

A NOVEL HEAT PUMP CYCLE DESIGN FOR VERY COLD WEATHER DISTRICT HEATING



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TABLE OF CONTENTS

- 1 EXECUTIVE SUMMARY**
- 2 DISTRICT HEATING IN EUROPE**
EU HEAT PUMP STUDIES
- 3 HEAT PUMP ANALYSIS**
REFRIGERANT SECTION
HEAT PUMP ADVANCED CYCLE CONFIGURATIONS
COMPRESSOR TYPES AND PERFORMANCES
- 4 RESULTS AND DISCUSSIONS**
CYCLE CONFIGURATION COMPARISON
- 5 CONCLUSIONS**
- 6 REFERENCES**

EXECUTIVE SUMMARY

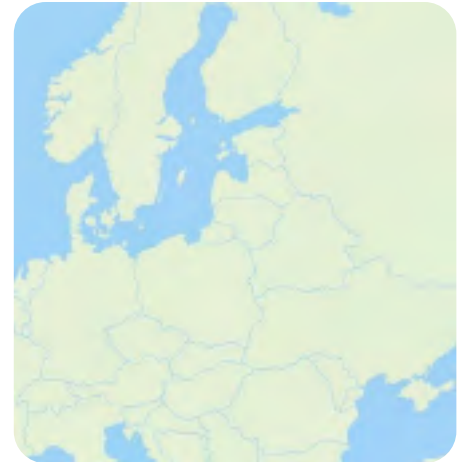
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District heating (DH) provides around 12% of the residential and commercial heat in the EU, with about 70% of it being generated from fossil fuels. It is crucial to prioritize the replacement of fossil fuel-fired boilers in district heating as part of the decarbonization process. There is a consensus within the energy sector that integrating emission-neutral electricity with heat pumps can significantly contribute to decarbonizing Europe.

Existing district heating networks have several thousand kilometers of pipes installed, and the supply water temperature can be as high as 115°C in Nordic countries and in Central Europe (e.g., Poland).

The analysis of thermodynamic cycles for heat pumps indicates the potential for developing heat pumps capable of replacing boilers in very high-temperature heat, offering economic and ecological benefits. A partial cascade system with deep subcooling has been identified as the best configuration due to its high efficiency, low cost and component availability.

This paper analyzes the impact of cycle architecture, especially the ones with deep subcooling, and the choice of refrigerant on the investment and operational costs of high-temperature district heating heat pumps designed for year-round operation with cold to very cold winters and mild summers, such as Nordic countries and in Central Europe.



The supply water temperature can be as high as 115°C in Nordic countries and in Central Europe.



DISTRICT HEATING IN EUROPE

2

District heating networks will play a crucial role in Europe's now legally binding mission to reach Net Zero by 2050, and on delivering the REPower EU objective to reduce dependency on Russian fossil fuels. Decarbonizing heating is a high priority – but a complex one.

EU HEAT PUMP STUDIES

Decarbonization policy in Europe encourages replacing gas boilers with heat pumps [1]. According to Connolly et al. [2], existing urban district heating capacity for heating and domestic hot water totals more than 250 TWh/year. Connolly et al. [2] concluded that a considerable reduction in fossil fuel consumption and carbon dioxide emission can be achieved if the district heating networks are widely used in European cities.

David et al. [3] show that heat pumps are already part of district heating sources in several European countries. For example, significant installed capacities have existed in Sweden since the 1980's with most of the installed capacities delivering hot water at temperatures up to 90°C. They mention the temperature lifts and COPs of existing capacities. The maximum temperature lift is 80°C with COP close to 3.

Arpagaus et al. [4] reviewed the current state-of-the-art high-temperature heat pumps (HTHPs) with heat sink temperatures in the range of 90 to 160°C. More than 20 HTHPs from 13 manufacturers have been identified on the market that can provide heat sink temperatures of at least 90°C.

Bergamini et al. [5] compared the thermodynamic performance of various HTHPs based on a vapor compression cycle using water, ammonia, propane and isobutane with natural gas boilers to assess their technical feasibility and competitiveness as future replacements. First-law and second-law thermodynamic analyses were performed for single-stage, two-stage, and cascade architectures. They concluded that HTHPs can competitively replace gas boilers in delivering heat of up to 180°C with temperature lifts of up to 160°C, and two-stage cycles were able to achieve the highest efficiencies.

Robinson et al. [6] discussed the design and optimization of turbo compressors using computational fluid dynamic and finite element mechanical analysis techniques. They show that compressor polytropic efficiency values higher than 80% can be achieved by optimizing the design for the refrigerant and the operating conditions.

Decarbonization policy in Europe encourages replacing gas boilers with heat pumps.

Jaatinen-ärri et al. [7] combined the analyses of a 500 kW two-stage heat pump cycle and centrifugal compressor designs. The study qualitatively assesses the impact of refrigerant (hydrocarbons and hydrofluoroolefins) and operating conditions (temperature lift and evaporation temperature) on the compressor dimensions and rotational speed, and its isentropic efficiency. The study concluded that high-temperature heat pump equipment could be feasible for heat pump temperature lifts of around 100°C. The study shows that the R1233zd(E) compressor operates at a lower rotational speed and mach number with a smaller impeller diameter than the pentane compressor. This simplifies the R1233zd(E) compressor design and reduces its overall size. R1233zd(E) and isopentane compressors can achieve isentropic efficiencies between 78 to 83%. The isentropic efficiency depends on the temperature lift.

While high-temperature heat pump cycles have been studied by previous researchers, district heating systems demand high flexibility of operation in areas that have very cold winters. Figure 1 shows the bin temperature data for Helsinki showing that 4% of the occurred hours are at temperatures lower than -10°C. During such periods, there is increased heat demand and thermal losses in the piping network, the supply temperature must be raised to up to 115°C. A higher supply temperature doesn't have a significant impact on the heat generation efficiency of both gas and electrical boilers.

Reducing the circulating water flow rate in a district heating pipes network allows the use of smaller pipe diameters and delivers the required heating capacity to more end users.

However, the need for an increase in heat generation coupled with an increase in supply temperature complicates the design of heat pump systems. The most basic setup would involve a heat pump to consistently generate hot water up to 90°C and, as needed, operate a booster gas or electric boiler to further heat the water to the desired temperature.

There are a few practical challenges with this architecture. Firstly, using both heat pumps and boilers increases the capital investment and maintenance cost. Secondly, the higher heating demand and hot water temperature needs during low ambient induces in some cases a peak demand on electricity networks when the electricity is most expensive. Similarly, gas or electricity prices are also high during extreme winter.

It is technically challenging to economically achieve a water temperature rise of up to 80°C and a hot water supply temperature of up to 115°C with a standard single-stage heat pump compared to the combination of a heat pump and an electrical boiler. The same heat pump must be capable of delivering hot water at 77°C (Figure 1) in summers with a temperature lift of less than 30°C (Figure 2).

Mateu-Royo et al. [8] analyzed several advanced high-temperature heat pump configurations with several low GWP refrigerants for industrial waste heat recovery. Configurations like single-stage cycle with economizer, single-stage with ejector, and two-stage cycle with either an economizer or a flash tank perform less when applied to the discussed district heating network (Figure 2).

The heat pump cycle needs to adjust for variations in the demanded water temperature and be able to deliver heat at variable temperature lifts. Configurations with an economizer, a flash tank, or an ejector are less beneficial when there is a possibility to achieve high subcooling by exchanging heat with the district heating water return flow.

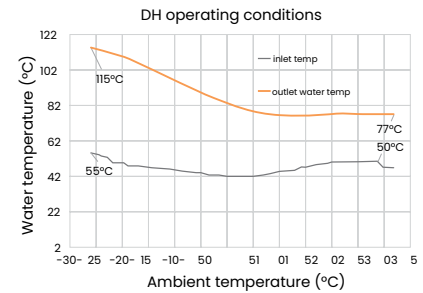


Figure 1. Temperature occurrence by hours and required water temperature rise as a function of the ambient temperature

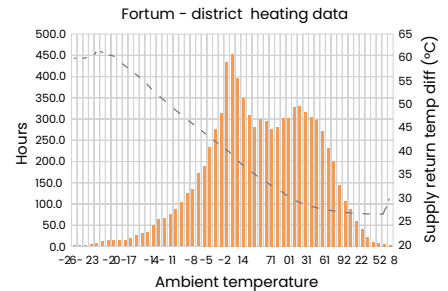


Figure 2. Delivered water and average return water temperatures from the district heating network

HEAT PUMP ANALYSIS

3

The selection of heat pump refrigerant and cycle architecture plays a significant role for the investment and operational costs of high-temperature district heating heat pumps operating year-round in regions with cold to very cold winters and mild summers.

REFRIGERANT SELECTION

Table 1 lists the refrigerants selected for this assessment. These refrigerants are employed within district heating infrastructures.

REFRIGERANT	NORMAL BOILING POINT (°C)	CRITICAL TEMPERATURE (°C)	CRITICAL PRESSURE (MPa)	ASHRAE CLASSIFICATION
Ammonia	-33.3	132.2	11.3	B2L
R1234ze(E)	-18.9	109.3	3.6	A2L
Isobutane	-11.7	134.6	3.6	A3
Butane	-0.5	151.9	3.8	A3
R1233zd(E)	18.3	165.6	3.6	A1

Table 1: Properties of selected fluids

HEAT PUMP ADVANCED CYCLE CONFIGURATIONS

For district heating heat pump systems operating in extremely cold climates, cycle configuration needs to be adjusted according to the district heating demands. For this reason, the following cycle configurations are selected to investigate their potential for this application (Figures 3 to 7).

Configuration A

The reference design (Figure 3) is a single-stage cycle heat pump with very high (deep) subcooling and an electrical boiler. During moderate winters and summers, the heat pump can meet the heating demand. However, in cold ambient periods, the heat pump cannot deliver hot water at the required supply temperature.

The electrical boiler is used to heat the water to the supply temperature of up to 115°C. Table 2 summarizes the maximum supply temperatures for each working fluid.

FLUID	R1234ZE	NH3	ISOBUTANE	BUTANE	R1233ZD(E)
Critical T (°C)	109	132	134	151.9	165.6
Maximum water supply temperature (°C)	89	92	115	115	115

Table 2: Properties of selected fluids and maximum water supply for single-stage cycle

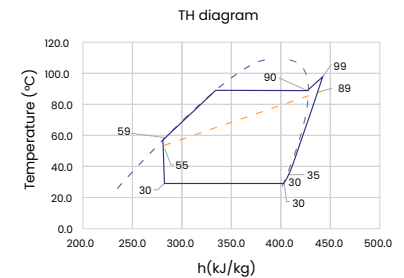


Figure 3. Single-stage cycle with deep subcooling

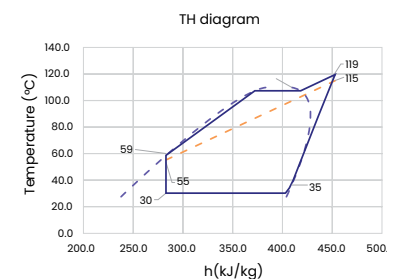


Figure 4. Supercritical cycle at high temperature with deep subcooling

Configuration B

Figure 4 represents a single-stage transcritical cycle. This configuration allows the refrigerant pressure at the compressor outlet to exceed its critical pressure and provide high water temperature when required. Deep subcooling is also used at subcritical conditions to increase COP. For this configuration, the same single-stage unit can cover the complete range of supply temperature, and the electrical boiler can be eliminated. The selected fluids should have a critical temperature lower than 115°C to achieve supercritical conditions. If the fluid critical temperature is higher than 115°C, the system will operate at subcritical conditions.

Configuration C

Figure 5 represents a two-stage cycle with a flash tank: This configuration allows cycles with higher pressure ratios and is an improvement over the single-stage cycle. After the first expansion, a flash tank allows vapor phase recovery and injection between the two compressors. The remaining subcooled liquid in the flash tank is expanded to the evaporator. This configuration can deliver hot water at high temperatures depending on the selected refrigerant. The electrical boiler is used, if needed, to increase water temperature up to 115°C during very cold ambient temperatures.

Configuration D

Figure 6 represents a cascade cycle with deep subcooling: The purpose of this cycle is to cover the production of water at very high temperatures by using two different fluids. The first stage will achieve a maximum temperature increase up to 90°C and the second stage will deliver on-demand hot water up to 115°C for DH.

Water is used for both stages to subcool the refrigerant at the condenser's outlets. Suction superheat in the high-stage cycle is achieved by using either an internal heat exchanger within the high-stage cycle or by exchanging heat with the hot discharge vapor of the low-stage cycle.

Configuration E

Figure 7 represents a partial cascade cycle with deep subcooling: This configuration uses two cascaded loops, but the high-stage system only partially recovers heat from the low-stage loop. The cascade condenser only partially condenses the refrigerant of the low-stage cycle, and the condensation is completed by heating the DH water. After condensation, the refrigerant is further subcooled by preheating the DH water return flow. The DH water flow is further heated by the high-stage loop. Deep subcooling can be achieved in the high-stage cycle. Like configuration D, an additional suction line heat exchanger is provided in the high-stage cycle.

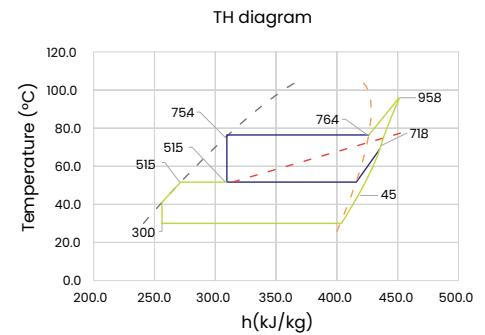


Figure 5. Two-stage cycle

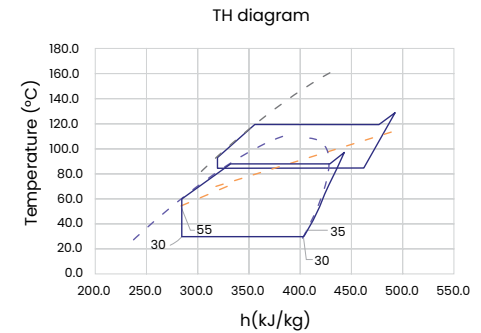


Figure 6. Cascade cycle with deep subcooling

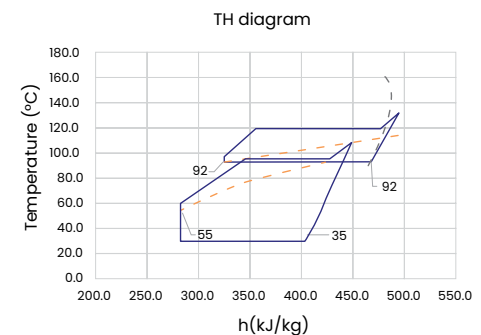


Figure 7. Partial cascade cycle with deep subcooling

Modelling Details

The simulation results of the advanced cycle configurations (Figures 3 to 7) are based on thermodynamic calculations with COP optimization. The evaporator operating conditions are fixed for all the cycles. The condensing temperature is calculated as a function of the water inlet/outlet temperature, cycle limitations, fixed temperature pinch through heat exchangers, and COP optimization.

The heat exchanger approach (HEA) is the minimum temperature difference between the refrigerant flow and water flow through the heat exchanger. The HEA is assumed equal to 4°C for all heat exchangers and all configurations.

At the heat source level, evaporator superheat is equal to 5°C, and condenser subcooling is equal to 1°C for all configurations. Configurations D and E use heat exchangers between first-stage discharge and second-stage suction to superheat the second-stage fluid and prevent wet compression and compressor damage.

In a single-stage cycle with deep subcooling, configuration A (Figure 3),

initially, the saturated vapor condensing temperature value is assumed to be higher than the water outlet temperature. For each condensing saturated vapor temperature, iterations on compressor capacity are done to match the fixed pinch point temperature difference between the condensing saturated vapor temperature and water temperature. Multiple iterations of the condensing temperature deliver the target water exit temperature to the DH network.

In the supercritical cycle at high temperature and deep subcooling, configuration B (Figure 4), the minimum condenser pinch temperature is fixed as input. Iteration was done on condenser/cooler pressure to get the right pinch value between the two temperature profiles.

In a two-stage cycle with a flash tank, configuration C (Figure 5), the mid-pressure is calculated as a function of the evaporating and condensing pressures. The condensing temperature is used for pinch minimization between the refrigerant and water flows. The pinch loop calculations are repeated up to calculation convergence.

In the cascade cycle with deep subcooling, configuration D (Figure 6), The low-stage condensation temperature is used for COP optimization. For each low-stage condensation temperature, iteration on the cooling capacity needed for the high-stage converges when the outlet water temperature matches the fixed outlet water demand. For each low-stage condensing temperature and high-stage cooling capacity, iterations are done on the low-stage compressor displacement to match the heating capacity to the high-stage demand.

In partial cascade cycle with deep subcooling, configuration E (Figure 7), The low-stage condensation temperature is used for COP optimization. For each condensation temperature, iteration on the heating capacity needed for the second stage converges when the outlet water temperature matches the fixed outlet water demand. The lowest-level iterations are on the first stage compressor capacity used for temperature pinch matching through the condenser.

COMPRESSOR TYPES AND PERFORMANCES

Compressors have a major impact on heat pump performance. Not only nominal or design heat pump operating conditions should be considered but also the whole annual operating conditions function of the outside ambient temperature. In this study, the pressure ratio and compressor rotating speed are reviewed for different compressor types to understand their impact on the compressor and heat pump's annual efficiency.

According to each cycle configuration, the pressure ratio across the compressors will vary between very low to very high-pressure ratios as the ambient temperature varies. Heating demand is also a function of the outside temperature (e.g., during summer temperatures) when heat demand is lowest, and compressors need to either frequently start and stop or run at reduced speed. The water temperature rise is a function of the ambient temperature in the district heating systems. Figure 1 shows that the required water temperature rise increases from a low of 25°C in the summer to as high as 60°C on the coldest winter days. Consequently, good compressor efficiency over a large range of pressure ratios and heating power demands is highly desired.

While there is an interest in using centrifugal compressors for heat pump applications, they may not be suitable when very high-pressure lift, such as in supercritical configurations. Also, published performance data of centrifugal compressors are scarce, this paper will focus on piston and screw compressors only.

Compressors have a major impact on heat pump performance.

Compressor Models

For compressor modelling, the focus is on real compressor power consumption versus isentropic compression. The compressor efficiency is defined as the ratio of isentropic enthalpy difference between discharge and suction over the real compressor electricity consumption. The real compressor electricity consumption is assumed to be equal to the product of the real enthalpy difference between discharge and suction and the mass flow rate.

To calculate compressor efficiency, the manufacturer's polynomial compressor equations are used inside an iterative routine function of the compressor efficiency. Calculated efficiency is plotted as a function of the pressure ratio for different compressor speeds.

Figures 8 and 9 show respectively piston and screw compressor performance curves using R-1234ze(E) refrigerant with an operating envelope suitable for heat pumps up to 90°C.

For the specific screw compressor (Figure 9), reducing compressor speed decreases the compressor efficiency. The highest efficiency is achieved at a pressure ratio of around 2.5 for displacement between 350–175 m³/h. Speed reduction from 700 m³/h to 175 m³/h reduces the maximum efficiency value by 40%.

The compressor efficiency is also impacted by the pressure ratio between the discharge and the suction pressure (Figures 8 and 9)

Compressor Efficiency Equation

For the cycle configurations used in this study, we selected a piston compressor efficiency trend curve function of pressure ratio. This curve will be used with all the configurations and all the refrigerants. This scenario is used because there is no available compressor (in manufacturer compressor selection software) suitable for the selected cycle configurations and refrigerants. Also, the impact of speed variation is excluded when executing configuration comparison for the same reason.

Compressor volumetric efficiency is assumed to be equal to the ratio of mass flow rate over the product of compressor displacement and the suction density. Compressor displacement and mass flow rate are calculated using compressor manufacturer data. Suction density is calculated as a function of the operating conditions. Calculated volumetric efficiency values are fitted with a linear equation function of the pressure ratio.

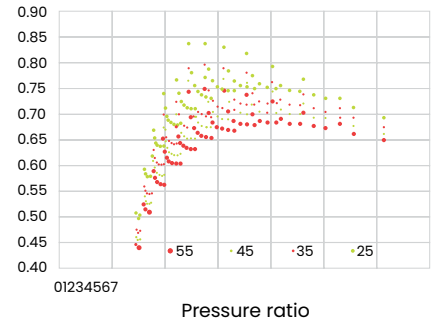


Figure 8. R1234ze(E) piston compressor efficiency plotted function of pressure ratio and compressor rotating frequency

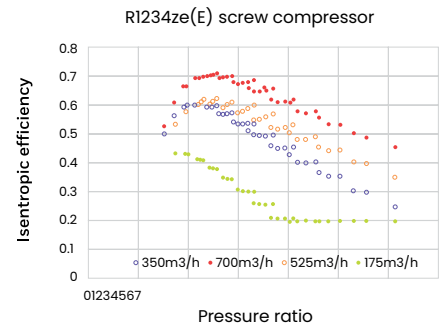


Figure 9. R1234ze(E) screw compressor efficiency plotted function of pressure ratio and compressor displacement

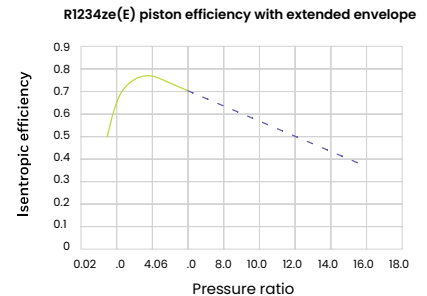


Figure 10. Piston efficiency at a fixed speed

RESULTS AND DISCUSSION

4

Utilizing data provided by the heat pump cycle configurations and compressor efficiency studies, it is now possible to compare different cycle configurations using various working fluids.

CYCLE CONFIGURATION COMPARISON

The task of this paper is to compare different cycle configurations using various working fluids. The annual performances and maximum compressor displacements are considered as main indicators. Each model uses the DH profile shown in Figures 1 and 2 as input data. The annual performance is calculated as the ratio of the heat delivered over the electricity consumption (SCOP).

For heat source, this paper considers only the case of heat recovery from a data center or any other heat source between 30 to 40°C, such as condensers of chillers and industrial waste heat. The HP should deliver 550 kW at -26°C outside ambient temperature and the DH water flow is assumed constant through the year.

SCOP Configuration Performance Evaluation

First, to determine the best configuration for our DH case study, we perform the analysis using the same refrigerants. The SCOP (Figure 11) is the ratio of heat delivered to the DH over the total electricity consumption including the electrical heater, when needed.

The reference configuration with deep subcooling is penalized at very low ambient because the electrical heater is needed to deliver hot water at 115°C.

The two-stage configuration has the lowest SCOP because the mid-stage vapor-liquid separator doesn't allow deep subcooling using DH water return flow. Consequently, the SCOP is penalized.

Electrical Power Peak Demand

Electrical power peak demand is an important parameter to be considered when selecting a heating system configuration. From SCOP analyses, partial cascade or supercritical configurations can reduce annual electricity consumption by only 7% (Figure 12). The analysis of power demand reveals that although the reference configuration has a high SCOP, it has a significantly high electricity demand during very cold weather conditions, which occurs for a few hundred hours annually. This leads to a very high-power demand, peaking at up to 200%. Therefore, using advanced high-efficiency configurations can help reduce peak electricity demand during times of high electricity prices. Additionally, the reduced peak load has a positive impact on the electrical grid infrastructure and installation cost.

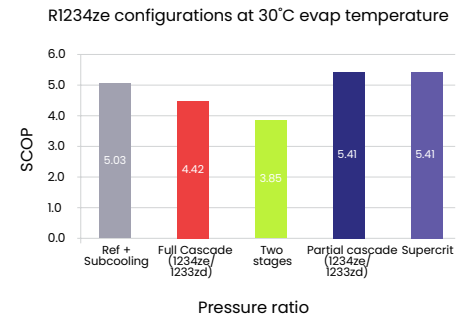


Figure 11. SCOP of DH configuration using R1234ze(E) or R1234ze(E) and R1233zd(E) as working fluids

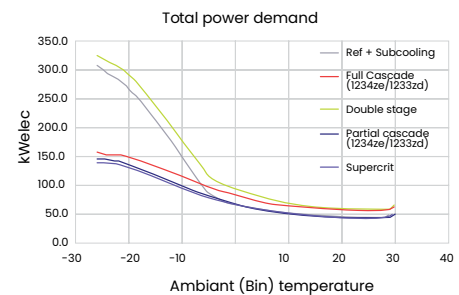


Figure 12. Electrical power demand

Components and Technologies

Predesign thermodynamic cycle analysis gives the potential advantages and disadvantages for each configuration. Advanced configurations need additional components like compressors, heat exchangers, valves and special control systems. This can have an important impact on the final heat pump cost. Some configurations are infeasible either because of technical constraints or the unavailability of components. For example, the supercritical configuration operates at very high condensing pressures and pressure ratios, which cannot be supported by current compressors and heat exchangers.

Configuration Performance Evaluation With Different Fluids

Different fluids are used for configuration evaluation according to the selected heat source scenario. Only annual performances and compressor displacements are compared (Figure 13).

For the specific DH case study analyzed in this paper, the SCOP values show the same trend for all the working fluids. Two-stage and cascade configurations are the least suitable options for this application. SCOPs of reference configuration with deep subcooling are either lower or equal to supercritical configuration SCOPs. These SCOPs are impacted by the fluid's critical temperature and pressure ratio. The supercritical and partial cascade configurations show equivalent results.

In the context of supercritical configurations, the choice of working fluid has minimal influence on the calculated SCOP. However, in situations requiring elevated water temperatures, the utilization of ammonia leads to a notable increase in both discharge pressure and temperature. Specifically, in a supercritical cycle, ammonia necessitates a compressor discharge pressure of 63 bar and a temperature of 209°C. Also, ammonia is corrosive to the usual construction material used with other refrigerant components. These conditions render selecting compressors and components notably demanding and more costly.

The size or displacement of the compressor is an important factor in determining the total heat pump cost. Compressor costs are higher for refrigerants with lower volumetric capacity. Refrigerants with lower volumetric capacity also require larger components, which increases heat pump costs. Figure 14 illustrates the comparison of compressor displacement for various fluids and configurations.

Impact of the Selected Fluids on the Investment and Operational Cost

Mateu-Royo et al. [08] correlated HP component costs to the required compressor displacement and heating capacity. 20% additional cost is mentioned for cascade configuration versus single-stage configuration. As the cost estimations are approximated, we consider that the heat pump's total cost is proportional to the required compressor displacement when using equivalent technologies.

The cost of risk mitigation due to increased flammability or toxicity of the fluid should be considered when calculating the total installed cost of the heat pump. These cost details considerations are beyond the scope of this paper.

We have considered that the refrigerant circuit cost is directly proportional to the compressor displacement (figure 14). For this reason, the R1234ze partial cascade configuration is 15% more expensive than the supercritical configuration. The isobutane and butane systems are 46% and 95% more expensive than the supercritical R1234ze(E) configuration, respectively.

Figure 15 compares the additional electricity costs for partial cascade systems with various working fluids to partial cascade systems with the R1234ze and R1233zd pair. For a constant electricity price of 0.5€/kWh, the partial cascade configuration with the R1234ze/R1233zd pair offers about a 1.1% reduction in electricity consumption compared to isobutane.

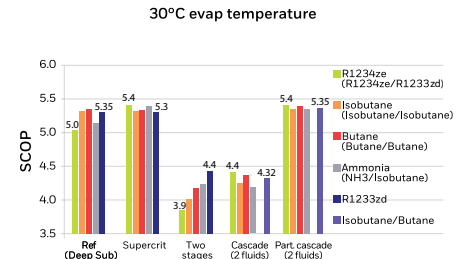


Figure 13. Impact of working fluids and selected configurations on the annual total electricity consumption

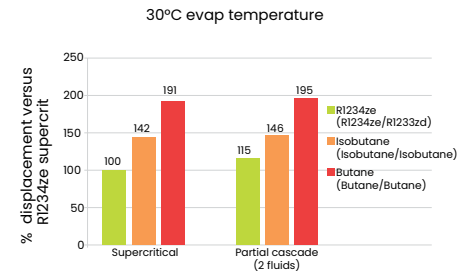


Figure 14. Compressor displacement difference for various fluids when using supercritical and partial cascade configurations

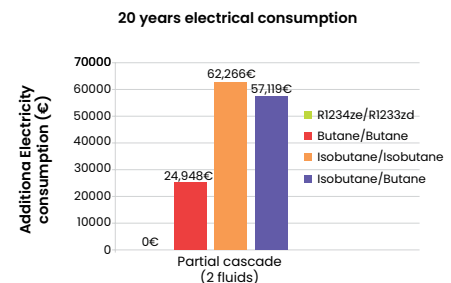


Figure 15. Additional electrical consumption versus R1234ze/R1233zd partial cascade configuration over 20 years

HP CAPEX and Operational Cost for Partial Cascade System

To estimate heat pump cost versus a single-stage standard heat pump configuration we assume that standard heat pump cost is divided into three parts:

- Refrigerant circuit costs: Includes compressors and heat exchangers
- Water loop costs: Includes water pipes, valves and hydraulic connections
- Installed costs: Includes footprint impact and risk mitigation costs

% COST PER KW	REFRIGERANT CIRCUIT	WATER CIRCUIT	INSTALLATION	ATEX & SAFETY REQUIREMENTS
Reference system using R1234ze	30	30	40	6
System with hydrocarbon	Calculated based on displacement and configuration	30	40	15

Table 3: Total CAPEX % cost function of the refrigerant circuit, water circuit, and installation

Heat pump total cost of a standard basic single-stage heat pump using A1 refrigerant is assumed to be 1000 €/kW installed heating capacity.

For cascade configurations, we consider a 20% increase in the refrigerant circuit cost versus single-stage reference configuration.

According to customer feedback, the total heat pump cost increases by 15 to 50% when A3 fluids are used versus A1 or A2L fluids. In this paper, we have considered 15% as the minimum cost increase when using A3 refrigerants. This cost increase includes ATEX and additional safety requirements not only for the heat pump itself but also for the whole surrounding. Footprint and building construction requirement costs are not considered as these are equipment and site-specific. Consequently, we have considered a 150€/kW cost increase when using A3 refrigerants versus other options.

	R1234ZE / R1233ZD	ISOBUTANE / ISOBUTANE	ISOBUTANE / BUTANE	BUTANE / BUTANE
Volumetric capacity impact	+15%	+46%	+49%	+95%
Cascade impact	+20%	+20%	+20%	+20%

Table 4: Volumetric capacity and cascade configuration impact on refrigerant circuit cost

Figure 16 shows the CAPEX increase for partial cascade systems with different fluids. This capex increase covers the components, configuration, and safety costs. Figure 17 shows the total CAPEX and OPEX costs over 20 years.

HP Cost Assumption for DH in Finland

To cover heating needs for the DH network in the Helsinki climate zone, Fortum investigated project costs based on HP and electrical boiler implementation (Figure 18). The quality of the equipment, electrical system and plant automation were required to be of high energy industry standard.

	CAPACITY KW	COST K€	WATER FLOW (l/s)	MAXIMUM TEMPERATURE °C	COST K€/KW
Heat pump plant	6000	3800	26.6	95	0.63
Electrical boiler plant	2400	800		115	0.33
Electrical grid		1400			

Table 5: Project CAPEX

Table 5 shows the costs of the heat pump plant, the electrical boiler plant and the electrical distribution module including the modular building and all auxiliary equipment but excluding foundation works.

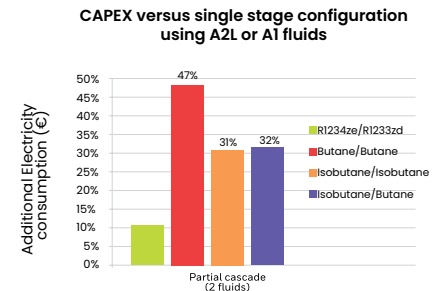


Figure 16. Total cost increase versus reference single-stage heat pump using A1 or A2L refrigerant

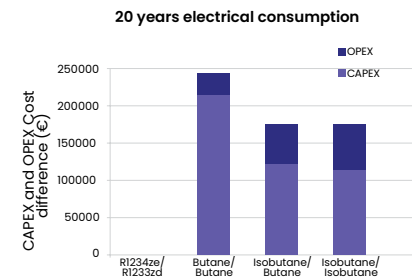


Figure 17. Cost increase over 20 years life span when using A3 versus A2L refrigerant for a 550 kW DH heat pump using heat recovery



Figure 18. Fortum plants, building for HP and boiler

Partial Cascade HP Option

A partial cascade system can be an option for this project. For this reason, total HP consumption and maximum electricity power are investigated to analyze the cost impact on CAPEX and OPEX and estimate payback scenarios.

Peak Electricity Load Reduction

Figure 19 shows calculation results according to Fortum operating conditions for the heat pump and boiler and for partial cascade heat pump options. The use of a partial cascade system allows the reduction of installed electrical power by 2000 kW and the elimination of electrical boiler plants.

CAPEX Cost Saving

According to the estimated peak load reduction (2000 kWelec), table 6 shows the impact on electrical grid investment cost reduction.

ELECTRICAL GRID WITHOUT BOILER

SYSTEM	COST/kW (K€/kW)	CAPACITY (kW)	COST (K€)	COST SAVING
Reference System		4600	1400	
Partial Cascade HP	0.304	2600	791	609

Table 6: Electrical grid cost saving

The total calculated cost savings is about 1,409 K€ for boiler cost plus electrical grid cost. Additional cost savings are computed as the plant automation cost reduction (-100k€) and power transfer fees (-37k€/year) which will also decrease the cost of the full HP option versus the hybrid HP and boiler option.

HP Cost increase

The use of a cascade system has an impact on heat pump cost versus reference options. According to prior art [08], a 20% cost increase can be expected as a quick estimation versus a single-stage system. We need to conduct further investigations in collaboration with HP manufacturers to improve cost estimation for available technologies.

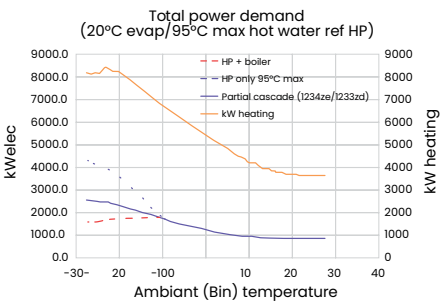


Figure 19. Electrical power consumption and heating power for selected configurations

CONCLUSION

5

This study demonstrates that a heat pump alone can supply very high-temperature hot water to meet all district heating needs, even in very low ambient temperatures. By using heat recovered from data centers, a SCOP of around 5 can be achieved using various advanced configurations and fluid combinations.

The **partial cascade configuration** is the most promising design, delivering the highest SCOP with available piston compressors. The use of a partial cascade helps reduce the pressure ratio of the compressors compared to the supercritical configuration and allows for the use of oil-free centrifugal compressors, which offer higher system efficiency at full and part loads. This has a positive impact on the heat pump SCOP and increases the efficiency gap with other options.

Besides the availability of components to build up the heat pump, the combination of **R1234ze(E) and R1233zd(E)** shows an equipment size reduction advantage thanks to the higher volumetric capacity of R1234ze versus hydrocarbon at low evaporating temperatures.

The A2L and A1 classification of R1234ze and R1233zd, respectively, also have a beneficial impact on component selection and implementation cost. Consequently, the thermodynamic properties and safety classification give a **major cost advantage** for HFO over hydrocarbon fluids for this district heating application.

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6

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