http://dx.doi.org/10.18462/iir.nh3-co2.2017.0037

# HIGH-TEMPERATURE HEAT PUMPS BASED ON NATURAL WORKING FLUIDS TO PRODUCE DISTRICT HEATING FROM INDUSTRIAL WASTE HEAT

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## ABSTRACT

The amount of industrial waste heat produced in Europe annually corresponds to the total annual heating demand in buildings. Common barriers for utilizing this heat are the lack of required infrastructure, i.e. a district heating (DH) network, and too low temperature of the waste heat. To address the last-mentioned barrier, high-temperature heat pumps may be applied to upgrade the heat. This study evaluates the possibility of utilizing a heat pump to supply DH using industrial wastewater at 40 °C as a heat source. Different heat pump technologies based on natural refrigerants were compared at steady state operating conditions: single- and two-stage ammonia, and ammonia-water absorption-compression (hybrid) heat pumps. The highest COP as well as the lowest operating pressure were obtained with a two-stage hybrid heat pump. Moreover, the COP for hybrid heat pumps was hardly affected by an increase in the desired DH supply temperature.

*Keywords:* High-temperature heat pumps; ammonia heat pumps; hybrid heat pumps; district heating; waste heat utilization

## 1. INTRODUCTION

It has been estimated that the amount of industrial excess heat rejected annually in Europa is roughly equal to the buildings heat demand, and in many cases, the waste heat sources are located in areas with high heat demand density (Persson, Möller et al. 2014). Waste heat is often regarded as a source of energy, which is difficult to utilize in for instance power generation. The role of electricity in solving the future energy problems is however often overestimated; 46 % of the global energy demand is related to heating and cooling (IEA 2012). District heating (DH) is a significant technology in enabling efficient and economical utilization of waste heat sources to cover the buildings heating demands (Laajalehto, Kuosa et al. 2014). Common barriers for utilizing the waste heat sources are that in many European cities, there are no district heating (DH) networks for transporting the heat, or the heat may be available at a too low temperature level.

High temperature heat pumps are an important enabling technology for upgrading low-temperature waste heat sources to higher temperature levels. Heat pumps are to some degree already employed in DH production, using various heat sources, such as sea water (Grinrød 2012), sewage (Friotherm 1995, Sveinall 2008) or industrial waste heat from e.g. data centers (CNN 2011, Fortum 2016). Natural refrigerants, such as ammonia, has been employed as the refrigerant in some cases (Grinrød 2012), while often HFCs have been the preferred alternative (Friotherm 1995, Sveinall 2008).

This study evaluates the possibility of utilizing a heat pump to supply DH by lifting the return temperature at 70-75 °C to the desired level of 85-100 °C, using industrial wastewater at 40 °C. The desired heating capacity was 4 MW. Different heat pump technologies based on natural refrigerants were compared at steady operation conditions: single- and two-stage ammonia heat pumps, as well as single- and two-stage ammonia-water absorption-compression (hybrid) heat pumps.

## 2. HEAT PUMP TECHNOLOGIES

Single- and two-stage ammonia and ammonia-water absorption-compression (hybrid) heat pumps were evaluated for different supply and return temperatures for DH, at a heating capacity of 4 MW, utilizing an industrial waste water stream with a temperature of 40 °C and a volume flow of 40 l/s (144 $m^3$/h). In the calculations, the following assumptions were made:

- Constant isentropic compressor efficiency $\eta_{is}$ of 0.7
- Heat loss in the compressor of 5 %; no compressor cooling
- 2 K temperature difference ($\Delta T_{pinch}$) between the refrigerant and water at condenser inlet and evaporator outlet

The evaluated heat pump technologies are discussed in more detail below.

### 2.1 Ammonia heat pumps

For ammonia as the refrigerant, three different heat pump technologies were considered: (1) plain single-stage, (2) single-stage heat pump with economizer and de-superheater, and (3) two-stage heat pump with de-superheater. A plain single-stage heat pump is inappropriate for the current problem as the compressor discharge temperature becomes too high. Moreover, expansion losses for this case are especially high, since the condensation temperature is high and not far from the critical temperature of ammonia (132 ° C). Plain single-stage heat pump was nevertheless included in the calculations for the sake of comparison.

In the economizer solution, shown in Figure 1 (a), the fluid coming out of the condenser is divided into two, one of which is throttled down to a lower pressure level (economizer pressure). In a heat exchanger (the economizer), heat from the fluid coming straight from the condenser is utilized to vaporize the throttled fluid, which in turn is fed into the compressor at the economizer pressure, which is somewhat higher than the evaporation pressure. The solution has three advantages:

1. Compression temperature is reduced by the gas which is fed from the economizer
2. Two-stage throttling reduces the throttling losses
3. The liquid flowing into the evaporator is subcooled, which reduces the throttling losses and hence increases the specific evaporation enthalpy.

These advantages become evident from the *Log(p)-h* diagram shown in Figure 1(b). In the calculations, it was assumed that injection from the economizer to the compressor occurs when approximately 13 % of the pressure lift (equivalent to ¼ of compressor volume) has been completed.

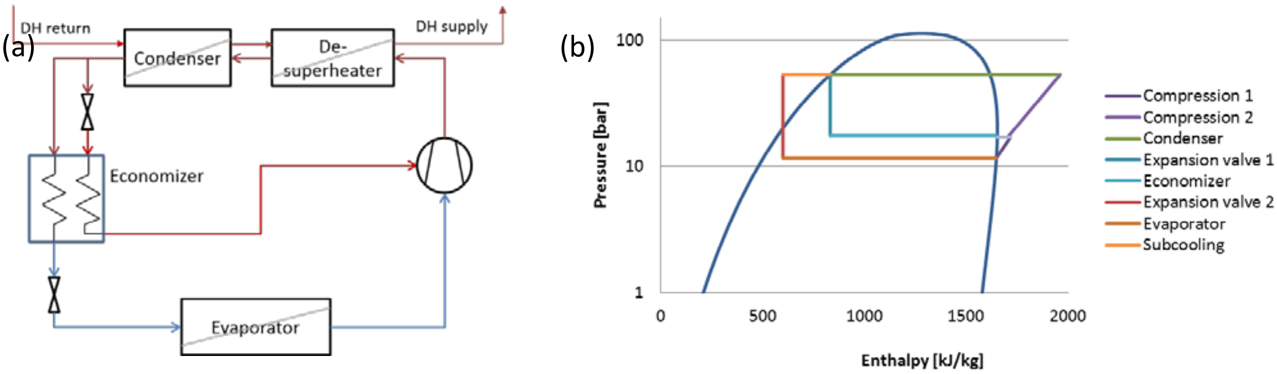


**Figure 1** Single-stage $NH_3$ heat pump with economizer: (a) principle diagram and (b) *Log(p)-h* diagram.

In addition to an economizer, a de-superheater was included in the system as shown in Figure 1 (a). With this, the high temperature of the superheated vapour leaving the compressor can be utilized, allowing a reduced condensing temperature (and pressure), and hence reduced compressor work. The condenser pressure was chosen as the lowest possible pressure such that a temperature difference ($\Delta T_{pinch}$) of 2 K was maintained between water and ammonia at the condenser inlet. The system could be further improved by using an internal heat exchanger and/or a sub-cooler, however these alternatives have not been considered here.

By using two-stage compression and expansion with an intermediate pressure vessel for cooling between the two compressors, the compression temperatures and expansion losses are reduced. The principle scheme and the *Log(p)-h* diagram for this solution are shown in Figure 2. Also in this solution, a de-superheater was included.

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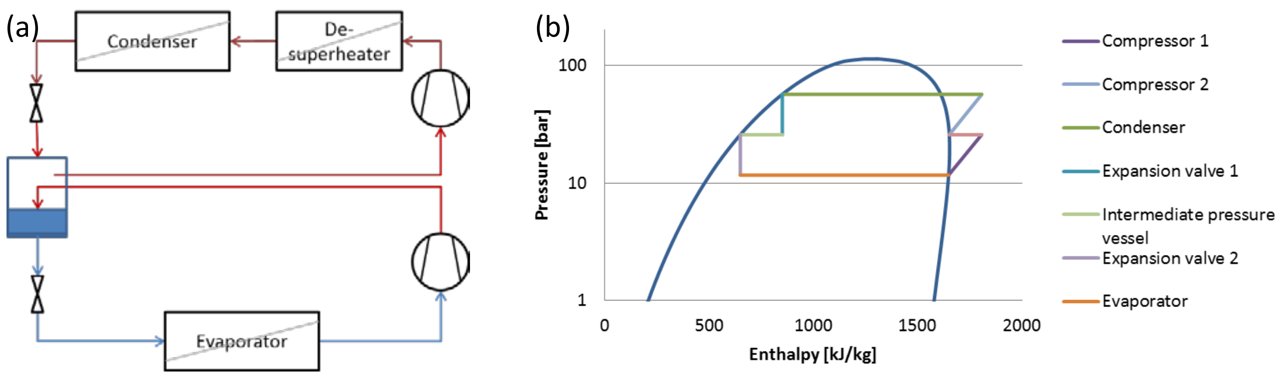


**Figure 2** Two-stage $NH_3$ heat pump with de-superheater: (a) principle diagram and (b) *Log(p)-h* diagram.

Assuming a heating capacity of 4 MW and a COP of 3, the compressor power becomes approximately 1.3 MW and the evaporator capacity 2.7 MW. With a volume flow of 40 l/s, this implies a temperature glide of 16 °C for the wastewater. With such a high temperature glide for the heat source, it is more sensible with parallel connection of two heat pumps as shown in Figure 3. In the comparison with hybrid heat pumps, parallel-connected (single- and two-stage) $NH_3$ heat pumps were hence employed. With two heat pumps connected in parallel, the total COP for heating is calculated as

$$COP_{H,tot} = \frac{Q_{H,tot}}{Q_{H,1}/COP_{H,1} + Q_{H,2}/COP_{H,2}} = \frac{Q_{H,tot}}{W_1 + W_2} \tag{1}$$

where the total delivered heat $Q_{H,tot}$ = 4 MW and heat delivered by the individual heat pumps is $Q_{H,1} = Q_{H,2}$ = 2 MW. $COP_{H,1}/COP_{H,2}$ and $W_1/W_2$ denote the COP and compressor power of the individual heat pumps, respectively.

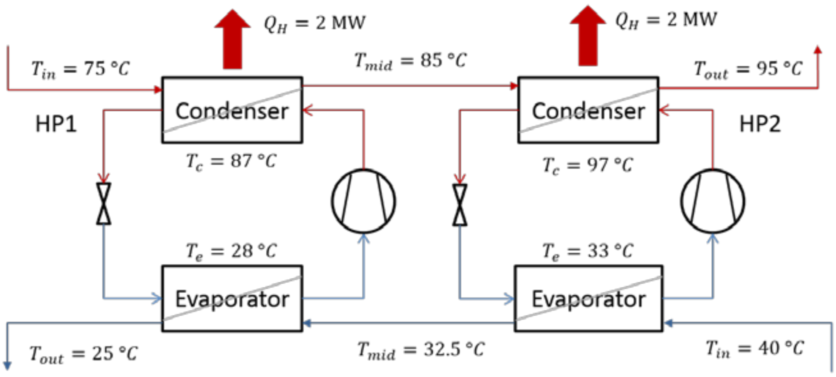


**Figure 3** A diagram for parallel connection of two heat pumps (HP1 & HP2), here with a temperature glide of 15 °C for the heat source, as well as DH supply and return temperatures of 95 and 75 °C, respectively.

## 2.2 Ammonia-water hybrid heat pump

By mixing ammonia with water, the average boiling point becomes higher, and therefore, a higher condensing temperature is achieved with lower operating pressure and standard equipment. Furthermore, by mixing two fluids with different saturation temperatures, evaporation and condensation become non-isothermal processes, and the temperature glide can be adjusted to the external heat source/sink by adjusting the composition. This renders ammonia-water hybrid heat pump suitable for processes with large temperature differences on both sides (Nordtvedt, Horntvedt et al. 2011).

In an absorption-compression heat pump, also called a hybrid heat pump, the pressure lift is obtained with ordinary mechanical compression of vapour. Principle diagrams for single- and two-stage hybrid heat pumps are shown in Figure 4. After the desorber/evaporator, the liquid passes into a separator, from which the solution rich in water is pumped through a solution heat exchanger (SHX) for internal heat recovery, and the ammonia vapour is compressed to higher pressure. The solution rich in water is then mixed with ammonia at high pressure in the absorber, where heat is released during the absorption of ammonia. The solution leaving the absorber, rich in ammonia, releases heat in the internal heat exchanger and is thereafter expanded to a

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lower pressure. In the desorber, heat is added from an external heat source, causing ammonia to vaporize from the solution, and the process is repeated. In a two-stage hybrid heat pump (Figure 4 (b)), an additional SHX is placed in between the two compressors, delivering heat to the solution before it enters the absorber and hence cooling down the ammonia vapor.

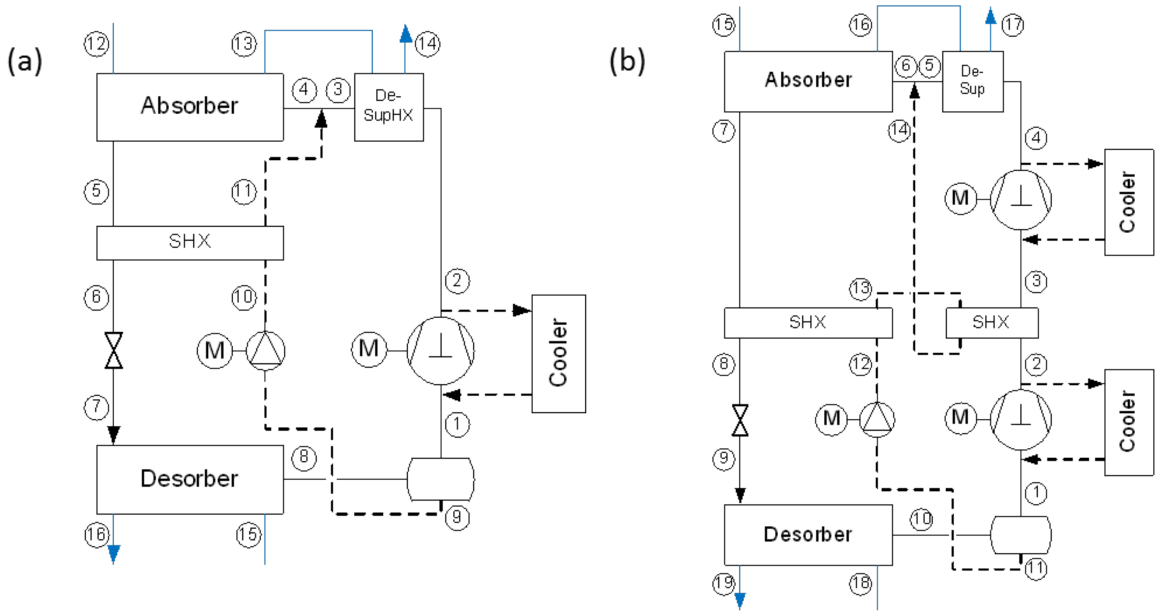


**Figure 4** (a) Single-stage and (b) two-stage hybrid heat pump with de-superheater. Compressor cooler was not considered in the calculations.

Both single- and two-stage hybrid heat pumps have been evaluated, both equipped with a de-superheater. An operating (high) pressure of 22.5 bar was assumed in the calculations, so that standard 25-bar compressors could be used. By increasing the high pressure, the volume flow at the compressor inlet is reduced because of the increased suction pressure, and the pressure ratio in the heat pump is reduced. The COP will thus increase. Water content of the suction gas will also be reduced by increasing the suction pressure. The following additional assumptions were made for the calculations for the hybrid heat pump:

- Thermal efficiency for the de-superheater was set to 0.85
- $\Delta T_{pinch}$ in SHX ($T_5$-$T_{11}$ in Figure 4 (a)) is 2 K
- $\Delta T_{pinch}$ in absorber is 2 K (location of pinch point varies, but it is usually located at the absorber inlet, i.e. $T_4$-$T_{13}$ in Figure 4 (a)/one-stage or $T_6$-$T_{16}$ in Figure 4 (b)/two-stage)
- $\Delta T_{pinch}$ in desorber is 2 K ($T_8$-$T_{15}$ in Figure 4 (a) or $T_{10}$-$T_{18}$ in Figure 4 (b))
- $\Delta T_{pinch}$ in the intermediate cooler ($T_3$-$T_{14}$) is 2 K (two-stage hybrid heat pump)

As a hybrid heat pump is naturally suited for processes with high temperature glide due to the non-isothermal evaporation/condensation processes, parallel connection was not necessary for the hybrid heat pumps.

## 3. RESULTS

The three different ammonia heat pumps (single-stage with and without economizer, and two-stage) were compared with single- and two-stage hybrid heat pumps under the following conditions:

- Three different temperature glides for the heat source, $\Delta T_{source}$: 10, 15 and 20 °C
- Return temperature from DH at 75 °C, and three different supply temperatures, $T_{supply}$: 85, 90 and 95 °C
- Return temperature for DH at 70 °C and supply at 100 °C. This case was included to investigate an extreme case with a high temperature glide at both heat source and sink.

Table 2 shows the results for COP, discharge temperature and operating pressure for the three different temperature glides at evaporator side, return temperature for DH at 75 °C, and three different supply

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temperatures: 85, 90 and 95 °C. Figure 5 shows additionally the COP as a function of DH supply temperature for a return temperature of 75 °C and a temperature glide of 15 °C at the heat source.

**Table 1** COP, discharge temperature and operating pressure for the different heat pump solutions at three different temperature glides at evaporator side, return temperature for DH at 75 °C, and three different supply temperatures: 85, 90 and 95 °C.

| | | **$\Delta T_{source}$ = 10 °C** | | | | | **$\Delta T_{source}$ = 15 °C** | | | | | **$\Delta T_{source}$ = 20 °C** | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **$T_{supply}$ [°C]** | **Heat pump solution** | **COP** | **Discharge temp. [°C]** | | **Operating pressure [bar]** | | **COP** | **Discharge temp. [°C]** | | **Operating pressure [bar]** | | **COP** | **Discharge temp. [°C]** | | **Operating pressure [bar]** | |
| **95** | **$NH_3$ 1-stage** | **3.63** | 167 | 179 | 45.4 | 55.4 | **3.41** | 178 | 184 | 45.3 | 55.3 | **3.21** | 189 | 189 | 45.1 | 55.2 |
| | **$NH_3$ 1-stage with economizer** | **3.74** | 164 | 175 | 45.5 | 55.5 | **3.51** | 175 | 180 | 45.3 | 55.4 | **3.30** | 186 | 185 | 45.1 | 55.3 |
| | **$NH_3$ 1-stage** | **3.86** | 122 | 132 | 46.5 | 56.7 | **3.64** | 126 | 134 | 46.3 | 56.6 | **3.44** | 131 | 136 | 46.2 | 56.6 |
| | **Hybrid 1-stage** | **4.06** | 193.8 | | 22.5 | | **3.86** | 196.3 | | 22.5 | | **3.63** | 210.4 | | 22.5 | |
| | **Hybrid 2-stage** | **4.24** | 133.2 | 132.9 | 22.5 | | **4.09** | 137.5 | 137.1 | 22.5 | | **3.83** | 147 | 145 | 22.5 | |
| **90** | **$NH_3$ 1-stage** | **3.82** | 163 | 170 | 43.8 | 51.1 | **3.58** | 173 | 175 | 43.6 | 51.0 | **3.36** | 185 | 180 | 43.5 | 50.9 |
| | **$NH_3$ 1-stage with economizer** | **3.93** | 160 | 166 | 43.8 | 51.1 | **3.68** | 171 | 171 | 43.7 | 51.0 | **3.50** | 178 | 176 | 41.9 | 50.9 |
| | **$NH_3$ 1-stage** | **4.07** | 119 | 126 | 44.5 | 51.9 | **3.82** | 123 | 128 | 44.4 | 51.9 | **3.60** | 128 | 130 | 44.3 | 51.8 |
| | **Hybrid 1-stage** | **4.35** | 175.8 | | 22.5 | | **4.00** | 184.6 | | 22.5 | | **3.67** | 200.4 | | 22.5 | |
| | **Hybrid 2-stage** | **4.46** | 127 | 126.9 | 22.5 | | **4.15** | | | | | **3.83** | 147 | 145 | 22.5 | |
| **85** | **$NH_3$ 1-stage** | **4.04** | 158 | 160 | 42.1 | 46.8 | **3.77** | 169 | 165 | 42.0 | 46.8 | **3.53** | 181 | 170 | 41.9 | 46.7 |
| | **$NH_3$ 1-stage with economizer** | **4.15** | 156 | 157 | 42.1 | 46.8 | **3.87** | 166 | 162 | 42.0 | 46.8 | **3.62** | 178 | 167 | 41.9 | 46.7 |
| | **$NH_3$ 1-stage** | **4.30** | 116 | 119 | 42.6 | 47.3 | **4.02** | 120 | 121 | 42.5 | 47.3 | **3.78** | 125 | 123 | 42.4 | 47.2 |
| | **Hybrid 1-stage** | **4.47** | 164.9 | | 22.5 | | **4.15** | 173.9 | | 22.5 | | **3.70** | 189 | | 22.5 | |
| | **Hybrid 2-stage** | **4.59** | 121.7 | 122.9 | 22.5 | | **4.20** | 122.9 | 131.1 | 22.5 | | **3.84** | 125 | 138 | 22.5 | |

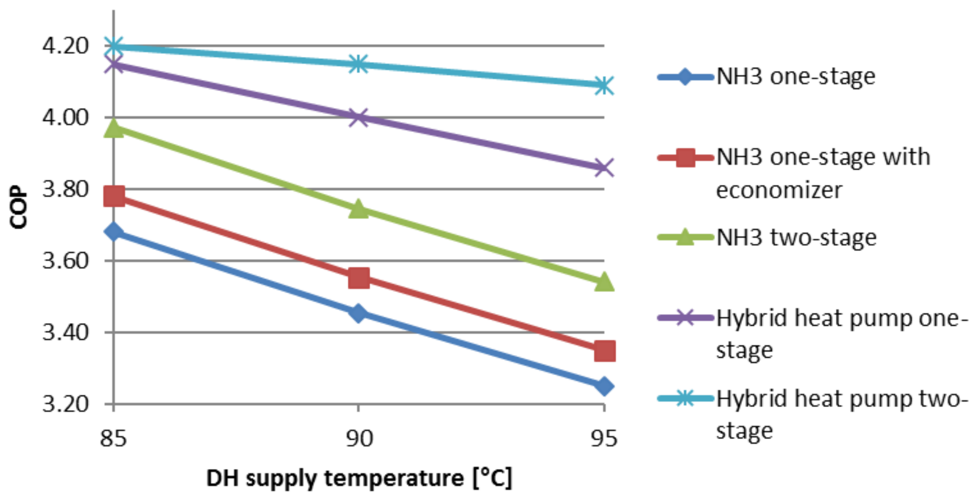


**Figure 5** COP as a function of DH supply temperature for a return temperature of 75 °C, assuming a temperature glide of 15 °C for the heat source (waste water).

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Table 3 shows the results for COP, discharge temperature and operating pressure for three different temperature glides for the heat source, return temperature for DH of 70 °C, and a supply temperature of 100 °C.

**Table 2** COP, discharge temperature and operating pressure for the different heat pump solutions at three different temperature glides at evaporator side, return temperature for DH at 70 °C, and a supply temperature of 100 °C.

| | **$\Delta T_{source}$ = 10 °C** | | | | | **$\Delta T_{source}$ = 15 °C** | | | | | **$\Delta T_{source}$ = 20 °C** | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Heat pump solution** | **COP** | **Discharge temp. [°C]** | | **Operating pressure [bar]** | | **COP** | **Discharge temp. [°C]** | | **Operating pressure [bar]** | | **COP** | **Discharge temp. [°C]** | | **Operating pressure [bar]** | |
| **$NH_3$ 1-stage** | **3.57** | 164 | 186 | 44.3 | 59.1 | **3.36** | 174 | 191 | 44.0 | 58.9 | **3.17** | 186 | 196 | 43.7 | 58.7 |
| **$NH_3$ 1-stage with economizer** | **3.67** | 161 | 182 | 44.3 | 59.2 | **3.45** | 172 | 187 | 44.1 | 59.0 | **3.26** | 182 | 192 | 43.8 | 58.9 |
| **$NH_3$ 1-stage** | **3.77** | 121 | 138 | 45.7 | 61.1 | **3.56** | 125 | 140 | 45.6 | 61.0 | **3.37** | 130 | 142 | 45.4 | 60.9 |
| **Hybrid 1-stage** | **3.88** | 202.7 | | 22.5 | | **3.73** | 208 | | 22.5 | | **3.56** | 215 | | 22.5 | |
| **Hybrid 2-stage** | **4.06** | 138.9 | 139.8 | 22.5 | | **4.03** | 140.5 | 138.6 | 22.5 | | **3.86** | 143 | 143 | 22.5 | |

## 4. DISCUSSION

Based on the results in Table 2 and Table 3, two-stage hybrid heat pump stands out with the highest COP, the lowest operating pressure and a relatively low discharge gas temperature for all the operating conditions. The COP for hybrid heat pump is further hardly affected by an increase in the supply temperature, as opposed to the other solutions, as can be seen from Figure 5. From solutions with ammonia as refrigerant, the two-stage heat pump with a de-superheater has the highest COP and the lowest discharge gas temperature. The COP of two-stage hybrid heat pump was 4 % higher than the COP of two-stage ammonia heat pump at a DH supply temperature of 85 °C, and 12 % higher at a supply temperature of 95 °C, assuming a return temperature of 75 °C and a temperature glide of 15 ° C for the heat source.

Under the most extreme conditions, with a temperature glide of 30 °C for the heat sink (DH return/supply 70/100 °C) and 20 °C for the heat source, the COP of the two-stage hybrid heat pump is 15 % higher the COP of the two-stage ammonia heat pump. Under these operating conditions, the operating pressure of hybrid heat pump is still 22.5 bar, while the operating pressure of the second two-stage ammonia heat pump is 60.9 bar.

For a hybrid heat pump with a mixture of water and ammonia as the refrigerant, the temperature difference between water and refrigerant remains low through the entire heat exchange process due to the gliding temperature in evaporation and condensation, as discussed in section 2.2. With a temperature glide of 10-20 °C in both the evaporator and condenser, the hybrid heat pump has a logarithmic mean temperature difference (LMTD) between 2 and 4 K in both heat exchangers. For the same conditions, ammonia heat pumps have a LMTD at 9-14 K in the evaporator and 11-26 K in the condenser.

A low LMTD ensures low exergy losses in the heat exchange process, but at the same time, this necessitates an increased heat exchanger area. Assuming a heat transfer coefficient of 1200 W/(m²K), a heat exchange area of 119 m² is required at a LMTD of 14 K and 556 m² at a LMTD of 3 K to transmit 2000 kW of heat. In practice, the LMTD of a hybrid heat pump will be slightly higher than the values employed in the calculations here. It should also be mentioned that in a hybrid heat pump the heat transfer coefficient may be lower, as in the absorber/desorber both mass- and heat transfer take place, in contrast to a conventional evaporator/condenser, where only heat transfer takes place.

Finally, in the present study, a constant isentropic compressor efficiency of 0.7 was assumed. This is obviously not realistic; in reality, the isentropic efficiency depends on the pressure ratio, and this should be taken into consideration in a further study.

## 5. CONCLUSIONS

In this study, different heat pump solutions, including single- and two stage heat pumps with ammonia as the refrigerant, as well as single- and two-stage ammonia-water hybrid heat pumps, have been compared for the purpose of DH production. Return temperatures for DH of 70-75 °C, and supply temperatures from 85 up to 100 °C were investigated. The heat source was an industrial wastewater stream at 40 °C, and the desired heating capacity was 4 MW. In this comparison, two-stage hybrid heat pump stands out with the highest COP, the lowest operating pressure and a relatively low discharge gas temperature. The COP for hybrid heat pump is further hardly affected by an increase in the supply temperature, as opposed to the other solutions. From solutions with ammonia as refrigerant, the two-stage heat pump with a de-superheater has the highest COP and lowest discharge gas temperature. The COP of two-stage hybrid heat pump was 4-15 % higher than the COP of the two-stage ammonia heat pump, depending on the temperature glide at the heat source/sink. Moreover, the operating pressure of a hybrid heat pump can be kept low, at 23 bar, whereas for the two-stage ammonia heat pump, the operating pressures from 47 up to 61 bar were required to achieve the desired supply temperature levels.

The properties of ammonia-water mixture give the possibility to adapt the temperature glide of the working fluid in condensation and evaporation processes to the temperature glide of the heat source and sink, respectively. As a result, the driving temperature difference between warm and cold side of the heat exchanger can be kept low throughout the heat exchanger in a counter flow heat exchange process. Ammonia-water mixture will however have different heat exchange properties in condensation/evaporation as opposed pure ammonia with regards to e.g. heat transfer coefficients, and larger heat exchange areas are required, which should be taken into consideration in a further study. Moreover, in the present study, a constant isentropic compressor efficiency was additionally assumed, regardless of the pressure ratio.

To conclude; in a further feasibility study, two-stage hybrid and ammonia heat pumps should be considered, and realistic, varying compressor efficiencies and heat transfer coefficients should be employed in the calculations. The cost of heat exchangers and other components should additionally be taken into account.

## ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of The Research Council of Norway and the industrial partners Statoil Petroleum AS, Hydro Aluminium AS, Statkraft Varme AS, Vedde AS, member of TripleNine Group Vedde AS, Mars GmbH, TINE SA, Cadio AS, Hybrid Energy AS and EPCON Evaporation Technology AS through the grant NFR-243679 (HeatUp).

## REFERENCES

1. CNN, 2011, Helsinki's underground master plan. Retrieved 21.3.2016, from http://www.huffingtonpost.com/2011/02/19/helsinki-underground-data-center_n_823091.html.
2. Fortum, 2016, Bahnhof Pionen - Profitable recovery by district heating. Retrieved 18.3.2016, from http://www.opendistrictheating.com/pilot/bahnhof_pionen/.
3. Friotherm, 1995, Energy from sewage water – District heating and district cooling in Sandvika.
4. Grinrød, J. M. 2012, Høytemperatur fjernvarme med ammoniakk som kuldemediet. *Fjernvarmedagene*, Oslo, Norway, Norsk Fjernvarme.
5. IEA, 2012, Energy Technology Perspectives 2012: Pathways to a Clean Energy System.
6. Laajalehto, T., M. Kuosa, T. Mäkilä, M. Lampinen and R. Lahdelma, 2014, Energy efficiency improvements utilising mass flow control and a ring topology in a district heating network. *Applied thermal engineering* **69**(1): 86-95.
7. Nordtvedt, S. R., B. R. Horntvedt, J. Eikefjord and J. Johansen, 2011, Hybrid heat pump for waste heat recovery in Norwegian food industry, *10th International Heat Pump Conference*.

http://dx.doi.org/10.18462/iir.nh3-co2.2017.0037

8. Persson, U., B. Möller and S. Werner, 2014, Heat Roadmap Europe: Identifying strategic heat synergy regions. *Energy Policy* **74**: 663-681.
9. Sveinall, O. 2008, Analyse av kloakkbasert R134a-varmepumpeanlegg i fjernvarmesystem, Master's Thesis, Norwegian University of Science and Technology.