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# Design and optimization of the heat exchanger network for district heating ammonia heat pumps connected in series

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## ABSTRACT

Denmark presents ambitious climate policies, and in order to fulfil these visions electrically driven large-scale heat pumps (HP) are often mentioned as an important technology for future district heating (DH) systems. To reach the high temperatures needed in current DH systems, the suggested HP installations become complex systems, where heat transfer between the HP cycle and the heat sink takes place at several temperature levels. In this study the heat exchanger network (HEN) between a HP installation consisting of two serially connected two-stage ammonia HP units and a heat sink being heated from 50 °C to 80 °C was investigated. The study applied pinch analysis to estimate the highest attainable Coefficient of Performance (COP) with the given HP configuration. Based on the result of the pinch analysis, a HEN reaching the highest COP was suggested and compared with COPs obtained with three other solutions for a HEN. The result revealed an estimated highest COP of 3.46. The three other design suggestions yielded reductions in the COP of -2.3%, -2.0%, and -1.8% compared to the highest. From this it was concluded that the HEN has an influence on the COP, and that the pinch analysis can be used to estimate the highest COP for a given HP installation. Furthermore, the COP obtained by practical installations was accordingly shown to come close to the target.

Keywords: Large-scale heat pump, Ammonia system, District heating, Heat exchanger network, COP, Pinch analysis

## 1. INTRODUCTION

Denmark presents ambitious climate policies, as the national goal is to have a 100% renewable energy supply by 2050, and a 100 % renewable supply for electricity and heating by 2035 (The Danish Government, 2013). In order to fulfil this target, electrically driven large-scale heat pumps (HP) are often mentioned as an important technology for production of district heating (DH) (Danish Council on Climate Change, 2017), (Energy Commission, 2017). As highlighted from several perspectives, large-scale HPs have some great advantages in a future energy system, e.g., 1) utilizing power from renewable sources, primarily wind, which will be extensively expanded in them coming years, 2) recovering energy from renewable low temperature heat sources and industrial waste heat, and 3) balancing the electricity grid as an effective power-to-heat tool, and thereby providing flexibility between energy sectors (Lund et al., 2016), (Averfalk et al., 2017).

However, the introduction of HPs in large scale also faces a number of barriers. Among these are the relatively high cost of heat compared to existing CHP solutions, limited knowledge regarding performance of heat sources, variations in daily operation, and optimal design. In (Ommen et al. 2015a) it was shown how the choice of refrigerant in relation to the conditions of the source and sink will influence the system capacity in terms of Volumetric Heating Capacity, the performance in terms of COP, and the feasibility in terms of Net Present Value.
Strict environmental legislation aiming at natural refrigerants and the need for a high forward temperature (up to 90 °C) in the current Danish DH system have made ammonia a preferred refrigerant for HP installations. When a large temperature glide, defined as the temperature difference between return and forward temperature of the sink, is needed, two cycles may be connected in series. It was shown by (Ommen et al., 2015b) how this can have a beneficial impact on the COP. The system then consists of 12 heat exchangers (HEX) for integration with the heat sink. As the heat is rejected from the HP cycles to the sink at several temperature levels and phase conditions, the system becomes complex. Hence, the design of the HEN taking care of this heat transfer will

have an influence on both operating parameters, e.g. condensing pressure and amount of subcooling, and the investment cost in terms of needed HEX area and number of HEXs.

A well-known approach for the design of a HEN is the use of a *pinch method* based on predefined pinch points of the heat exchangers, as seen in e.g. (Zoughaib, 2017). With this method, the optimal amount of subcooling, as well as the condensing pressure, can readily be determined for simple cycles, and from this a calculation of the COP can be made. Figures 1a and 1b show a basic example of a configuration of a HP cycle, where the this method is applied.

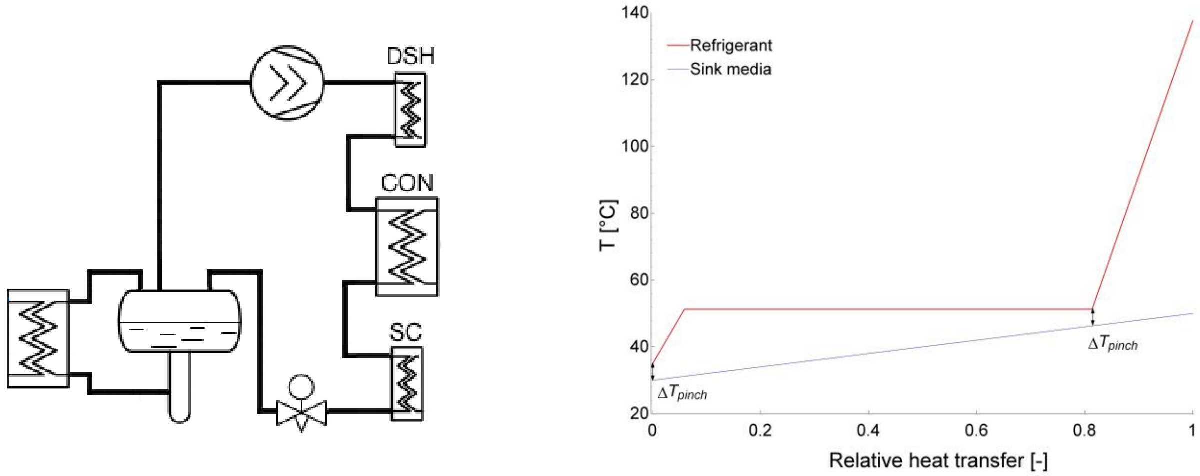


Figure 1a and 1b: Principle sketch of a simple HP cycle and the belonging $\dot{Q},T$ –diagram, with operation conditions determined by the pinch method (Zoughaib, 2017).

However, for a more complex system, where the HP cycle also includes a desuperheater from a lower pressure stage, as well as cooling of the compressors, e.g., oil cooling for screw compressors, the degrees of freedom in the design of the HEN increases. This makes it difficult to predict how the proposed HEN influences the COP, and when an optimal HEN based on the pinch method of (Zoughaib, 2017) is found.

Figure 2 shows a sketch of the simple serial HEN solution (subsequently referred to as HEN (1)) and the related $\dot{Q},T$-diagram, for the complete HP installation considered in this study. Here the pinch method, as illustrated in Figure 1b was applied to build the HEN. In table 1, the defined pinch temperature difference between the key temperatures determining the operation conditions and the heat sink appears for all the heat exchangers.

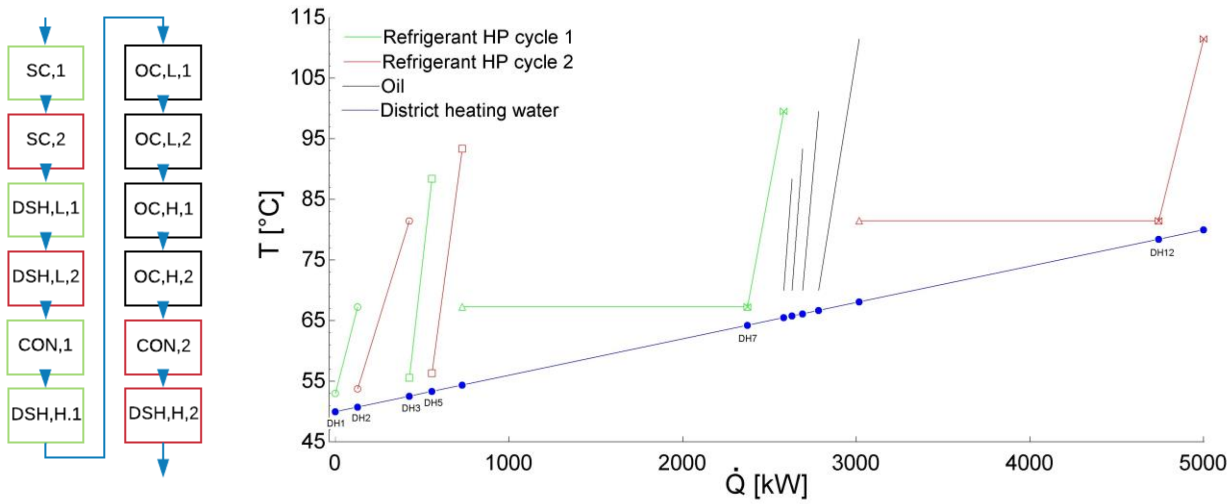


Figure 2: HEN (1) and the corresponding $\dot{Q},T$-diagram. The numbers on the cold stream refer to the temperatures used to define the pinch temperature difference seen in Table 1.

Table 1. Key temperatures and the integration with the heat sink.

| Key temperature | Designation | Associated pinch temperature difference |
|---|---|---|
| $T_{\mathrm{SC,out,1}}$ | Temperature after subcooler in HP cycle 1 | $\Delta T_{\mathrm{p,1}} = T_{\mathrm{SC,out,1}} - T_{\mathrm{DH,1}}$ |
| $T_{\mathrm{SC,out,2}}$ | Temperature after subcooler in HP cycle 2 | $\Delta T_{\mathrm{p,2}} = T_{\mathrm{SC,out,2}} - T_{\mathrm{DH,2}}$ |
| $T_{\mathrm{DSH,L,out,1}}$ | Temperature after desuperheater in HP cycle 1 | $\Delta T_{\mathrm{p,3}} = T_{\mathrm{DSH,L,out,1}} - T_{\mathrm{DH,3}}$ |
| $T_{\mathrm{DSH,L,out,2}}$ | Temperature after desuperheater in HP cycle 2 | $\Delta T_{\mathrm{p,4}} = T_{\mathrm{DSH,L,out,2}} - T_{\mathrm{DH,4}}$ |
| $T_{\mathrm{CON,1}}$ | Condensation temp in HP cycle 1 | $\Delta T_{\mathrm{p,5}} = T_{\mathrm{CON,in,1}} - T_{\mathrm{DH,6}}$ |
| $T_{\mathrm{CON,2}}$ | Condensation temp in HP cycle 2 | $\Delta T_{\mathrm{p,6}} = T_{\mathrm{CON,in,2}} - T_{\mathrm{DH,12}}$ |
| $T_{\mathrm{m,1}}$ | Saturation at intermediate pressure in HP cycle 1 | - |
| $T_{\mathrm{m,2}}$ | Saturation at intermediate pressure in HP cycle 2 | - |

The method applied in this study used a complete *pinch analysis* (Linhoff et al, 1986) of the integration between the heat sink and the heat rejection from the refrigerant in combination with an optimization algorithm, in order to determine optimal operating conditions for the HP system by determining the highest possible COP. The COP target found with the pinch analysis (subsequently referred to as $\mathrm{COP_{opt}}$), can be used as a good benchmark for COPs obtained with other proposed designs for a HEN.

The results presented in this work are based on the pinch analysis and four practical, suggested designs for a HEN for a 5 MW ammonia HP installation. The HP system considered was a design case for the DH system in Copenhagen, Denmark, utilizing sea water as heat source.

## 2. METHOD

Determination of the $\mathrm{COP_{opt}}$ for benchmarking as well as the examination of different designs of the HENs was based on a thermodynamic model of the HP cycle including both the heat source and heat sink. The model was implemented in Engineering Equation Solver (EES) (F. Chart, 1992). The model included mass and energy balances for all units of the cycle.

### 2.1 System description

The HP installation consisted of two serially connected two-stage ammonia HP cycles. The cycle configuration, including the naming of the heat exchangers can be seen in Figure 3. Each of the cycles includes subcooler, SC, condenser, CON, desuperheater at high stage, DSH,H, and low stage, DSH,L. The compressors were all screw compressors, requiring an oil separator and subsequent oil cooling at both stages OC,H and OC,L. Furthermore, the HP cycles consisted of a flooded evaporator connected to a low-pressure separator, a flash intercooler, and two expansion valves. All heat exchangers in connection to the heat sink were considered to be configured with counter flow.

The total heating capacity of the HP installation was 5 MW, with a share of 45 % capacity for HP cycle 1 and 55 % for HP 2. Table 2 presents the values used for different input parameters to the HP model. The index *i* indicates whether a component belongs to HP cycle 1 or 2. The assumed values are based on what can be estimated from the installations seen in industry, e.g. (EUPD project 2016), (Hoffmann et al. 2011) and from estimated state-of-the-art for the applied technologies. The heat transfer between the heat sink and the ammonia in the HEXs depended only on the temperature conditions. A pinch temperature difference of 3 K was used consistently for all heat exchangers. Furthermore, the same relatively conservative value of an isentropic efficiency was assumed for all compressors. It was assumed that this value could always be obtained for a compressor in the design condition with the pressure ratios applied. The calculation of heat rejected in the oil coolers was based on a required oil flow for each compressor as well as the inlet temperature of the oil. Thermal equilibrium between refrigerant and oil was assumed at the compressor discharge (Rane et al. 2016).

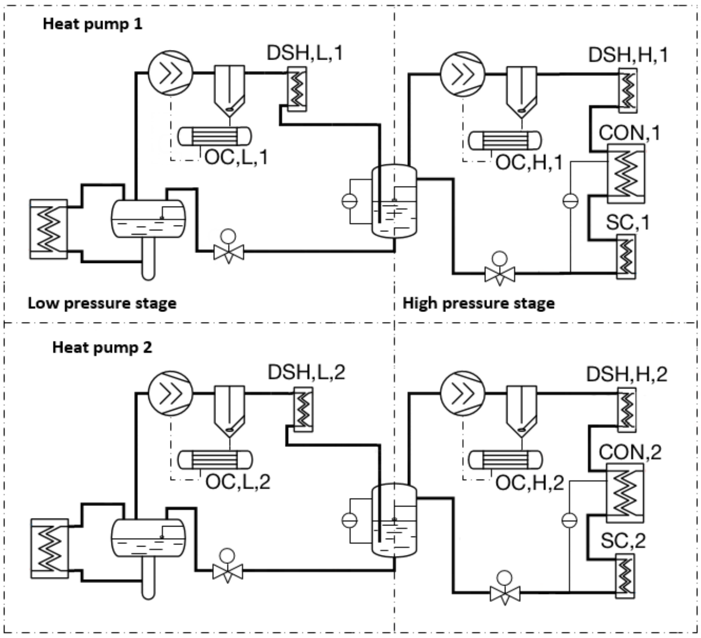

Figure 3: Principle sketch of the HP installation.

At the design condition the sea water is at low temperature. To avoid freezing, a low temperature glide of the heat source was required, and hence the heat source was connected in parallel to the evaporator of each HP cycle. The evaporation pressures were defined by a pinch temperature difference (3 K) between the outlet of the sea water and the ammonia. The source was considered not to limit the capacity, meaning that the required flow was always available. The heat exchange between the heat source and evaporator remained the same for all suggested solutions. Pressure losses were neglected for all parts of the system.
The intermediate pressures in each HP cycle were subject to an optimization. The optimal pressures were determined for each solution.

Table 2. Values applied for different parameters used in the model of the HP installation.

| Designation | Parameter | Value | Unit |
|---|---|---|---|
| Isentropic compressor efficiency | $\eta_{\mathrm{is,i}}$ | 0.75 | - |
| Oil inlet temperature | $T_{\mathrm{o,in,i}}$ | 70 | ℃ |
| Oil volume flow, low stage | $\dot{V}_{\mathrm{oil,l,i}}$ | 85 | l/min |
| Oil volume flow, high stage 1 | $\dot{V}_{\mathrm{oil,h,1}}$ | 100 | l/min |
| Oil volume flow, high stage 2 | $\dot{V}_{\mathrm{oil,h,2}}$ | 180 | l/min |
| Source inlet temperature | $T_{\mathrm{sw,in,i}}$ | 4 | ℃ |
| Source outlet temperature | $T_{\mathrm{sw,out,i}}$ | 0.5 | ℃ |
| Sink inlet temperature | $T_{\mathrm{DH,in}}$ | 50 | ℃ |
| Sink outlet temperature | $T_{\mathrm{DH,out}}$ | 80 | ℃ |
| Dimensioning capacity of system | $\dot{Q}_{\mathrm{DH}}$ | 5 | MW |

### 2.2 Pinch analysis

In order to obtain a target of the $\mathrm{COP}_{\mathrm{opt}}$ of the HP installation, a pinch analysis and an optimization algorithm, was applied to the system. Basically a conventional pinch analysis performs an energy balance between total heat capacity flow rate of heating and cooling demands at all temperature levels and allows excess heat from a higher temperature to be utilized at lower levels. Hereby the minimum demand for external heating and cooling supply may be determined for a given minimum temperature difference, the pinch point temperature difference (Kemp, I., 2007).
The pinch analysis of the heat pump cycle differed from conventional pinch analyses because the HP system in itself is balanced by the compressor power, such that all the heating and cooling demands between the hot and cold streams were recovered internally, and external utility was not considered as an option.

The pinch analysis of the HP was integrated with an optimization of the COP to determine the target, i.e., the highest possible efficiency of the system. The objective function was the COP of the total HP system, which inherently evaluated the performance of the heat recovery.

The pinch analysis was based on discretizing the whole temperature range between the highest compressor discharge temperature and the district heating inlet temperature.
With streams defined as above, the overall temperature difference was divided into smaller temperature intervals with the size of $\Delta T_{\text{increment}}$ by Eq. (1)

$$\Delta T_{\text{increment}} = \frac{T_{\text{DSH,in,H,2}} - T_{\text{DH,in}}}{(n-1)} \tag{1}$$

where $T_{\text{DSH,in,H,2}}$ represents the hottest temperature found in the system, $T_{\text{DH,in}}$ the coldest, and $(n-1)$ the number of desired intervals.

In addition it was necessary to add a small variation of the condensing temperature in each condenser, in order to avoid zero heat capacity for these streams in the pinch analysis. This was solved by Eq. (2), where a temperature difference of $\Delta T_{\text{increment}}$ was subtracted from the inlet temperature of each condenser.

$$T_{\text{CON,out,i}} = T_{\text{CON,in,i}} - \Delta T_{\text{increment}} \tag{2}$$

For each temperature interval, the sum of heat transferred from the hot streams and to the one cold stream was calculated by Eq. (3)..

$$\dot{Q}_{\text{hot}} = \Delta T_{\text{increment}} \cdot \sum \dot{C}_{\text{hot}} \; ; \dot{Q}_{\text{cold}} = \Delta T_{\text{increment}} \cdot \dot{C}_{\text{cold}} \tag{3}$$

The temperature difference between the summation of the hot stream outlets and the inlet of the cold stream was then found by Eq. (4) for every temperature interval.

$$\Delta T = T_{\text{hot.out}} - T_{\text{cold,in}} \tag{4}$$

To obtain a global pinch point with a given minimum temperature difference, $\Delta T$ from Eq. (4) was constrained for every interval with a minimum value of 3 K.

With the above pinch analysis implemented in the model, an optimization of the COP was performed in EES, applying the Variable Metric Method. The optimization was based on HEN (1) seen in Figure 2.
The variable changes in the search for $\text{COP}_{\text{opt}}$, and a global pinch point temperature difference of 3 K, were the values of the pinch temperature differences defined in Table 1, together with the intermediate saturation temperatures $T_{\text{m,1}}$ and $T_{\text{m,2}}$. The outcome of the optimization was then a new set of values for $\Delta T_{\text{p1}-6}$, $T_{\text{m,1}}$ and $T_{\text{m,2}}$, defining the optimal key temperatures and thus the optimal operating conditions of the HP cycles.

As with a conventional pinch analysis, the result then implies that it should be possible to design a HEN from the streams determined by the optimization (eg. streams now defined by the achieved key temperatures), where the minimum pinch temperature difference is respected in all heat exchangers. In this case it also allows the design of a HEN (subsequently referred to as $\text{HEN}_{\text{opt}}$) where the $\text{COP}_{\text{opt}}$ could be obtained for the HP installation. The approach used for designing the network was to split the hot streams into intervals at the condensing temperatures. This ensured that heat transfer before a condenser only happened at temperatures lower than the related $T_{\text{con}}$.

**2.3 Comparison of HENs**
The result of the pinch analysis and the suggested $\text{HEN}_{\text{opt}}$ were compared with three other HENs and the COPs obtained with these solutions. HEN (1) was the serial connection of all HEXs shown in Figure 2. HEN (2) and HEN (3) were inspired by HENs seen applied in industry (EUPD project 2016), (Hoffmann et al. 2011). A sketch of these is shown in Figures 6a and 6b. The operating conditions of the HP cycles connected to HEN (1), (2) and (3) were again determined by the *pinch method* to reach the highest possible COP for the given

configuration. In order to make comparable solutions the minimum value of the pinch temperature difference applied in the solutions was also 3 K.

## 3. RESULTS

The pinch analysis integrated with optimization resulted in a COP of 3.46 for the complete cycle. The composite curve for this solution is presented in Figure 4, and the key temperatures can be seen in Table 5. Furthermore, the belonging condensation temperatures for HP cycle 1 and HP cycle 2 is also shown in Figure 4.

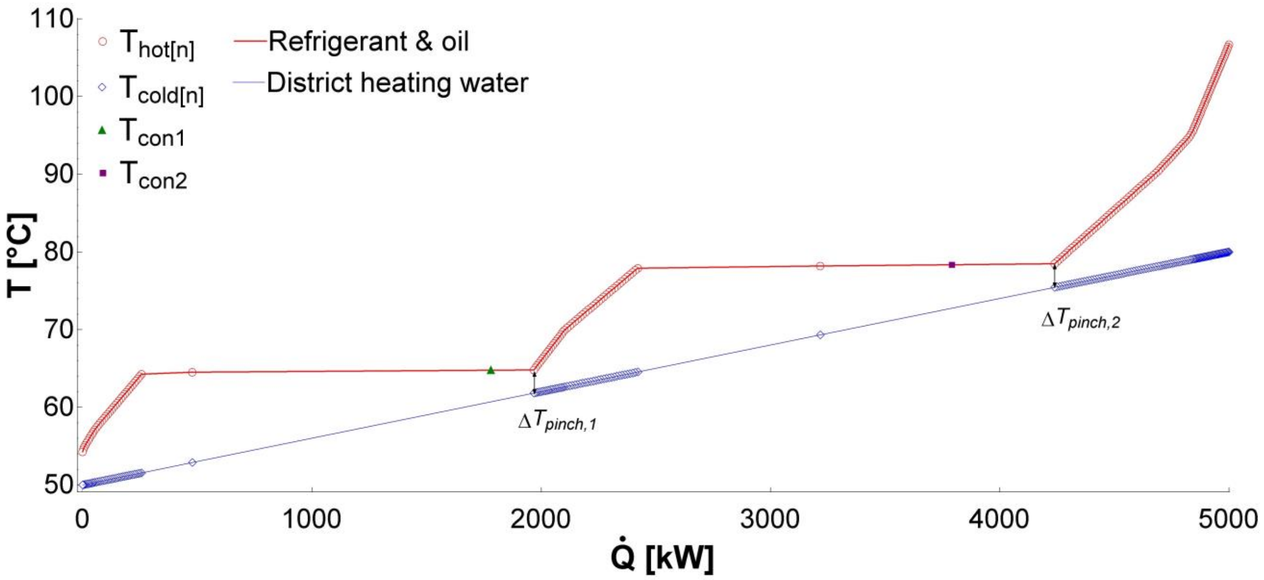


Figure 4: The composite curves and condensation temperatures obtained with pinch analysis and optimization of COP.

Figure 5 shows the layout of the HEN and the temperatures of the cold stream for the suggested $HEN_{opt}$. This network reaches the target COP by use of 22 HEX. The large amount of heat transferred at constant temperature in the condensers caused challenges in the network design and resulted in splitting all stream at the condensation temperatures.

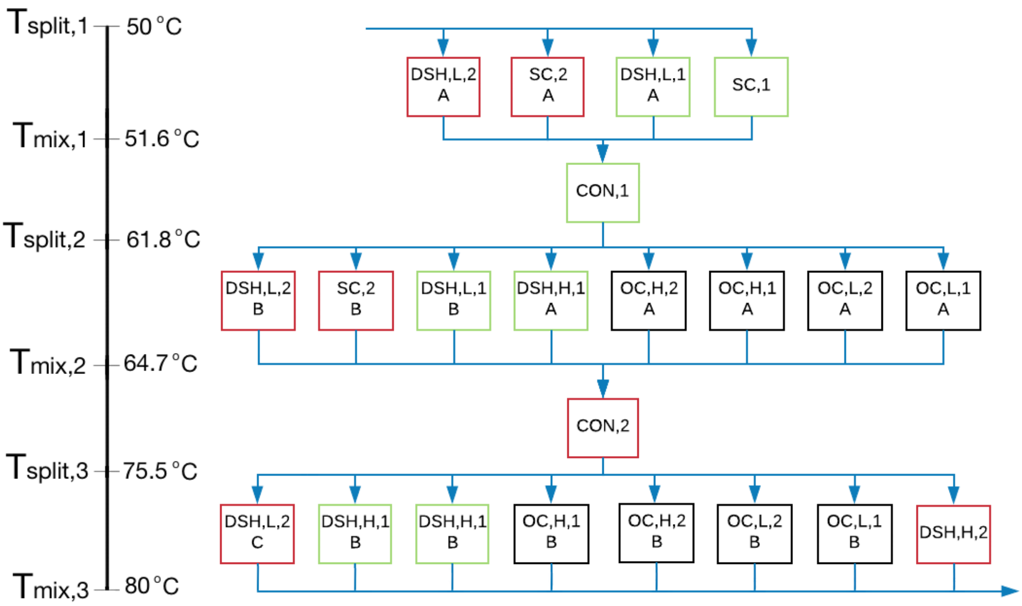


Figure 5: Sketch of $HEN_{opt}$ for the optimized HP cycles. Temperatures of the cold streams can be seen to the left.

Table 3 shows the temperature differences between the hot and cold streams in each HEX. The resulting pinch temperature difference of the outlet of the HEX CON,2/inlet of the last 8 HEXs is just below 3 K. This can be explained by the mismatch between the condensation temperature and the temperature forming the second pinch point, as shown in Figure 4.

Table 3. Temperature differences at the inlet and outlet of each HEX in $HEN_{opt}$.

| HEX cycle 1 | ΔT inlet | ΔT outlet | $\dot{Q}$ [kW] | HEX cycle 2 | ΔT inlet | ΔT outlet | $\dot{Q}$ [kW] |
|---|---|---|---|---|---|---|---|
| SC,1 | 4.3 | 13.2 | 96.0 | SC,2 A | 4.7 | 13.2 | 106.2 |
| DSH,L,1 A | 6.8 | 13.2 | 33.5 | DSH,L,2 A | 7.5 | 13.2 | 36.5 |
| CON,1 | 13.2 | 3.0 | 1694.0 | CON,2 | 13.7 | 2.9 | 1794.0 |
| DSH,H,1 A<br>DSH,L,1 B | 3.0 | 13.7 | 95.7<br>54.9 | SC,2 B<br>DSH,L,2 B | 3.0 | 13.7 | 151.7<br>65.4 |
| OC, L,1 A<br>OC,H,1 A | 8.2 | 13.7 | 21.8<br>25.6 | OC, L,2 A<br>OC,H,2 A | 8.2 | 13.7 | 21.9<br>46.9 |
| DSH,L,1 C<br>OC,L,1 B | 2.9 | 10.5 | 47.2<br>31.3 | DSH,L,2 C<br>OC,L,2 B | 2.9 | 15.6 | 79.3<br>44.8 |
| DSH ,H,1 B<br>OC,H,1 B | 2.9 | 14.7 | 98.7<br>50.0 | DSH,H,2<br>OC,H,2 B | 2.9 | 26.7 | 242.7<br>158.3 |

HEN (1), HEN (2) and HEN (3) are shown in Figures 2, 6a and 6b, respectively. Table 4 lists the corresponding temperature differences between the hot and cold streams in each HEX. As seen in Figure 6a and 6b, the cold stream was split up and mixed several times in HEN (2) and (3). The share of mass flow through each HEX was determined by a desired isothermal mixing in the downstream mixing point. This implied that the outlet temperature of the HEXs adding flow to the same mixing point would enter the point with the same temperature. An exception from this approach can be seen for the HEX SC,2 in HEN (3), where an optimal outlet temperature ($T_{DH,SC2}$) was investigated. A high outlet temperature introduced entropy generation in the mixing point, but also lowered the condensing temperature for the condenser (CON1,) being by-passed and therefore improved the COP. The outlet temperature of SC,2 (77.4 °C) was then defined by a pinch temperature difference.

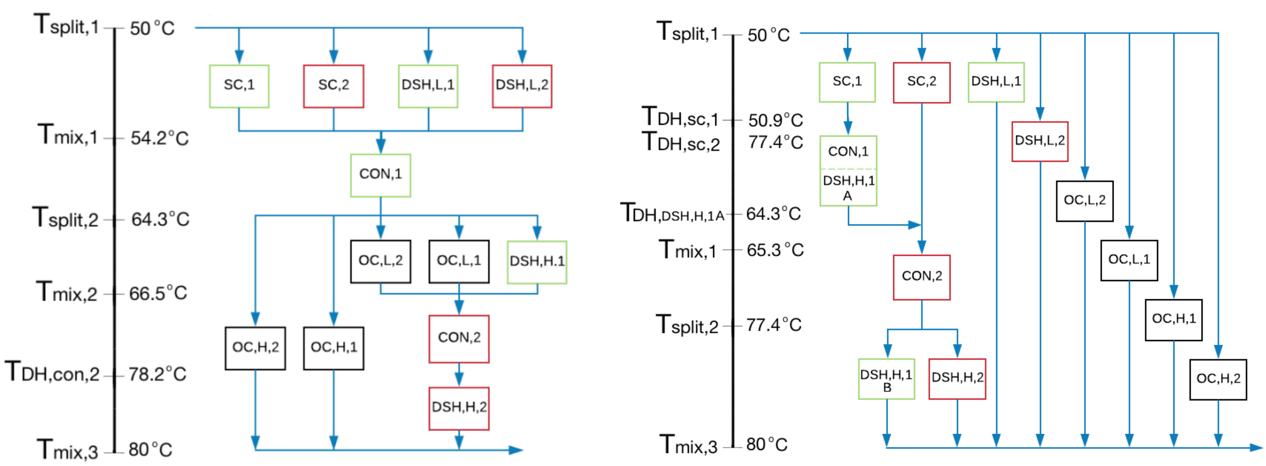


Fig 6a and 6b: A sketch of HEN (2) and HEN (3) used for comparison.

Table 4. Temperature differences at the inlet and outlet of the HEXs shown in Figure 2, 6a and 6b.

| HEX cycle 1 | ΔT inlet | ΔT outlet | $\dot{Q}$ [kW] | HEX cycle 2 | ΔT inlet | ΔT outlet | $\dot{Q}$ [kW] |
|---|---|---|---|---|---|---|---|
| **HEN (1)** | | | | | | | |
| SC,1 | 3.0 | 17.3 | 128.4 | SC,2 | 3.0 | 29.1 | 297.9 |
| DSH,L,1 | 3.0 | 35.1 | 130.1 | DSH,L,2 | 3.0 | 39.0 | 172.9 |
| CON,1 | 12.9 | 3.0 | 1643.0 | CON,2 | 13.3 | 3.0 | 1725.0 |
| DSH,H,1 | 3.0 | 34.0 | 208.8 | DSH,H,2 | 3.0 | 31.5 | 258.6 |
| OC,L,1 | 4.5 | 22.6 | 47.9 | OC,L,2 | 4.2 | 27.2 | 61.1 |
| OC,H,1 | 3.9 | 32.8 | 91.7 | OC,H,2 | 3.3 | 43.4 | 234.7 |
| **HEN (2)** | | | | | | | |
| SC,1 | 3.0 | 12.8 | 128.4 | SC,2 | 3.0 | 26.7 | 300.7 |
| DSH,L,1 | 3.0 | 33.8 | 140.7 | DSH,L,2 | 3.0 | 37.6 | 183.1 |
| CON,1 | 12.8 | 3.0 | 1633.0 | CON,2 | 14.8 | 3.0 | 1714.0 |
| DSH,H,1 | 3.0 | 33.2 | 207.8 | DSH,H,2 | 3.0 | 31.6 | 258.8 |
| OC,L,1 | 5.7 | 21.9 | 47.8 | OC,L,2 | 5.7 | 25.6 | 57.7 |
| OC,H,1 | 5.7 | 19.6 | 92.0 | OC,H,2 | 5.7 | 31.6 | 235.8 |
| **HEN (3)** | | | | | | | |
| SC,1 | 3.0 | 15.6 | 121.6 | SC,2 | 3.0 | 3.0 | 292.2 |
| DSH,L,1 | 3.0 | 6.6 | 132.9 | DSH,L,2 | 3.0 | 14.0 | 194.0 |
| CON,1 | 15.6 | 3.0 | 1649.0 | CON,2 | 15.2 | 3.0 | 1727.0 |
| DSH,H A,1 | 3.0 | 16.2 | 211.4 | DSH,H,2 | 3.0 | 29.8 | 249.4 |
| DSH,H B,1 | 3.0 | 19.6 | 113.8 | - | - | - | - |
| OC,L,1 | 20,0 | 6.6 | 43.2 | OC,L,2 | 20,0 | 14.0 | 62.6 |
| OC,H,1 | 20,0 | 19.6 | 91.7 | OC,H,2 | 20,0 | 29.8 | 224.8 |

Different operating conditions of the HP cycles for each solution are shown in Table 5. As expected, the highest COP was obtained when the condensation pressures were lowest, as this resulted in reduced compressor work. The condensing temperatures with the optimized HEN were between 1.7 K and 2.5 K lower for HP 1 and between 2 K and 3.1 K lower for HP 2, compared to the other HENs. Furthermore it can be seen from the values of the intermediate temperatures, $T_{m,1}$ and $T_{m,2}$, as well as the PRs, that $HEN_{opt}$ comes with the highest intermediate pressure. Comparing the different solutions shown in Table 5, it can be seen that the COP obtained with HEN (1), (2), and (3), was -2.3%, -2.0% and -1.2%, respectively, lower than the COP obtained with the optimized HEN. These solutions are accordingly close to the target, which documents the appropriate design of the cycles. But it also documents the value of having a target for the design based on an ideal solution.

Table 5. COP and operation conditions for the HP cycles with different HENs.

| | $\mathbf{HEN_{opt}}$ | **HEN (1)** | **HEN (2)** | **HEN (3)** |
|---|---|---|---|---|
| $T_{dsh,in,H,1}/T_{con,1}/T_{sc,out,1}$<br>$T_{dsh,in,L,1}/T_{m,1}$ [°C] | 94.7 / 64.8 / 54.3<br>90.5 / 33.7 | 99.5 / 67.2 / 53.0<br>88.4 / 32.6 | 99.6 / 67.3 / 53.0<br>88.3 / 32.6 | 99.5 / 66.5 / 53.0<br>86.6 / 31.6 |
| $T_{dsh,in,H,2}/T_{con,2}/T_{sc,out,2}$<br>$T_{dsh,in,L,2}/T_{m,2}$ [°C] | 107 / 78.4 / 54.7<br>95.6 / 35.8 | 111 / 81.5 / 53.8<br>93.4 / 34.7 | 112 / 81.2 / 53.0<br>92.1 / 34.0 | 110 / 80.4 / 53.0<br>93.9 / 35.0 |
| $PR_{L,1}$ / $PR_{H,1}$ [-] | 3.3 / 2.3 | 3.2 / 2.5 | 3.2 / 2.5 | 3.1 / 2.5 |
| $PR_{L,2}$ / $PR_{H,2}$ [-] | 3.5 / 2.9 | 3.4 / 3.2 | 3.3 / 3.2 | 3.4 / 3.1 |
| $p_{con,1}/p_{con,2}$ [bar] | 29.3 / 40.0 | 31.1 / 42.7 | 31.1 / 42.5 | 30.6 / 41.8 |

*Table 5 continued*

| | | | | |
|---|---|---|---|---|
| $\dot{W}_{\mathrm{L.1}}/\dot{W}_{\mathrm{H.1}}$ [kW] | 336 / 270 | 321 / 298 | 321 / 297 | 311 / 302 |
| $\dot{W}_{\mathrm{L.2}}/\dot{W}_{\mathrm{H.2}}$ [kW] | 416 / 425 | 398 / 462 | 391 / 465 | 404 / 446 |
| $\dot{W}_{\mathrm{tot}}$ | 1447 | 1479 | 1474 | 1463 |
| COP<br>Change in COP [%] | **3.46**<br>**0** | 3.38<br>*-2.3 %* | 3.39<br>*-2.0 %* | 3.42<br>*-1.2 %* |

### 3.1 Discussion

The target COP determined by the pinch analysis was found to be a higher than the practical solutions. The difference was, however, small and practical considerations may absolutely justify the realized configurations. One important difference is that the three practical networks have 12 HEX compared to the 22 needed for the optimal configuration. A high COP will have a positive impact on the operation cost for a HP installation. This will contribute to an improvement in the lifetime economy of the installation. However, as mentioned in the introduction, to fully understand the impact on the lifetime economy, the investment cost of the HEN will also need to be considered. This includes both the numbers of HEXs needed, as well as the expected heat transfer area.

The method applied in this study has some limitations that cause the obtained $COP_{opt}$ to be seen as an estimation of a true maximum COP. A brief discussion of the limitations can be seen below.

One limitation already mentioned is the mismatch between condensing temperatures and division of the temperature intervals in the pinch analysis, which led to a smaller pinch temperature difference in $HEN_{opt}$ than the minimum global pinch temperature difference applied in the pinch analysis. The issue could may be solved with an increased amount of intervals. This would, however, increase the number of equations in the thermodynamic model significantly, and as the difference between the global minimum pinch temperature and the actual minimum pinch temperature difference seen is very small (3.0 K vs 2.9 K), the influence on the COP obtained, is expected to be negligible.

In order to fully investigate the possibilities for heat integration, the approach used for determining the outlet temperature of SC,2 in HEN (3), could also be applied for the other streams where a condenser is by-passed. This was the case for OC,H,1+2 in HEN (2), and for DSH,L,1+2, and all OCs in HEN (3). If applied the maximum temperature of the cold stream should be limited by a pinch temperature difference to the inlet of the hot streams. However, in this study the maximum outlet temperature of a cold stream was limited to 80 °C, due to the definition of the cold stream in the case study. A further development of the pinch analysis would be needed, in order to include higher temperatures of the cold stream. It is expected that this could have a small positive influence on the COP.

The calculated COP of the three practical cycles differs from the numbers in the realized cycles (EUPD project 2016), (Hoffmann et al. 2011) because of other values of the parameters were used in the calculations for allowing comparison between the four solutions. The assumed values for component parameters may not be valid for all configurations; for example may the efficiency of compressors depend on the pressure ratio.

The minimum temperature difference of 3 K was chosen based on experience. The choice of an arbitrary minimum temperature difference is one of the challenges of pinch analysis in general, not only in this case, in particular when economic feasibility is the target of the optimization. However, the value may easily be adjusted for sensitivity analysis of the solution.

## 4. CONCLUSIONS

The results illustrated an application of pinch analysis coupled with optimization of COP for a complex heat pump system including two serially connected two-stage heat pumps for producing district heating based on seawater as low temperature source. The applied pinch analysis for an ammonia HP installation resulted in an estimation of a $COP_{opt}$ of 3.46 for the system. Furthermore a design of a HEN was presented in order to obtain this COP. The optimal COP was compared with COPs obtained with three other HEN designs. The analysis revealed a COP -2.3%, -2.0% and 1.2% lower than the optimal COP, respectively.

The method needs further development in order to give a more precise $COP_{opt}$ and to fully investigate the possibilities for heat integration. Furthermore, the cost of the HEN can be included in future work to compare the economy of the different design solutions.

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## NOMENCLATURE

| | | | |
|---|---|---|---|
| $p$ | pressure (bar) | HP | Heat Pump |
| $T$ | temperature (°C) | DH | District Heating |
| $\dot{W}$ | compressor work (kW) | DSH | Desuperheater |
| $\dot{Q}$ | Capacity (kW) | CON | Condenser |
| $\dot{C}$ | Heat capacity rate (kW/K) | SC | Subcooler |
| $\Delta T$ | Temperature difference (K) | OC | Oil cooler |
| HEN | Heat Exchanger Network | H | High |
| COP | Coefficient of Performance | L | Low |
| PR | Pressure Ratio | i | Heat pump index (1 or 2) |
| HEX | Heat Exchanger | | |

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