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Exergetic evaluation of heat pump booster configurations in a low temperature district heating network

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Abstract:

In order to minimise losses in a district heating network, one approach is to lower the temperature difference between working media and soil. Considering only direct heat exchange, the minimum forward temperature level is determined by the demand side, as energy services are required at a certain temperature. As domestic hot water is required at a temperature range where legionella is no longer a threat, forward temperatures in a traditional low temperature district heating network cannot be lowered beyond approximately 55 °C. One solution is to boost the temperature of the forward tap water stream with a heat pump, as the remaining heat demands are often not required at temperature levels as high as the tap water. The scope of this work is to evaluate the power consumption and second law efficiency of booster heat pumps for tap water production in a low temperature district heating network. The heat pump and storage arrangement is evaluated based on a tapping sequence from the Danish standards (DS439). Based on an initial investigation of possible designs, three configurations have been chosen for the evaluation. Of the three heat pumps, two are implemented on the primary side to boost the network stream, and one is intended to increase the temperature of the tap water directly. Results show that one of the three configurations are superior to the two remaining, when considering temperature levels of forward stream between 35 °C and 47 °C. The overall results remain the same regardless of heat exchanger sizes and the isentropic efficiency of the compressor used in the heat pump. The superior configuration shows exergetic efficiencies higher than 0.5 when forward temperatures is around 45 °C.

Keywords:

Domestic hot water, Exergy analysis, Heat pumps, Low temperature district heating.

1. Introduction

Using district heating in urban areas is a measure to increase overall energy efficiency and reduce consumption of fossil fuels. These systems are implemented in many northern cities and even rural areas where incineration plants provide surplus waste heat. As the market value of heat is increasing (due to numerous reasons - mainly due to increase in fuel prices), so is the interest in lowering the losses affiliated with transportation of heat. One simple measure is to reduce the temperature of the network, as this reduces the driving potential of the heat loss in the distribution system.

Novel parts of existing Danish district heating networks tend to be built with a forward temperature of around 60-55 °C [1] as this is the lowest temperature for which direct conversion into domestic hot water is possible. Domestic hot water (hot tap water) and space heating are the common heat demands in residential areas, of which the domestic hot water constitute approximately one third of the combined consumption [2]. Lowering the forward temperatures of the district heating network could potentially be beneficial, if only a small amount of electricity is required to increase the temperature of the tap water, while the temperature is high enough to provide space heating without using additional means. In this way heat losses of the combined district heating stream can be minimized while using only a small amount of electricity to boost the temperature of a minor part. Many of the new networks are coupled to the existing district heating networks. In case of the build of a completely new network and production unit (combined heat and power plant or district

heating boiler) several effects may be experienced from changing the temperatures levels of both forward and return in the network [3]. Changed production or efficiencies of these production units are not considered in this paper, as the entire production facility and district heating network must be changed for these effects to become realized.

Several heat pump solutions have been considered in the on-going research affiliated with this paper. Below the most promising candidates are evaluated based on electricity consumption, district heating network considerations and exergetic efficiency.

2. Concept considerations for low temperature DH systems.

2.1. Main obstacles

In trying to reach a lower supply temperature in the district heating system - beyond 55 °C, new steps must be taken to utilise the heat, as several constraints appear in this temperature range. In residential areas, the load for the district heating system consists mainly of two parts; space heating and hot tap water.

For space heating, the temperature difference between indoor heaters and the room temperature is minimised when using the lowered temperature in the system. Assuming a constant heat demand, the low temperature difference requires larger surfaces for heat transfer. In these situations floor heating is often utilised. Still quite some temperature difference is needed, as building materials are often inferior to slim iron constructions in terms of heat transfer. A minimum of 15 K higher floor heating inlet temperatures, compared to the required room temperature is considered a requirement in this evaluation [4]. In addition to this heat transfer consideration, the flow rates and pressure losses in both the district heating network and the house installations must be considered before choosing the appropriate temperature levels.

Considering the tap water requirements, the main issues are related to the bacterium “Legionella”. To prevent problems with bacteria two simple measures can be taken. Either the hot tap water must exceed a predefined temperature limit where the bacteria can no longer exist when stored, or the tap water is not to be stored after being heated. Either way, some constraints are encountered.

Additionally the Danish building standard must be met, where hot tap water is assumed at two temperature levels – 45 °C and 40 °C, respectively, differentiated by their use in kitchens or bathrooms. Even with small pinch temperature differences in the heat exchanger network, it is unlikely that forward temperatures in district heating can be reduced below 50 °C without considering heat pumps or other efforts to increase the temperature of tap water. In order to evaluate an overall conversion efficiency of systems with very low forward temperature (below 50 °C), small heat pump installations for individual houses are considered.

2.2. Different implementation schemes

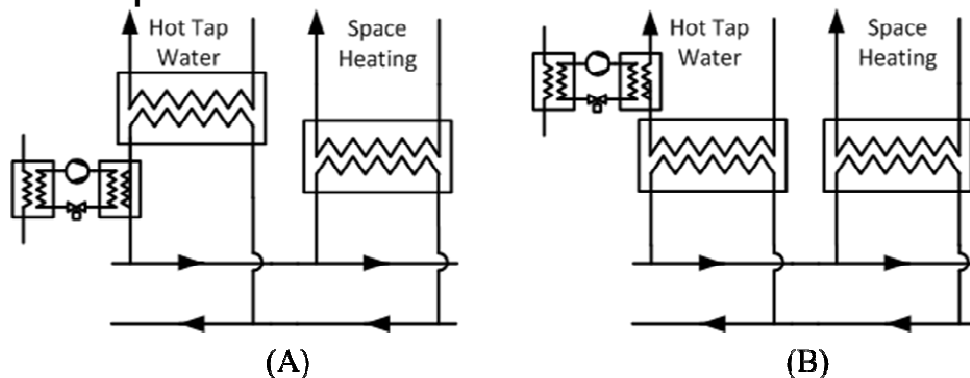


Fig. 1. Two different implementation schemes: (A) Heat pump on primary side of the tap water heat exchanger. (B) Heat pump on secondary side of the tap water heat exchanger.

In individual house installations for low temperature networks, heat pumps can be implemented in two different operating schemes, either to boost the temperature level of the district heating water prior to heat exchange with the tap water (named “primary”), or to boost the tap water temperature after the district heat network heat exchanger (“secondary”).

Within these two schemes, many individual concepts are plausible. Several different conceptual ideas have been tested and evaluated, based on “back of the envelope” calculations. The three most promising concepts are presented in this paper. This focus is to evaluate the most promising candidates in terms of energy efficiency. The evaluated systems consist principally of the tap water heat exchanger, heat pump and the storage system. The evaluation is considering both first and second laws of thermodynamic.

The results presented are intended for further analysis, as the impact of reducing supply temperature will influence the entire district heating system, among others; space heating requirements, pressure losses, cost of implementation and dimensioning of the piping system. Tap-water corresponds to between one half and one third of the combined heat consumption in the house.

2.3. Assumptions

The calculations are based on the assumptions presented in Table 1. Assumptions are made based on estimates of state of the art technology for a small decentralized heat pump producing hot tap water by use of low temperature district heating network.

Table 1 - Assumptions for low temperature district heating network heat pump

Variable	Assumption
Pinch temperature in Tap-water HEX ($Q_{MAX}=32$ kW)	8 [K]
Initially assumed forward temperature of DH network	40 [°C]
Initially assumed return temperature of DH network	22 [°C]
Refrigerant	R134a
Isentropic efficiency of compressor	0.5 [/]
HEX pinch temperature difference in both Condenser and Evaporator	2.5 [K]
Hot tap water	45 [°C]
Tap water in	10 [°C]
Minimum temperature if water stored on secondary side	58 [°C]

In the conducted calculations heat exchange between district heating water and tap water is assumed to have a constant pinch temperature of 8 K, as high flow rates occur in the tap water system. The assumed pinch temperature corresponds to the highest flow of tap water, but is assumed constant across the entire range of tap water flows. In practice the temperature difference would decrease at lower flow.

As the temperature difference between the forward and return stream of the district heating network is reduced (by a factor of minimum 2) [2] while assuming no change in the demand profile, significantly higher flow rates are required in the district heating network. Furthermore the high flow rates in the system will require high heat exchanger area and intermittent operation of the heat pump. To reduce these issues, storage of hot water is introduced in each scenario. The storage is regarded as a means to lower heat exchanger sizes and service life of components and will as such require an economic optimisation, which is not part of this paper.

In order to dimension the different heat pumps and storage tank sizes, the heat demand profile from DS439 is used [5]. As the recovery time for the system (storage empty -> storage full) is not expected to exceed 3 hours, only the time interval between 6.00 AM and 7.05 AM (morning showers and cooking) is considered in these calculations, as the time until next tap is almost 2 hours according to the standard. Only for the tapping sequence from 6.00 AM to 7.05 AM the full capacity of the heat storage will be needed. The preceding hours are assumed without any tapping,

thus the storage can be full before the tapping sequence. In the calculations presented below, the interaction between heat pump and storage tank is dimensioned to allow a “refilling” (leaving a heated volume of water corresponding to the desired) in two hours.

Regarding heat storage and heat pump on either primary or secondary side, some assumptions are introduced:

- With heat pump and storage on primary side of the network, only the tapping temperature dictates the temperature of the storage in the calculations.
- Employing the Heat pump on the secondary side of the system, the tap water is stored at high temperatures. Concern must be regarded towards legionella, so the heat pump system must be able to prevent and even remove the bacteria. Taking into account some of the heat losses that may emerge in a real system, the heat pump must deliver the tap water at minimum 58 °C.

The profile presented in figure 2 corresponds to the tapping and refilling profile considered. The concept considered in the figure corresponds to configuration A, but identical profiles are experienced in the two remaining configurations. The tapping sequence is assumed to correspond to a tapping temperature of 45 °C during the entire profile (this is a small offset from standard – where some are 40 °C).

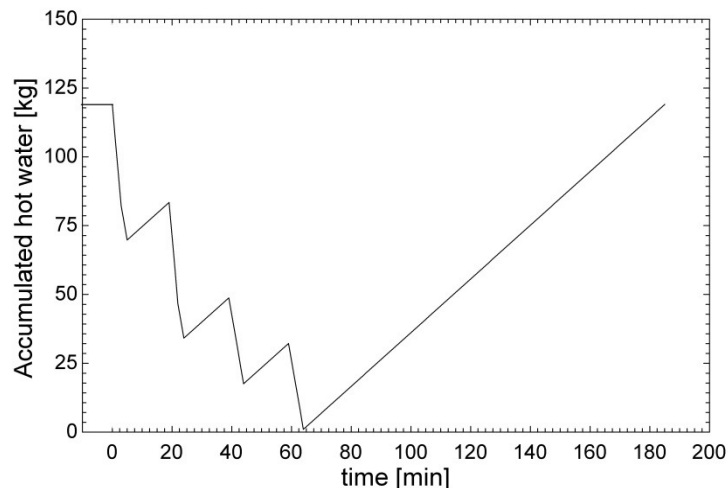


Fig. 2. Tapping and refilling sequence considered.

It is assumed that the heat pump is in operation from the initial tapping sequence and until the heated water volume is restored. The heat pump is working continuously during the tapping procedure in order to reduce the required amount of stored hot water. The tapping and refilling sequence is presented in Fig. 2. Thus proper dimensioning of the heat pump capacity can reduce the required volume of storage. Heat loss from the hot storage of water is neglected, as an almost equivalent amount of stored hot water is required in all the configurations at equally comparable temperature levels.

3. Method

Numerical models have been implemented in Engineering Equation Solver (EES) [6], corresponding to each individual heat pump implementation scheme. Operation assumptions are listed either in Table 1 or in the section considering each individual heat pump solution. The calculation of the state of all streams is primarily based on energy and mass balances. Pressure losses in heat exchange and pipes are neglected throughout the paper.

Heat exchange is modelled according to Nellis and Klein [7] using pinch temperatures in heat exchangers both with and without phase change. The used formulation of pinch point results in

lowered condensation pressure as the pinch point is not assumed at either end of the considered heat exchanger, but at the location of minimum temperature difference.

Calculation of the exergetic efficiency is based on the formulation of physical exergy presented in Bejan et al [8]. There are no changes in chemical composition of the working media or district heating media, leaving only changes in the physical exergy of each separate stream:

$$E_i^{PH} = \dot{m}_i((h_i - h_0) - T_0(s_i - s_0)), \quad (1)$$

Massflow \dot{m}_i , enthalpy h_i and entropy s_i is based on the above mentioned EES calculations for each concept. The dead state is based on $t_0 = 10^\circ\text{C}$ and $p_0 = 1\text{ bar}$, from where h_0 and s_0 can be calculated for the working media. The dead state is related to the cold tap water at ambient pressure. Exergetic efficiency is modelled according to the formulation in Bejan et al. [8]. As exergetic efficiency is calculated as a relative term, the location of the dead state does not matter for the final results presented [9].

4. Individual concepts and initial calculations

4.1. A (primary side)

The heat pump is modelled according to the simplified PI-diagram presented in Figure 3. The forward stream supplies DH water for both the evaporator and the condenser. The two streams are mixed in the return flow, combining the residue heat from the evaporator and tap water HEX. During tapping, heated water is removed from the hot layer in the stratified tank, heat is transferred in the tap water heat exchanger and returned to the cold bottom layer in the tank. This is done to avoid high mass flows of district heating water in the heat pump condenser and in the district heating network. During recharging heated water is filled in the tank, displacing the bottom cold layer, which is returned to the District heating network.

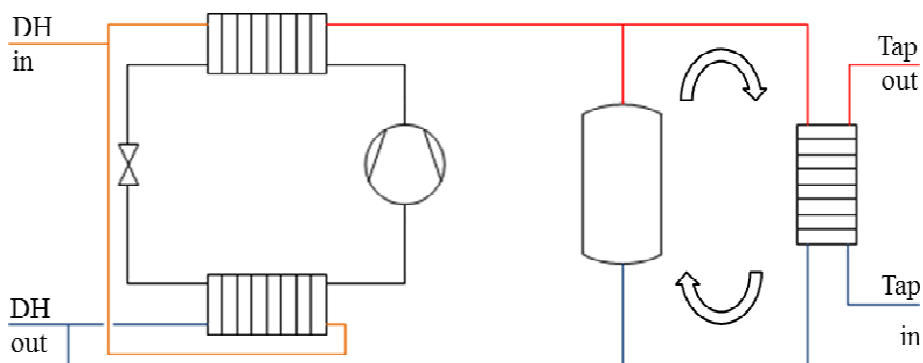


Fig. 3. Simplified diagram of A, with arrows to indicate the short circuit during tapping

Table 2. Initial calculations of variant A based on information from table 1.

Variant	\dot{V}_{DH} [m ³ /h]	Condenser [kW]	P [kW]	Heat pump COP [l]	Water Volume [m ³]	Exergetic eff. [l]
A	0.107	0.89	0.157	5.62	0.118	0.44

The results shown in Table 2 indicate that condenser capacity of approximately 0.9 kW is required in order to boost the temperature of the district heating water. The temperature levels in both the condenser and the evaporator allow the heat pump to operate with a COP of approx. 5.6. The combination of the tapping profile and the temperature of the boosted storage dictate the required amount of water in the storage.

4.2. B (primary side)

Source heat for the heat pump system can also be supplied from other sources than the forward district heating line. Heat can be extracted from space heating return flow, or even from the system return line. High temperatures in the heat supply for the evaporator is of cause an advantage in order to minimize the temperature lift between condenser and evaporator. The advantage of this system is a reduction in the district heating network forward flow compared to the variant A. This is achieved by increasing the temperature difference between DH forward and return beside the assumptions in Table 1.

To allow evaluation of introducing additional “waste” heat before the evaporator, two different calculations is performed in this variant.

- ‘B1’ is only using the return stream from either the tap water heat exchanger or the storage tank.
- ‘B2’ is an additional amount of return flow (most likely from the space heating circuit) with temperature 22 °C and mass flow corresponding to the assumption that the district heating requirement of a house can be divided into 2/3 space heating and 1/3 tap water [2]. The additional flow is subject to some uncertainties, as it is not always likely that the space heating flow is available when the tap water is required. On the other hand, utilising the space heating return flow would enable a lower return temperature than the one otherwise considered, which is dictated by the space heating heat transfer.

Figure 4 presents the simple flow diagram. The concept is quite similar to A, except for the addition of surplus waste heat prior to the evaporator.

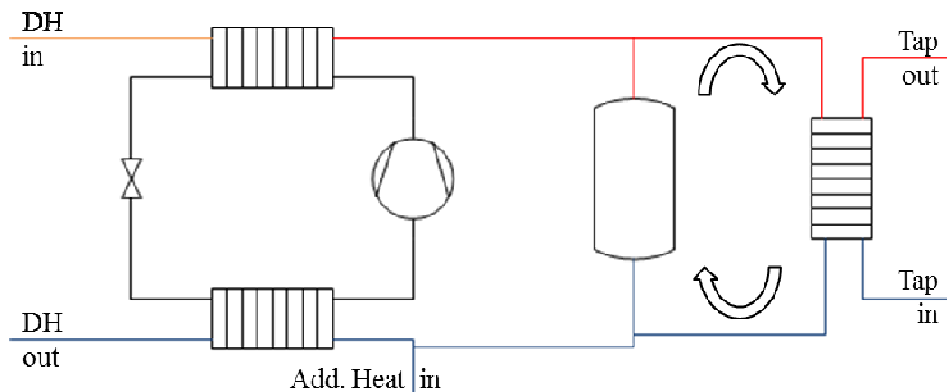


Fig. 4. Simple diagram of variant B (primary), with arrows to indicate the “short circuit” during tapping

Table 3. Initial calculations of variant B based on information from table 1.

Variante	\dot{V}_{DH} [m ³ /h]	Condenser [kW]	P [kW]	Heat pump COP [l]	Water Volume [m ³]	Exergetic eff. [l]
‘B1’	0.059	0.89	0.252	3.52	0.118	0.38
‘B2’	0.059	0.89	0.207	4.27	0.118	0.42

Table 3 shows the initial calculations of both variant B1 and B2. The condenser load is in both cases equal to the one presented in table 2, as the hot DH water stream for the condenser is identical to the one in variant A. Due to the changed temperature levels of the evaporator in both B1 and B2 the heat pump COP is changed, which calls for a higher electricity consumption. Considering available surplus heat (according to ‘B2’), the system efficiency improves, as this reduces the temperature lift between heat pump sink and source. As the system changes only influence the evaporator, the new system provides similar effects with a variation in forward temperature. In

cases with no additional heat requirements in the house, the heat pump unit will operate on only the return stream from the tap water heat exchanger.

As the two systems have similarities in operation, only the ‘B1’ system is considered for further analysis, as this system composes the simple solution, where both streams from Figure 1 are not available in the same location due to practical constraints.

4.3. C (secondary side)

The last variant proposes the most efficient solution for boosting the tap water with the heat pump (secondary side implementation). The configuration of this system allows preheating to be utilized in an efficient way, where the high flow rates of the tap water does not influence the temperature lift. The forward stream of the district heating network is supplied both to the evaporator of the heat pump and the heat exchanger for preheating of tap water. In modelling the system the preheater was considered both as a tap water heat exchanger (pinch temperature in tap-water HEX = 8 K) or as a separate type (pinch temperature in HEX = 2.5 K = Condenser pinch temperature). As only a limited constant stream of tap water is heated, the heat exchanger (named ‘preheater’) was assumed to resemble the condenser based on the load profile. The pinch temperature defines the thermal load of the heat pump, and as such the losses in this heat exchanger must be minimised for efficient water heating.

The simple diagram of B2 is presented below in Figure 5. The arrow represents the continuous heating of tap water, which is independent of the tapping stream. The high flow from the Tapping procedure will only affect the amount of hot water in the stratified tank.

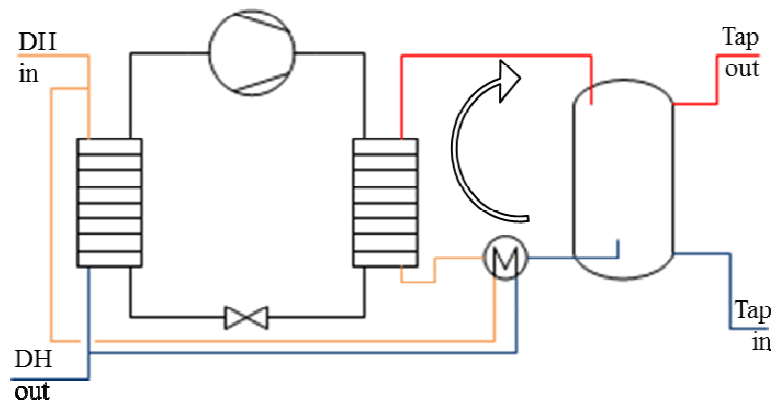


Fig. 5. Simple diagram of C, the arrow represents the continuous heating of tap water through the heat pump.

Table 4. Initial calculations of variant C based on information from table 1.

Variant	\dot{V}_{DH} [m ³ /h]	Condenser [kW]	P [kW]	Heat pump COP [∕]	Water Volume [L]	Exergetic eff. [∕]
C	0.105	1.02	0.193	5.26	0.086	0.40

Table 4 reveals a slightly increased heat pump condenser load is in variant C compared to the primary configurations. The increased load is due to the heat exchanger losses introduced in the secondary solution, where the district heating water is directly used (without temperature loss) in variant A and B. The configuration has a slightly lower requirement of DH flow than configuration A, and a lower storage volume than the both A and B.

As the heated water in the tank is hotter than the desired tapping temperature, cold tap water is mixed with the hot tap water during tapping, as is common practise when using district heating water today.

4.4. Evaluation

Based on table 2 to 4, a simple evaluation of electricity consumption and exergetic efficiency is possible. However, a variation of some of the parameters from table 1 may reveal changes in performance of the different booster configurations.

Heat exchanger sizes is a major interest, as the assumptions in Table 1 may not prove the economic optimum in later calculations. Other economic evaluations may include improvement in isentropic efficiency of the heat pump compressor, which may become possible through the use of different compression technologies and/or development of a compressor specifically designed for the temperature levels of the booster heat pump.

Heat exchange pinch temperature difference: In the evaluation of different heat exchangers, an increase in pressure losses from a decrease in pinch temperature difference is neglected. Such pressure losses would only affect the heat pump performance, as the pressure difference between forward and return DH stream is controlled at the district heating central, and as such not included in this paper.

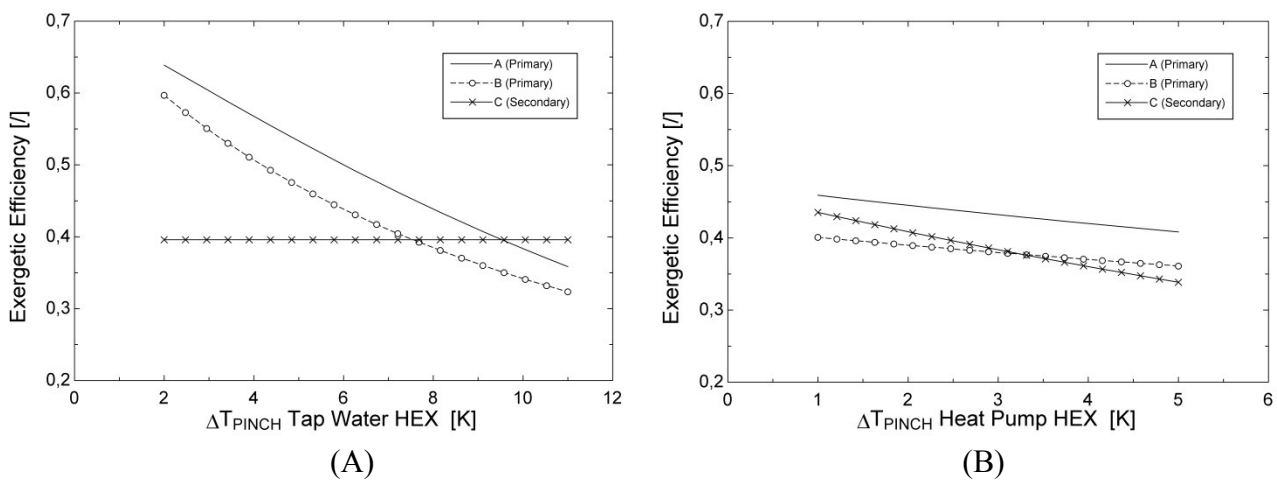


Fig. 6. (A) The impact of tap water HEX pinch temperature on the 3 proposed configurations. (B) Impact of pinch temperature difference in evaporator and condenser on the 3 proposed configurations.

From Figure 6 (A) it is clear that the tap water HEX performance will influence the efficiency of Variant A and B, indicating that with poor heat exchange in these system, optimal performance will shift from variant A to variant C (as described in section 4.3, the heat exchanger in configuration C is not regarded as a tap water heat exchanger due to the constant flow rate of the HEX). The steeper gradient of variant C in Figure 6 (B) is due to a higher number of HEX controlled by this pinch temperature difference (same explanation as in Fig. 6 (A))

Isentropic efficiency of heat pump compressor: The evaluation presented in Figure 7 cover a broader band of isentropic efficiency than what is reasonable to expect. A compressor for high temperature heat pumps in the condenser capacity range expected and at a reasonable cost is unlikely to have a higher efficiency than 0.65 [] [9].

The evaluation presents the COP (coefficient of performance) for the heat pump pack and exergetic efficiency for the combined system with variable isentropic efficiency of the compressor. It is noticeable from Figure 7 (B), that an increase of isentropic efficiency above 0.65 [] changes the relation between variant B and C. Configuration A has the highest performance of the three in both fig. 7 (A) and (B).

The difference between the first and the second law evaluation of performance is the influence of the condenser load on the electricity consumption. The increased load for the heat pump in configuration C is mainly due to the pinch temperature differences discussed above.

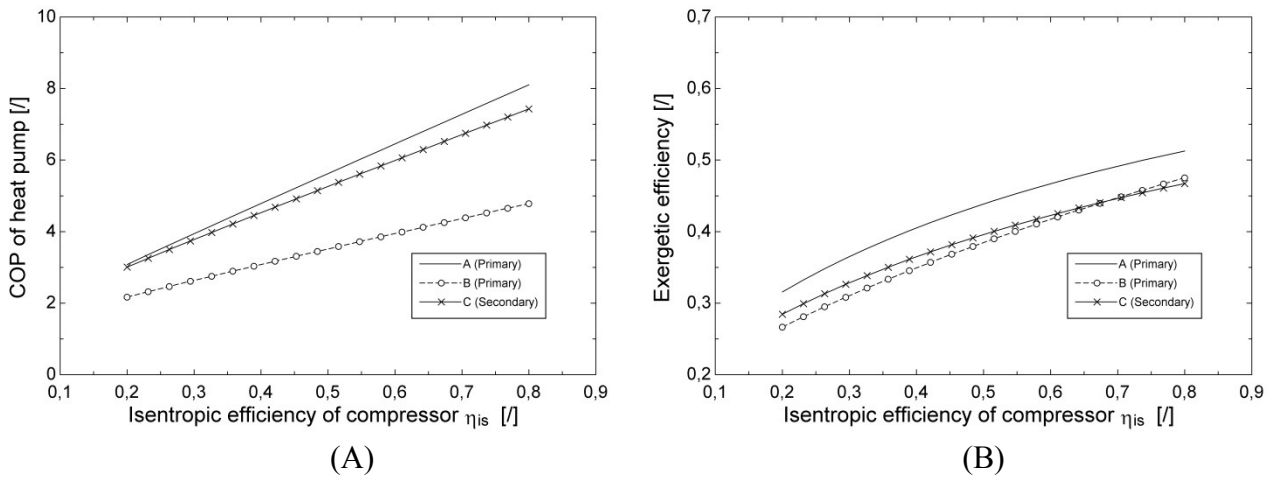


Fig.7. (A) COP of heat pump with variable isentropic efficiency for each of the 3 proposed configurations. (B) Exergetic efficiency of individual configurations with variable isentropic efficiency.

5. Results

5.1. Variation of forward temperature of the DH network

The forward temperature of the DH has a high impact on the system performance, as the temperature is directly linked to the heat pump capacity and temperature lift in all the different configurations. Figure 8 shows the variation of the described configurations in terms of both volume flow of district heating water and electricity consumption, with variation in forward temperatures of the district heating network. Electricity consumption is presented as a function of the product – this is to represent how much power (and the remaining heat load) is required in order for the system to produce one [kWh] of hot tap water at 45 °C according to the assumptions explained above and the Danish building standard.

Heat is calculated on the basis of enthalpy difference between forward and return temperatures (in the case of Figure 8 the return temperature can be found in Table 1). The full heat content between forward and the lowered return temperature can be found by subtracting the curve of variant B from Figure 8 (B) from the product (energy balance calculation where the product is 1).

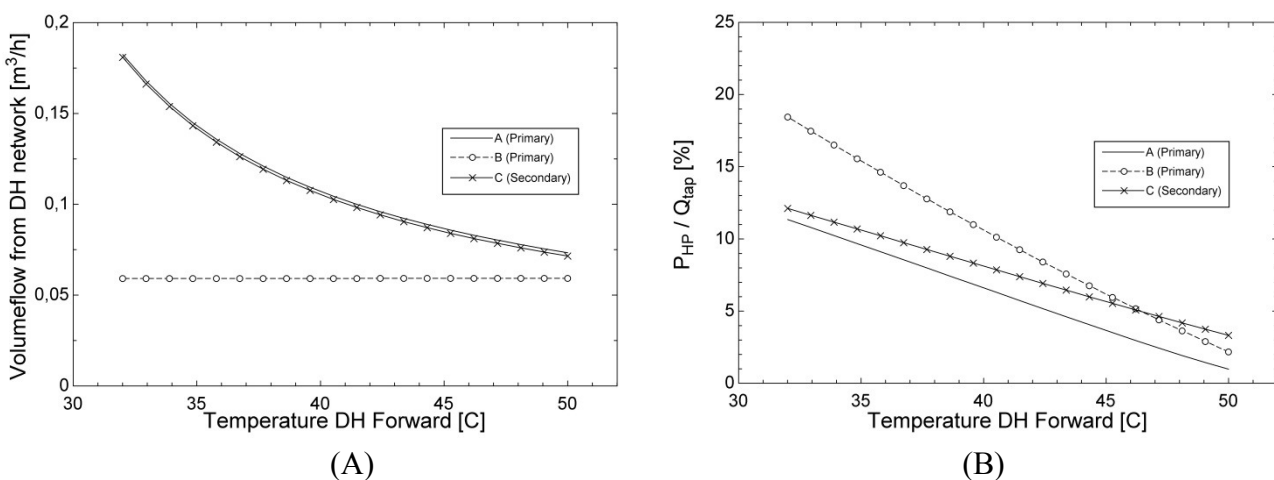


Fig. 8. (A) Required volume flow of hot DH stream with variable forward DH temperature. (B) Relation between electricity consumption and product with variable forward DH temperature.

When the DH forward temperature approaches the temperature required for tap water heating (53 °C for primary side when considering a pinch temperature of 8 K, 58 °C for secondary side

according to the assumptions of Table 1), the consumption of electricity is reduced significantly; while almost the full energy flow is required from the district heating network. This is due to the significantly reduced thermal load in each of the heat pump configurations. As discussed above, configuration C has a slightly higher condenser load as in variant A, thus increasing the electricity consumption for the heat pump correspondingly at all temperature levels.

5.2. Variation of return temperature of the DH network

Changes in return temperatures are highly important, as not only the electricity consumption of the heat pump booster configuration is affected, but also the temperature difference between the forward and return of the district heating network. Thus the optimal heat pump must perform with high efficiency in a range of high temperature differences between forward and return temperatures.

Assuming tap water at 10 °C, and a finite heat exchanger (8 K), 18 °C is the lowest reachable temperature for the return water in the district heat system by direct heat exchange. Lower temperatures can only be achieved by using the heat pump evaporator to cool the stream further, which in this study only is considered in variant B.

With an increase in return temperature of the district heating network, power consumption is reduced as the evaporation temperature of the heat pump refrigerant can be increased. An evaluation of the heat pump characteristics with a change in return temperature is considered in Figure 9 (constant forward temperature corresponding to Table 1). From the curvature of variant A and C in Figure 9 (B) it is clear that an optimum exists if the district heating water from Figure 9 (A) has a change in value.

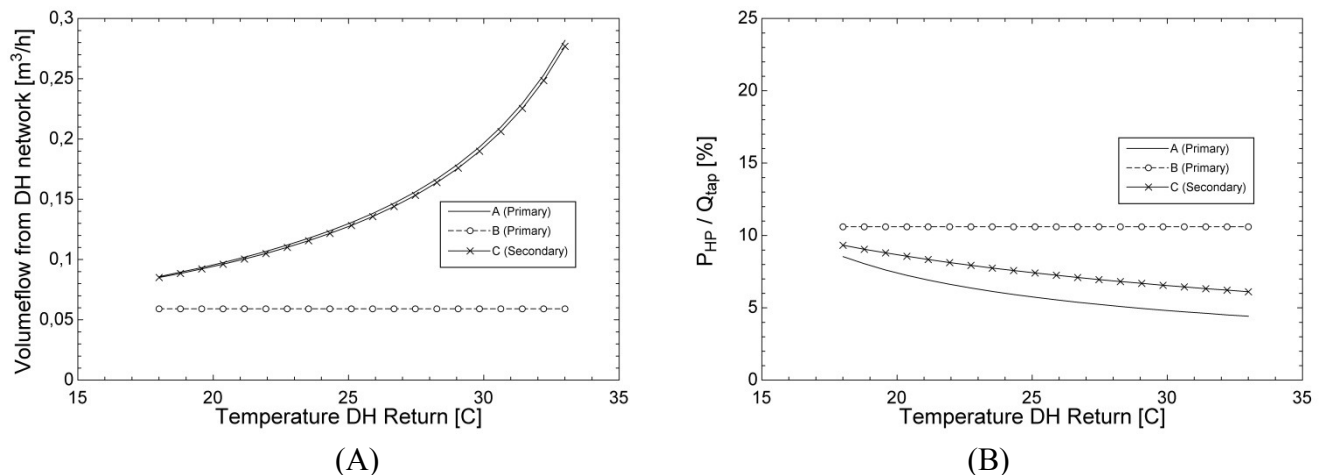


Fig. 9. (A) Required volume flow of hot DH stream with variable forward DH temperature. (B) Relation between electricity consumption and product with variable return DH temperature

The differences between configurations A and C are quite hard to spot in Figure 9. In principle it does not make sense to display variant B, as the return temperature of the DH network is not controlled. The visible changes in Figure 9 (A) correspond to the previously addressed wish to show the differences in flow of DH water required to fulfil the tapping process.

5.3. Comparison of results using exergy

Exergy is used as another way to evaluate the performance of the different concepts. In this evaluation the different temperature levels of the district heating network is evaluated. Exergy is furthermore a good evaluation parameter when more than one fuel stream combine into only one product, as optimum between the different fuel streams is easily spotted.

The lower electricity consumption of configuration A is rewarded in the calculation of exergetic efficiency throughout the entire range of forward and return temperatures considered in the paper. Considering the initial calculations, and the sensitivity study of heat exchanger performance and

isentropic efficiency, the distribution between the performances of the individual configurations is distinct.

Figure 10 shows the exergetic efficiency of the three different variants, considering both the forward and the return temperatures. From Figure 10 (B) it seems that the second law efficiency is not improved with a return temperature above 25 °C in either of the cases, because the trade-off between reductions in electricity consumption is no longer compensating the increased exergy content of the heat consumption. The influence of pressure losses on exergy destruction is not considered in the systems and would lower efficiency further at the higher temperature due to higher flow rates. Increasing the forward temperature seems to be beneficial to the point where heat pump is no longer needed in the system. This is further discussed in section 6.

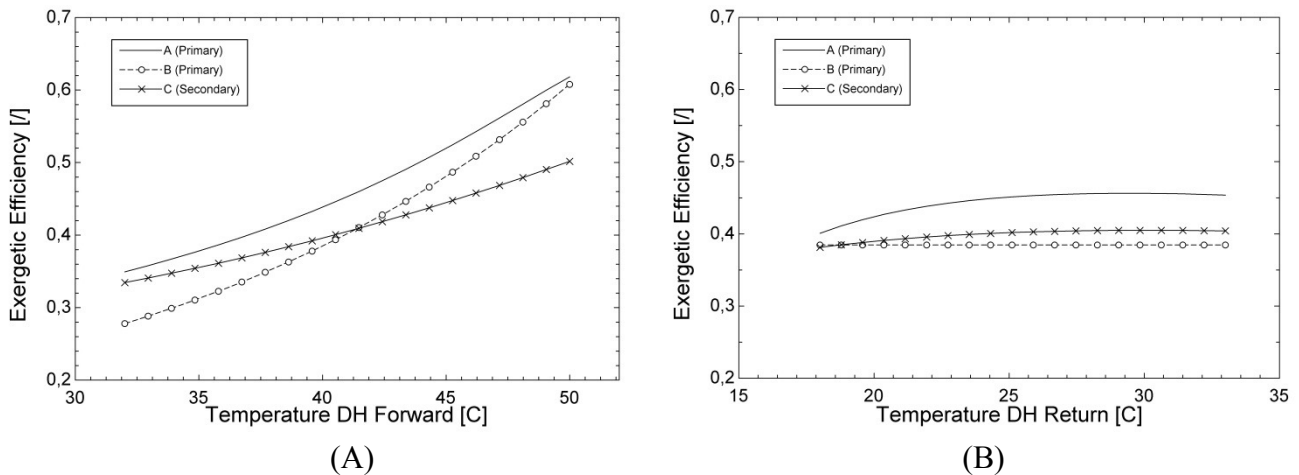


Fig. 10. (A) Exergetic efficiency of individual configurations with variable DH forward temperature. (B) Exergetic efficiency of individual configurations with variable DH return temperature.

Variant B performs well with a low temperature return stream, or very high forward temperatures. Allowing this configuration additional heat from the space heating as proposed in section 4.2 might improve the performance of the configuration considerably, but in the temperature regime proposed in the above calculations, the configuration is not advantageous in any part of the temperature span considered.

5.4. Constant temperature difference between forward and return

As it is not easy to find the optimal forward temperature from the above calculations, an additional calculation has been performed with a constant temperature difference (18 K) between forward and return of the district heating network. This is to rule out the coinciding effects of very high temperature lifts in the heat pump in one end and high thermal heat pump load in the other end of the studied temperature range.

In figure 11, most of the range is clearly covered by the configuration A. Only at very high temperature levels configuration B is advantageous. The secondary system C is inferior in the entire range.

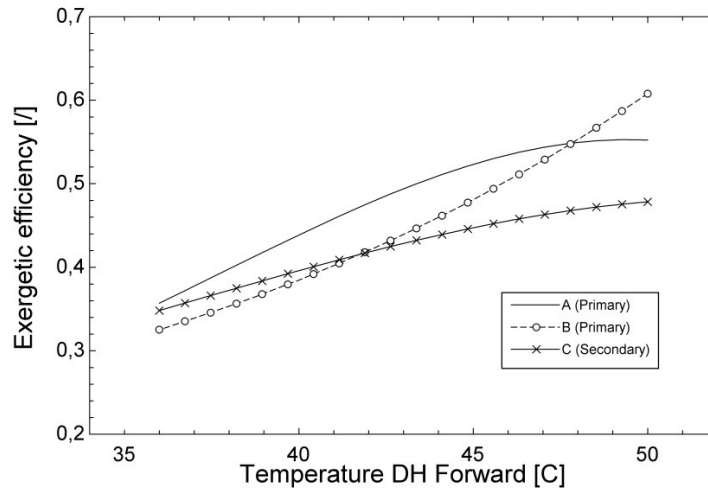


Fig. 11. Exergetic efficiency of the individual configurations with constant temperature difference between forward and return line of the district heating network.

6. Discussion

Several system configurations have been considered in the initial work of the project based on forward and return temperatures corresponding to table 1. Of the investigated systems, the three configurations presented in this paper have provided the best performance. It is not unlikely that other energy efficient solutions can exist. The three variants have been chosen based on the criteria considered in the overall project, not only to satisfy energy efficiency, but also to comply with e.g., state of the art technology and DH network considerations.

Considering both Fig. 8 and Fig. 10 (A), the optimal operation temperature of the district heating forward is not easily determined. The method used in fig. 11 show, that with a reasonable temperature difference throughout the range (18 K), the thermal load is the important factor to observe, as the COP is (almost) constant. Determination of optimal forward temperature of a low temperature district heating network will therefore not depend on the heat pump booster unit, but rather on external factors such as heat losses in the distribution network, sustainable sources and optimum production criteria for the combined heat and power plant.

This effect is also shown in In Fig. 10 (B), as the exergetic efficiency levels out without consideration to the improvement in COP from increasing the evaporator temperature.

From the same figure it is clear, that with a constant forward temperature (40 °C), the return temperature has an optimum (25 °C – 30 °C), which presumably would not be beneficial for the remaining network.

Consulting Figure 10 (A) it is clear that the exergetic efficiency decreases with consumption of electricity in the heat pump configurations. When approaching the temperatures where direct heat exchange is possible, the exergetic efficiency increases, as heat losses are not considered in the network.

If the heat pump booster unit is used in a system where it is coupled with a traditional district heating network, changes in the heat and power prices can be neglected. The reason for this is that the new system does not significantly change the operating conditions of the combined heat and power plant or the capacity of the transmission line in the district heating network. In this case only the heat losses in the novel DH system, and the increased end user capacity of the DH system from using lower temperatures can be compared with the additional electricity consumption.

7. Conclusion

Three heat pump schemes were singled out for evaluation in a low temperature district heating network in order to increase tap water temperature to meet the Danish standard. Out of the three

heat pumps, two are used to boost the network temperature prior to heat exchange with the tap water, while the third is used to boost the temperature of the heated tap water. Variant A was found to be the most efficient configuration in the temperature range considered. In the expected temperature range the heat pump has an exergetic efficiency between 0.4 [/] and 0.6 [/>. Variant B proved that power consumption might not become significantly increased, if the heat pump is used to actively lower the temperature of the return flow as source heat. This would allow for lower flow rates to meet the tap water requirements.

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Nomenclature

\dot{E} Time rate of exergy, kW

h Enthalpy, kJ/kg

\dot{m} Mass flow rate, kg/s

p Pressure, kPa

P Electricity, kW

Q Heat, kW

s Entropy, kJ/(kg*K)

t Temperature, C

\dot{V} Volume flow rate, m³/h

subscripts

0 Dead state

i Index (component)

DH District heating network

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