ECOS 2020: Alternative solutions for the optimal integration of decentralized heat-pumps in district heating/cooling networks

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Abstract:

The new low-grade district heating (DH) networks operate at temperature levels of around 8-12°C and use decentralized heat pump (HP) units to raise the temperature in buildings for heating and domestic hot water (DHW) production. Such advanced networks have the advantage of being able to serve both for heating (in winter) and cooling (in summer) but correspond to significant pressure losses because of the low differential temperature on the primary network. Also, the performance of the heat pumps depends strongly on the temperature of the network. A high pinch differential temperature between the DH network and the temperature level of energy supply for heating and hot water process in the building corresponds to an important consumption of electricity in the decentralized HP units.

This paper presents new solutions and configurations of integrating heat-pumps in advanced district heating/cooling networks in order to minimize the pinch temperature while allowing high differential temperatures on the primary network, then minimizing both the flow rate on the networks and the total exergy losses.

Thermodynamic models of heat-pumps have been developed in order to simulate different configurations of integration where the performances in term of exergy efficiency on the substation are determined in function of the temperature of the network and the differential temperature on the primary network. The models have been applied: first to simulate the performance of a reference heat-pump system based on a classical “anergic” low-temperature (8/12°C) advanced heating/cooling network and also to compare with the proposed integrated HP system. Results show that an exergy efficiency of the substations units of more than 55% can be achieved for the proposed concepts, reducing electricity consumption of about 20% compared to the “anergic” DH network decentralized heat-pumps.

Keywords:

Heat pumps, Exergy, District Heating, Sustainability, ECOS Conference.

1. Introduction

In Europe, the heating and cooling demand in the residential sector is responsible for a share of about 40% of the final overall energy usage [1]. With new regulations on new building energy performance [2] and on the refurbishment of the existing building stock [3] the energy consumption in Europe by 2050 could show a reduction in the heating demand between 20% and 30% and a rise of about three times of the cooling demand compared to 2006 values [1].

The use of district heating (DH) could help lower both primary energy consumption and local emissions to cover the heating and cooling demand for buildings [4][5]. Traditional DH systems consist of centralized power stations that feed hot water or steam into pipes to distribute heat in urban areas. High-temperature DH systems still suffer from significant heat losses and high installation costs. With the fall of electricity prices and simultaneously the increase in market share of fluctuating power sources such as PV and wind energy, large centralized heat pumps started being used in DH mainly in Sweden and later also in Norway [6]. In these district heating networks, mainly waste water, sea and river water but also industrial waste heat are used as sources for the heat pumps. However, this type of DH also involves a few drawbacks which can limit their application in certain cases. For example, they are running at temperatures relatively far from the ambient temperature, giving rise to non-negligible thermal losses and operating with low-efficiency centralized heat pumps.

The urban environment offers several other options of waste heat, though at lower temperatures. A simple example is refrigeration units in shopping malls. In principle, these systems could also be connected to a high-temperature network through a high-temperature heat pump. However, this type of source alone typically does not justify the installation of a traditional district heating network. An alternative option to consider is a low-temperature network, with average operating temperatures of around 20 °C. The main advantages of using low-temperature network coupled to decentralized heat pumps are the decrease in thermal losses and the possibility of directly recovering low-temperature waste heat. The main drawbacks of such systems are several [7]: large diameter of pipes because of the low differential temperature between supply and return pipes (high mass flow rate on the primary network), high investment costs for substations based on HPs compared to traditional heat exchangers and high electricity cost for HPs related to the primary energy consumption.

This paper presents some possibilities of integration of HPs in a case study network and which increase the efficiency of substations while increasing the differential temperatures on the primary network. Current integration solutions are shown, followed by a proposed new integration

2. Heat pumps in district heating (DH)

Heat-pump (HP) system uses electrical or mechanical energy to transfer heat from a low temperature environment (cold source) to supply heat to a hot source (sink) with higher temperature levels. The most basic heat pumps use the atmosphere as a cold source. For better performance and higher capacity, heat pumps can also use the ground water or geothermal energy. The process by which heat transfer occurs is as follows: in a closed loop process a refrigerant working fluid is compressed, producing hot steam at high pressure (high temperature). Then the steam is condensed by supplying heat to the water hot source system (for space heating and/or DHW). In an expansion valve, the pressure and the temperature of the refrigerant are reduced making the evaporation at low temperature possible by recovering heat from a cold source. Finally, the vapored working fluid is used to feed the compressor at low pressure and low temperature.

Many previous works, carried out in several European countries, notably in Switzerland [8][9], Great Britain [10] but also in countries such as Denmark and Sweden [11][12][13] were able to demonstrate the application of heat-pumps to supply heat to traditional networks (hot water with temperature levels in general above 60°C), hence relatively high temperature levels compared to ones required for heating new and high-performance buildings (below 35°C). In addition to the linear thermal losses related to the high differential temperature between the networks and the environment (cold source), the heat transfer processes that occurred on the substations (decentralized heat exchangers connected to the network system to supply heat to buildings) are clearly inefficient with significant exergy losses.

Recent developments have led to a so-called “anergic” network operating at low temperature levels (~ 8/12 °C) and using decentralized heat pump (HP) substation units for heating and domestic hot water production. These HPs make it possible to raise the temperature to approximately 35°C for space heating and 60°C for the preparation of hot water. Such advanced “anergic” networks have also the advantage not only of being able to serve for cooling (in summer) all connected buildings by using simple heat exchangers but also to be able to integrate and recycle low temperature energy sources such as residual hot water, thermal losses from HPs and local renewable energy sources (solar thermal, biomass and geothermal). Some recent examples of operational “anergic” networks have been implemented in Zurich by Amstein + Walthert Engineering [14] (including the district heating networks of the Polytechnic institute, the Airport and the districts of Richti and FGZ) and also developed by Goupe E for the City of Tour de Peilz, which is fed by water from the Lake [15].

Based on the future evolution of the district heating network technology, there are some new concepts in development using refrigerant like carbon dioxide (CO_2) as a heat transfer fluid [16]. The use of such condensing fluid, with high pressure and low viscosity compared to water, has the advantage of developing the network pipes with smaller-diameter and with lower pressure drop. In addition, the performance of heat pumps that are used to meet heating and hot water requirements in buildings will be significantly improved, thereby reducing electricity consumption, ie at least 10% compared to the "anergic" water network [16][17]. An exergy efficiency of the cycles of these heat pumps of the order of 53% is estimated in a case study corresponding to the realization of such networks in the district of Rues Basses in Geneva. Nevertheless, much larger investment and operating costs than the anergic water network are required for this type of CO_2 network because of leakages and risks of transporting CO_2 at very high pressure eg. in the order of 50 bars. A high-performance level of sealing system is mainly required over a long distance of pipes with high-pressure refrigerant liquid and steam.

Currently, most of the DH are third-generation one. Technically, this system is based on hot water, which usually reaches a temperature of less than 90 °C. The water is transported by central pumping stations. In contrast to previous generations, the transport pipes, like most other components, are no longer insulated on site, but are prefabricated and pre-insulated in factories and can then be sunk directly into the ground on site. Compared to their predecessors, the distribution stations have a compact design and the heat exchangers are made of stainless steel. The heat is generated in central combined heat and power plants, in decentralized combined heat and power plants, by waste incineration plants and biomass power plants, supplemented by peak load boilers. Geothermal and solar thermal energy is also fed sporadically into such networks as a supplement.

The 4th generation DH is currently being developed. This generation is designed to integrate high share of variable renewable energy into DH by providing high flexibility to the electricity system. Lund et al. [18] have mentioned different possibilities that these DH must follow for example by recycling heat from low-temperature sources and integrate renewable heat sources such as solar and geothermal heat. Depending on the temperatures on the source and sink side, the heat pumps can operate with different efficiency. [8][19] shows that the average COP in the plant is 3.74, a COP between 5.4 and 6.5 can be achieved by using a different system (e.g. use of high temperature sources or increase in the return temperature network).

Zvingilaite et al. [20] proposed a 40°C low temperature DH water system using a booster HP for domestic hot water production. It operates by splitting the DH flow in two different streams: one going to the condenser of the booster HP to increase the temperature at 53°C for hot water production and one going to the evaporator for the source of the HP. Ostergaard et al. [21] performed models and simulation of a centralized HP combined with booster HPs by using high temporal resolution of the COP values for HP and the DH network losses. The authors found that The DH systems with booster HPs have a better annual performance compared to systems with individual HPs, as the increase of performance during the rest of the year would far compensate the relatively poorer performance of the HP booster system during the summer. Nevertheless, the authors have mainly used simplified models of HPs to calculate the COP by using a theoretical Lorentz efficiency multiplied by a constant real or exergy efficiency (of around 40% for boosters HP and 50% for the centralized HP). In this article, we develop a generic detailed model of exergy efficiency of HPs in function of the temperature of the network (cold source) by considering the isentropic efficiency of the compressor and the pinch temperatures on heat-exchangers (evaporator and condenser). Alternative solutions of using centralized HPs combined with decentralized booster HPs utilizing the DH water as heat source are analyzed and compared with respect to energy and exergy efficiencies. The temperature levels of the DH network are between 30 and 40°C. As opposed to different works referred to the literature, the integration of booster HPs systems in the DH network is realized in a way that allows production of hot water with high temperature levels (60°C) while allowing a high differential temperature on the primary network.

3. Case study of Bluefactory DH systems

3.1. An alternative solution of integrating booster HP in DH

BlueFactory is a fast-developing district within the city of Fribourg (Switzerland). The goal is to become a zero-emission neighborhood by 2040. It is in this context that the BlueCad project foresees the implementation of an intelligent heating network which wants to promote the integration of renewable energies as well as an energy efficiency. The concept of the BlueCad network is to use two different networks, a heat network and an anergic one (Fig. 1). The first is used for heating, while the second is an anergic recovery network that serves as a cooling source for the heat pumps as well as domestic hot water production (DHW), building cooling and ventilation. The network operates with two different regimes according to the seasons.

In winter, buildings need to be heated. Therefore, it is mainly the heating network that will be used. The heat is supplied by the three heat pumps in the production plant. However, the heat pumps are limited by their cold source, which is the anergic network. As a result, they only deliver 75% of annual needs.

In summer, it is the anergic network that will be mainly used. This network used energy from the geothermal probes as well as waste, clear and spring water. Therefore, the temperature of this network varies throughout the year.

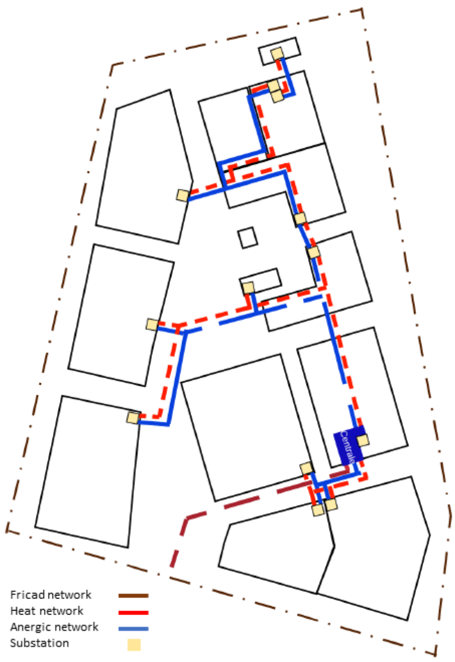

Fig. 1 BlueCad schematic network

As mentioned above, the heat network is supplied by three heat pumps. The heat pumps operate in parallel and use the anergic network a cold source. In addition to the heat pumps, a hot water tank serves as storage and helps to smooth the water temperature.

The operation of substations is relatively simple. In winter the heat network is used to heat the building. The heat from the network is recovered through a heat exchanger. In summer, the anergic network is used instead of the heat one to cool the buildings.

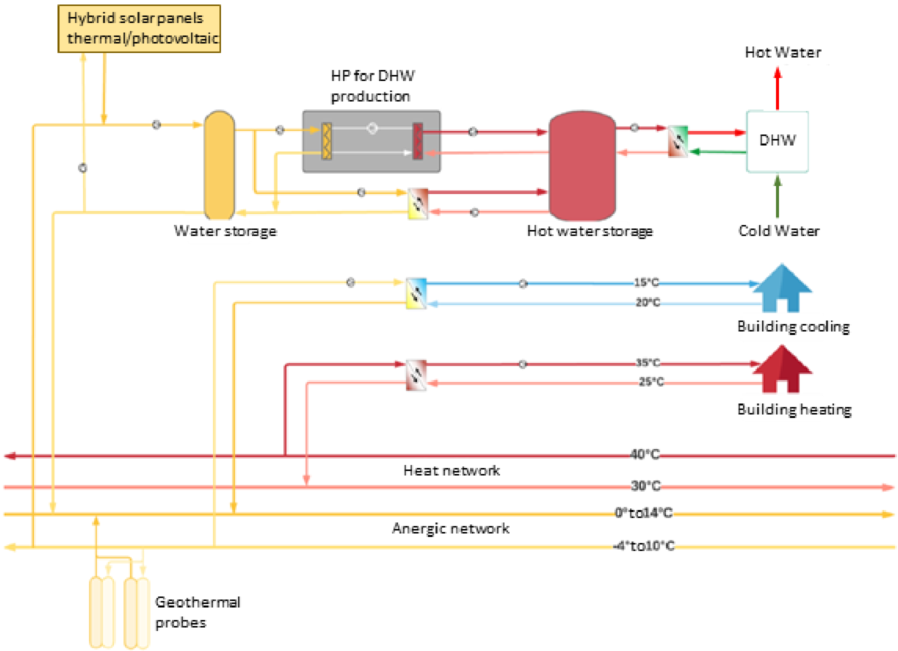

Fig. 2 Schematic representation of the initial variant substations

This network is also used for the DHW, the anergic network serves as a cold source for a heat pump. Geothermal probes are placed under some buildings and feed the anergic network. Therefore, they not only supply the building under which they are located, but also share their energy with the entire network. The temperatures presented in the Fig. 2 represent what network temperatures should look like in 2040.

The existing solution, proposed above, does not allow for a heat network using only renewable energies. Indeed, heat pumps are limited by their cold source and can only supply 75% of BlueFACTORY's annual needs. The remaining 25% comes from the Fricad network, which is the district heating network of Fribourg. However, this network uses 85% [24] of imported fossil or non-renewable energy; by 2035 the network is expected to use 54% of renewable energy.

Another difficulty encountered is the use of a low-temperature network. Usually higher volume flows and thus sometimes larger pipe diameter should be used in order to transport the corresponding heat in networks with low temperatures, due to smaller possible spread between flow and return.

3.2. New integration of decentralized HPs for high-differential temperature on the primary network

In the configuration shown in Fig. 3, the anergic network is only used as a cold source by the three heat pumps in the power plant. Moreover, the temperatures of heat networks are 30°C for the forward line and 15°C for the return one, which corresponds to a differential temperature of 15°C instead of 10°C for the initial variant. A higher temperature difference means a lower flow rate and therefore lower pressure losses and power consumption of the pumps [25]. In order to reach a higher temperature difference, it should be possible to increase the forward line temperature up to 40°C.

The substation described below operates only with the heat network, which is used for heating, cooling and DHW production. Firstly, the hot water goes through a heat exchanger (HX) which heats it up, then it is heated a second time using a water-to-water heat pump to reach 60°C. The next HX is used for building heating. The water the passes through two successive heat exchangers, the second one is also used for building heating. After that the water is cooled down by the same heat pump that was previously used to heat it. The cooling of the buildings is done by the following HX. The water then has the possibility to recover different energy flows (thermal solar, DHW waste). The last step is to pass the water through an air-source heat pump in order to obtain the desired temperature on the return line. However, these last two elements are optional.

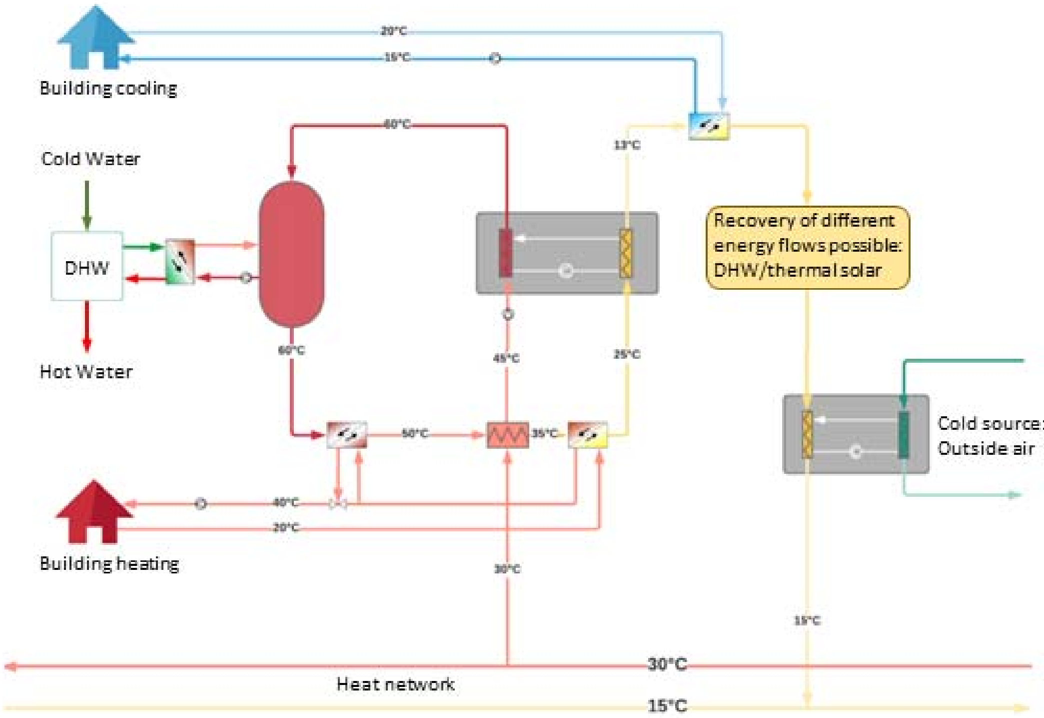

Fig. 3 Schematic representation of the new variant substations

4. Thermodynamic model of heat pumps in DH

4.1. Exergy efficiency in function of the temperature of the network

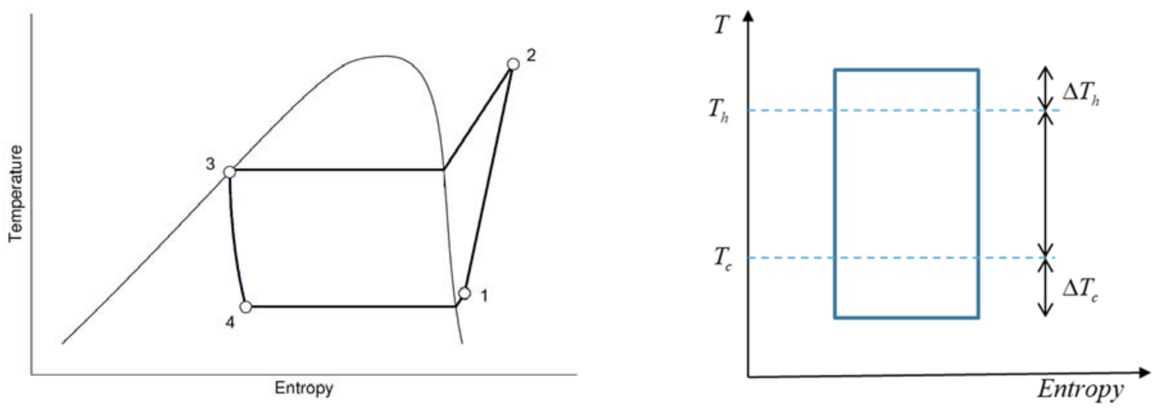

Fig. 4 : a) Real thermodynamic cycle of a heat pump, temperature versus entropy, b) illustration of pinche temperatures

Depending on the temperatures of the cold source and the hot sink side, the heat pump can operate with different efficiency. The highest possible efficiency or maximum coefficient of performance (COP_{max}) of a HP is defined by Eq. (1):

$$COP_{\max} = \frac{T_h}{T_h - T_c} \qquad (1)$$

Where T_h is the temperature of the heat sink and T_c the temperature of the heat source in Kelvin. As the real thermodynamic cycle of a heat pump is different from the ideal cycle (Fig. 4), the next step is to convert the COP_{max} of the ideal HP into the loss-related COP_{real} of the real HP by using the exergy efficiency of the cycle. For this purpose, the exergic efficiency η , which is also a function of the cold and hot temperatures, need to be determined for a specific operating point and multiplied by the corresponding COP_{max} as described in Eq. (3).

$$COP_{real} = \eta\left(T_h, T_c\right) \cdot COP_{\max} \qquad (3)$$

The exergy efficiency η of the HP can be obtained from the exergy balance equation and includes all losses that occur in the refrigerant circuit of the HP. It can be determined by the following equation:

$$\eta\left(T_h, T_c\right) = \frac{e_{qh} - e_{qc}}{e_K} \qquad (4)$$

Where e_{qh} represents the heat-exergy transferred to the sink by the refrigerant (condenser), e_{qc} the heat-exergy received by the refrigerant from the cold source (evaporator) and e_K the electrical exergy received by the compressor. Equation (4) can be expanded in order to obtain Eq. (5):

$$\eta\left(T_h, T_c\right) = \frac{\left(1 - \frac{T_a}{T_h}\right) \cdot q_h - \left(1 - \frac{T_a}{T_c}\right) \cdot q_c}{e_K} \qquad (5)$$

Where T_a represents the reference atmospheric temperature, q_c the heat received at the evaporator and q_h the heat provide from the condenser. Considering the energy balance equation of the HP cycle $(q_h = q_c + e_K)$, Equation (5) can be easily simplified to Equation (6):

$$\eta\left(T_h, T_c\right) = 1 - \frac{T_a}{T_h}\left[1 - \frac{q_c}{e_K} \cdot \left(\frac{T_h}{T_c} - 1\right)\right] \qquad (6)$$

In this simulation, we consider a saturated fluid temperature (T_1) at the inlet of the compressor and a constant (or mean-value) specific heat (c_p) between the inlet and outlet of the compressor, so the work supplied to the compressor and the heat received at the evaporator can be defined by Equations (7) and (8) respectively:

$$e_K = \frac{c_p T_1}{\eta_{Ks}} \cdot \left(\frac{T_{2s}}{T_1} - 1\right) \qquad (7)$$

$$q_c = T_1 \cdot \Delta s \qquad (8)$$

Where T_{2s} and η_{Ks} represent respectively the outlet isentropic temperature and η_{Ks} the global isentropic efficiency of the compressor respectively and Δ_s represents the differential entropy of the evaporation process. Combining the Eq. (7) and (8) and assuming that T_{2s} is equal to the condensing temperature of the refrigerant, the ratio q_c/e_K can be writen by Equation (9):

$$\frac{q_c}{e_K} = \eta_{Ks} \cdot \Delta s / c_p \cdot \frac{1}{\left(\frac{T_h + \Delta T_h}{T_c - \Delta T_c} - 1\right)} \qquad (9)$$

Where ΔT_h and ΔT_c represent the pinches differential temperature at the evaporator and condenser, respectively (see Fig. 4). Replacing the ratio q_c/e_K in Eq. (6) by the value in Eq. (9), the generic exergy efficiency η of a HP can be obtained by Equation (10):

$$\eta\left(T_h,T_c\right)=1-\frac{T_a}{T_h}\left[1-\eta_{Ks}\cdot\Delta s/c_p\cdot\frac{\frac{T_h}{T_c}-1}{\frac{T_h+\Delta T_h}{T_c-\Delta T_c}-1}\right] \tag{10}$$

It can be seen that the exergy efficiency of HPs is function of the temperature of the network (cold source), the isentropic efficiency of the compressor, the pinch temperatures on heat-exchangers (evaporator and condenser) and also the characteristic of the refrigerant which is represented by the ratio $\Delta s/c_p$.

4.2. Characteristic charts of Δs over c_p

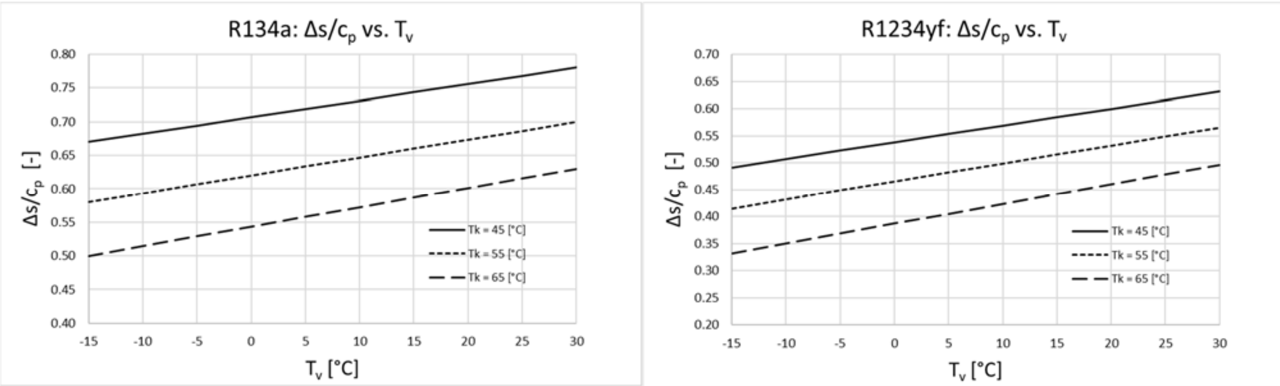

Fig. 5 Δs/cp versus the temperature at the evaporator: a) R134a, b) R1234yf

The ratio $\Delta s/c_p$ could be expressed as a function of the temperatures at the evaporator (heat source) T_v and condenser (heat sink) T_k in the HP. By using CoolProp [28], it is possible to extract this value for different evaporation and condensation temperatures. In Fig. 5, this ratio is displayed as a function on the temperature at the evaporator. The two graphs show that the relation between $\Delta s/c_p$ with the temperature at the evaporator as well as the one with the temperature at the condenser are linear. by interpolation, it is then possible to determine the Δs over c_p ratio as a function of the two temperatures.

5. Performance comparison with respect to energy and exergy efficiencies

5.1. Performance of HPs and results

By knowing the temperatures at the evaporator and condenser as well as the pinches temperatures, it is possible to characterize the performances and the efficiency of different HP. Fig. 6 represents the COP and the efficiency of three different heat pumps, based on water, air and glycol respectively. The same trends are visible for all HP, the COP increases when the difference in temperature decreases while the exergy efficiency shows the reverse behavior. However, the chart presented in Fig. 6 is only valid for a single heat source temperature.

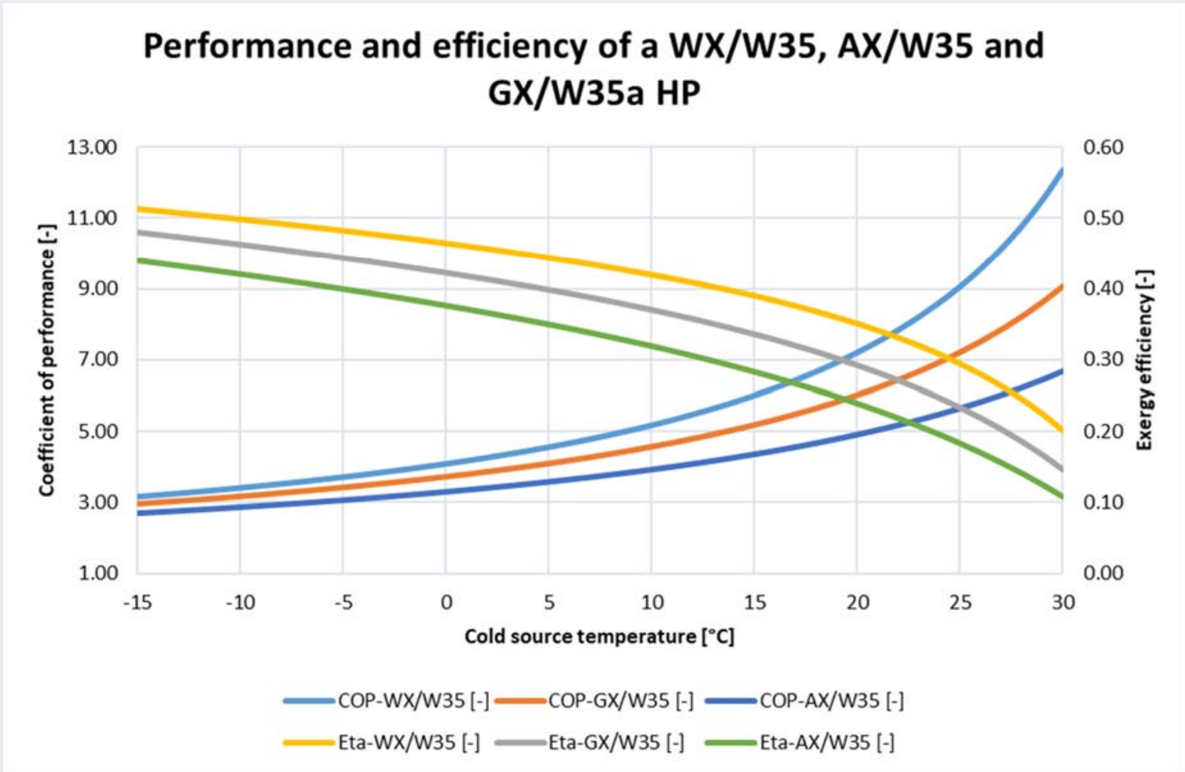

Fig. 6 Performance and efficiency of different HP

By using the characteristic chart developed in the section 4.2 as well as the Eq. (2), (3) and (7), it is possible to define the COP_{real} and the exergy efficiency for a wide source of heat sink (T_k) and heat source (T_v) temperatures, see Fig. 7. Thus, using the graphs in Fig. 7, it is possible to show that the COP_{real} of the HP proposed in the new variant shows an improvement of between 24 and 54% compared to the initial variant. Indeed, the new temperature difference allows the COP_{real} to reach 3.77, when using the anergic network, the COP does not exceed 3.05 or 2.46 (depending on the temperature of the anergic network). Moreover, the temperatures of heat network are 30°C for the forward line and 15°C for the return one, which corresponds to a temperature delta of 15°C instead of 10°C for the initial variant. A higher temperature difference means a lower flow rate and therefore lower pressure losses and power consumption of the pumps.

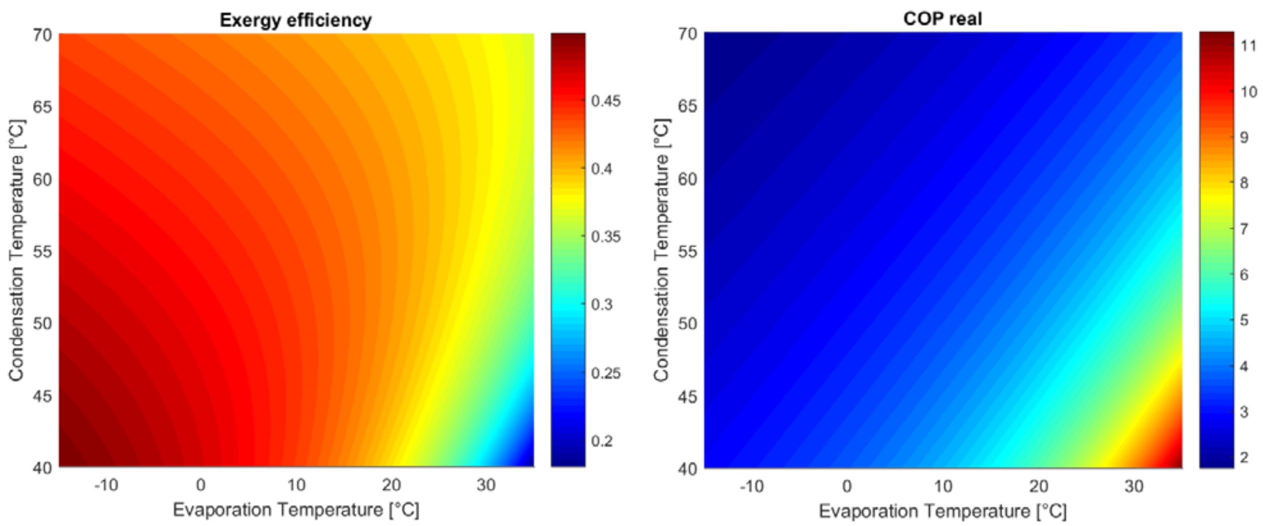

Fig. 7 Characteristics of a R134a based HP as a function of the evaporation and condensation temperature with a pinch of 8°C at the evaporator and 5°C for the condenser: a) Exergy efficiency, b) COP_{real}.

5.2. Integration of HPs and results

Three different integration variants for the case study of BlueFactory have been studied. Their main characteristics are presented in Table 1. The centralized HPs used in variant 1 and 3 use an anergic network (~ 7/10 °C) in which the heat comes from geothermal probes.

Table 1 Summary of the three variants

Variant	Building heating demand	DHW demand	DH supply temperature [°C]	DH return temperature [°C]
1	Centralized HPs	Centralized HPs	60	45
2	Decentralized HPs	Decentralized HPs	10	7
3	Centralized HPs	Decentralized HPs	40	30

The heat demand for the heating of buildings represents 5’640 MWh/year and the DHW demand 1’920 MWh/year. The COP of the different HPs have been determined using the equations described in section 4. In order to determine the total electrical consumption, the heat losses in the pipe as well as the electrical consumption of circulation pumps are calculated. The results of this analysis are presented in Table 2.

The use of the third variant allows a reduction in electricity consumption of about 34% compared to the first one. The second variant makes it possible to greatly reduce heat losses, but the small temperature difference (3°C) requires a large flow rate and therefore higher head losses, which are reflected in the consumption of the pumps. The third variant seems to offer a good compromise between the heat losses and the head losses.

Table 2 Results of the three variants

Variant	Centralized HPs COP [-]	Decentralized HPs COP [-]	Yearly electrical consumption [MWh]	Electricity savings
1	2.85	-	4 119	-
2	-	2.85	3 968	3.7%
3	4.58	5.56	2 720	34.0%

6. Conclusion

The purpose of this paper was to analyze various HPs integration proposals in a DH form an exergy point of view. And then to estimate the COP of the different variants. The development of the exergy efficiency formula has highlighted the influence of the ratio Δs over c_p on HPs efficiency. This ratio has been calculated for different fluids. A linear relationship between evaporating and condensing temperatures made it possible to express this ratio as a function of HPs operating conditions. The performance calculation showed an improvement of the COP between the initial variant and the new one which varies between 24 and 55% depending on the operating conditions. Further investigations are needed to identify the influence of the new substations on the whole DH as well as the influence of operating temperatures on the efficiency of the HP’s compressor.

Nomenclature

Roman symbols

c specific heat, J/(kg K)

E exergy, J

q heat transferred to the system, J/kg

T temperature, K

W work provided to the system, J

Greek symbols

η efficiency

Subscripts and superscripts

c cold source

h hot source

in that enters the system

is isentropic

k condenser

out that goes out of the system

p at constant pressure

v evaporator

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