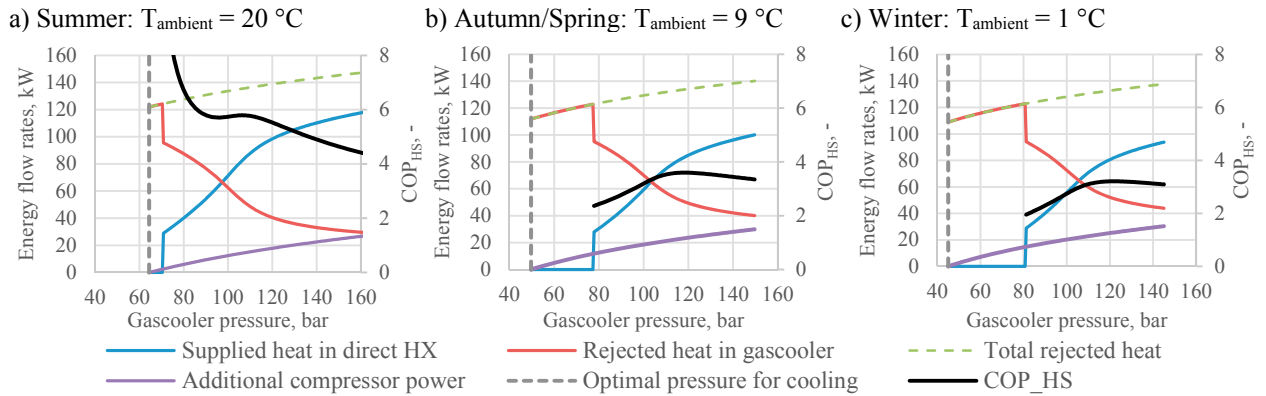Additional compressor power — – – – Optimal pressure for cooling — COP_HS

Figure 2: Comparison of the additional power in the refrigeration system ($\dot{W}_{comp} - \dot{W}_{Tamb,opt}$) and the heat flows rejected to environment in the gascooler and to the DH in the direct heat exchanger for a cooling capacity of 100 kW

Table 2: Performance of different scenarios at different boundary conditions for a cooling load of 100 kW

	Operating the refrigeration system at increased pressure			Cascade Heat Pump		
	Summer	Autumn/Spring	Winter	Summer	Autumn/Spring	Winter
$T_{ambient}$	20.0 °C	9.0 °C	1.0 °C	20.0 °C	9.0 °C	1.0 °C
CO2 refrigeration system:						
$T_{gascooler,exit}$	25.0 °C	14.0 °C	10.0 °C	28.7 °C	28.7 °C	28.7 °C
Optimal gascooler pressure for cooling:	64.3 bar	50.0 bar	45 bar	64.3 bar	50.0 bar	45 bar
Design gascooler pressure for heat supply:	105.0 bar	118.6 bar	120.8 bar	70 bar	70 bar	70 bar
Additional power ($\dot{W}_{comp} - \dot{W}_{Tamb,opt}$):	14.1 kW	23.3 kW	25.3 kW	5.1 kW	15.8 kW	18.9 kW
Heat supply in direct HX $\dot{Q}_{dHX}$:	81.4 kW	83.8 kW	81.2 kW	32.6 kW	32.6 kW	32.6 kW
Cascade heat pump system:						
Compressor power $\dot{W}_{HP}$:	-	-	-	23.5 kW	23.5 kW	23.5 kW
Heat supply by cascade HP $\dot{Q}_{HP}$:	-	-	-	116.5 kW	116.5 kW	116.5 kW
COP of cascade heat pump:	-	-	-	4.9	4.9	4.9
Sum of supplied heat:	**81.4 kW**	**83.8 kW**	**81.2 kW**	**149.1 kW**	**149.1 kW**	**149.1 kW**
COP for heat supply COP_{HS}:	**5.8**	**3.6**	**3.2**	**5.2**	**3.8**	**3.5**

The second considered option for supplying heat to a LTDH network consisted of a cascade heat pump that recovers the part of the heat from the CO_2 system, which is rejected at temperatures that are too low for directly supplying the heat. A parameter variation has shown, that the system performance considering both the heating and the cooling of the entire system is maximal, when the gascooler of the CO_2 refrigeration system is operated at 70 bar, corresponding to subcritical conditions.

Table 2 shows a summary of the performance of the two heat supply technologies. The cascade heat pump using Ammonia as a working fluid operates with a COP of 4.9 and supplies 117 kW, while 33 kW are supplied to the DH

network by direct heat exchange. The additional power that is consumed in the refrigeration system to operate at 70 bar is higher during winter. This results in values for the coefficient of performance for heat supply COP_{HS} of 3.5 in winter, 3.8 in autumn/spring and 5.2 at summer conditions. While the cascade heat pump shows COP_{HS} that are higher than the solution supplying the heat directly with an increased gascooler pressure during winter and autumn/spring, the COP_{HS} is lower at summer conditions. The total amount of supplied heat is almost twice as large for the cascade heat pump solution throughout the year.

3.2. Working fluid optimization of cascade heat pump

In the arrangement of the cascade heat pump, the CO_2 stream entered the evaporator at approximately 50 °C to evaporate the pure working fluid of the cascade heat pump at around 24 °C. The heat transfer among this enlarged temperature difference corresponds to irreversibilities and constitutes the potential improvements that can be obtained by selecting a mixed working fluid that changes the temperature during evaporation and thereby better matches the temperature profile of the CO_2 during heat rejection.

Table 3 shows the selected screening results and includes different pure fluids with a competitive COP, while Ammonia shows the largest volumetric heating capacity VHC, indicating the most compact compressor. The results show furthermore, that the mixtures can achieve COP as high as 5.7, corresponding to an increase of 15 %.

Figure 3 shows the temperature-heat-diagram for the thermodynamic cycle of Ammonia (left) and 90 % DME / 10 % Hexane (right) and the direct heat transfer. The CO_2 stream cools down while heating part of the DH stream, before cooling further down and condensing while heating the working fluid of the heat pump cycle. Ammonia evaporates and condenses at a constant temperature while the mixture changes temperature in the two-phase zone. Due to the good match between the mixed working fluid of the cascade heat pump and the heat source and sink, the irreversibilities during heat transfer are decreased and the overall performance is improved.

Table 3: Performance of different pure and mixed working fluids for the cascade heat pump

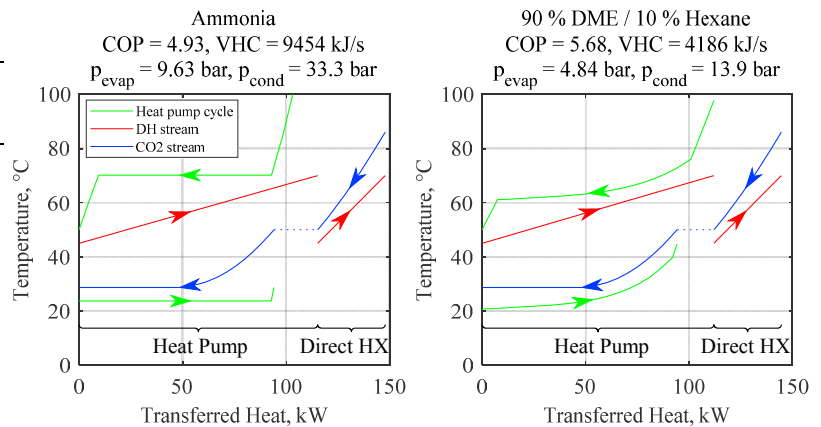| Working Fluid	COP	VHC
	-	kJ/m³
Propane	4.79	6,519
DME	4.96	4,923
Butane	4.92	2,179
R-1234ze(E)	4.84	4,097
Ammonia	4.93	9,454
90 % DME / 10 % Hexane	5.68	4,186
80 % DME / 20 % DEE	5.48	3,831
10 % Hexane / 90 % R-1234ze(Z)	5.65	1,327

Figure 3: Temperature heat diagram of cascade heat pump working with Ammonia (left) and 90 % DME / 10 % Hexane (right) incl. direct HX

The cascade heat pump using the pure working fluid ammonia consumed 23.5 kW of electrical power for supplying 116.5 kW to the DH network, which corresponds to a COP of 4.93. Utilizing the mixture of 90 % DME / 10 % Hexane, the required power can be reduced to 19.7 kW while supplying 112.0 kW of heat, corresponding to a COP of 5.68.

3.3. Economic evaluation of heat supply technologies

An economic comparison for the case of a refrigeration unit with an average cooling capacity of 100 kW is shown in Figure 4. The approach to operate the CO_2 refrigeration system at an increased gascooler pressure is compared to investing in a cascade heat pump that will work at a higher COP and be able to deliver more heat. The total annual cash flows include the operating cost, the income from the heat sales and the levelized investment cost and correspond

to the annual surplus. The annual cash flows of the options using the cascade heat pump exceed the cash flows for the option to operate the refrigeration system at an increased gascooler pressure for heat prices above 35 €/MWh. The total annual cash flows of the different cascade heat pump solutions are relatively similar. The Danish excess heat taxation impacts the mixture heat pump negatively as the tax per supplied MWh is highest, when delivered at a higher COP. The taxes on the heat sales account for 12 % of the heat sales for the heat pump using a mixture, while accounting for 9 % for the other solutions with a lower COP. The total electricity cost includes 22 % of taxes for all solutions.

The investment cost of the ammonia heat pump was firstly estimated based on the specific cost per supplied heating capacity of 671 €/kW as specified in section 2.3, and the heat pump sizing in Table 2, which results in a total investment of 78,000 €. Considering the development of a modular standard unit, a decrease in manufacturing cost, as well as in cost for installation can be expected. Assuming that the specific investment cost per unit supplied heat can be decreased to 400 €/kW yields an investment of 46,600 €. Assuming a lifetime of 20 years and an effective interest rate of 5 %, the investment costs correspond to an annual cash flow of 6,260 €/year and 3,740 €/year, respectively. Assuming a specific investment cost of 600 €/kW for a standard unit using the mixture 90 % DME / 10 % Hexane, the total investment is expected to be 67,200 €, corresponding to an annual cash flow of 5,390 €/year. The investment for the establishment of the DH substation for the case in which the refrigeration system is directly connected to the DH network was estimated with 17,000 €, corresponding to an annual cash flow of 1,360 €/year.
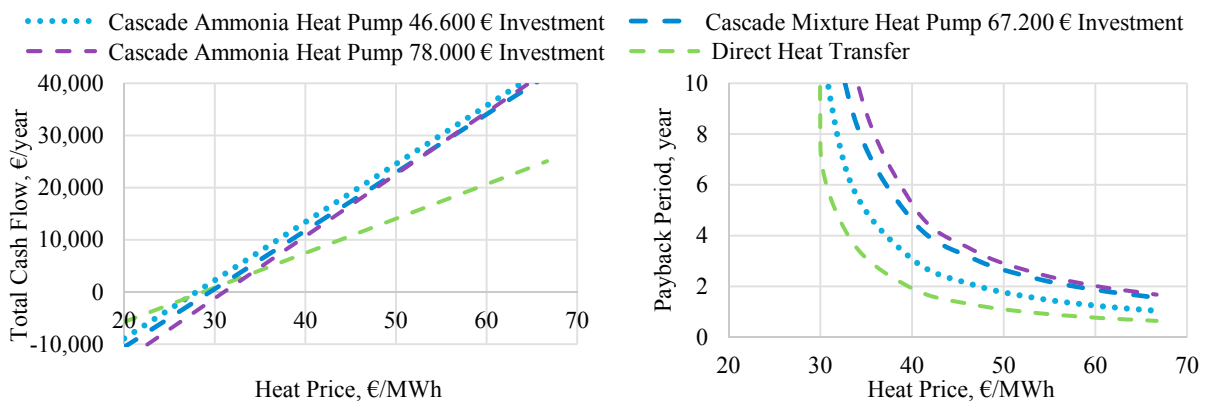

Figure 4: Comparison of total cash flows (left) and simple payback time (right) for the different heat supply technologies

A simple payback period was calculated comparing the investment cost for each solution to its annual income. The second diagram of Figure 4 shows that the lowest payback periods are obtained for the solution in which the heat is directly transferred from the refrigeration system. The payback times for this solution drop below 4 years for 33 €/MWh and reach values below 2 years for heat prices above 40 €/MWh. The solutions using a cascade heat pump show longer payback periods while the standard unit using Ammonia approaches the payback periods of the direct heat recovery. Other performance indicators that account for the entire life time, such as the heat generation cost or the net present value indicate a higher profitability of the cascade heat pump solutions, which is consistent with the findings from [5,22]. The estimation of the investment cost for the establishment of the DH connection implied uncertainties, which have the largest impact on the solution with the direct heat supply from the refrigeration system.

4. Discussion

Both of the presented solutions imply advantages and disadvantages and the considerations as well as uncertainties. The cascade system has shown different beneficial effects, such as the supply of heat to the DH network, the potential to flexibly consume electricity and to improve the performance of the refrigeration system. It may therefore be concluded that such a setup makes sense from a socioeconomic perspective, while the distribution of the benefits to different parties raises difficulties for viable business models. It could be possible, that either the utility company owns and operates the cascade heat pump and is compensated for providing access to a heat sink for the refrigeration system by the supermarket or the supermarket owns the cascade heat pump and sells the heat to the DH network.

In order to conclude further an evaluation of the possibilities to use supermarkets as heat source for DH networks from a socioeconomic point, a comparison to a conventional operation of the DH network and the supermarket refrigeration system would have to be conducted. A central heat pump using the ambient as a heat source has to cover a slightly larger temperature difference but might have better efficiencies due to larger capacities.

It may furthermore be noted, that the external preconditions for acting as a heat supplier are more beneficial during the winter time. During summer, the heat demand and district heating forward temperature requirements are generally lower and the availability of alternative competing heat sources is higher, which may result in lower feed-in tariffs for heat supply to DH networks. Nevertheless, the amount of heat that a supermarket could offer is larger during summer, due to an increased on-site heat demand during the winter period, which generally should be prioritized. Considering these two contradicting effects, a decreased economic potential may be expected. Considering the on-site requirements for heating would generally result in a decreased amount of heat that can be supplied to DH. Since the on-site heat demands are site specific, the impact of this aspect has to be analyzed for specific case studies. The promising performance of the cascade heat pump indicates however, that this solution might be as well promising for covering the on-site heating demands of the supermarket.

The analysis assumed an average cooling demand of the refrigeration system, both during the day and throughout the year. Adding an additional evaporator at the accumulator would enable using the compressors independently of the cooling demand and enable a continuous heat supply. Especially systems equipped with parallel compression would benefit from installing additional evaporators, due to increase utilization of the equipment [17].

5. Conclusions

The study analyzed two different possibilities to recover excess heat from CO_2 refrigeration units from supermarkets for supply of LTDH networks. The two analyzed approaches were either operating the refrigeration system at an increased gascooler pressure to directly transfer the heat or installing a cascade heat pump that recovers the heat that is rejected at temperatures being too low for recovering it directly. It was furthermore studied if it is thermodynamically and economically viable to employ a zeotropic working fluid mixture in the cascade heat pump.

The results showed that the direct heat transfer appears to be thermodynamically promising during the summer months, when the system operates at already high gascooler pressures and the required power to reach the point of operation that enables heat recovery is low. At autumn/spring and winter periods, the cascade heat pump solution showed higher thermodynamic performances. The optimization of the working fluid of the cascade heat pump showed, that a mixture of 90 % DME / 10 % Hexane is expected to yield a COP that is 15 % higher than for using Ammonia. The cascade heat pump could supply more heat at an increased performance during winter and autumn/spring and the economic evaluation showed that the increased incomes could compensate the additional investment, when considering the entire lifetime of the investment. The direct heat supply solution showed the lowest payback times, while the additional investment for the mixture did not cause much longer payback times.

The economic analysis furthermore showed that the recovery of excess heat appears to be an economically viable solution, while the cascade heat pump is more promising for DH prices above 35 €/MWh to 40 €/MWh. The payback periods for the cascade heat pump reached values below 4 years at a DH price of 37 €/MWh under the assumption that a standard unit with low specific investment cost could be produced. Based on the economic analysis it may be concluded that the presented solutions could become economically promising if the required equipment can be supplied as cost efficient standard units.

Acknowledgements

References

[1]   Lund H, Werner S, Wiltshire R, Svendsen S, Thorsen JE, Hvelplund F, et al. 4th Generation District Heating (4GDH). Integrating

smart thermal grids into future sustainable energy systems. Energy 2014;68:1–11. doi:10.1016/j.energy.2014.02.089.

[2]   Ommen T, Markussen WB, Elmegaard B. Lowering district heating temperatures - Impact to system performance in current and future Danish energy scenarios. Energy 2016;94:273–91. doi:10.1016/j.energy.2015.10.063.

[3]   Ommen T, Eric J, Brix W, Elmegaard B. Performance of ultra low temperature district heating systems with utility plant and booster heat pumps. Energy 2017. doi:https://doi.org/10.1016/j.energy.2017.05.165.

[4]   Köfinger M, Basciotti D, Schmidt RR, Meissner E, Doczekal C, Giovannini A. Low temperature district heating in Austria: Energetic, ecologic and economic comparison of four case studies. Energy 2016. doi:10.1016/j.energy.2015.12.103.

[5]   Zühlsdorf B, Meesenburg W, Ommen TS, Thorsen JE, Markussen WB, Elmegaard B. Improving the performance of booster heat pumps using zeotropic mixtures. Energy 2018;154:390–402. doi:https://doi.org/10.1016/j.energy.2018.04.137.

[6]   Bühler F, Petrović S, Karlsson K, Elmegaard B. Industrial excess heat for district heating in Denmark. Appl Energy 2017;205:991–1001. doi:10.1016/j.apenergy.2017.08.032.

[7]   Bühler F, Petrović S, Holm FM, Karlsson K, Elmegaard B. Spatiotemporal and economic analysis of industrial excess heat as a resource for district heating. Energy 2018;151:715–28. doi:10.1016/j.energy.2018.03.059.

[8]   Bühler F, Petrović S, Ommen T, Holm F, Pieper H, Elmegaard B. Identification and Evaluation of Cases for Excess Heat Utilisation Using GIS. Energies 2018;11:762. doi:10.3390/en11040762.

[9]   Lund R, Persson U. Mapping of potential heat sources for heat pumps for district heating in Denmark. Energy 2016;110:129–38. doi:10.1016/j.energy.2015.12.127.

[10]   Gullo P, Hafner A, Cortella G. Multi-ejector R744 booster refrigerating plant and air conditioning system integration - A theoretical evaluation of energy benefits for supermarket applications. Int J Refrig 2017;75:164–76. doi:10.1016/j.ijrefrig.2016.12.009.

[11]   Pardiñas Á, Hafner A, Banasiak K. Novel integrated CO2vapour compression racks for supermarkets. Thermodynamic analysis of possible system configurations and influence of operational conditions. Appl Therm Eng 2018;131:1008–25. doi:10.1016/j.applthermaleng.2017.12.015.

[12]   Hafner A, Försterling S, Banasiak K. Multi-ejector concept for R-744 supermarket refrigeration. Int J Refrig 2014;43:1–13. doi:10.1016/j.ijrefrig.2013.10.015.

[13]   Arias J, Lundqvist P. Heat recovery and floating condensing in supermarkets. Energy Build 2006;38:73–81. doi:10.1016/j.enbuild.2005.05.003.

[14]   Kauko H, Kvalsvik KH, Hafner A. How to build a new eco-friendly supermarket - Report 3 - SUPERSMART 2016. http://www.supersmart-supermarket.info/ (accessed May 22, 2018).

[15]   Nöding M, Fidorra N, Gräber M, Köhler J. Operation Strategy for Heat Recovery of Transcritical CO2 Refrigeration Systems with Heat Storages. Proc. ECOS 2016, Portoroz, Slovenia: 2016.

[16]   Sawalha S. Investigation of heat recovery in CO2trans-critical solution for supermarket refrigeration. Int J Refrig 2013;36:145–56. doi:10.1016/j.ijrefrig.2012.10.020.

[17]   Funder-Kristensen T, Finn L, Larsen S, Thorsen JE. Integration of the hidden refrigeration capacity as heat pump in smart energy systems. 12th IEA Heat pump Conf., Rotterdam: 2017, p. 1–10.

[18]   Adrianto LR, Grandjean P-A, Sawalha S. Heat Recovery from CO2 Refrigeration System in Supermarkets to District Heating Network. Proc. 13th IIR Gustav Lorentzen Conf. Nat. Refrig., Valencia: 2018. doi:doi:10.18462/iir.gl.2018.1385.

[19]   Nekså P. CO2 heat pump systems. Int J Refrig 2002;25:421–7. doi:10.1016/S0140-7007(01)00033-0.

[20]   Klein SA. Engineering Equation Solver 2017. http://fchart.com/.

[21]   Zühlsdorf B, Jensen JK, Cignitti S, Madsen C, Elmegaard B. Analysis of temperature glide matching of heat pumps with zeotropic working fluid mixtures for different temperature glides. Energy 2018;153:650–60. doi:https://doi.org/10.1016/j.energy.2018.04.048.

[22]   Zühlsdorf B, Bühler F, Mancini R, Cignitti S, Elmegaard B. High Temperature Heat Pump Integration using Zeotropic Working Fluids for Spray Drying Facilities. 12th IEA Heat pump Conf., Rotterdam: 2017, p. 1–11.

[23]   Regeringen - Energi Forsynings- og Klimaministeriet. Energi - til et grønt Danmark 2018. http://efkm.dk/media/11873/energiudspil_2018.pdf (accessed May 29, 2018).

[24]   Danish Energy Agency, Energinet. Technology Data for Energy Plants for Electricity and District heating generation 2018. https://ens.dk/en/our-services/projections-and-models/technology-data (accessed June 7, 2018).

[25]   Danish Energy Agency, Energinet. Technology Data for Individual Heating Installations 2018. https://ens.dk/en/our-services/projections-and-models/technology-data (accessed July 11, 2018).