Cool ways of using low grade Heat Sources from Cooling and Surplus Heat for heating of Energy Efficient Buildings with new Low Temperature District Heating (LTDH) Solutions.

Deliverable no.:	**D2.11**
Name of deliverable:	**Working paper on optimising cascade couplings for optimal use of low temperature sources**
Revision no.:	**1**

Due date of deliverable:	M8
Actual submission date:	M9
Start date of project:	1. October 2017
Duration:	48 Months

Organisation name of lead contractor for this deliverable: Kraftringen Energi AB & COWI A/S

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This project has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement No 767799

Dissemination level		
PU	Public	X
PP	Restricted to other programme participants (including the Commission Services)	
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Scope of deliverable

This deliverable describes an analysis of energy efficiency in heat pump systems that recover waste heat from low temperature heat sources for District Heating.

Context of deliverable

Heat recovery of waste heat available at low temperatures is a key element in both of the demo sites in the COOL DH project.

Perspective of deliverable

The approach and the output from this working paper will form a base when designing the heat recovery systems in the COOL DH project in Lund (SE) and Høje-Taastrup (DK).

Involved partners

Kraftringen Energi AB has prepared this paper together with COWI A/S, who has performed the simulations. Setting up the simulation cases was a joint effort.

English summary

This report studies parameter variations of cascade couplings of heat pumps used to produce district heating based on low temperature heat sources of 7-23 °C as basis for demonstration projects in Brunnshög district of Lund in Sweden and in Høje-Taastrup in Denmark.

The study concludes that the way to reach high energy performance is to put effort into several various aspects. When combined, these efforts will combine into large potential energy savings.

The main findings in this study was the highest overall COP_H is achieved when similar heat pumps are working under similar conditions. This means that each heat pump in the system should work with the same temperature differences on both the hot and cold side. It was also concluded that lower district heating supply temperatures results in higher COP_H values, and that this parameter has significant influence on the heat pump performances making it more feasible for low temperature district heating. Furthermore, a theoretical study of the refrigerants used as working fluid in the heat pumps showed that R717 (Ammonia) resulted in the highest COP_H of the refrigerants investigated.

The simulations described in this report are based on real life systems, with a number of assumptions and simplifications to reduce the complexity of the calculations. However, the models are still considered close to real life operation, and thereby the relationships between COP and system inputs varied in the parametric analyses referred to in the appendix can be transferred from the theoretical study to real life situations.

List of content

Introduction

Recovery of waste heat for heating purposes is a huge subject that could be (and has been) a subject for book writing. In this context we are concentrating on how to design efficient heat recovery heat pump processes for the particular cases we can foresee at the two demo-sites for this project: The city of Høje-Taastrup in Denmark and the Brunnshög area in the city of Lund in Sweden.

We believe that the results we obtain here are relevant also in numerous other cities and areas in much of Europe. The temperature levels of the waste heat are similar to waste heat temperatures that occur in many places. And the temperature levels of heating and tap water systems are also similar to those found elsewhere. And the basic thermodynamics behind heat pump processes are certainly the same everywhere.

We have divided our study into three steps and consequently this report into three main sections. The first section treats some theoretical but important steps to take into consideration. In the second section, we describe the specific cases we have studied and calculated. In the third section we present the results from our calculations and simulations.

Site descriptions
Brunnshög and MAX IV in Lund

In the north-eastern part of the city of Lund in southern Sweden, a new area for research, residential and commercial buildings and recreation is under construction. This is one of the demo-sites for low temperature district heating that is the main topic of this project. The first research facility to be established in the area was the large-scale material research facility MAX IV. Kraftringen owns and operates the cooling and heat recovery central at MAX IV. The core process equipment here are the heat pumps. These are the subject for a part of this report.

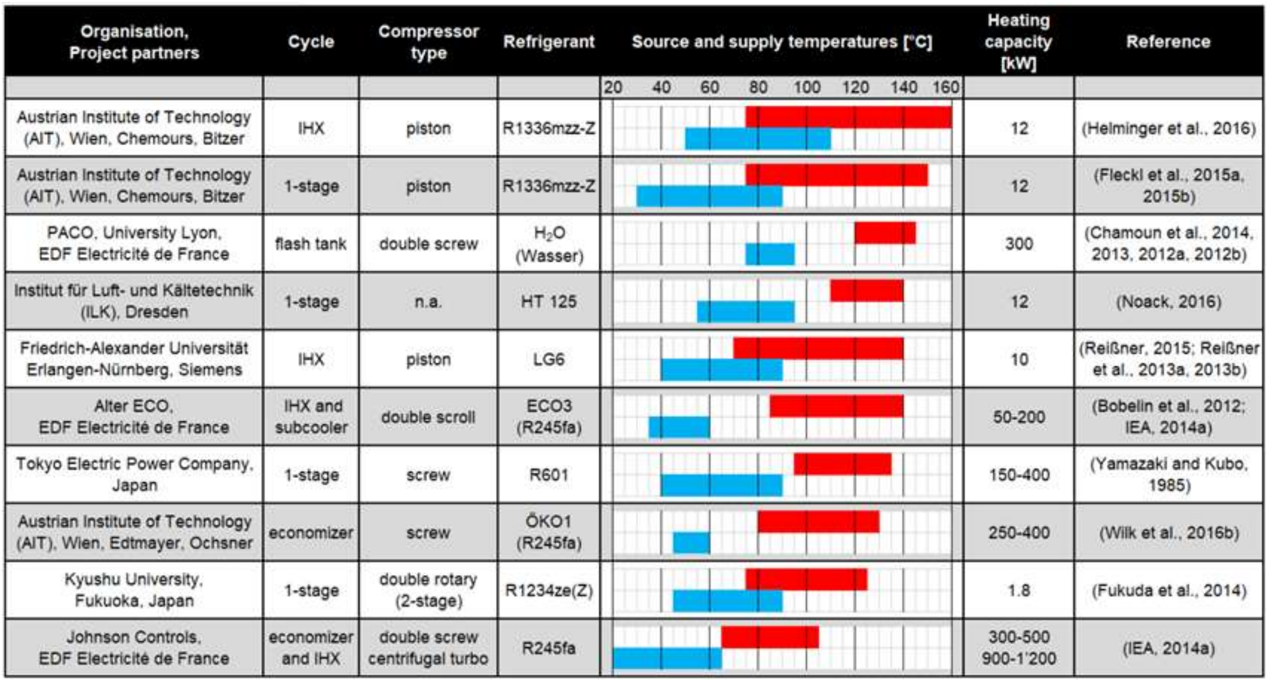
Høje –Taastrup and City2

Høje –Taastrup is located in the greater Copenhagen area, and is another demo-site for this project. Here the local energy company Høje –Taastrup Fjernvarme is recovering heat from low temperature waste heat. In the present project, more waste heat is to be recovered from the City2 shopping mall. The present cooling machines will be supplemented by heat pumps in order to increase the temperature level of the recovered heat. These heat pumps is another subject for this report.

Method

We wanted this report to give us valuable input for the design of the systems in Lund and Høje –Taastrup, but also that it should be a good resource when optimising other heat pump installations. To fulfil these ambitions we have used input from several sources.

First of all, we already have a substantial experience by ourselves. Kraftringen has installed and operates large-scale heat pumps for heating and/or cooling at six production sites, with capacities ranging from 1 to 25 MW. COWI has acted as consultant in multiple projects involving heat pumps with different applications, hence achieving experience how heat pumps can be integrated in other systems. Heat pumps can be designed to operate in a wide variety of applications with different hot and cold source temperatures, heating capacity, compressor type, refrigerant etc. A short selection of various heat pump systems is shown in Figure 1, showing systems with different design parameters[1]. This gives an understanding of the extensive opportunities for applying heat pumps.



Organisation, Project partners	Cycle	Compressor type	Refrigerant	Source and supply temperatures [°C]	Heating capacity [kW]	Reference
				20 40 60 80 100 120 140 160		
Austrian Institute of Technology (AIT), Wien, Chemours, Bitzer	IHX	piston	R1336mzz-Z		12	(Helminger et al., 2016)
Austrian Institute of Technology (AIT), Wien, Chemours, Bitzer	1-stage	piston	R1336mzz-Z		12	(Fleckl et al., 2015a, 2015b)
PACO, University Lyon, EDF Electricité de France	flash tank	double screw	H_2O (Wasser)		300	(Chamoun et al., 2014, 2013, 2012a, 2012b)
Institut für Luft- und Kältetechnik (ILK), Dresden	1-stage	n.a.	HT 125		12	(Noack, 2016)
Friedrich-Alexander Universität Erlangen-Nürnberg, Siemens	IHX	piston	LG6		10	(Reißner, 2015; Reißner et al., 2013a, 2013b)
Alter ECO, EDF Electricité de France	IHX and subcooler	double scroll	ECO3 (R245fa)		50-200	(Bobelin et al., 2012; IEA, 2014a)
Tokyo Electric Power Company, Japan	1-stage	screw	R601		150-400	(Yamazaki and Kubo, 1985)
Austrian Institute of Technology (AIT), Wien, Edtmayer, Ochsner	economizer	screw	ÖKO1 (R245fa)		250-400	(Wilk et al., 2016b)
Kyushu University, Fukuoka, Japan	1-stage	double rotary (2-stage)	R1234ze(Z)		1.8	(Fukuda et al., 2014)
Johnson Controls, EDF Electricité de France	economizer and IHX	double screw centrifugal turbo	R245fa		300-500 900-1'200	(IEA, 2014a)

Figure 1: Different usage of heat pumps. Presented on "International Workshop on High Temperature Heat Pumps", in Copenhagen 9th September 2017.

It is forbidden by Danish law to apply synthetic refrigerants except HFO's (hydrofluoroolefin) with low global warming potential (GWP) values, e.g. R1234ze. Ammonia is commonly used for larger scale systems, i.e. larger than 500 kW, while hydrocarbons such as isobutene and propane is used for smaller scale systems. CO_2 can be applied for medium sized systems, e.g. between 200 kW and 500 kW[2].

The actual work within the current started with a thematic workshop, held in Lyngby, Denmark on April 5th 2018. The participants in this workshop included also external participants e.g. from the Danish Institute of Technology, giving valuable input to this study.

A simulation tool is very useful when analysing the various parameters. The thermal simulations of the heat pump systems where conducted using the commercial software Engineering Equation Solver, EES. EES can solve multiple equations of any type numerically. The great advantage of the software is the built-in thermodynamic properties for numerous fluids such as water and refrigerants R134a and R1234ze.

Equations for energy and mass conservation were set up in EES for each node in the heat pumps systems in the different cases, in order to carry out a parametric study of different system parameters. Since we have operational data for the heat pumps at MAX IV, we could calibrate the simulation model using different operation data as inputs, and obtained good results.

[1] Cordin Arpagaus, Fréderic Bless, Jürg Schiffmann and Stefan S. Bertsch (2017), *Review on High Temperature Heat Pumps – Market Overview and Research Status.*

[2] Danish Technological Institute, *Heat pumps and Generic first assessment tool and heat pump development at DTI*.

We could therefore simulate the MAX IV installation, varying several parameters. The heat pump model was then used for a parametric study on two other cases: City 2 shopping mall in Høje-Taastrup and a theoretical case with multiple heat pumps in cascade.

Short theory

Heat pump theory and technology is dealt with in text books in all relevant engineering educations, and we will not include a long section on this subject in this report. However, a short summary is relevant to include.

Basically, the efficiency of a heat pump is defined as the quota between the heat output from the condensing side of the heat pump and the power consumption in the compressor. The efficiency is labelled Coefficient of Performance, COP. In this report this refers to the Carnot COP.

The COP of a heat pump is affected by many of different factors, but the highest theoretically possible COP is defined by properties of the working media in the heat pump. After some simplifications, this maximum COP is calculated as a quota between the condensing temperature on the warm side of the heat pump and the temperature difference between this condensing temperature and the evaporator temperature on the cold side of the heat pump:

$$\mathrm{COP_{Carnot}} = \frac{T_{\mathrm{Cond}}}{T_{\mathrm{Cond}} - T_{\mathrm{Evap}}} \qquad \text{(Temperatures in Kelvin)}$$

In reality, the actually achievable COP of a heat pump is around 50-70 % of this value.

The equation above implies:

- The crucial temperatures are the forward cooling temperature and the forward heat recovery temperature.
- The return temperatures are less important.
- It is more important to try to increase the cooling temperature than to decrease the heating temperature.
- If there are cooling demands at different temperatures, or the heating demands can be differentiated by temperature, it might be a good idea to divide into separate cooling and/or heating systems and install separate heat pumps.

Further studying of the equation above reveals that in systems with a large temperature difference between the supply and return temperatures, it is a good idea to divide the heat pump systems into several steps with cascade coupling. This is typically the case with district heating applications.

A heat pump for District Heating is a big investment. It is important to consider the relation between the investment cost and the operating cost. For the type of heat pump applications we discuss in this report, the heat pump investment is in the same range as two years of operational costs. This means that it could be a good idea to purchase a heat pump that is twice as expensive, if the energy efficiency can be increased by say 20 %.

Case studies

We have divided the results section into two parts. The first part is dealing with the present set-up of the MAX IV heat pump installation, and the consequences resulting from the preparations we made before the installation there. The second part is dealing with simulation results of the two sites and a theoretical installation.

The three simulation cases of systems with heat pumps are described below.

Case 1 – MAX IV: Existing plant where heat pumps are used to deliver cooling for an application and transfer the heat to the district heating system. The application with the cooling demand is the particle accelerator located at the University of Lund, Sweden. The cooling demand is approximately 1.3 MW and three heat pumps are coupled to the system to transfer the heat from the cooling system to the district heating system.

Case 2 – Høje-Taastrup: A theoretical example of heat pumps used for cooling at the shopping mall City 2 in Høje-Taastrup.

Case 3 – Cascade coupled heat pumps supplying district heating: A theoretical example of how heat pumps can be used in cascade using different heat sources to supply the district heating network. The electrical input to the heat pumps is in this case presumed to come from renewable sources, e.g. solar or wind power.

Results

This chapter starts with an evaluation of the initial set-up of the MAX IV installation. The results from the three simulation case studies investigated in this project are presented individually. This chapter is followed by a chapter with a discussion and recommendations based on the findings from this study.

Evaluation, MAX IV setup

When designing the MAX IV energy recovery process, we started by identifying all cooling demands and their temperature requirements. This will not be treated here. We ended up with two temperatures for the cooling demands, one at 7 °C and one at 23 °C. We wanted to recover this heat to our district heating network at 80 °C.

After discussions with suppliers and as a result of the procurement process, we installed heat pumps which are nicely cascade-coupled both on the cold side and the warm side. Flow diagrams are found in the following section.

We started operations of the first heat pumps in June 2014 and recovered 20 GWh to the district heating network in 2017. Therefore, we now have enough data to make a preliminary evaluation of the efficiency measures, even though we expect these figures to increase during the upcoming years with the gradual extension of the MAX IV facility.

Current data:

Cooling capacity: 2 MW at 7 °C and 1 MW at 23 °C

Full-load operating hours: 5500 h/yr

COP-values: 3 and 3.7, respectively

Cooling production: 16500 MWh/yr

Power consumption: 7500 MWh/yr

If all cooling would have been supplied in a single 7 °C-system, the COP would have been reduced to 3 for all recovered heat. Calculations using the equation above show that the COP would have been 2.7 without the cascade coupling. The resulting power consumption then would have been 9700 MWh/yr.

The savings thus are around 2200 MWh/yr of electricity. With the current electricity price of 700 SEK/MWh in Sweden, this corresponds to yearly savings of 1.5 MSEK. Since the investment cost for these types of heat pumps is around 3 MSEK/MW cooling, it seems as the energy savings measures we undertook in the design phase pays off.

Results Simulation Case 1 – MAX IV

A principal diagram of the heat pump system modelled in the MAX IV case is shown in Figure 2.

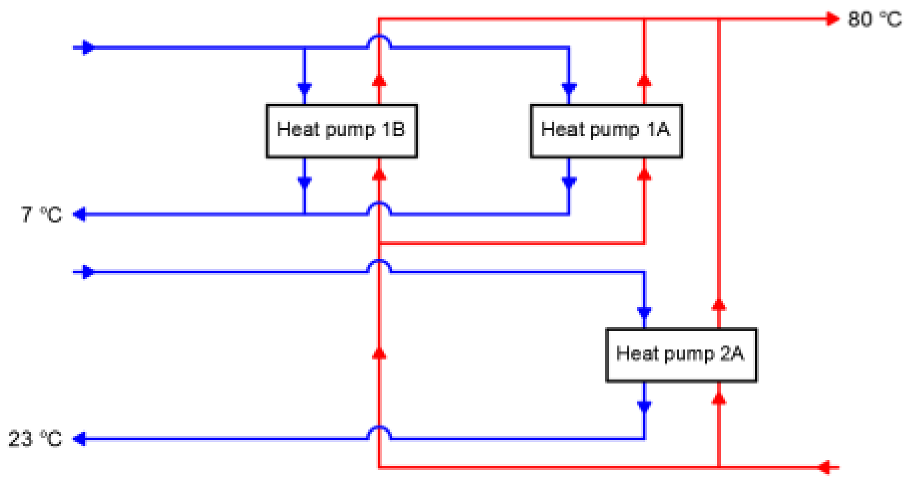


Figure 2: Principal diagram of the heat pump system modelled for the MAX IV case.

The principal diagram in Figure 2 shows the heat pump configurations in MAX IV. The district heating line is shown with red in the figure, where the supply line is in the top right and the return line in the bottom right. The cooling system is divided in two separate systems: a cold side connected to Heat pump 1A and 1B and a warmer side connected to Heat pump 2A. Each of the three heat pumps shown in Figure 2 consist of four individual heat pumps placed in cascade.

These individual heat pumps are similar in principal. A principal diagram of Heat pump 1B is shown in Figure 3. Heat pump 1B is selected for the calibration and the parametric study, as the findings are considered valid for Heat pump 1A and 2A as well.

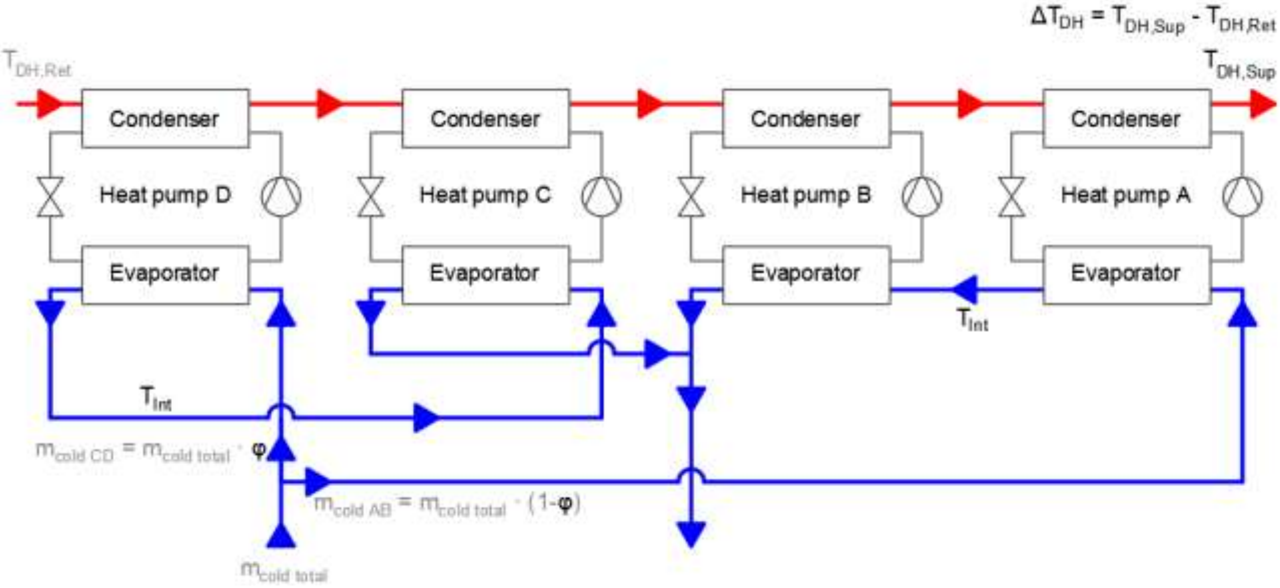


Figure 3: Principal diagram of Heat pump 1B on MAX IV.

The selected parameters shown with black in Figure 3 are investigated in the parametric study presented later.

Calibration

As mentioned in the methods chapter, the model of MAX IV was calibrated with operation data. Various temperatures and flow distributions were used as model input, and the corresponding outputs was compared with the operation data. The results from the calibration are shown in Figure 4. Please note that the Heat pump D was not operating at the time of retrieving the operation data, which is included in the model for calibration. Heat pump D is included in the parametric study.

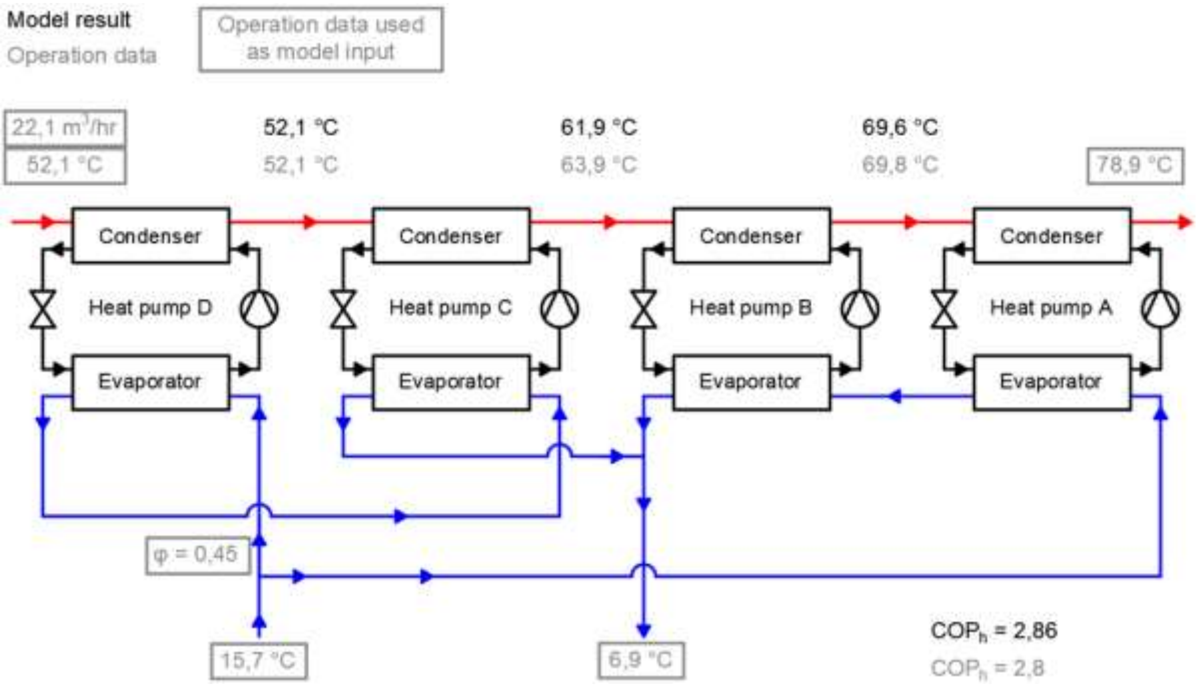


Figure 4: Operation data and values from calibration simulation of MAX IV Heat pump 1B.

Figure 4 shows that the results from the calibration simulation corresponded well with the operation data. This simulation was used to adjust pinch temperatures and temperature differences in the heat pump compressor and evaporator.

The deviations are due to the assumptions and simplifications made in the model in EES. For example, all compressor efficiencies are fixed to 0.7, and heat and pressure losses are not included in the model. Any bypass due to the temperature control is not included. Despite the assumptions and simplifications in the model, it is considered sufficiently accurate to use for the parametric study.

Parametric study

The parametric study involved variation of five parameters and investigating the effect on the coefficient of performance, COP, of the heat pumps. The five parameters investigated in the parametric study are divided into one simulation run with reference values and four individual simulation runs varying each parameter. The variables are described in Table 1. The location of the variables are shown in the principal diagram in Figure 2 on Page 9. The physical meaning of the parameter φ is seen in Figure 2 in the lower left corner, as it is defining how the cold water is split between heat pumps C and D ($m_{cold\ CD}$) and heat pumps A and B ($m_{cold\ AB}$). If φ = 1, all cold water is towards heat pumps C and D and vice versa. If φ = 0.5, the cold water is split evenly between the heat pumps.

Simulation case no.	Description	Input values				
		φ [-]	T_{Int} [°C]	$T_{DH,Sup}$ [°C]	ΔT_{DH} [°C]	Refrigerant
A	Reference calculation	0,5	11	80	30	R134a
B	Variation of cold water flow distribution (φ)	[0.0 – 1.0]	11	80	30	R134a
C	Variation of intermediate temperature on cold water side (T_{Int})	0.5	[8 - 14]	80	30	R134a
D	Variation of district heating (DH) supply line temperature ($T_{DH,Sup}$) and temperature difference between district heating supply and return line (ΔT_{DH})	0.5	11	[50 - 80]	[20 - 40]	R134a
E	Variation of heat pump refrigerant	0.5	11	80	30	R134a R1234ze R600a R717

Table 1: Overview of the parametric study on Heat pump 1B in MAX IV.

The values for the parameters in each of the simulation cases are shown in Table 1. The range of which each parameter is varied is shown with square brackets, and the constant values of the remaining variables are similar to the reference Simulation case A.

The reference case is based on the operation data, but values have been rounded for simplification. Results of COP_H from Simulation case A are presented in Table 2, and is used as reference in the parametric study. For convenience, the reference values of COP_H is shown with grey in the following tables.

Simulation case no.	Description	Input					Output
		φ [-]	T_{Int} [°C]	$T_{DH,Sup}$ [°C]	ΔT_{DH} [°C]	Refrigerant	COP_H [-]
A	Reference calculation	0,5	11	80	30	R134a	2.95

Table 2: Results from Simulation case A, which is the reference case in the parametric study.

The reference value for COP_H in the parametric study varies slightly from the calibration case, as the input parameters have been rounded. The influence of each parameter is discussed in the following pages.

The first parametric study, Simulation case B, is varying the cold water flow distribution to the heat pumps, which is denoted φ, from 0 to 1. Results of COP_H from this study is presented in Figure 4, and the table with values is included in Table 13 in the Appendix.

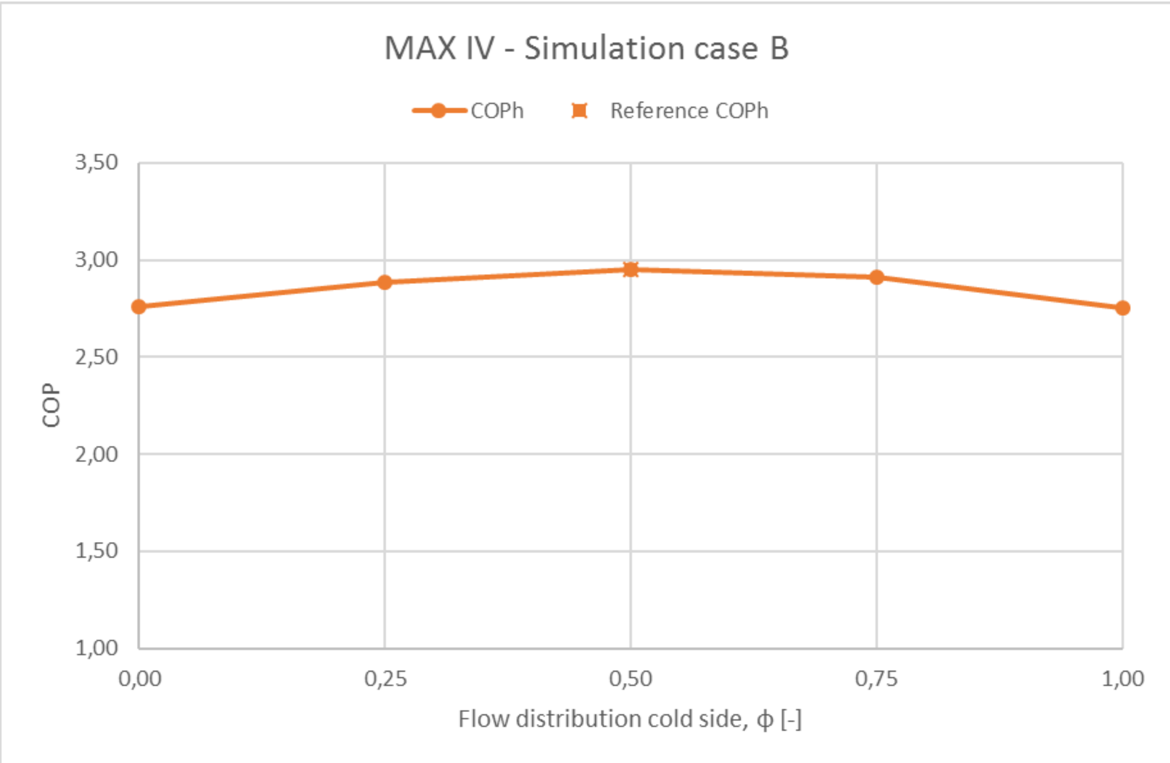


Figure 5: Graphical presentation of the results from Simulation case B showing variation of the cold water flow distribution, ϕ.

The results of the parametric study of the cold water flow distribution, φ, in Figure 4 shows the calculated COP_H values that ranges from 2.75 to 2.95. The highest value is achieved when φ = 0.5 that ensures an even cold water flow distribution between all heat pumps. This shows that multiple heat pumps in cascade all working under the same conditions delivers a higher COP_H, as the heat transfer is carried out in steps.

The second parametric study, Simulation case C, is varying the intermediate temperature on the cold water side, T_{Int}, between evaporator C and D and between evaporator A and B. The intermediate temperature is varied from 8 °C to 14 °C, which is considered the maximum possible range, as the supply and return temperatures on the cooling circuit are fixed in the model to 7 °C and 15 °C respectively. Results of COP_H from this study is presented in Figure 5, and the table with values is included in Table 14 in the Appendix.

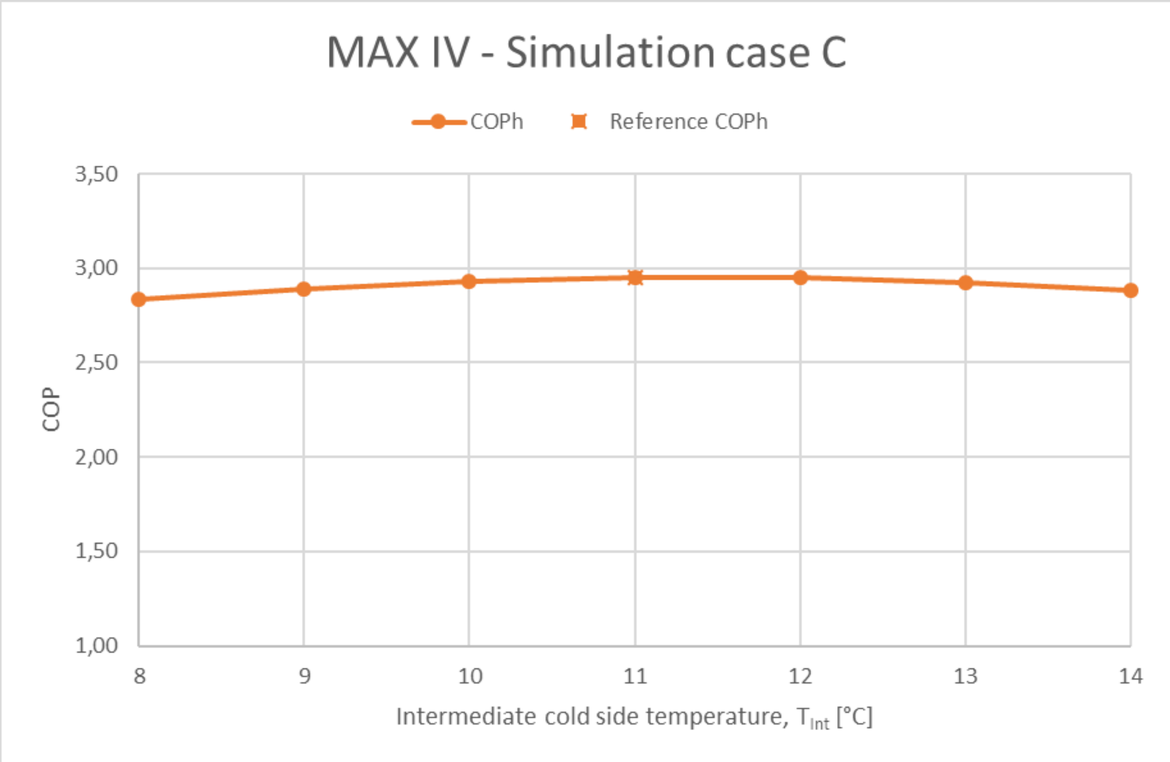


Figure 6: Graphical presentation of the results from Simulation case C showing variation of the intermediate temperature on cold waterside, T_{Int}.

Calculated COP_H values from the parametric study of the intermediate temperature on the cold water side, T_{Int}, in Figure 5 ranges from 2.84 to 2.95. The maximum COP_H value is found when T_{Int} = 11 °C, which is the reference case. A value of 11 °C gives a temperature difference of 4 °C on each heat pump evaporator, as the cold water supply and return temperatures are 7 °C and 15 °C respectively. Again, this shows that the highest COP_H is achieved when all heat pumps, which are similar, in the cascade are working under the same conditions, which in this case means equal temperature difference across the evaporator.

The third parametric study, Simulation case D, is varying both the district heating (DH) supply line temperature, $T_{DH,Sup}$, from 50 °C to 80 °C and temperature difference between district heating supply and return line, ΔT_{DH}, from 20 °C to 40 °C. Lowering the district heating supply temperature is particularly interesting in this project involving low temperature district heating networks. Results of COPH from this study is presented in Figure 6, and the table with values is included in Table 15 in the Appendix.

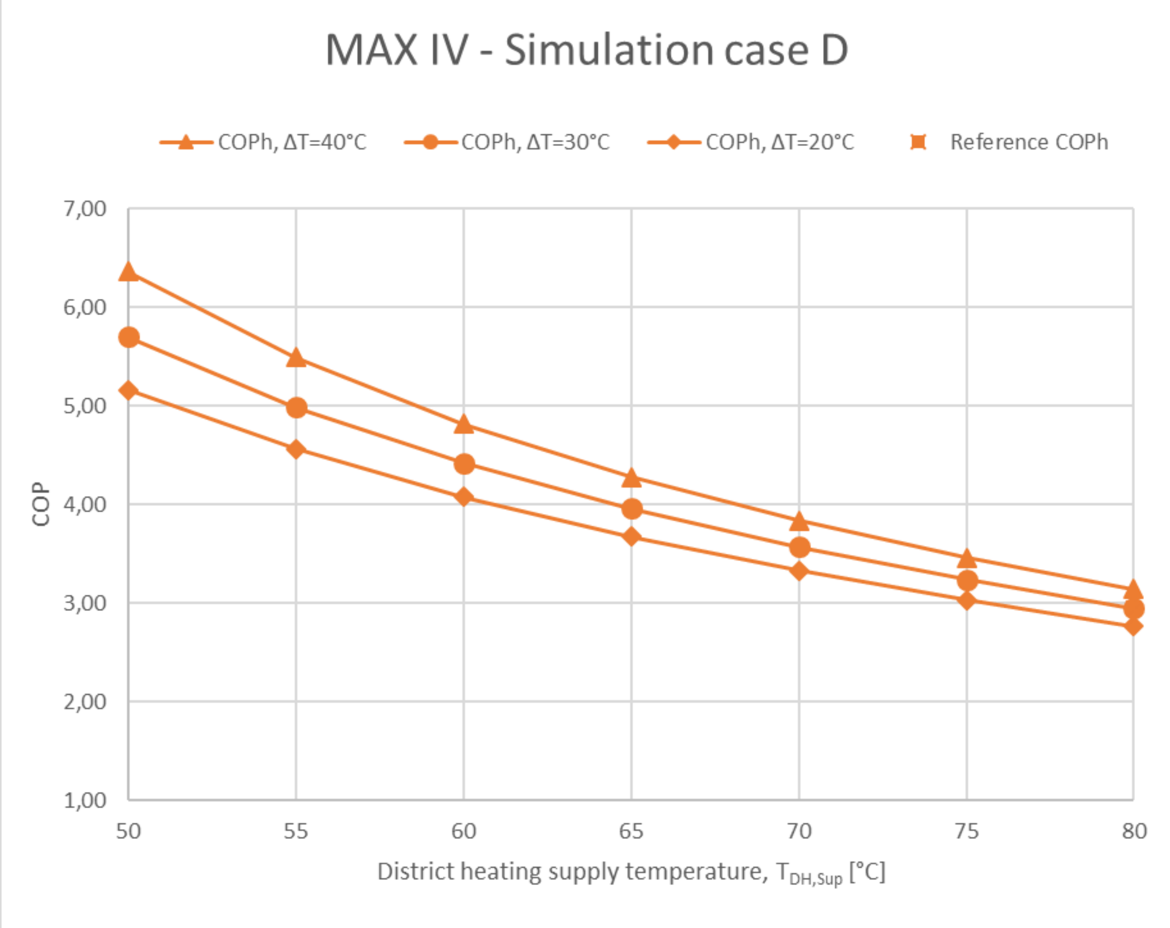


Figure 7: Graphical presentation of the results from Simulation case D showing variation of the district heating supply temperature, $T_{DH,Sup}$, and the temperature difference between district heating supply and return line, ΔT_{DH}.

Figure 7 shows different combinations of DH supply temperature and DH temperature difference, and the COP_H ranges from 2.77 to 6.37. COP_H increases significantly when the DH supply temperature is lowered, which is expected as the temperature difference between the hot and cold sides decreases. A large DH temperature difference also results in higher COP_H.

The fourth and last parametric study, Simulation case E, is analysing different refrigerants as working fluid in the heat pumps. This study is considered more theoretical than practical, as changing the refrigerant in the existing system is not necessarily possible. Furthermore, some refrigerants possibly does not operate optimally under the conditions fixed in the MAX IV study.

The four refrigerants investigated in Simulation case E are R134a, R1234ze, R600a (Isobutane) and R717 (Ammonia).

Results of COP_H from this study is presented in Table 3.

Simulation case no.	Description	Input: φ [-]	Input: T_{Int} [°C]	Input: $T_{DH,Sup}$ [°C]	Input: ΔT_{DH} [°C]	Input: Refrigerant	Output: COP_H [-]
E	Variation of heat pump refrigerant	0,5	11	80	30	R134a	2.95
						R1234ze	2.94
						R600a	3.00
						R717	3.26

Table 3: Results from Simulation case E showing different refrigerants as working fluid in the heat pumps.

The calculated COP_H values in Table 3 ranges from 2,94 to 3,26, and the highest calculated COP_H is for R717, which is ammonia. Considering the environmental aspects, the natural refrigerants Isobutane and Ammonia are more desirable as well.

Partial conclusion

The heat pump configuration in MAX IV has now been analysed and different input parameters were varied in order to study the effect on the COPH of the heat pump. The main conclusions from each simulation case are summed up in Table 4.

Simulation case no.	Description	Conclusion
A	Reference calculation	-
B	Variation of cold water flow distribution (φ)	Four heat pumps working in cascade instead of only two resulted in a higher COP_H. The highest COP_H was found when all four heat pumps had equal mass flow in the cooling circuit.
C	Variation of intermediate temperature on cold water side (T_{Int})	Again, this shows that the highest COP_H is achieved when all heat pumps in the cascade are working under the same conditions, which in this case means equal temperature difference across the evaporator.
D	Variation of district heating (DH) supply line temperature ($T_{DH,Sup}$) and temperature difference between district heating supply and return line (ΔT_{DH})	COP_H increases significantly when the DH supply temperature is lowered. A large DH temperature difference also results in higher COP_H.
E	Variation of heat pump refrigerant	Refrigerant R717 (ammonia) resulted in the highest COP_H compared to the other refrigerants analysed.

Table 4: Conclusions from the simulation cases of MAX IV.

For the Brunnshög demo site, where MAX IV will be the main production facility for several years to come, it especially promising to look forward to an increase in COP from 3 to 4 with the production of low temperature district heating.

Results Case 2 – City 2, Høje-Taastrup

A principal diagram of the heat pump system modelled in the Høje-Taastrup City 2 case is shown in Figure 7. On the right hand side of the figure the process water for comfort cooling is cooled in the primary heat pump, which is a piston driven compression heat pump. The cold water supply temperature, $T_{cold,Sup}$, is 6 °C, and the return temperature, $T_{cold,Ret}$, is 16 °C. Energy is transferred from the primary HP to the secondary HP via an intermediate water circuit, driven by a circulation pump which power consumption is included in the COP calculations. The district heating water is heated from 35 °C to 55 °C in the secondary heat pump driven by a screw compressor.

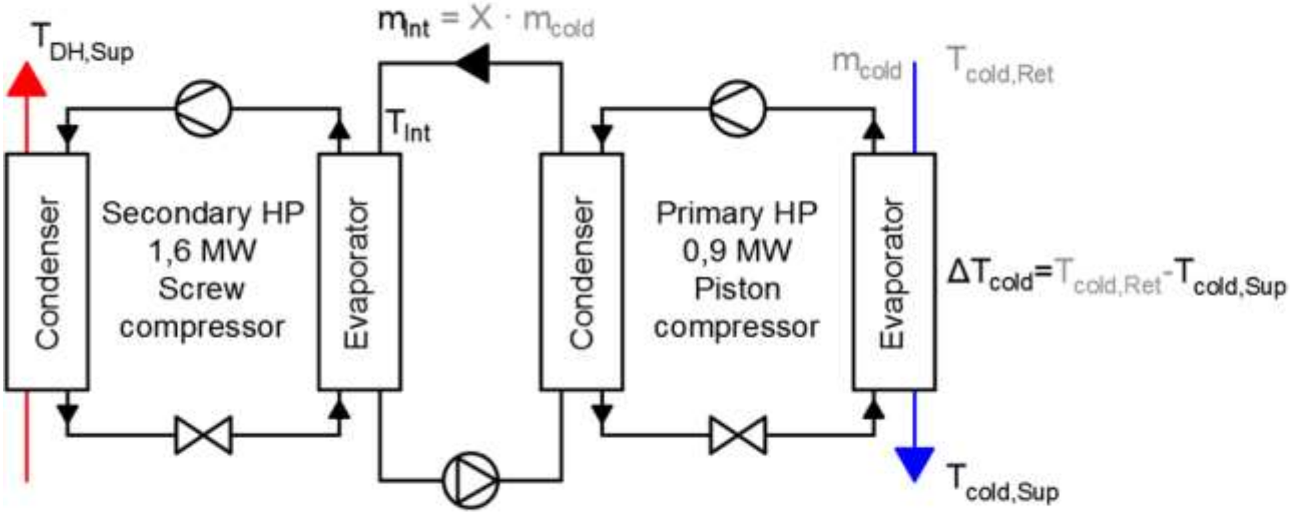


Figure 8: Principal diagram of the heat pump modelled for City 2, converting a peak load cooling machine into a secocondary heat pump operating in cascade.

In this simulation case, the compressor efficiencies and the pump power consumption are not fixed. The compressor efficiency is depending on the load level, and follows the curves depicted in Figure 17 included in the Appendix. The pump power consumption is defined according to the known pump affinity laws[3], which is expressed by:

$$\left(\frac{Q_1}{Q_2}\right)^3 = \frac{P_1}{P_2} <=> P_2 = \left(\frac{Q_2}{Q_1}\right)^3 \cdot P_1$$

P denotes the power consumption and *Q* denotes the volume flow rate. The circulation pump is a assumed to be a Grundfos TPE 150-170/4-S pump[4], with a corresponding power, P_1, of 12 kW and a volume flow rate, Q_1, of 350 m^3/h, which is the flow rate in the reference case, and a Δp of 1 bar. Thereby, the corresponding power, P_2, at a certain flow rate, Q_2, can be calculated. The power, P_2, increase with the power of three to the ratio of the flow rate. E.g. by increasing the flow rate with a factor of two, the power increases with a factor of eight. The losses in the electric motor and circuit is not modelled.

[3] Munson et al. (2013), *Fluid Mechanics 7th edition*
[4] https://product-selection.grundfos.com/product-detail.product-detail.html?from_suid=1528195219527047685654667618094&pumpsystemid=386271560&qcid=386274134

Parametric study

The parametric study consists of a variation of seven parameters and investigating the effect on the COP_H of the heat pumps. Similar to the previous case with MAX IV, the first simulation run determines the reference values, followed by six simulations runs with varying parameters. The variables are described in Table 5, and the location of the variables are shown in the principal diagram in Figure 7.

The meaning of the parameter "Load level" is percentage of maximum cooling capacity, which is 2.5 MW. The parameter m_{Int} denotes the mass flow in the intermediate circuit, and the value of this flow is set as twice the mass flow of the cooled process water in the reference case.

Simulation case no.	Description	Input values						
		Load level [%]	$T_{cold,Sup}$ [°C]	ΔT_{cold} [°C]	m_{Int} [kg/s]	T_{Int} [°C]	$T_{DH,Sup}$ [°C]	Refrigerant
A	Reference calculation	80	6	10	$2 \cdot m_{cold}$	30	55	R134a
B	Variation of load level	[40 - 100]	6	10	$2 \cdot m_{cold}$	30	55	R134a
C	Variation of cooling supply temperature and temperature difference between cooling supply and return line	80	6 & 8	[4 - 12]	$2 \cdot m_{cold}$	30	55	R134a
D	Variation of flow in the intermediate circuit	80	6	10	$[0,5 - 10] \cdot m_{cold}$	30	55	R134a
E	Variation of intermediate temperature	80	6	10	$2 \cdot m_{cold}$	[25 - 35]	55	R134a
F	Variation of district heating supply line temperature	80	6	10	$2 \cdot m_{cold}$	30	[45 - 65]	R134a
G	Variation of heat pump refrigerant	80	6	10	$2 \cdot m_{cold}$	30	55	R134a R1234ze R600a R717

Table 5: Overview of the parametric study on the heat pumps in the City 2, Høje-Taastrup case.

Table 6 shows the calculated values of COP_H from the reference calculation.

Simulation case no.	Description	Input values	Output values COP_H
A	Reference calculation	*See Table 5*	3.55

Table 6: Results from Simulation case A, which is the reference case in the parametric study of City 2, Høje-Taastrup.

The parametric study, Simulation case B, is investigating the effect of the load level of the heat pumps by varying this from 40 % to 100 %. Results of COP_H from this study is presented in Figure 9, and the table with values is included in Table 16 in the Appendix.

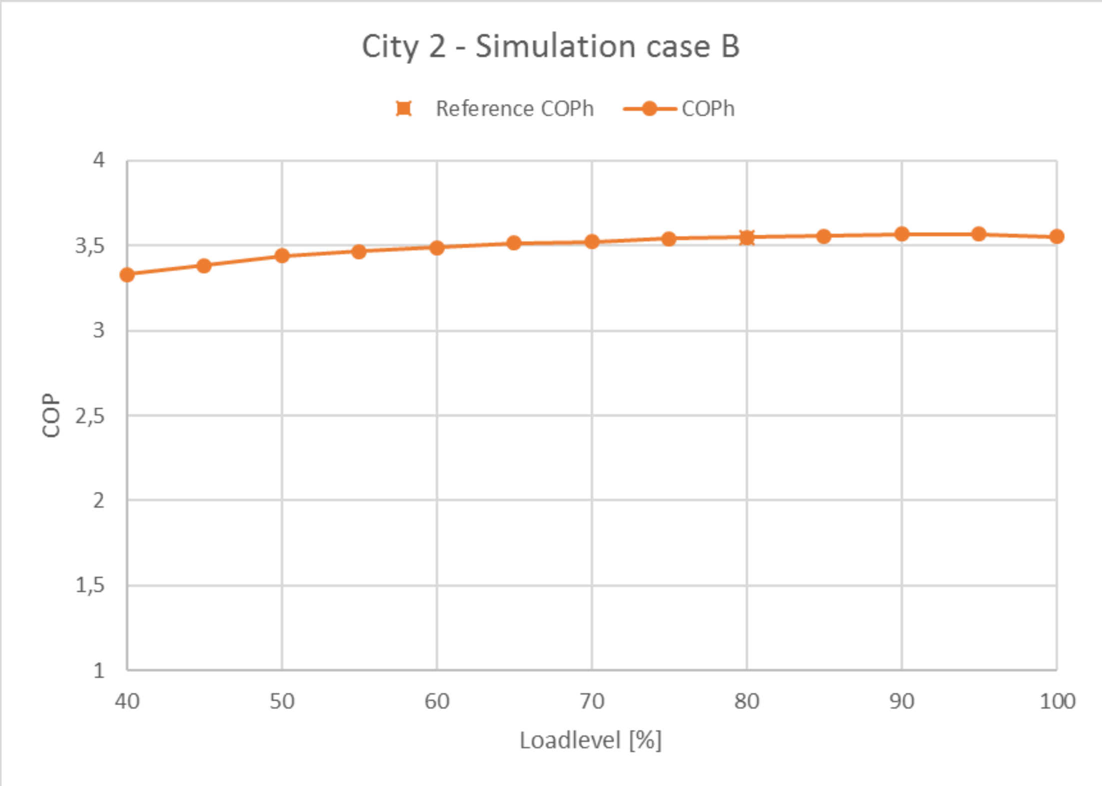


Figure 9: Graphical presentation of the results from Simulation case B showing variation of the load level.

Figure 9 illustrates an optimum load level at approximately 90 % load. At low load levels, the efficiency of the compressors is low which reduces the COP_H, but a lower load also results in lower pump power consumption, which increases the COP_H. However, in this setup the power consumption of the compressor are more dominating than the pump power, and therefore the COP_H decreases at low load levels.

A study of the cooling supply temperature, $T_{cold,Sup}$, and temperature difference between cooling supply and return line, ΔT_{cold}, is investigated in Simulation case C. Results of COP_H from this study is presented in Figure 10, and the table with values is included in Table 17 in the Appendix.

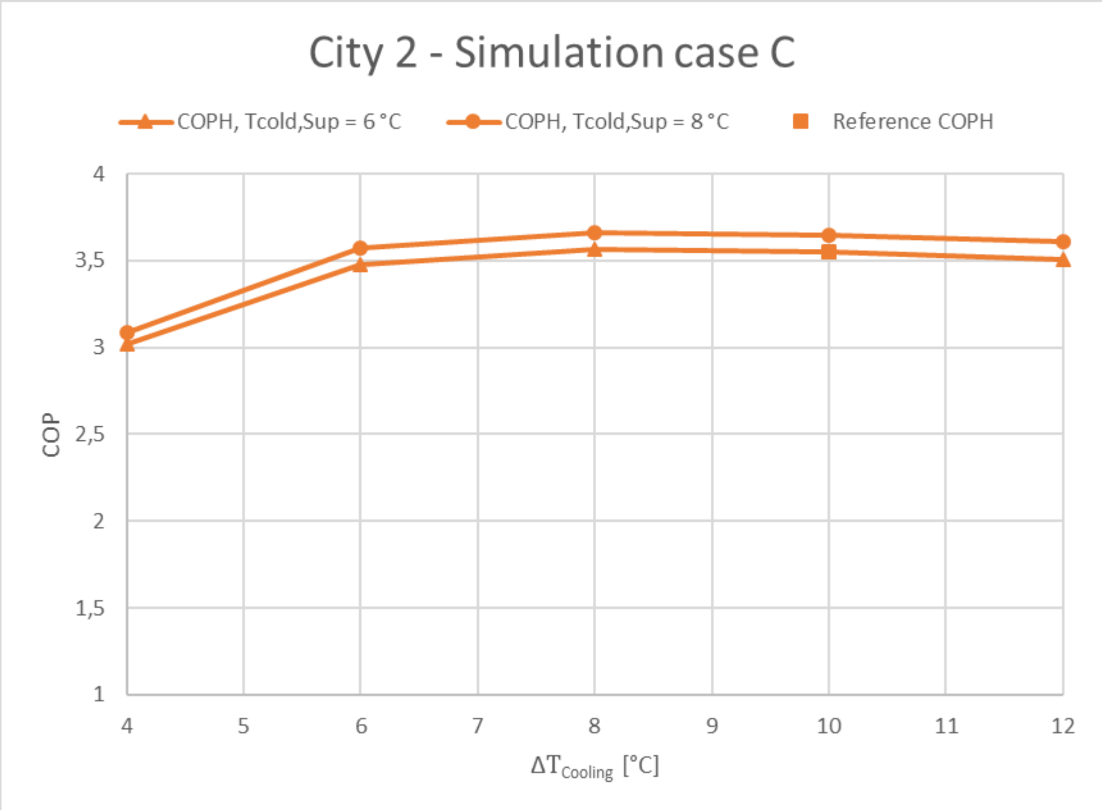


Figure 10: Graphical presentation of the results from Simulation case C showing variation of the cooling temperature set.

Figure 10 shows the calculated COP_H for different cooling temperature sets, and the optimum is found at temperature difference of 8 °C. It can also be extracted that a higher cooling supply temperature is advantageous, as expected.

It is important to remember the coupling between the flow rate in the cooling circuit, m_{cold}, and in the intermediate circuit, m_{Int}, as m_{Int} is fixed to 2 times m_{cold}. When ΔT_{cold} decreases the mass flow rates m_{cold} and m_{Int} increase, and this has an impact on the pump power consumption, which is accounted for when calculating COP_H. The magnitude of this correlation is based on a number of assumptions in this study, and thus only the characteristics of the curves in Figure 10 should be considered.

The relationship between the mass flow rates m_{cold} and m_{Int} is further investigated in Simulation case D, where the ratio between these mass flows, *X*, are varied from 0.5 to 10. As seen in Figure 8, the relationship between the mass flows are defined as: $m_{Int} = X \cdot m_{cold}$. Results of COP_H from this study is presented in Figure 11, and the table with values is included in Table 18 in the Appendix.

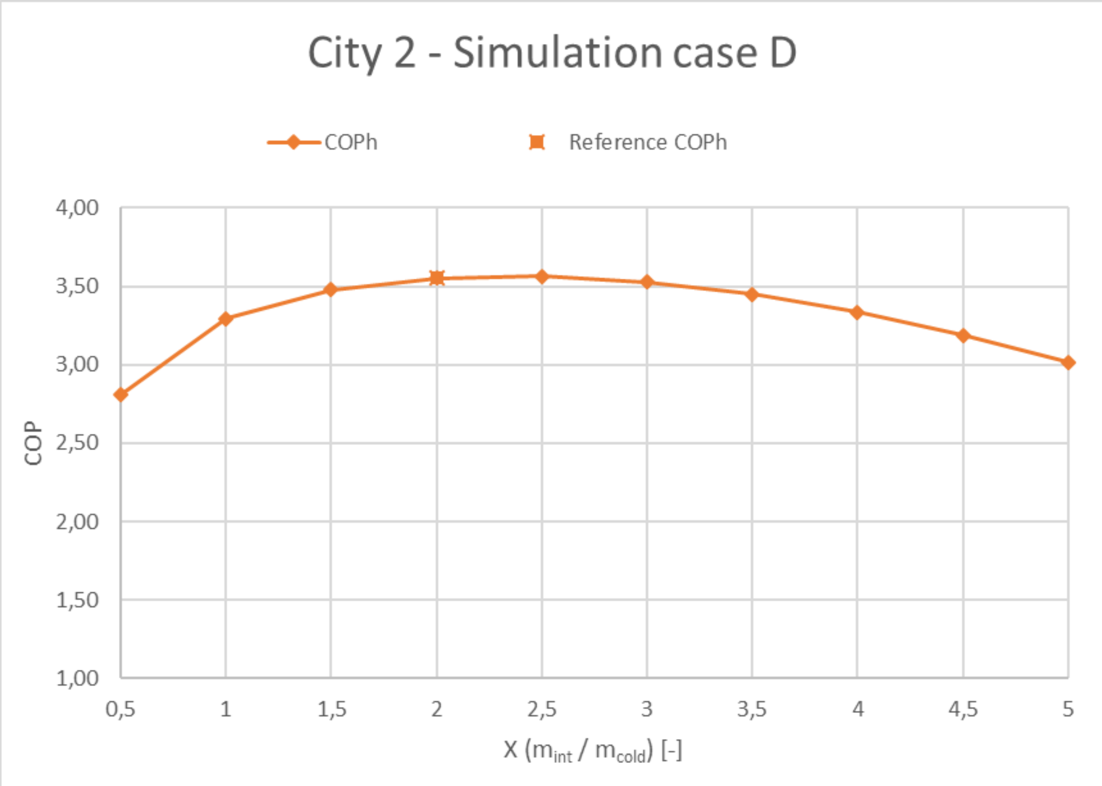


Figure 11: Graphical presentation of the results from Simulation case D showing variation of flow in the intermediate circuit.

Figure 11 shows that the intermediate flow rate has a significant dependency on the calculated COP_H in this modelled system. The optimum is close to the reference case, which is expected as the chosen pump was selected based on the parameters from the reference case. In this system, an intermediate flow lower than 1.5 times the cooling mass flow is not desired, as the COP_H drops significantly at lower *X*-values. A high flow in the intermediate circuit gives a higher COP_H, but at some point the power consumption becomes dominating and the COP_H decreases. The value of *X* at which the decrease in COP_H occur, depends on the pump selected to operate the intermediate circuit. Similarly, the rate of change of COP_H also depends on the design point of the selected pump.

The influence of the intermediate temperature, T_{Int}, at the inlet to the evaporator of the Secondary heat pump is investigated in Simulation case E. This temperature is varied from 25 °C to 35 °C, and results of COP_H from this study is presented in Figure 12, and the table with values is included in Table 19 in the Appendix.

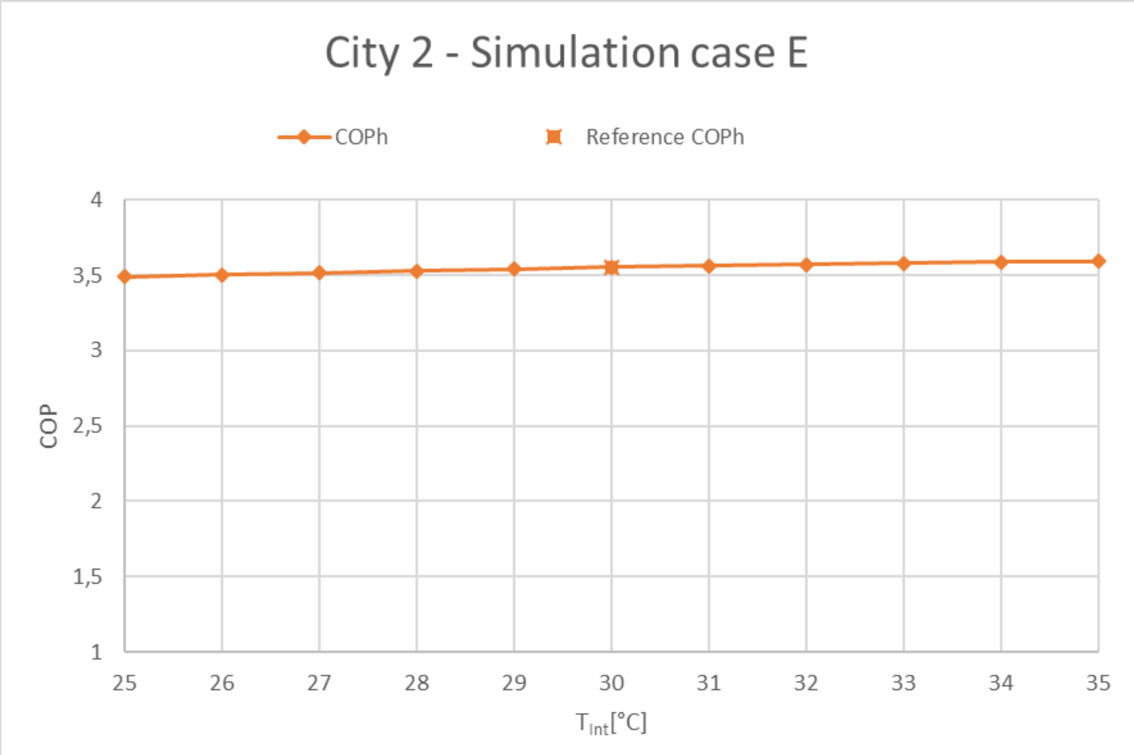


Figure 12: Graphical presentation of the results from Simulation case E showing variation of intermediate temperature.

From Figure 12 it can be seen that COP_H slightly increases with higher intermediate temperatures, but the COP_H is not very sensitive to changes in the intermediate temperature. A high intermediate temperature reduces the temperature difference between the district heating circuit and the intermediate circuit, which increases the COP_H of the Secondary HP. However, this reduces the COP_H of the Primary HP, but as this is smaller than the Secondary HP, the overall COP_H of the system increases slightly.

The district heating supply temperature, $T_{DH,Sup}$, is investigated in Simulation case F, where it is varied from 45 °C to 65 °C. Results of COP_H from this study is presented in Figure 13, and the table with values is included in Table 20 in the Appendix.

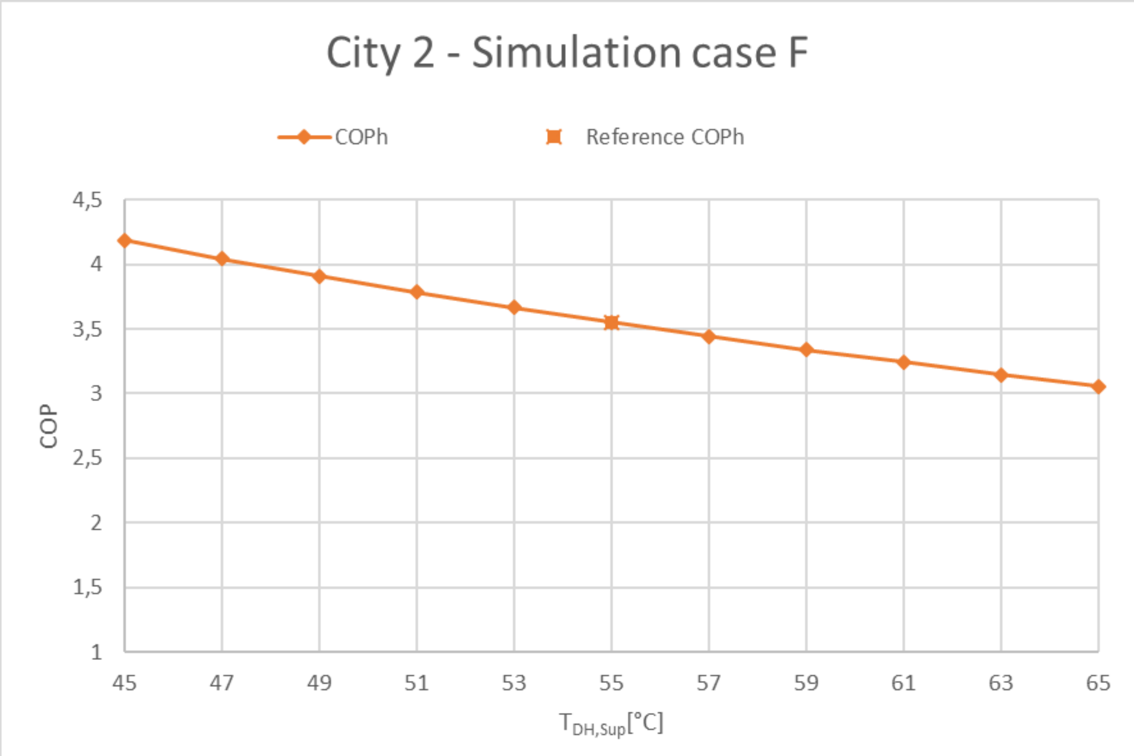


Figure 13: Graphical presentation of the results from Simulation case F showing variation of district heating supply temperature.

Figure 13 shows the relation between COP_H at different district heating supply temperatures. Similar to the MAX IV case, a low district heating supply temperature result in higher COP_H of the system, as the Secondary HP is operating at lower temperature differences. If the district heating supply temperature is decreased from 55 °C to 45 °C, then the COP_H increases by nearly 18 %.

The last parametric study carried out for the HP model of City 2 in Høje-Taastrup is Simulation case G, where different refrigerants are investigated as working fluid in the heat pump. All other parameters are fixed, and thus the working conditions of each refrigerant are not necessarily suitable.

The four refrigerants investigated in Simulation case G in the City 2 model are similar to those in Simulation case E for MAX IV: R134a, R1234ze, R600a (Isobutane) and R717 (Ammonia).

Results of COP_H from this study is presented in Table 7.

Simulation case no.	**Description**	**Input values**	**Refrigerant**	**Output values COP_H**
G	Variation of heat pump refrigerant	*See Table 5*	R134a	3.55
			R1234ze	3.58
			R600a	3.61
			R717	3.61

Table 7: Results from variation of heat pump refrigerant.

The calculated COP_H values shown in Table 7 only ranges from 3.55 to 3.61, which is considered insignificant. The difference between the refrigerants are smaller in this case compared to the calculation for MAX IV. This can be due to the lower district heating temperature in the City 2 case, which result in higher COP values, and thereby the influence of the refrigerant is less significant. As for the MAX IV case study, the natural refrigerant R717 (ammonia) results in the highest calculated COP_H.

Partial conclusion

The heat pump configuration City 2, Høje-Taastrup, has now been analysed and different input parameters were varied in order to study the effect on the COP_H of the heat pump. The main conclusions from each simulation case are summed up in Table 8.

Simulation case no.	Description	Conclusion
A	Reference calculation	-
B	Variation of load level	Optimum load level at approximately 90 % load. Low load levels result in low compressor efficiency but low pump work. The power consumption of the compressor are more dominating than the pump power, and therefore the COP_H decreases at low load levels.
C	Variation of cooling supply temperature and temperature difference between cooling supply and return line	Higher cooling supply temperature results in higher COP_H. The optimum ΔT was found to 8 °C in this study, but is depending on the pump selected.
D	Variation of flow in the intermediate circuit	A high flow in the intermediate circuit gives a higher COP_H, but at some point the power consumption becomes dominating and the COP_H decreases. In this system, an intermediate flow lower than 1,5 times the cooling mass flow is not desired, as the COP_H drops significantly at lower *X*-values.
E	Variation of intermediate temperature	COP_H slightly increases with higher intermediate temperatures, but the COP_H is not very sensitive to changes in the intermediate temperature.
F	Variation of district heating supply line temperature	Low district heating supply temperature result in higher COP_H of the system.
G	Variation of heat pump refrigerant	Refrigerant R717 (ammonia) resulted in the largest COP_H compared to the other refrigerants analysed.

Table 8: Conclusions from the simulation cases of City 2, Høje-Taastrup.

Results Case 3 – Cascade coupled heat pumps supplying district heating

The following section presents a conceptual model of cascade coupled heat pumps supplying district heating. The setup could represent a city with an existing district heating network and facilities nearby, which have a need for cooling or other low-grade heat source. In this model, a range of 10 km is considered for the simulations. The system should preferably be located close to renewable power production sites, such as solar or wind power. This cascade coupling case will be compared to a scenario with only one facility and the entire temperature increment is carried out in one heat pump.

The DH supply temperature is set to 75 °C and the ΔT_{DH} to 45 °C. The volume flow rate is assumed to be 380 m³/h and the U-value[5] 0.4 W/(m*K) respectively, which correspond to a DN300 pre-insulated steel pipe with app. 60 mm insulation (series 1).

Heat loss is included in this model and calculated based on a soil temperature of 5 °C, the U-value and the pipe length. The pipe lengths, heat sink temperatures and system setup are as depicted in Figure 13.

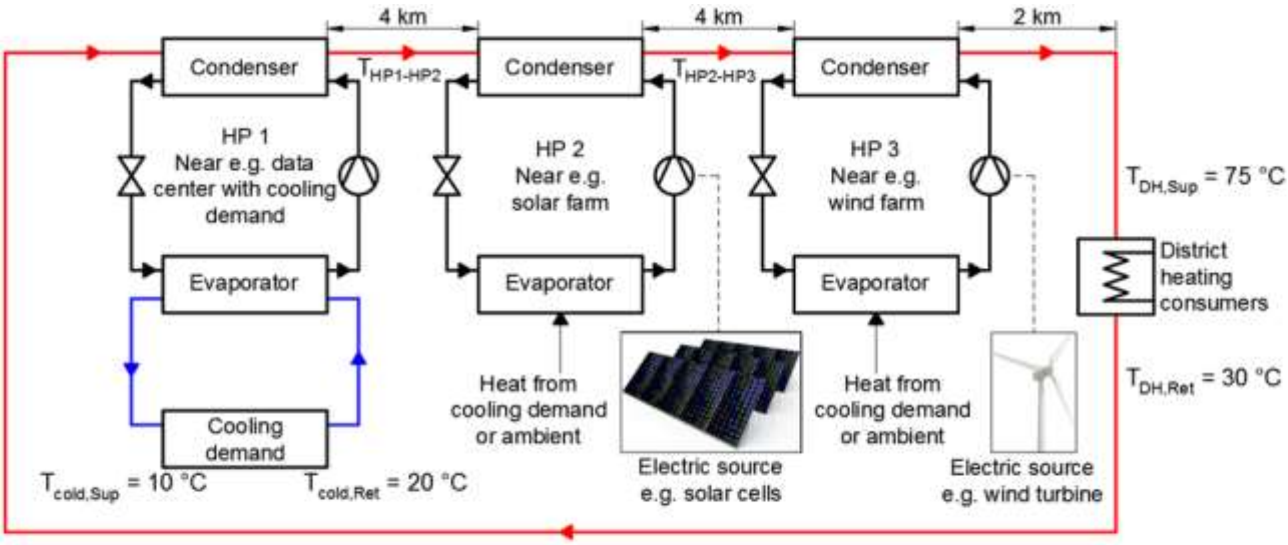


Figure 14: Principal diagram of cascade coupled heat pump system.

As this case is made to study the possible benefits of multiple heat pumps in cascade on a district heating pipe network, a reference case have been made with only HP 1 supplying the district heating network. The inputs are shown in Figure 14, and the heat pump refrigerant is R134a. Results of COP_H from this study is presented in Table 9.

Simulation case no.	Description	Input values	Output values COP_H
A	Reference calculation Single heat pump	*See Figure 14*	2.66

Table 9: Results from Simulation case A, which is the reference case in the parametric study of the cascade coupled heat pump system.

The COP_H is calculated to 2.66 when HP 1 is the only heat pump in the system as seen in Table 9.

[5] Isoplus.dk, Single pipes - design - laying rules

Parametric study

The different simulations in the parametric study for the cascade coupled heat pump system is shown Table 10. The intermediate temperature between the heat pumps are studied, i.e. $T_{HP1\text{-}HP2}$ and $T_{HP2\text{-}HP3}$, including the refrigerant used as working fluid in the heat pumps.

Simulation case no.	**Description**	**Input values**		
		$T_{HP1\text{-}HP2}$ [°C]	**$T_{HP2\text{-}HP3}$ [°C]**	**Refrigerant**
B	Variation of $T_{HP1\text{-}HP2}$	[30 - 60]	60	R134a
C	Variation of $T_{HP2\text{-}HP3}$	45	[45 - 75]	R134a
D	Variation of heat pump refrigerant	45	60	R134a R1234ze R600a R717

Table 10: Overview of the parametric study on the heat pumps in the cascade coupled heat pump system.

Simulation case B studies the intermediate temperature between HP 1 and HP 2, $T_{HP1\text{-}HP2}$, by varying this from 30 °C to 60 °C, while keeping the temperature between HP 2 and HP 3 fixed at 60 °C. The results for COP_H are shown in Figure 15.

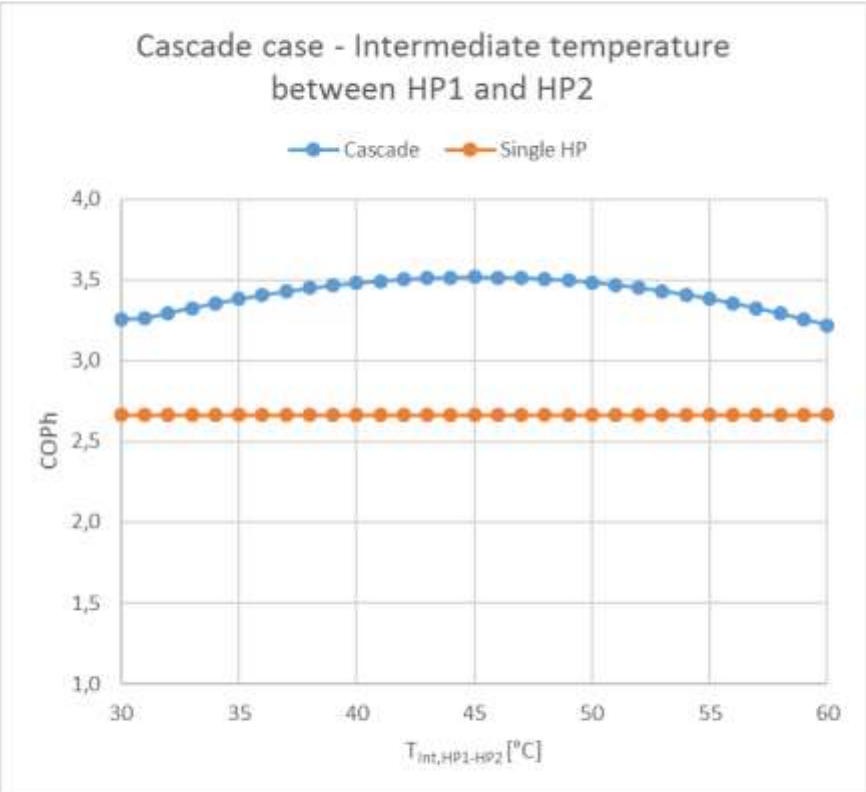


Figure 15: Results from the parametric study of the intermediate temperature between HP1 and HP2, $T_{Int,HP1\text{-}HP2}$. $T_{Int,HP2\text{-}HP3}$ is fixed to 60 °C in this case.

Figure 14 shows that the optimum temperature is at 45 °C and that the difference between the lowest and highest COP_H is around 0.3 for the cascade coupled heat pump system. The highest calculated COP_H is 3.52, which is 0.85 higher than the reference case with only one heat pump.

When $T_{HP1\text{-}HP2}$ = 45 °C, the temperature increase is similar for each of the heat pumps, which results in higher COP_H. When $T_{HP1\text{-}HP2}$ = 30 °C, HP 1 does not influence the system, and thus resembling the case of only two heat pumps operating, i.e. HP 2 and HP 3. The temperature increase across HP 2 is from 30 °C to 60 °C and for HP 3 it is 60 °C to 75 °C, which resulted in a calculated COP_H = 3.26. Similarly, when $T_{HP1\text{-}HP2}$ = 60 °C, HP 1 increases the temperature from 30 °C to 60°C and HP 3 from 60 °C to 75 °C. HP 2 does not increase the temperature of the district heat network in this case, and the calculated COP_H was 3.22.

Simulation case C presents a similar study by varying $T_{Int,HP2\text{-}HP3}$ from 45 °C to 75 °C, and fixing the temperature between HP 1 and HP 2, $T_{Int,HP1\text{-}HP2}$, to 45 °C. The results for COP_H are shown in Figure 16.

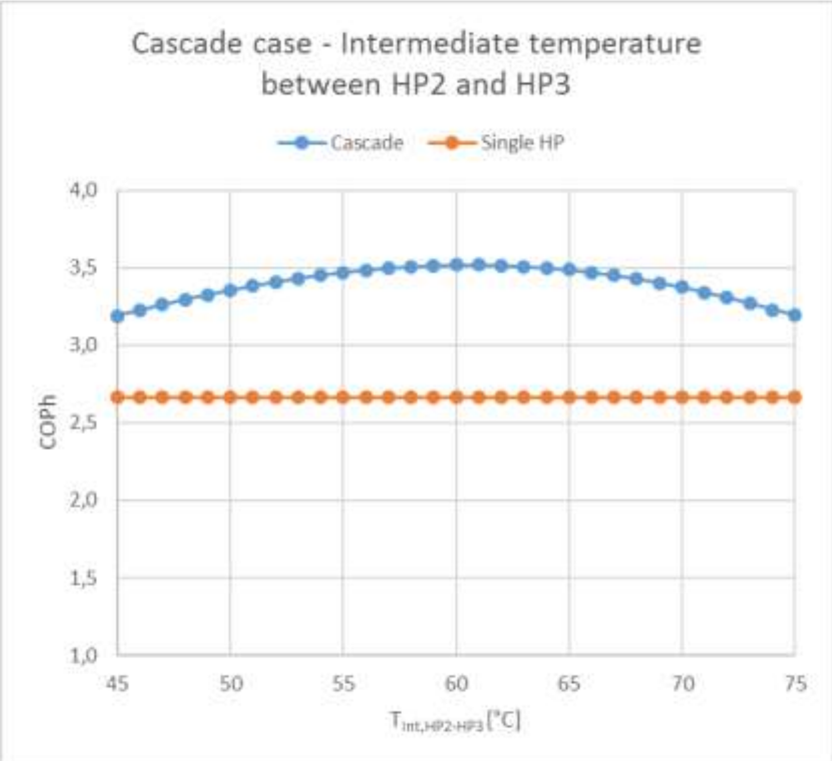


Figure 16. Results from the parametric study of the intermediate temperature between HP2 and HP3, $T_{Int,HP2\text{-}HP3}$. $T_{Int,HP1\text{-}HP2}$ is fixed to 45 °C in this case.

The findings in Figure 16 are similar to those in Figure 14. The highest COP_H (3.52) is achieved when the temperature increase are similar on each heat pump, i.e. when $T_{Int,HP2\text{-}HP3}$ = 60 °C.

The last parameter investigated is the refrigerant in Simulation case D. The four refrigerants investigated in in the cascade coupled heat pump model are similar to those in the MAX IV and the City 2 model: R134a, R1234ze, R600a (Isobutane) and R717 (Ammonia). Results of COP_H are shown in Table 11.

Simulation case no.	**Description**	**Input values**	**Refrigerant**	**Output values COP_H**
D	Variation of heat pump refrigerant	*See Figure 14*	R134a	3.52
			R1234ze	3.51
			R600a	3.64
			R717	3.79

Table 11: Results from variation of heat pump refrigerant.

As for the MAX IV case and the City 2 case, the results in Table 11 shows that R717 (Ammonia) results in the highest calculated COP_H of the refrigerants investigated in this study. The COP_H for R717 is 3.79 which is 0.28 higher than for R134a.

Partial conclusion

The cascade coupled heat pump configuration supplying district heating has now been analysed and different input parameters were varied in order to study the effect on the COP_H of the heat pump. The main conclusions from each simulation case are summed up in Table 12.

Simulation case no.	**Description**	**Conclusion**
A	Reference calculation	-
B	Variation of $T_{HP1-HP2}$	Highest COP_H was found when $T_{HP1-HP2}$ = 45 °C, as the temperature increase in this case is similar for all of the heat pumps.
C	Variation of $T_{HP2-HP3}$	Highest COP_H was found when $T_{HP2-HP3}$ = 60 °C, as the temperature increase in this case is similar for all of the heat pumps.
D	Variation of heat pump refrigerant	Refrigerant R717 (ammonia) resulted in the largest COP_H compared to the other refrigerants analysed.

Table 12: Conclusions from the simulation cases of cascade coupled heat pumps.

Discussion and recommendations

One main conclusion is that there is not one major factor to consider when optimizing heat pump systems. Instead, the way to reach high energy performance is to put some effort into all the various aspects. When combined, these efforts will combine into large potential energy savings.

The main findings in this study was the highest overall COP_H is achieved when similar heat pumps are working under similar conditions. This means that each heat pump in the system should work with the same temperature differences on both the hot and cold side. It was also concluded that lower district heating supply temperatures causes higher COP_H values, and that this parameter has significant influence on the heat pump performances. Furthermore, a theoretical study of the refrigerants used as working fluid in the heat pumps showed that R717 (Ammonia) resulted in the highest COP_H of the refrigerants investigated.

The simulations described in this report are based on real life systems, with a number of assumptions and simplifications to reduce the complexity of the calculations. For example, the compressor efficiency was fixed in the model and heat losses in the pipe layout was neglected – except for the purely theoretical case with cascade-coupled heat pumps. Overall, the assumptions made in the heat pump models are omitting losses and thereby increasing the efficiency of the system and result in higher COP. However, the models are still considered close to real life operation, and thereby the relationships between COP and system inputs varied in the parametric analyses can be transferred from the theoretical study to real life situations.

Appendix

Result tables

Case 1 – MAX IV.

Simulation case no.	Description	Input					Output
		φ [-]	T_{Int} [°C]	$T_{DH,Sup}$ [°C]	ΔT_{DH} [°C]	Refrigerant	COP_H [-]
B	Variation of cold water flow distribution (φ)	0.00	11	80	30	R134a	2.76
		0.25					2.89
		0.50					2.95
		0.75					2.92
		1.00					2.75

Table 13: Results from Simulation case B showing variation of the cold water flow distribution, ϕ.

Simulation case no.	Description	Input					Output
		φ [-]	T_{Int} [°C]	$T_{DH,Sup}$ [°C]	ΔT_{DH} [°C]	Refrigerant	COP_H [-]
C	Variation of intermediate temperature on cold water side (T_{Int})	0.5	8	80	30	R134a	2.84
			9				2.89
			10				2.93
			11				2.95
			12				2.95
			13				2.93
			14				2.88

Table 14: Results from Simulation case C showing variation of the intermediate temperature on cold water side, T_{Int}.

Simulation case no.	Description	Input φ [-]	T_{Int} [°C]	$T_{DH,Sup}$ [°C]	ΔT_{DH} [°C]	Refrigerant	Output COP_H [-]
D	Variation of district heating (DH) supply line temperature ($T_{DH,Sup}$) and temperature difference between district heating supply and return line (ΔT_{DH})	0,5	11	50	20	R134a	5.16
				55			4.57
				60			4.08
				65			3.68
				70			3.33
				75			3.04
				80			2.77
				50	30		5.70
				55			4.99
				60			4.42
				65			3.96
				70			3.57
				75			3.24
				80			2.95
				50	40		6.37
				55			5.49
				60			4.82
				65			4.28
				70			3.84
				75			3.47
				80			3.14

Table 15: Results from Simulation case D showing variation of the district heating supply temperature, $T_{DH,Sup}$, and the temperature difference between district heating supply and return line, ΔT_{DH}.

Case 2 – City 2, Høje-Taastrup: A theoretical example of how heat pumps can be used for cooling at the shopping mall City 2 in Høje-Taastrup.

Simulation case	Description	Input values	Output values
		Load level [%]	COP_H
B	Variation of load level	40	3.33
		45	3.38
		50	3.44
		55	3.47
		60	3.49
		65	3.51
		70	3.52
		75	3.54
		80	3.55
		85	3.56
		90	3.57
		95	3.57
		100	3.55

Table 16: Results from Simulation case B showing variation of the cooling load level.

Simulation case	Description	Input values		Output values
		$T_{cold,Sup}$ [°C]	ΔT_{cold} [°C]	COP_H
C	Variation of cooling supply temperature and temperature difference between cooling supply and return line	6	4	3.02
			6	3.48
			8	3.56
			10	3.55
			12	3.51
		8	4	3.08
			6	3.57
			8	3.66
			10	3.65
			12	3.61

Table 17: Results from Simulation case C showing variation of the cooling supply temperature and the cooling temperature difference.

Simulation case	Description	Input values m_{Int} [kg/s]	Output values COP_H
D	Variation of flow in the intermediate circuit	$0.5 \cdot m_{cold}$	2.81
		$1.0 \cdot m_{cold}$	3.29
		$1.5 \cdot m_{cold}$	3.48
		$2.0 \cdot m_{cold}$	3.55
		$2.5 \cdot m_{cold}$	3.56
		$3.0 \cdot m_{cold}$	3.53
		$3.5 \cdot m_{cold}$	3.45
		$4.0 \cdot m_{cold}$	3.33
		$4.5 \cdot m_{cold}$	3.19
		$5.0 \cdot m_{cold}$	3.02

Table 18: Results from Simulation case D showing variation of flow in the intermediate circuit.

Simulation case	Description	Input values T_{Int} [°C]	Output values COP_H
E	Variation of intermediate temperature	25	3.49
		26	3.50
		27	3.52
		28	3.53
		29	3.54
		30	3.55
		31	3.56
		32	3.57
		33	3.58
		34	3.59
		35	3.59

Table 19: Results from Simulation case E showing variation of intermediate temperature.

Simulation case	Description	Input values	Output values
		$T_{DH,Sup}$ [°C]	COP_H
F	Variation of district heating supply line temperature	45	4.18
		47	4.04
		49	3.91
		51	3.78
		53	3.66
		55	3.55
		57	3.44
		59	3.34
		61	3.24
		63	3.15
		65	3.06

Table 20: Results from Simulation case F showing variation of intermediate temperature.

Isentropic efficiency for different types of compressors

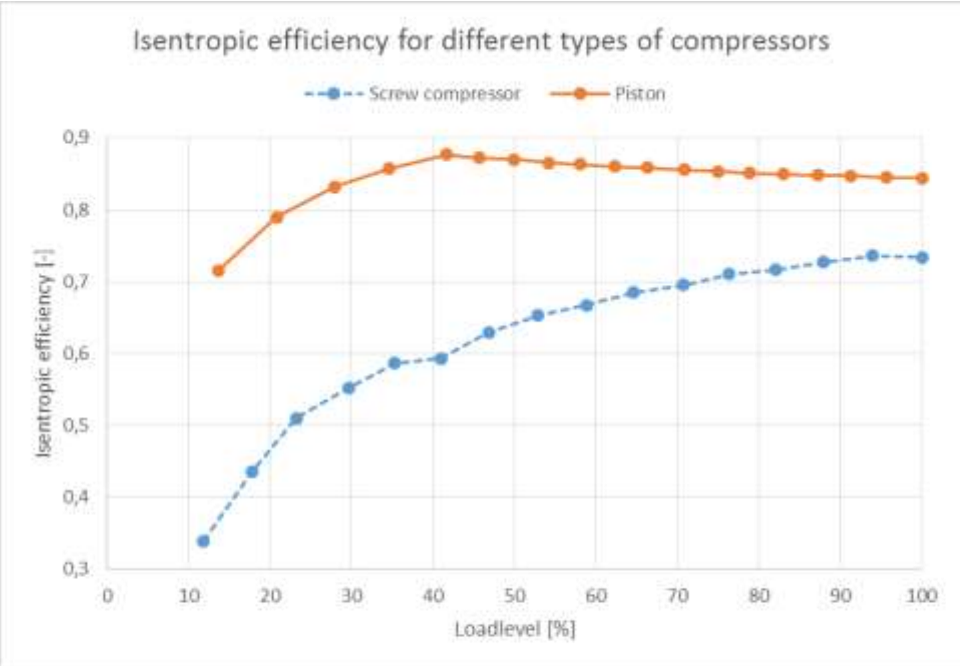


Figure 17: Isentropic efficiency as a function of load level for piston and screw compressor[6].

[6] Sacco, Andrea (2018), *Part Load Behavior of a Large-Scale Ammonia Heat Pump supplying District Heating network. http://tesi.cab.unipd.it/59695/*