High temperature heat pump integration into district heating networks

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Abstract

This study illustrates the potential of high temperature heat pumps (HTHPs) integration into district heating network (DHN) through a twofold approach, using DHN as a heat sink and source. It is used as a heat sink of HTHP that uses waste heat from the supermarket's refrigeration system as a heat source whereas it is used as a heat source to HTHP that provides heat to industrial applications. When the DHN acts as the heat sink, the integrated system provides a coefficient of performance (COP) of the waste heat recovery (WHR) system between 3.2 and 5.4, reducing the operating costs between 50% and 100% with an average price ratio of 2.25 compared with the standard CO_2 refrigeration system. If the DHN is the heat source, the integrated system provides a COP from 2.8 to 5.7 for a heat sink of 110 oC. The alternative low-GWP refrigerants assessment illustrates that HC-290, HFO-1234ze(E) and HFO-1234yf were considered the ideal candidates to replace the HFC-134a, whereas HCFO-1233zd(E) and HCFO-1224yd(Z) were the most promising low-GWP refrigerants to replace HFC-245fa. Finally, the environmental results showed that the utilisation of the DHN as the heat sink in the integrated system solution produces about 60% lower equivalent CO_2 emissions than the DHN generation mix. Moreover, using DHN as the heat source, the equivalent CO_2 emissions can be reduced up to 98% in Sweden compared to conventional natural gas boilers. Hence, the combination of HTHPs and the DHN represents a step forward in the mitigation of climate change through the utilisation of sustainable energy conversion technologies.

Keywords: Sustainability; CO_2; Environmental analysis; HTHP; waste heat recovery

Nomenclature

a, b, k_e, k_s, k_1, k_2 Pierre's correlations constants COP coefficient of performance

(-) E_a annual energy consumption (kWh) h specific enthalpy (kJ kg^{-1}) m refrigerant mass flow rate (kg s^{-1}) n lifespan of the vapour compression system (years) L annual refrigerant leakage rate (kg $year^{-1}$) P pressure (MPa) p price (€ MWh^{-1}) Q thermal loads (kW) T temperature (oC) V volumetric flow rate (m^3 s^{-1}) W electric power consumption (kW)

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VHC volumetric heating capacity (kJ m^{-3})

Greek symbols ε effectiveness (-) η efficiency (-) Δ variation α recycling factor of the refrigerant (%) β indirect emission factor (kgCO_2 kWh^{-1})

Subscripts c compressor em electromechanical iso isentropic k condenser disch compressor discharge temperature sink heat sink r reduced ref reference working fluid suc suction source heat source vol volumetric

Abbreviations DHN district heating network EES engineering equation solver GWP global warming potential HC hydrocarbon HCFO hydrochlorofluoroolefin HFC hydrofluorocarbon HFO hydrofluoroolefin HR heat recovery HTHP high-temperature heat pump IHX internal heat exchanger LT low temperature MT medium temperature NBP normal boiling point ODP ozone depletion potential PR pressure ratio WHR waste heat recovery

1. Introduction

The growing concern about climate change requires a transformative change, integrated with sustainable development to limit the increase in global temperature to 1.5°C above pre-industrial levels. Among other measures, the Intergovernmental Panel on Climate Change proposes the adopting of low-emission innovations as heat pumps and district heating and cooling as climate mitigation behaviour [1]. Werner [2] defined district heating as the use of 'local fuel or heat resources that would otherwise be wasted, in order to satisfy local customer demands for heating, by using a heat distribution network of pipes as a local market place'. The traditionally excess heat resources for district heating networks (DHN) are combined heat and power (CHP), Waste- to-Energy plants and industrial processes. However, there are other local heat resources with a lower thermal level that can be used as potential heat sources for district heating networks, instead of being rejected to the ambient and wasting the available energy [3]. Moreover, the wind and photovoltaics intermittency can be efficiently managed at an affordable cost, using transformative

energy technology connected to district heating and cooling networks, as heat pumps and thermal storage. An optimised use of these technologies contributes to the management of the intermittent generation in some European countries [4].

Supermarkets are the most energy-intensive commercial buildings, in which the refrigeration system is the largest energy user [5]. The standard CO_2 trans-critical booster system has become a well-established refrigeration solution in Scandinavian supermarkets as a single compact unit [6]. The critical function of the refrigeration systems is to preserve the food products (fresh or frozen) by absorbing the heat from the cabinets at low temperature levels and rejecting it at a higher temperature level; usually to the ambient. Although part of this rejected heat can be recovered to provide the space heating demand of the supermarket's building [7], between 5% and 45% of this heat is still wasted, releasing it to the ambient via a condenser/gas cooler, as represents Fig. 1, which is a representation of the results presented by Karampour et al. [8] and Sawalha [7]. However, there is an opportunity to recover this low-grade waste heat through heat pump technology [9].

Fig. 1. Breakdown of the thermal loads in a CO_2 refrigeration system in average size supermarket based on the modelling proposed by Karampour et al. [8] and Sawalha [7].

Heat pumps are based on a sustainable technology that uses the thermodynamics principles to revalorise heat with a smaller amount of electric energy used [10]. Recent heat pump prototypes have extended the operating ranges (up to 90 oC) of this technology efficiently, becoming possible to produce heating at temperatures up to 140 oC [11]. This technological improvement modified the classification of the heat pumps that operate at high temperature conditions, becoming known as high temperature heat pumps (HTHPs) [11]. Therefore, waste heat rejected by the supermarket refrigeration system can be recovered using the HTHP technology in order to upgrade this heat to useful temperature levels, which will have enough quality to be injected into the local district heating network. The integration of HTHPs into DHN makes the distribution more versatile because HPs are considered as environmental systems to produce heat for DH systems, with proper implementation, design and control [12]. Moreover, the integration of flexible HPs into DHN has been recognised as a significant step to a 100% renewable energy system, converting excess clean electric energy in heat [13]. In this case, the DHN is used as a heat sink in the HTHP integration, although it can also be used as a heat source for other applications [14].

Another interesting application of the HTHP integration into DHN can be the substitute of natural gas boilers in the industrial sector, using the DHN as a heat source. Approximately, 3200 TWh of the annual EU energy use corresponds to the industrial sector, for which in average, about 30% represents the consumption in the form of electricity, and the remaining 70% corresponds to the heating consumption. Food, chemical and paper industries are energy-intensive sectors that require heat up to 150 °C for their industrial processes (e.g. for drying, evaporating, distilling) [15]. The heating demand in these industries is usually satisfied with fossil fuel-fired technology as natural gas boilers. Nevertheless, HTHPs becomes a promising and competitive technology for boiler substitution in delivering the heat of up to 180 °C [16]. Since the ambient temperature

becomes not viable as a potential heat source for energy-efficient industrial HTHPs, DHN can be a potential heat source for this technology [17]. The integration of HTHPs into DHN becomes a promising combination to replace a gas boiler in the industrial sector. A recent potential application proposed by Sartor, et al. [18] is the integration of HTHPs in the low-temperature district heating network. Besides increasing the efficiency and flexibility of the DHN, it reduces the heat losses up to 1,900 MWh per year.

HTHPs, mainly based on vapour compression technology, typically use hydrofluorocarbons (HFCs) as working fluids (refrigerants), which have high values of global warming potential (GWP) [11]. However, in 2014, EU Regulation No. 517/2014 [19] gradually limited the acquisition and utilisation of HFCs, establishing market quotas to their consumption and limiting the maximum GWP in some applications in order to increase the share of low GWP refrigerants. Given this situation, the commonly called fourth generation of refrigerants appeared to replace HFCs [20]. Due to their critical temperature, HFC-134a is used in HTHPs for heating applications until 90 oC whereas HFC-245fa is considered the reference fluid for high temperature applications up to 140 oC. Therefore, different low GWP alternatives should be found for these two refrigerants if HTHPs expect to get anywhere in the industrial sector.

Regarding the potential alternatives to HFC-134a, Maiorino et al. [21] proposed R-152a as alternative low GWP refrigerant for domestic refrigerators, showing refrigerant charge and energy consumption reductions. Mota-Babiloni et al. [22] performed a drop-in replacement of HFC-134a with HFO-1234yf and HFO-1234ze(E) in vapour compression liquid chiller, proving that the HFC-134a can be replaced by both very low GWP hydrofluoroolefins (HFOs). Palm [23] realised a comprehensive investigation of the use of hydrocarbons in heat pumps, illustrating that HC- 600a and HC-290 become suitable alternatives to HFC-134a. Finally, Mota-Babiloni et al. [24] investigate the bend R-513A as a sustainable alternative to HFC-134a in refrigeration systems, with and without internal heat exchanger. Although these potential low-GWP alternatives to HFC-134a are widely studied for refrigeration systems, more studies are required to analyse and ensure the operation in high temperature conditions.

Then, as low-GWP alternatives to HFC-245fa, HCFO-1233zd(E) and HCFO-1224yd(Z) are considered the most promising candidates to replace HFC-245fa in HTHPs and organic Rankine cycles (ORCs) [25,26]. Moreover, HFO-1336mzz(Z) is widely studied as an alternative to HFC- 245fa [27,28], although presents slightly different thermophysical properties to HFC-245fa compared to the previously mentioned alternatives [29]. However, the blend R-514A, composed by HFO-1336mzz(Z) and trans-1,2-dichloroethylene (t-DCE), was developed [30] to reduce the thermodynamic differences between HFO-1336mzz(Z) and HFC-245fa. R-514A was studied by Majurin et al. [31] and showed a great potential to replace HFC-245fa. Natural refrigerants must be named to complete the list of candidates considered to replace HFC-245fa in high temperature applications. The hydrocarbons can be found among this family of refrigerants, specially HC-600 [32,33] and HC-601 [34,35].

Based on the above discussed, the integration of HTHP into DHN appears as an innovative and efficient solution to contribute to climate change mitigation in different areas. Moreover, the recent developments in heat pump technology make it possible to operate in higher heat sink and source temperatures with the use of HTHPs. The combination of HTHPs and DHN represents a step forward in the development of sustainable integrated systems. Thus, the purpose of this paper is to investigate the integration of HTHPs into DHN in two different scenarios, using the DHN as a heat sink in one scenario and heat source in the other. When the DHN is acting as the heat sink, the HTHP recovers, revalorises and injects the heat rejected by the refrigeration system of a supermarket. A thermo-economic optimisation of the refrigeration system and HTHP has been done to maximise the benefits of this system integration. Moreover, the DHN is used as the heat source for HTHPs in the industrial sector to replace the conventional natural gas boilers. Finally, alternative low-GWP refrigerants to HFC-134a and HFC-245fa are evaluated as sustainable candidates in HTHP systems.

The structure of this paper is organised as follows. Section 2 describes the integrated systems, the proposed scenarios and the alternative refrigerants. Section 3 introduces the modelling assumptions and simulation methodology. Section 4 presents the thermo-economic optimisation and results of the systems integration, along with the alternative low-GWP refrigerants evaluation and environmental analysis. Finally, the last section contains the most relevant conclusions of this work. The results of this study provide a comprehensive view of the potential of integrating HTHPs into DHN to decarbonise different urban and industrial environments.

2. System description

2.1 Research scenarios The schematic of both proposed scenarios to integrate HTHPs into DHN is shown in Fig. 2. In the first scenario, the DHN is acting as a heat sink, and it is presented in Fig. 2a. The scenario in which the DHN is used as a heat source corresponds to Fig. 2b.

- DHN_{sink} scenario: In order to integrate HTHP system into the standard CO_2 trans-critical booster system, a new heat exchanger named waste heat recovery (WHR) is allocated between the gas cooler/condenser exit and the expansion valve inlet, acting as an evaporator of the HTHP. Bypassing the gas cooler/condenser makes the WHR absorb all the waste heat coming from the CO_2 refrigeration system. This absorbed heat is upgraded with the compressor to a useful thermal level, which can be injected into the DHN through the HTHP condenser. This scenario requires a thermo-economic optimisation of the CO_2 exit temperature from the WHR because a variation of this parameter influences the performance of both systems; i.e. refrigeration and HTHP systems. HTHP system in this scenario provides a heating production temperature up to 90 oC.

- DHN_{source} scenario: In this case, the HTHP evaporator absorbs the heat from the DHN supply line to revalorise it to thermal levels for which will be useful in the industrial processes (90-130 oC). In contrast to the previous scenario, the HTHP system should provide heating production temperatures up to 130 oC to satisfy the industrial thermal requirements.

Both HTHP systems include the use of internal heat exchanger (IHX). This component transfers the heat from the condenser outlet to the suction line, increasing the sub-cooling and superheat degree. Numerous studies illustrate that the use of IHX in HTHP systems, improves energy performance, and therefore, its use is highly recommended for high-temperature applications [29,36].

(a) (b) Fig. 2. Schematic of the integration of HTHPs into DHN: DHN_{sink} a) scenario with an HTHP heating production up to 90 oC and b) DHN_{source} scenario with an HTHP heating production up to 130 oC.

2.2 Alternative low-GWP refrigerants

Although has been widely discussed the high condensing temperature required for the heating production, HTHP technology operates also with high values of evaporating temperature, between 49-90 oC. Moreover, the operating temperature differences of the HTHP systems between both scenarios, distinct reference and alternative refrigerants should be considered for each scenario according to their thermophysical properties. For DHN_{sink} scenario, HFC-134a becomes the appropriate reference refrigerant to overcome the heating requirements up to 90 oC. Nevertheless, due to its high GWP (1430), different low-GWP refrigerants with GWP lower than 150 are proposed as potential candidates to replace it. These candidates have similar or higher critical temperature than the reference HFC-134a along with GWP values lower than 150. Table 1 shows the main thermophysical and transport properties along with the environmental parameters of the reference fluid and the suggested low-GWP alternatives. Although all the alternative refrigerants in this scenario have lower toxicity (ASHRAE class A), there is some difference in the flammability levels. Whereas the hydrocarbons have higher flammability (3), HFC-152a (2) presents lower flammability, becoming even lower for the HFOs (2L). The safety class becomes an important consideration to realise a proper refrigerant selection.

Table 1. Selected properties of HFC-134a and its suggested low-GWP alternatives [37]

Refrigerant

Molecular weight (g mol^{-1})

Saturated Vapour density (kg m^{-3})[a]

ASHRAE Safety Class [38]

HFC-134a (Ref.) 102.0 101.1 4.05 32.4 -26.0 0 1430 A1 HC-600 58.1 152.0 3.8 6.2 -0.5 0 4 A3 HC-600a 58.1 134.6 3.6 9.1 -11.7 0 20 A3 HC-290 44.1 109.4 3.6 20.6 -19.0 0 3 A3 HFO-1234yf 114.0 94.7 3.4 37.9 -29.5 0 <1 A2L HFO-1234ze(E) 114.0 109.4 3.6 26.3 -19.0 0 <1 A2L HFC-152a 66.0 113.3 4.5 18.5

-24.0 0 138 A2 a At saturated pressure of 25 °C.

T_{crit} (oC)

P_{crit} (MPa)

NBP (oC) ODP GWP_{100}

[38]

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On the other hand, DHN_{source} scenario has other thermal requirements to overcome the industrial heating demand at high temperature levels. Due to this fact, different refrigerants with a higher critical temperature that the previous presented should be required. In this case, HFC-245fa with a critical temperature of 154 oC becomes the references fluid in HTHP systems that operates with a heat sink temperature above 100 oC [11]. Owing to its high GWP (858), different alternative low-GWP refrigerants are proposed as potential candidates to replace this HFC. The thermophysical and transport properties, along with environmental parameters of the potential alternative working fluids to replace HFC-245fa, are presented in Table 2. Whereas R-514A is classified with higher toxicity (ASHRAE class B), the rest alternative refrigerants present lower toxicity (ASHRAE class B). About the flammability, the hydrocarbons have higher flammability (3) while HFO-1336mzz(Z), R-514A and the HCFOs shows no flame propagation (1). Thus, the refrigerants with A1 classification will require lower safety measures than the others, becoming an important consideration to the refrigerant selection.

Table 2. Thermophysical properties of HFC-245fa and its alternative low-GWP refrigerants

Refrigerant

Molecular weight (g mol^{-1})

Saturated Vapour density (kg m^{-3})^a T_{crit} (oC)

ASHRAE Safety Class [38]

HFC-245fa (Ref.)	134.0	154.0	3.65	38.68	15.1	0	858	B1
HC-601	72.2	196.6	3.37	8.93	36.1	0	5	A3
HC-600	58.1	152.0	3.80	22.45	-0.5	0	4	A3
R514A	139.6	178.0	3.52	22.78	29.1	0	2	B1
HFO-1336mzz(Z)	164.1	171.4	2.90	24.07	33.4	0	2	A1
HCFO-1233zd(E)	130.5	166.5	3.62	30.66	18.3	0.00034	1	A1
HCFO-1224yd(Z)	148.5	155.5	3.33	40.18	14.6	0.00012	<1	A1

a At saturated pressure of 75 °C.

Fig. 3 provides a clear view of the reference fluids, and their low-GWP alternative thought a T-s diagram. All the proposed candidates to replace HFC-134a possess similar or even higher critical temperature than the reference fluid, although different T-s curves, as shown in Fig. 3a. It should be highlighted that the T-s slope difference in each refrigerant, will provide different interesting behaviours in the performance analysis of these candidates. The T-s diagram of the alternative low-GWP refrigerants to replace HFC-245fa are presented in Fig. 3b. In this case, HCFOs, HFO and the blend R-514A show similar T-s curve to the reference HFC-245fa, whereas the hydrocarbons present remarkable differences. All the candidates provide interesting results of Coefficient of performance (COP) and Volumetric heating capacity (VHC), which will be presented and discussed in the results section.

(a) (b) Fig. 3. T-s diagram of the reference refrigerants and its potential low-GWP alternatives for a) DHN_{sink} scenario (Ref. HFC-134a) and b) DHN_{source} scenario (Ref. HFC-245fa)

P_{crit} (MPa)

NBP (oC) **ODP** **GWP_{100}**

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3. Methodology and modelling details

The computer modelling parameters, along with the methodology used in this study, are presented in this section. Stockholm (Sweden) has been selected as the reference case for both the climate conditions and the district heating characteristics. The performance analysis covers the winter period time, in which all ambient temperatures are below 10 oC.

3.1 Calculations procedure

The simulation results of the integrated systems exposed in this paper are calculated based on the methodology presented in Fig. 4. Each scenario configuration and refrigerant, along with the boundary conditions and assumptions are used as input parameters for the model. The modelling is realised with the software Engineering Equation Solver (EES) [39]. This software solves the thermodynamic equations of each configuration and the IHX optimisation. Moreover, both models, CO_2 refrigeration model based on Karampour et al. [8] and HTHP, are connected to realise the thermo-economic optimisation in the scenario DHN_{sink}. The IHX optimisation, based on the Golden Section Search algorithm implemented in EES, maximises the COP of each scenario and refrigerant, varying the IHX effectiveness without exceeding the maximum discharge temperature of each compressor, depending on the scenario. Finally, the performance parameters become the modelling outputs, which are compared with the reference refrigerant in order to calculate the relative difference of each parameter and therefore, realise the performance and environmental analysis.

Fig. 4. Methodology flow diagram for the modelling scenarios, IHX optimisation and alternative low-GWP refrigerant analysis.

3.2 Boundary conditions and assumptions

3.2.1 CO_2 refrigeration system in DHN_{sink} Scenario The standard CO_2 trans-critical booster system with heat recovery (HR) was modelled using the boundary conditions and assumptions proposed by Karampour, et al. [8] for a typical average size supermarket in Sweden. This model has been modified to bypass the gas cooler and to integrate the additional heat exchanger that connects the CO_2 refrigeration and the HTHP systems. The thermal loads of the supermarket CO_2 refrigeration system for each hour of the year based on Karampour et al. [8] and Sawalha [7] are presented in Fig. 5. The medium temperature (MT) cooling and space heating demands are dependent of the ambient temperature, whereas the low

temperature cooling demand remains constant throughout the year because glass lids typically cover the low temperature (LT) level freezers.

Fig. 5. Cooling and space heating thermal load for the CO_2 supermarket refrigeration system during the year.

3.2.2 District heating network

The temperature profile of DHN has been modelled as a function of the ambient temperature, using the climate data of Stockholm [40] and the selling prices of 2017 for DHN heat injection in this city [41]. The DHN price variation during the year can be observed in Fig. 6a. For this study, the analysis zone will consider the winter season for which the ambient temperatures are below 10 oC. The supply and return temperatures of the DHN are presented in Fig. 6b, along with the Stockholm BIN hours used to calculate the different energy indicators as COP, VHC along with economical parameters. The BIN method corresponds to a discrete group (BINS) of specific weather condition where, in this case, each bin contains the number of average hours of the occurrence of each specific ambient temperature during a year. The Open District Heating in Stockholm becomes trading market of the excess heat and therefore, a perfect way to inject the excess heat into DHN with two different supply temperature profiles [42]. Two supply temperatures are depending on the DHN generation and the area installed. In this study, the low supply temperature profile is considered in all the calculations.

(a) (b) Fig. 6. District heating network modelling assumptions: a) Selling prices and b) supply and return temperatures.

3.2.3 High temperature heat pumps systems

The use of an IHX is included in both HTHP models to provide the highest COP of the system in each operating condition. The sub-cooling degree at the condenser outlet is assumed 2 K while the superheating degree at the evaporator outlet considered is 5 K. The pitch point or temperature approach in the condensers and evaporator is assumed 5 K. The electro-mechanical efficiency of the compressors is assumed 95%. The isentropic and volumetric efficiencies in this model are calculated using Pierre's correlations (Eq.(1) and (2)), following the methodology used in previous studies [29,34].

$\eta_{vol} = k_1 \cdot (1 + k_s \cdot \frac{t_{2k} - 18}{100}) \cdot \exp(k_2 \cdot \frac{p_1}{p_2})$ (1) where symbols t_{2k} k_1 is , k_s the and inlet k_2 are compressor constants temperature with the value and of p1.04, $_1/p_2$ is the pressure ratio. The remaining 0.15, and -0.07, respectively [43]. $(\frac{\eta_{is}}{\eta_{vol}}) = (1 + k_e \cdot \frac{t_{2k} - 18}{100}) \cdot \exp(a \cdot \frac{T_1}{T_2} + b)$ (2) where (in Kelvin). T_1/T_2 The is the constants ratio between k_e, a and the b condensation are -0.1, -2.40 and and evaporation 2.88, respectively.
absolute temperatures ratio

About the HTHP system in the DHN_{sink} scenario, the temperature difference in the WHR heat exchanger from the CO_2 part depends on the de-superheater outlet temperature, which depends on the space heating demand, and the WHR outlet temperature, controlled by the HTHP. This difference has a great influence on both the system performance of the CO_2 refrigeration and the HTHP systems, and therefore, a techno-economical optimisation is required to find the optimum value of the inlet temperature to CO_2 expansion valve. The heat absorbed by the evaporator in this scenario depends on the inlet and outlet temperature in the secondary fluid. Whereas the temperature difference between the inlet and outlet in the secondary fluid has been set to 15 K, the inlet temperature of the secondary fluid will follow the temperature profile of the DHN supply line.

On the other hand, the DHN_{source} scenario is modelled using the supply DHN temperature profile as the inlet temperature of the secondary fluid in the HTHP evaporator. The temperature difference of the secondary fluid in the evaporator is assumed to be 15 K, whereas this temperature difference becomes 20 K in the condenser. The heat sink outlet temperature takes into account different values in order to perform a sensitivity analysis of this parameter depending on the thermal requirements of the industrial process. Isentropic and volumetric compressor efficiencies have been implemented in the HTHP model using experimental data from an HTHP prototype with HFC-245fa. The data points of the compressor efficiencies are curve fitted and represented in Eq. (3) and (4), and both efficiencies have been corrected for each alternative refrigerant following the methodology described by Mateu-Royo et al. [44].

$\eta_{iso}(PR) = -0.02509PR^2 + 0.1829PR + 0.4555$ (3)

$\eta_{vol}(PR) = -0.02467PR^2 + 0.2448PR + 0.3650$ (4)

3.3 Energy use analysis

Two main parameters are used to evaluate the benefit of the studied scenarios with each of the low-GWP alternative refrigerants, which are: the COP and the VHC. The COP for the CO_2 refrigeration system is defined by Eq. (5), whereas this parameter for the HTHP is defined using Eq. (6).

$COP_{CO2} = W^{Q}{}_{MT}{}_{MT} + {}^{+} W^{Q}{}_{LT}{}_{LT} + {}^{+} W^{Q}{}_{Fans}$

HR

(5)

$COP_{HTHP} = W^{Q}{}_{kC}$

(6)

Nevertheless, a new definition of the COP is proposed to evaluate the performance of the DHN_{sink} scenario, considering the HTHP compressor electricity consumption and the difference of the

electricity consumption in the using Eq. (7).

CO_2 refrigeration system ($\Delta W_{tot,CO2}$). This COP_{WHR} is defined

$COP_{WHR} = W_C\ Q_k$

$+ \Delta W_{tot,CO2}$

(7)

where Q_k represents the heat rejected in the HTHP condenser, W_C represents the compressor electric difference power of the consumption CO_2 refrigeration and ΔWsystem, $_{tot,CO2}$ represents operating the compressor electric power consumption with and without the integration of HTHP.

On the other hand, the VHC is used to evaluate the difference in heating per unit of volume between the reference working fluid and the low-GWP alternative refrigerants. This parameter is defined by Eq. (8).

where V_{suc} is the refrigerant volume flow VHC rate = (mV^{Q}_{suc} ^{3} k
s^{-1}).
(8)

The results obtained for the alternative low-GWP refrigerants evaluation are provided as a relative difference, comparing the reference fluid with each candidate refrigerant. Thus, Eq. (9) and (10) are used to calculate the relative performance difference for both parameters, the COP_{HTHP} and VHC.

$$\%COP = \frac{COP_{low-GWP} - COP_{Ref}}{COP_{Ref}} \cdot 100 \quad (9)$$

$$\%VHC = \frac{VHC_{low-GWP} - VHC_{Ref}}{VHC_{Ref}} \cdot 100 \quad (10)$$

Finally, the relative difference in operating cost is calculated using the Eq. (11)

$$\%OC = \frac{\sum_i (W_{C,i} + \Delta W_{tot,CO_2,i}) \cdot p_{DHN,i} \cdot R_{p,j} - \sum_i (W_{CO_2,i}) \cdot p_{DHN,i} \cdot R_{p,j}}{\sum_i Q_k \cdot p_{DHN,i}} \cdot 100 \quad (11)$$

where p_{DHN} is the district heating network price, R_p the price ratio between the DHN and electricity, of HTHP, i Wcorresponds $_{CO2}$ the power consumption of the CO_2 refrigeration system without the integration to the ambient temperature and j corresponds to different values of price ratios.

3.4 Environmental analysis

The equivalent CO_2 emissions analysis has been realised to provide an environmental perspective of the system integration benefits. Table 3 presents the different emission factor.

Table 3. Emission factor associated with different system processes.

Emission factor description	**Value (g CO_2 kWh^{-1})**
Sweden electricity consumption [45]	13.3
European Union electricity consumption [45]	295.8
Sweden DHN generation mix [2]	32.4
Natural gas consumption [46]	205

For the scenario DHN_{sink}, the emission factor associated with the electricity consumption is assumed 13.3 g CO_2 kWh^{-1} based on the European Environment Agency for Sweden [45]. In this case, the equivalent CO_2 emissions of the integrated system will be compared with the emission factor of the mix generation in the DHN with a value of 32.4 g CO_2 kWh^{-1} [2]. For the scenario

DHN_{source}, an equivalent CO_2 emissions comparison has been realised between the HTHP using DHN as a heat source and a natural gas boiler, providing both systems with the same amount of heat. The emission factor associated with the natural gas consumption is considered 205 g CO_2 kWh^{-1} [46], whereas the electricity emission factor is considered 295.8 g CO_2 kWh^{-1} [45] for the European Union and the above-stated value for Sweden, in order to compare two different perspectives.

3.5 Total equivalent warming impact (TEWI) evaluation

In order to evaluate the benefits of the proposed low GWP alternatives, Total Equivalent Warming Impact (TEWI) has been calculated for each refrigerant to quantify the equivalent CO_2 emission due to the energy consumption of the HTHP systems (indirect emissions) and to the accidental losses of refrigerant (direct emissions) [47]. TEWI method is calculated based on Eq. (12).

$$TEWI = GWP \cdot L \cdot n + GWP \cdot m_r(1 - \alpha) + n \cdot E_a \cdot \beta \quad (12)$$

Some necessary assumptions can be considered in order to realise the TEWI analysis. The lifespan of the system (n) is considered 15 years and the recycling factor of the refrigerant (α) is omitted for this evaluation. Considering HTHPs systems will be contained in the machinery room, an annual leakage rate of 5% of the total refrigerant charge is considered [48]. Finally, the indirect emission factor (β) considered is the electricity emission factor for the European Union. Using the Sweden electricity emission factor will provide the same relative results of each refrigerant evaluation with lower absolute values.

4. Results and discussion

This section includes the results and discussion of the HTHP integration into the DHN, dividing this section between the two proposed scenarios: DHN_{sink} and DHN_{source}.

4.1 DHN_{sink} Scenario

4.1.1 Thermo-economic optimisation

In this scenario, a thermo-economic optimisation of the WHR exit temperature is required in order to maximise the benefits of the systems integration. Low temperature at CO_2 expansion valve inlet of the refrigeration system results in higher COP_{CO2}; however, it means that the evaporation temperature of the HTHP becomes lower, resulting in a decrease of COP_{HTHP}. Therefore, the lower temperature at the expansion valve inlet of the refrigeration system results in more heat recovery in WHR heat exchanger.

Fig. 7 presents the economic benefits per operating hour, depending on the WHR exit temperature, the ambient temperature and the price ratio between electricity consumption and selling heat to DHN prices. Based on the results obtained, an increase of the WHR exit temperature increases the economic benefits of the integrated system. Moreover, the thermal loads, DHN prices and energy consumption are highly influenced by the ambient temperature and therefore, the economic benefits become affected by the ambient variable. Nevertheless, there is a clear tendency to obtain the maximum economic benefits, which will be located operating in ambient temperatures between 0 to 10 oC and fixing the WHR temperature at the highest level. The space heating demand of the supermarket's building is influenced by the ambient temperature, becoming lower with high ambient temperatures. Hence, the maximum economic benefit will be located operating in higher ambient temperatures where there is more waste heat available.

Fig 7. Thermo-economic optimisation and economic benefits per operating hour of the HTHP integration into DHN as a heat source.

Nevertheless, the WHR exit temperature influences the energy performance of both systems, as shown in Fig. 8. Whereas the COP of the CO_2 refrigeration system decreases with an augment of the WHR exit temperature, the COP of the HTHP is positively affected. It is important to highlight that the condensing temperature is a function of the ambient temperature due to the connection with the DHN. As the COP_{CO2} reduction is less influenced than the COP_{HTHP}, the COP_{WHR} increases with an increase of the WHR exit temperature. It is essential to remark that the CO_2 refrigeration system is assumed to recover the same heat to overcome the building heating demand with or without HTHP integration. As a result of the thermo-economic optimisation, the WHR exit temperature is set to 20 oC for the performance evaluation and the alternative low-GWP refrigerant evaluation.

(a) (b) (c) Fig 8. COP of the different integrated system depending on the WHR exit and the ambient temperatures for a) CO_2 refrigeration system, b) HTHP and c) Total integrated solution.

Fig. 9 illustrates that the HP heating capacity can be slightly modified with a variation of the WHR exit temperature. This variable heating capacity provides flexibility to the integrated HTHP to overcome a specific heating requirement only with a variation in the WHR exit temperature. At ambient temperatures between 0 to 10 oC, there is located most of the operating hours in this study and moreover, it becomes the region where this system provides the highest flexibility. For low ambient temperatures, the supermarket building requires a high amount of heat to overcome the space heating demand. Thus, CO_2 refrigeration system control reduces the heat rejected to the ambient in order to provide the required heating demand. It causes a decrease in the available waste heat and therefore, the HTHP system becomes less flexible at low ambient temperatures.

Fig. 9. Heating capacity of the HTHP system depending on the WHR exit and the

ambient temperatures

4.1.2 Performance evaluation

The evaluation of the integrated system has been carried out from two perspectives: analysing the performance of the refrigeration system, HTHP and WHR system along with the relative operating cost of the full integration. Fig. 10a presents the three different COP analysed, corresponding to the CO_2 refrigeration system, the HTHP and the integrated system as waste heat recovery solution. The COP_{CO2} has some variations with the ambient temperature due to the strategy control implemented in the system to recover the heating demand of the supermarket building [7]. As the ambient temperature decreases, the system is forced to run in transcritical conditions in order to generate more heat to fulfil the heating requirements.

Until a specific ambient temperature, the system change the control strategy continue being able to provide the necessary heat. More details of the control strategy implemented are explained by Sawalha [7]. About the HTHP, the COP_{HTHP} remains constant until a specific ambient temperature, where this performance parameter starts to decrease. This behaviour is caused by the DHN profile temperature, which remains constant in 60 oC in the supply line until approximately 90 oC in the lowest ambient temperatures. This increase in the thermal supply levels increases the temperature lift of the HTHP and therefore, produces a drop in the performance. Finally, the COP_{WHR} presents higher values than the COP_{HTHP} operating in warm ambient temperatures caused by the energy consumption reduction of the gas cooler fans in the CO_2 refrigeration system compared with the reference. The integration of HTHP requires the gas cooler bypass, and therefore, the fans in this component become disabled in the integrated system. Then, COP_{WHR} is being influenced by the drop of both CO_2 refrigeration and HTHP systems, until certain ambient temperature where the CO_2 requires high heating demand and the WHR system becomes benefited by this strategy control.

The majority of the operating hours in Stockholm are located in the high-efficiency zone of the WHR integrated system. Due to this fact, it is expected a potential benefit of this system integration in terms of operating costs. Fig. 10b illustrates the potential operating costs savings comparing the integrated system with the standard CO_2 refrigeration system without integration. This study is focused on the operating cost because the investment cost does not become representative to illustrate the potential of this technology due to the novelty of HTHP technology in the market.

The major operating hours are located between ambient temperatures of -5 to 10 oC as can observe in Fig. 10a. The relative difference in operating cost has been calculated using Eq. (11) where the DHN price dependency with the ambient temperature provides only one possible DHN price for each ambient temperature. However, there are multiple price ratios for each temperature and therefore, multiple electricity prices can be considered in a single ambient temperature. Hence, a price ratio range between 1.2 and 2.7 is considered in this analysis in order to limit the possible solutions and provide a realistic result. Therefore, operating savings between 50% and 100% should be expected for an average price ratio of 2.25. Therefore, this system integration shows an excellent potential to be implemented in most of the supermarkets that already have a DHN connection in order to obtain higher economic operating benefits. A correct heat selling price to DHN will promote the application of this WHR technology due to the higher benefit for the prosumers.

(a) (b) Fig. 10. Performance evaluation of the integrated system with the ambient temperature, using DHN as a heat source: a) COPs and b) relative difference in the operating cost.

4.1.3 Alternative refrigerants to HFC-134a This section presents the results of the potential candidates from two perspectives: looking at the performance (i.e. COP from Eq. (6)) and the installation and components size (i.e. VHC in Eq. (8)). The calculations are realised with the same operating conditions of the DHN_{sink} scenario. These results are plotted in Fig. 11 as a relative difference between the reference fluid and each alternative; using Eq. (9) for %COP and Eq. (10) for %VHC. HC-600a and HC-600 present the highest performance improvements with a %COP increase between 9% to 17%, depending on the ambient temperature. However, both refrigerants have lower VHC; around 40% for HC-600a and 60% for HC-600, and therefore, greater compressor size is required to provide similar heating capacities owing to their lower suction density, shown in Table 1. In contrast, HC-290 provides a slight increase of COP along with the highest VHC increase of about 38% requiring a smaller compressor for the same heating demand.

It can be observed from Fig. 11 that HFC-152a has equal or slightly lower VHC, but this is the only refrigerant analysed that presents a reduction of the COP, but only about 1%. The hydrofloroolefins (HFO's) on the other hand, both refrigerants have an increase of the COP without being highly influenced by the ambient temperature. HFO-1234yf has slightly similar or even higher VHC, while HFO-1234ze(E) has a reduction of 20%, which can be significant in the refrigerant selection.

HC-290 can be a proper candidate to replace HFC-134a in new design installations, providing a slight increase in efficiency and providing the required heating with a smaller compressor. Contrary, HC-600a can provide a significant COP increase but at the expense of larger compressors for the same heating demand. All these refrigerants have a greater or lesser degree of flammability, and therefore, it is required special security measures to operate with these refrigerants.

(a) (b) Fig. 11. Alternative low-GWP refrigerants comparison to replace HFC-134a: a) COP and b) VHC. The discussion in this section looks only at the efficiency and the size of the compressor; however, other factors should be taken into account in choosing the suitable refrigerant for the application, such as availability of components, safety requirements, and installation cost.

4.2 DHN$_{source}$ Scenario

4.2.1 Performance evaluation In this scenario, DHN is used as a heat source, and the interest becomes in the heating production to overcome the thermal requirements of the industrial process. Thus, the results in this section are the COP and heating capacity of the HTHP system integrated into DHN as a function of the ambient and the heat sink temperatures (Fig. 12), using HFC-245fa as reference fluid. The calculated performance parameters are obtained for the same evaporator capacity. In this case, the heat sink temperature becomes limited by the temperature lift, which is defined as the temperature difference between the inlet heat source and outlet heat sink. Until an ambient temperature of 0 oC, the supply line of the DHN has a temperature of 60 oC, and therefore, the heat sink is being limited to 113 oC. As the DHN supplies temperature increase as a function of the ambient temperature, the HTHP system can provide higher heat sink temperatures to overcome a significant range of industrial processes. Although the HTHP system has some limitations, the integration concept can provide heat sink temperatures above 100 oC with high efficiency. Thus, the integration of HTHPs into DHNs becomes a sustainable alternative to replace the natural gas boiler in the industry, overcoming the industrial thermal requirements with high efficiency and low-carbon emissions. Moreover, HTHPs have high operating flexibility, being able to modulate the heating capacity and therefore, provide the precise amount of heat required in each instant, as shown in Fig. 12b. This solution represents a sustainable low-carbon alternative to decarbonise the industrial sector and therefore, contribute to climate change mitigation.

(a) (b)

Fig. 12. Performance analysis of integrating HTHP into DHN (scenario DHN_{source}): a) COP_{HTHP} and b) heating capacity

4.2.2 Alternative refrigerants to HFC-245fa Similarly to the other scenario, the refrigerant HFC-245fa is used in the majority of the HTHPs and ORC system with the problematic of having high GWP of 858. Thus, it is necessary to analyse the alternative refrigerants available that have low-GWP and can be a proper substitute for the reference fluid HFC-245fa. Equal to the previous low-GWP refrigerant assessment, COP and VHC parameters are used as key indicators of the performance and installation size of each candidate, presented in Fig. 13.

HC-601, HFO-1336mzz(Z) and R-514A have the highest COP improvements. However, these refrigerants exhibit the highest VHC reduction and therefore, greater compressors are required to provide the same amount of heat. Contrary to the previous scenario, HC-600 manifests similar efficiency to HFC-245fa, but also, in this case, the highest increase of VHC, requiring a smaller compressor size than HFC-245fa. Finally, the HCFOs exhibit a balance between the COP and VHC indicators, being potential drop-in replacements in this application, specially HCFO- 1224yd(Z).

(a) (b) Fig. 13. Assessment for alternative low-GWP refrigerants to replace HFC-134a: a) COP and b) VHC.

4.3 Environmental analysis One of the main aspects of integrating HTHPs into DHNs is the potential to reduce the equivalent CO_2 emissions, becoming an additional resource in decarbonisation and climate change mitigation. Fig. 14a presents the CO_2 emissions of the DHN and the integrated system for DHN_{sink} scenario. In this figure, the CO_2 emissions of both systems are compared, providing the same amount of heat as a function of the ambient temperature. Significant CO_2 emission reductions of the integrated system compared to DHN can be appreciated in Fig. 15a, quantified at about 60% over the winter period.

On the other hand, Fig. 14b illustrates the environmental analysis concerning DHN_{source} scenario. In this case, the integrated system is compared with a natural gas boiler with an efficiency of 95% to overcome the same heating demand in the industrial processes, using two different electricity emissions factors. During the winter period, the integrated system can reduce the equivalent CO_2 emissions up to 98% for Sweden, whereas this reduction can up to 47% using the EU emission factor in order to have an average view. For the mentioned results, this industrial integrated solution can contribute to achieving the environmental targets of the European Union, aiming at decarbonisation of the industrial sector.

(a) (b) Fig. 14. Environmental analysis of the CO_2 equivalent emissions for the: a) DHN_{sink} scenario and b) DHN_{source} scenario.

4.4 TEWI evaluation

The use of alternative low-GWP refrigerants in HTHP systems can get better equivalent CO_2 emissions reductions due to the performance improve and lower GWP values compared to the reference refrigerants HFC-134a and HFC-245fa. Hence, TEWI evaluation illustrates the equivalent CO_2 emissions of each refrigerant, including the energy consumption and leakages. Fig. 15 presents the TEWI comparison for both scenarios, DHN_{sink} and $DHN_{sink.}$ Since most of the alternatives have low-GWP, TEWI analysis becomes more sensitive to the energy performance improvements of each refrigerant. HFC-152a shows lower equivalent CO_2 emissions than the other alternatives owing to its slightly high GWP value compared to the other low-GWP refrigerants. The rest low-GWP alternatives in both scenarios present similar equivalent CO_2 emissions reduction, and therefore, there is not a clear candidate only based on TEWI evaluation.

(a) (b) Fig. 15. Total equivalent warming impact (TEWI) evaluation for the: a) DHN_{sink} scenario and b) DHN_{source} scenario.

5. Conclusions

This work presented the integration of HTHP into DHN in two different situations, using the DHN as a heat sink and heat source. The target was to investigate the potential of this technology as an innovative and efficient solution to contribute to climate change mitigation. Thus, a twofold approach was followed:

1. Using the DHN as a heat sink, an integrated system solution was modelled in order to recover the waste heat from CO_2 refrigeration systems, upgrade and inject it into the DHN. A thermo-economic optimisation of the proper WHR exit temperature was realised to maximise the economic benefits of this integration. Moreover, alternative low-GWP refrigerants were assessed to replace the reference HTHP working fluid, HFC-134a.

Finally, an environmental analysis was provided to illustrate the potential equivalent CO_2 emission reduction in the DHN generation mix, adopting this solution.

2. Using the DHN as a heat source, HTHP modelling was based on experimental compressor data from an HTHP prototype in a wide range of heat sink temperatures. System parameters, such as COP and heating capacity, were calculated to provide a comprehensive performance evaluation of the HTHP integration. Besides, potential candidates with low-GWP to replace HFC-245fa were investigated as an environmental- friendly refrigerants alternative. Finally, an environmental analysis of this integration system was presented to illustrate its potential in the decarbonisation of the energy- intensive industry sector.

The following conclusions can be drawn from the results of this study:

• The integration of HTHPs into DHNs provides significant environmental and performance improvements, being an innovative and profitable system solution to decarbonise different sectors.

• Using the DHN as a heat sink, the COP_{WHR} results between 3.2 and 5.4, reducing the operating cost between 50% and 100% with an average price ratio of 2.25 compared with the standard CO_2 refrigeration system. With the DHN as the heat source, the integrated system provides a COP from 2.8 to 5.7 for a heat sink of 110 oC. A reasonable heat selling price to DHN will promote the application of this WHR technology due to the higher benefit for the prosumers

• HC-290, HFO-1234ze(E) and HFO-1234yf were considered the ideal alternative low-GWP refrigerants to replace the HFC-134a in new design installations. Moreover, HCFO- 1233zd(E) and HCFO-1224yd(Z) were the most promising low-GWP refrigerants to replace the reference working fluid, HFC-245fa.

• The environmental results, using the DHN_{sink} scenario, showed that the integrated system solution produces about 60% lower equivalent CO_2 emissions than the DHN generation mix over the winter period. Moreover, the DHN_{source} scenario can reduce the equivalent CO_2 emissions up to 47% using the EU carbon emission factor, whereas this reduction can be about 98% for Sweden. Therefore, this integrated system becomes a potential solution to replace the conventional natural gas boilers and reduce the greenhouses gas emission in the industrial sector.

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