Technical notes on selected topics in SVAF phase 1

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Ammonia for large scale refrigeration and heat pump facilities

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Ammonia as a refrigerant in large scale refrigeration and heat pump facilities is a natural choice. The technology is well known and proven by industry and ammonia has been known as refrigerant for more than 100 years. In general there are some benefits and disadvantages when considering ammonia for a large scale heat pump. The two-phase working domain is large $T_{max} - T_{min} = 156K$ and centred around $40°C \approx 17bar$, which makes it ideal for cooling and heating using ambient temperature as sink or source [1]. Figure 1 shows a logP-h diagram of ammonia where the large temperature interval can be seen. The blue cycle in figure 1 represents a heat pump cycle as shown in figure 2. The compression is an ideal isotropic compression and the diagram is only for illustration and does not represent the actual heat pumps for choosen for the SVAF project. Ammonia has a high specific enthalpy of evaporation and condensation [2], which results in a lower mass flow rate than using HFC's and HC's, this results in a slightly higher coefficient of performance. This is even though the specific volume is higher, which results in a lower density and a need of a higher swept volume by the compressors. The molecular weight of ammmonia is $0.007kg/mol$ which is low compared to HC's and HFC, so in order to obtain a high mach number the velocity should be $450m/s$ instead of $\approx 300m/s$ as it is for HFC's and HC's [1]. The piping used can be of a smaller diameter than for HFC's and HC's as the specific evaporation enthalpy is higher and the velocity is higher. It can also be seen from the energy balance, eq. 1. If the heat transfer rate ($\dot{Q}$) is constant and the specific enthalpy (Δh) increases then the mass flow rate ($\dot{m}$) decreases.

$$\dot{Q} = \dot{m}\Delta h \tag{1}$$

Ammonia's specific enthalpy of evaporation is the second highest for known refrigerants, only water has a higher specific enthalpy of evaporation, but the operation range of water is not suitable for the heat pumps considered in this project as evaporation temperature preferably should be $> 100°C$ to avoid vacuum. Considering the safety of the environment, ammonia does not contribute to ozone depletion (ODP) or global warming potential (GWP) when emitted to the atmosphere. The drawbacks of ammonia are considered to be the aggressive corrosion which prevents the use of cobber tubing and ammonia is poisonous and explosive at high concentrations. Thus the compressor has to be explosion proof and safety arrangements must be installed at the facility.

Drawbacks of technology for large scale ammonia heat pumps are primarily concerns of the compressor technology and the pressure range. The Danish district heating network operates at temperatures between $80°C$ and $110°C$ corresponding to condensing pressures of a heat pump between $42bar$ and $75bar$, respectively. The temperature and pressures are also illustrated in figure 1 as the red lines in the two-phase area. The large scale ammonia compressors have until recently primarily been developed for refrigeration purposes and not much effort has been put in developing ammonia compressors for high condensation pressures needed for the district heating network, though more compressor manufacturers have started to sell ammonia compressors specific designed for heat pumps. The high temperatures of the system therefore sets a limit of the compressors available on the market. The two most common compressor types used for ammonia are reciprocating- and screw compressors. Unfortunately turbo machinery is not available for ammonia as it has a low molecular weight of 0.017 kg/mole, were the ideal molecular weight for turbo-machinery is between $0.04 - 0.05kg/mole$ [1]. The low molecular weight results in the need of up to 6 times the compression steps of R134a or very small blades compared to the rotor diameter [3]. It is not impossible to make a turbo compressor for ammonia, but there are none commercially available at this time. In general the capacity of turbo machinery is much higher than for reciprocating and screw compressors which at the moment can be purchased with a heating capacity of up 12 MW depending on the condensing pressure.

Reciprocating compressors discharge temperature is not allowed to exceed $140°C$ to $180°C$ presumably due to material limitations in the valves of the compressor [4] [5], this is shown as the red lines in figure 1. This claim could be challenged by experiments and using other materials in the valves, as

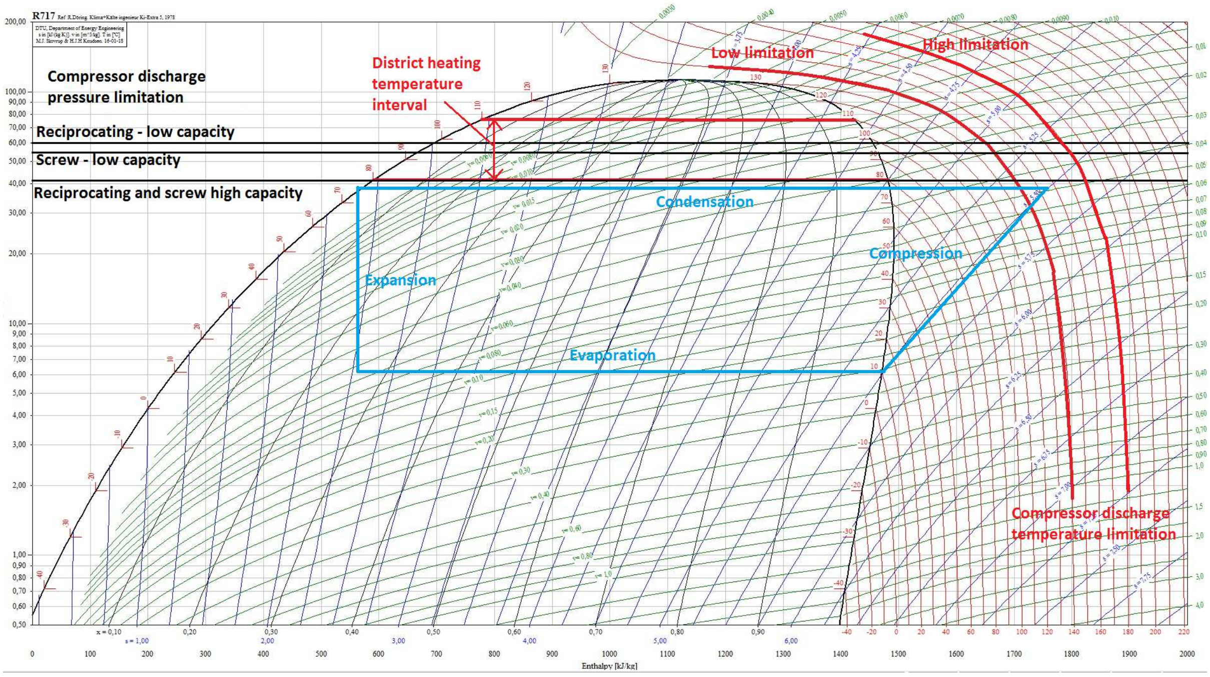

Figure 1: LogP-h diagram of hydrogen. The physical constraints of pressure and temperature of ammonia compressors for high temperature heat pumps

for a refrigeration facility the discharge gas would only in very few cases reach such high temperatures and the compressors used for heat pumps are the same as used for refrigeration. The limitation of the discharge gas, influence the maximum pressure ratio of a single stage compressor, which should not be higher than 5, if the discharge temperature should not exceed the limit. The capacities of reciprocating compressors vary with the pressure and discharge pressure. In general reciprocating compressors have a smaller capacity than screw compressors but they have a higher efficiency in part load operation. The oil in a reciprocating compressor should be selected carefully for the working temperature interval and the temperature of the discharge gas. Table 1 shows the known maximum capacities and pressure lift of reciprocating compressors for ammonia heat pumps. The limitations of reciprocating compressors are also shown in figure 1 as the black lines.

Screw compressors have a low efficiency if operated in part load but similar efficiency as a reciprocating compressor when operated at full load. The screw compressor capacity can be 8 times larger than a reciprocating compressor, which makes it a better choice for larger scale heat pumps. The oil cooling for lubricating the screw can be used as a heat source for the water at low temperature as the oil needs cooling and the amount of available energy is large enough to benefit from an oil cooler. The water is preheated in the oil cooler before entering the condenser and it can also be heated in an oil cooler after the condenser to take the water temperature up to or higher than the condensing temperature. The condensation pressure and load capacity are interdependent and with a higher discharge pressure the capacity decreases. See table 1 for maximum capacities and pressure lifts for screw compressors, the limitation are also shown in figure 1 as the black lines.

Table 1: Compressor data

Type	Reciprocating	Reciprocating	Screw	Screw
Pressure [bar]	60	40	42	55
Condensing temperature [$°C$]	98	78	80	92
Heat capacity [kW]	600	1500	12300	3200
Manufacture	Sabroe [6]	Sabroe [6]	Gea [7]	Sabroe [8]

Knowledge about Start-up and shut down of ammonia heat pumps is crucial in order to evaluate the possibility to use the heat pump as levelling source for fluctuating electricity production. After having consulted 3 people who works within the industry the main characteristics are yet to be clarified, the start up time can vary from 2 minutes to 1 hour [8] [9] [10]. Thus, the start-up time is going to be a part of the tests of the demonstration heat pumps to uncover how fast they start-up and the time to get into steady state operation.
The common way of starting a large heat pump is done by using a frequency control to make a ramp rate for the compressor. If the heat pump is controlled by an on-off design the compressor instantly starts up at full speed, all though full capacity is not reach until the refrigerant has been distributed in the system and the heat exchangers has reached operation temperature. When considering reciprocating compressors and screw compressors there are one major difference in the start-up and shut down. A reciprocating compressor keeps the pressure difference between condenser and evaporator when shut off, thus it does not need to retain the pressure difference upon start-up. When a screw compressor is shut off the refrigerant will make it spin backwards and the pressure between the condenser and evaporator will level out [8]. Thus, a screw compressor does need to retain the pressure difference upon start-up. The refrigerant will be collected in the evaporator, as it floats toward the coldest place and the condenser will only contain gas when a screw compressor is shut off.

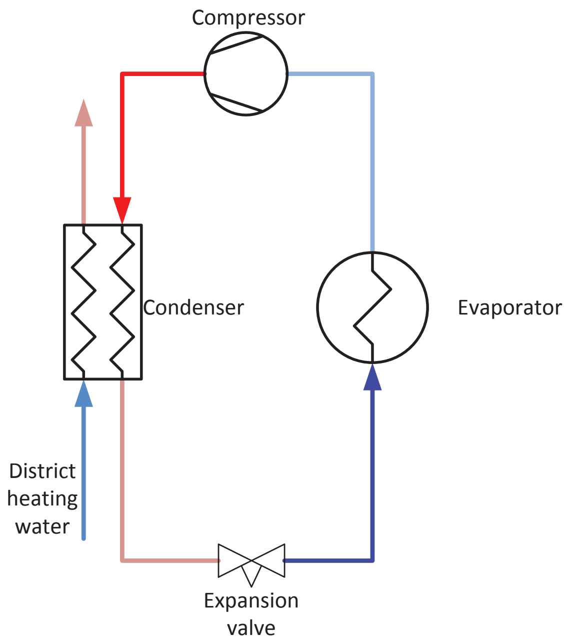

Figure 2: Process diagram of a simple one stage heat pump

A conclusion of the preliminary investigations is that there are unknown parameters concerning the limitations of ammonia equipment and the stress it is exposed to during start-up. At the moment it seems like the capacity limitation is set by the compressor technology available and the condensation pressure and temperature needed for heating the water. The capacity is limited by the condensation pressure as the different compressor technologies decreases in capacity as condensation pressure increases. The lack of turbo-machinery available for ammonia sets a capacity limit for the size of each compressor, more compressors are therefore needed to meet larger demands. It could therefore be of interest to investigate the possibility of making turbo-machinery for ammonia to decrease the number of compressors needed, all-though the properties of ammonia set a demanding limitation on ratio between blades and rotor and many compression steps are needed. The capacity of the commercially available compressors is decreasing as the condensation pressure is increasing, limiting the capacity or resulting in set-ups with multiple compressors to reach the required capacity. The experience with start-up with large ammonia refrigeration facilities shows that they typically are started using a ramp rate and that start-up time is not an issue as they can be planned ahead. Experimental test should be done in order to find out how fast the heat pumps can be started, especially if it is implemented in a flexible energy system where it is used for levelling out fluctuations in electricity production.

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Heat pump configurations and specifications

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1 Introduction and related work

Heat pumps (HPs) are proposed to co-supply heat for the Greater Copenhagen district heating (DH) network. Such integration of HPs, to supply a high degree of renewable energy to DH, has been the focus of several scientific journal papers [1, 2, 3, 4, 5, 6, 7]. For such applications, the temperature variation of either source or sink stream is typically of a magnitude, where serial connection of HPs, may provide an increase in the coefficient of performance (COP) [8]. On the other hand the economy of scale may suggest that the investment of a single unit is lower than two smaller units, when considering similar heat load. The most profitable solution may further vary with HP parameters such as sink temperature, temperature lift and temperature variation of sink and source streams.
Detailed thermo-economic models of various single stage vapour compression HPs (VCHP) were developed and investigated in [9]. The results were compared to similar results for the hybrid absorption-compression HP (HACHP) in [10]. The results show, that the best available technology (based on the lowest net present value (NPV)) typically depends on the performance and investment of the HP systems at the specific layout of the sink/source process streams. Besides the thermodynamic performance of the cycle and working fluid, it is important to consider the application limits of the individual components. The possible benefits of integrating several HPs in series were presented in [11]. The analysis is performed for vapour compression HPs using economic scenarios relevant for industrial integration/application. For such a case, the increased performance does not economically compensate for the increase in investment at the expected technical lifetime of the plant.
The specific work performed by DTU Mechanical Engineering within this project has been in close collaboration with HOFOR, Innotherm and DTI.

2 Changed economic optimum for HP in DH

For utility production, different taxation schemes are used for electricity and heat production in Denmark. Taxes for electricity are placed on the consumption (for which deductions may be granted for industry), whereas taxes for heat are placed on the fuel. To obtain low heat production prices, the utility companies are forced to select utility plants with low consumer cost, where fuel (eg. electricity) cost, market price of co-produced utilities (if any), O&M, taxes as well as investments are included in the calculation. In this respect, HPs for DH compete with many different technologies, where the competitive position focusses on heat cost, in which respect energy efficiency is only one of the significant parameters. In the current taxation scheme, HPs for DH are taxed according to one of two methods depending on the COP of the unit. A graphical representation is presented in Fig. 1, based on current legislation [12]. The intersection between the electric boiler law (2008) and the taxation for heat pump operating under DH conditions (2012) is approximately at COP = 2.3 (-). In the analysis, all heat pumps are operated according to the changed rules from 2012. According to previous analysis [13], the share of consumer cost for HPs related to taxes vary from 40 % to 55 %, where cost efficient units are typically above 50 %.
The increased heat production cost of DH HPs, compared to industrial HPs for process heat (with tax deduction), changes the economic optimum for HPs operated in series. Two examples are presented in Fig. 2. It is shown, that the benefit of serial connection depends mainly on the economic case and the temperature glide of HP sink and source streams. All other relevant economic parameters were similar to those presented in [11]. For the case of DH, the results show that serial connection of two HPs are preferable for both of the presented sink and source temperature glides. At low source temperature glide, the benefit of serial connection is reduced to an insignificant increase considering the uncertainties of the analysis. At sink and source temperature glide of 20 K, the economic benefit is exceeding 5 %.
The two proposed HPs are quite different with respect to both sink and source temperature glides. The geothermal HP has temperature glide exceeding 20 K for both sink and source (approximately 30 K for both). The above analysis indicates, that the proposed setup (with vapour compression HPs)

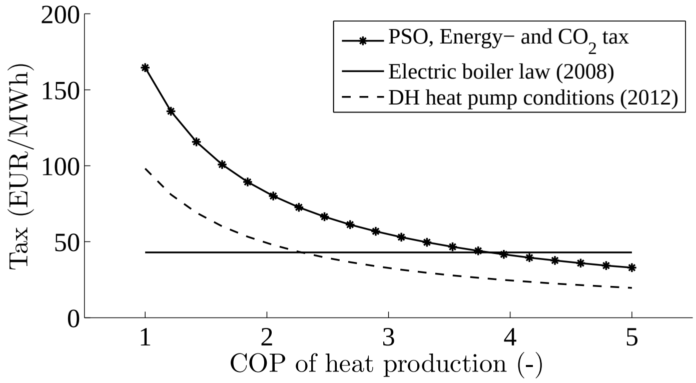
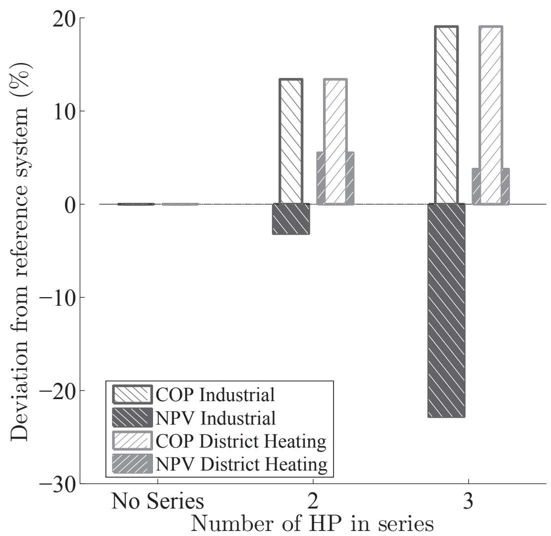
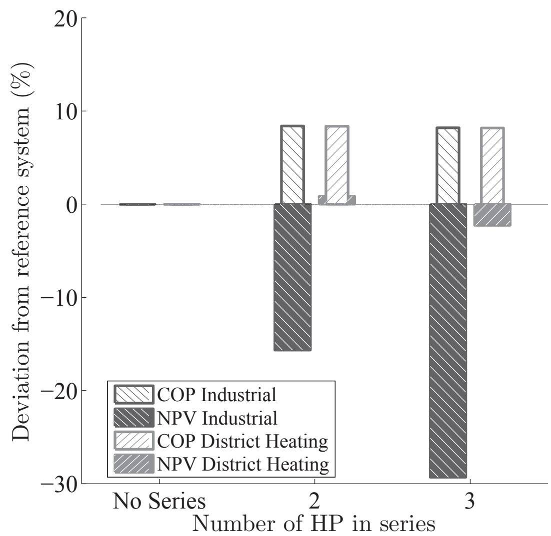


Figure 1: Taxation schemes for electric heat pumps



(a) Serially connected HPs with overall sink temperature glide at 20 K and source temperature glide at 20 K



(b) Serially connected HPs with overall sink temperature glide at 40 K and source temperature glide at 10 K

Figure 2: Example of different economic performances of the number of HPs in series for two relevant cases in Danish context. The benefit of serial connection depends mainly on the economic case and the temperature glide of HP sink and source streams.

should utilise two (or alternatively three) HPs in series. A simplified example of vapour compression HP integration with a geothermal heat source is presented in Fig. 3. The units are grouped by their integration with the heat source. The flow of sink stream of HP1, Oil heat exchanger (HEX) and direct HEX are mixed before being heated to the final specifications by HP2. Further detail to configurations are presented in Section 3 and 4.
For the second setup, the waste- and sea-water HP configuration, the sink and source configuration resembles the configuration used for Fig. 2b, although winter operation utilising sea-water as heat source may result in further reduction of the source temperature glide. In this case, it is indicated by the analysis, that two HPs is preferable, but that one HP may also present a relevant case for further investigation.

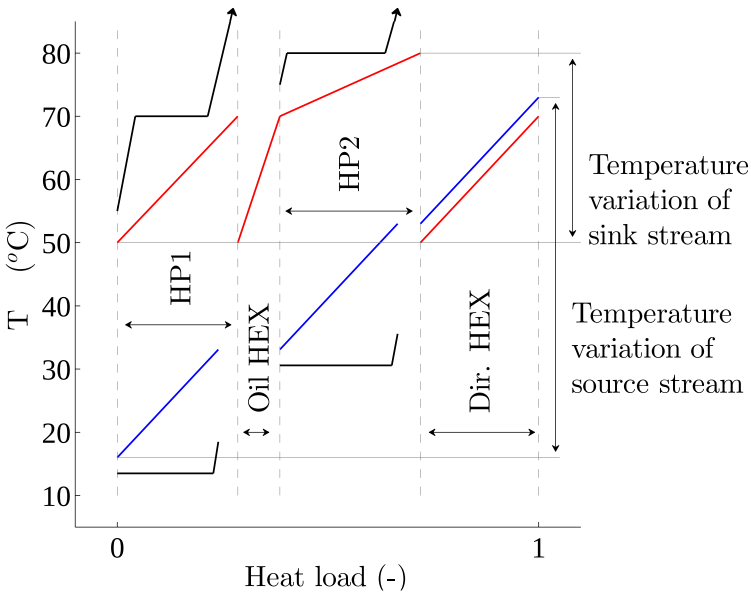


Figure 3: Simplified example of vapour compression HP integration with an geothermal heat source in a temperature heat load diagram.

3 Geothermal HP configuration

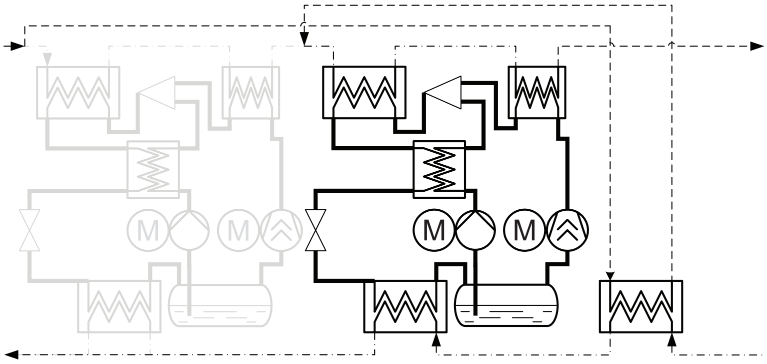
Due to the location of the allocated geothermal well, as well as the current layout of the DH network, the heat supplied by this unit is subject to specific capacity requirements, in terms of high supply temperature, for a share of the yearly operation hours. The requirements for high DH forward (HP sink) temperature, reduced the technically feasible configurations significantly. Whereas the design temperature (at 80 °C) is applicable for vapour compression HPs (with high share of oil cooling of the compressors), the additional requirement for operation at further increased temperature quickly exceeds the technical constraints.

At high sink outlet temperatures an ammonia-water HACHP may operate at lower pressure and discharge temperature levels, and it is possible to match the temperature glide of the absorption/desorption process in order to minimise entropy generation caused by heat transfer over a finite temperature difference, compared to the vapour compression HP utilising R717 [14, 15, 16]. The advantages result in high energetic performance, and technically feasible HP systems in the sink temperature range 70°C to 120 °C using current day refrigeration equipment. Due to the composition of the mixed working fluid, as well as the configuration of the solution circuit, the HACHP has two additional degrees of freedom for system design [10].

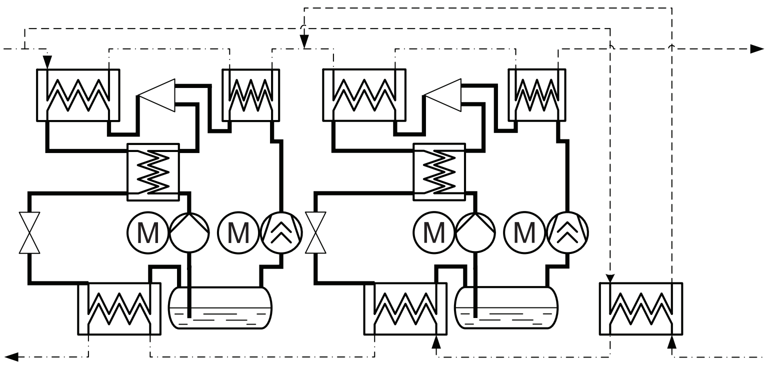
An energetic analysis showed that two designs were of particular interest, based on the general assumption to integrate direct heat exchange between the geothermal source and the DH network where applicable. The differences in design focused on whether the heat exchanger was balanced (similar heat capacity on both sides) or unbalanced. The energetic analysis further showed, that with little variation in the assumptions of the HPs cycle efficiency, the recommendation could shift to the other configuration. Based on such uncertainty, a full analysis of the consumer cost of heat were performed for each system, with detailed estimates of component sizing and efficiencies.

The investigated layouts are presented in Fig. 5, and consist of either one or two heat pumps in series, as well as one direct HEX. The designs are derived from the recommendations of Section 2 and literature [11], and the temperature variations of sink and source resembles those presented in Fig. 3. The results in terms of energetic performance and consumer cost (present value (PV) of the unit within it's technical lifetime) are presented in table 1. The proposed designs represent configurations with optimized working fluid composition and recirculation ration according to the considered technical constraints. The economic assessment includes the HP with installation and control, but without building, DH piping or connections to the heat source.

It is shown that the two heat pumps operated in series present a higher combined COP and lower PV



(a) 1 unbalanced HEX and 1 HACHP



(b) 1 balanced HEX and 2 HACHP

Figure 4: Principle sketch of the main components of two investigated geothermal HP layouts.

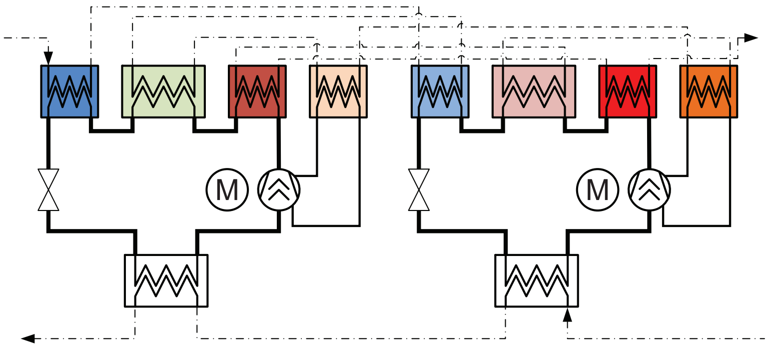
than for only one unit. The '1 balanced HEX and 2 HACHP' improve performance by approximately 8 % on consumer cost of heat and up to 20 % on COP. For either of the cases, the economic optimum (as well as achievable COP) is constrained by the technical constraints of current components. With reduced technical constraints (no limitation on discharge temperature) the PV of '1 unbalanced HEX and 1 HACHP' decreases by approximately 8 %, whereas the PV for '1 balanced HEX and 2 HACHP' only reduces by less than 1%. However, the PV of '1 balanced HEX and 2 HACHP' remain to provide the lowest PV regardless of the technical constraints, and is thus recommended.
It should be noted, that the current cost estimates and technical constraints are based on the reciprocating compressors without oil cooling. However, it is expected that the final design will be utilising oil cooled screw compressors. Optimal utilisation of heat from oil cooling adds further complexity to the integration, but does not influence the optimal cycle configuration as the temperature requirements for each unit is typically unchanged, as proposed in Fig 3.

Table 1: Energetic and economic performance of the two proposed layouts for the geothermal HP unit. The presented results do not cover the investment cost of the direct (balanced or unbalanced) HEX.

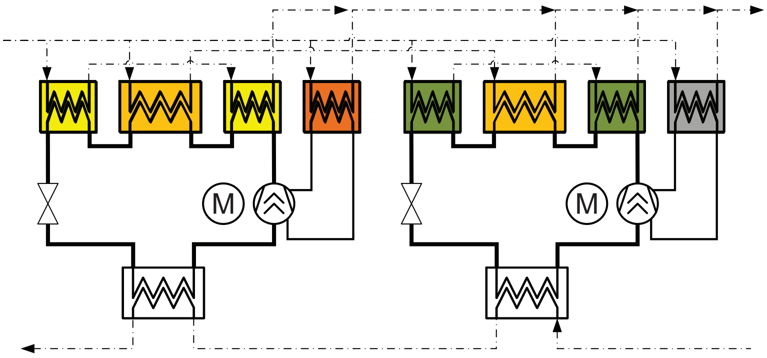
Configuration	$\dot{Q}$ (MW)	$P_{\max}$(bar)	COP (-)	PV (€/MW)	INV (€/MW)
1 unbalanced HEX and 1 HACHP	5.0	34	3.3	3.438.000	352.000
1 balanced HEX and 2 HACHP	5.0	34	3.9	3.108.000	485.000
- *HACHP1* -	1.5	25	4.1	934.000	179.000
- *HACHP2* -	3.5	36	3.8	2.175.000	306.000

4 Waste- and sea-water HP configuration

For waste- and sea-water HP unit, the proposed designs represent technically feasible versions of two HPs in series, according to the recomendation from Fig. 2. Two designs were investigated based on experience and proposals for improvement. Single stage HP units are considered for the analysis, but it is likely that the energetic and economic performance further benefits from utilisation of two stage units at high temperature lifts. The analysis addresses the significant possibilities of interconnection with a high number of heat exchangers for the heat sink. By understanding the interaction between components it is possible to assess the influence of off-design performance as well as design criteria. The presented units may be further optimised by challenging the current assumptions for oil temperatures etc.



(a) Serial configuration 1. All HEX are connected in series. Colour scale is used to show the temperature of the sink flow leaving the individual heat exchangers.



(b) Serial configuration 2. HEX are connected in series two and two. Colours indicate which of HEXs are coupled. The heated flows are subsequently mixed.

Figure 5: Principle sketch of the main components of two investigated waste- and sea-water HP layouts. Single stage heat pumps units are considered, but the energetic and economic performance may benefit further from utilisation of two stage units at high temperature lift.

Similarly as for the geothermal HP unit, the results are presented in table 2 in terms of energetic performance and consumer cost (PV of the unit in the technical lifetime). Current cost estimates are based on reciprocating compressors, but with the needed oil coolers included in the investment. The presented results represents technically feasible units in terms of pressure and temperature constraints for the utilised components.
The results show that in terms of investment the two proposed serial configurations exceed the investment of a single HP by approximately 25 %, but that the increased investment is returned in both cases by the increase in performance. The reduction in PV is as high as 10 % for one of the presented cases. The small differences between the two serial configurations do not allow for a detailed decision, without careful knowledge of influences such as part load performance. It is further believed that the potential for improvement for 'serial configuration 2' is higher than for 'serial configuration 1', due to the flexibility of fewer components in series.

Table 2: Energetic and economic performance of the three proposed layouts for the waste- and sea-water HP unit. The economic assessment does not include its building, DH piping or connections to the heat source.

Configuration	$\dot{Q}$ (MW)	$P_{\max}$ (bar)	COP (-)	PV (€/MW)	INV (€/MW)
No Series (1 HP)	5	39	3.0	4.281.000	333.000
Serial configuration 1	5	43	3.1	3.825.000	421.000
Serial configuration 2	5	44	3.1	3.994.000	415.000

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COMPARISON OF INTEGRATION OF ABSORPTION AND VAPOR COMPRESSION HEAT PUMP IN COMBINED HEAT AND POWER PLANT

In the central CHP plants the possibility of varying generation between electricity production and cogeneration of electricity and heat is exploited by utilizing the extraction steam from the turbine and generate heat for the district heating system.

Amager Power Station Unit 3 is designated with the same process as Avedøreværket Unit 1. For this plant a process model was developed in 2003 [Elmegaard, 2003] in connection with the Power Plant Simulator Contest proposed for the conference ECOS 2003 by Energi E2 and DTU.

Operation at full load and full heat production is illustrated in Figure 1

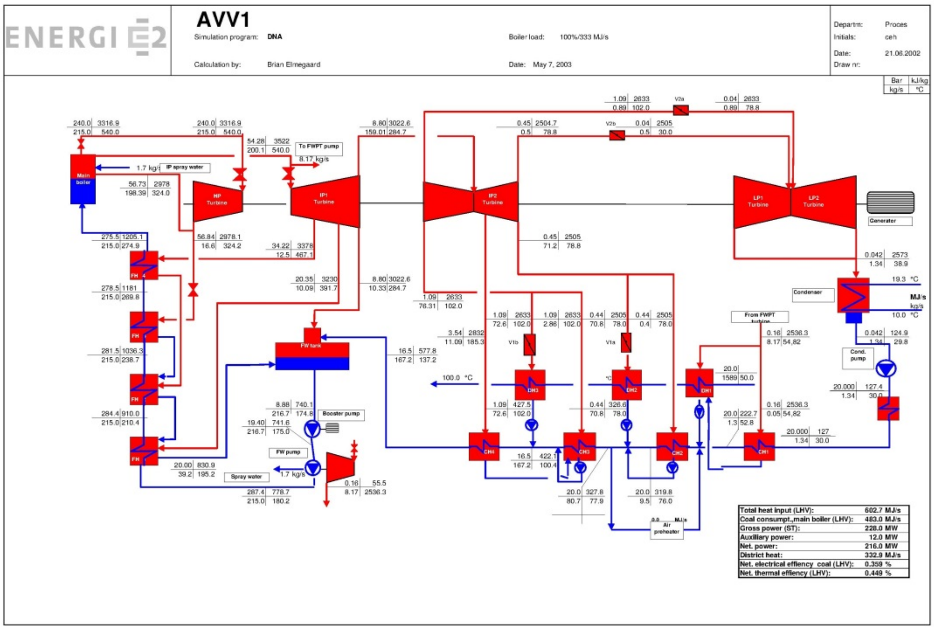


Figure 1 Avedøreværket Unit 1 process at 100% load and 100% district heating

The operating map of the plant is illustrated in the P-Q diagram in Figure 2. For heat input from 40% to 100% the diagram illustrates how the system can provide up to 250 MW of electricity in condensing mode and switch to 216 MW of electricity/333 MJ/s heat in back pressure mode.

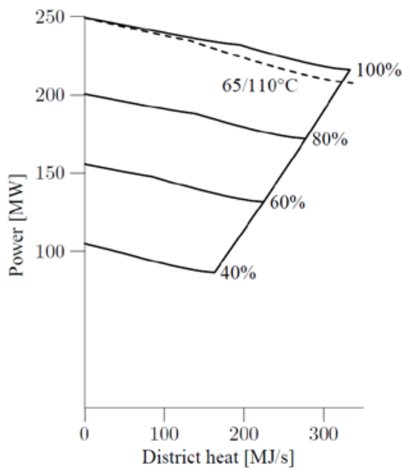


Figure 2 P-Q-diagram for AVV unit 1

In order to exploit geothermal energy in the Copenhagen district heating network, a facility plant linked to Amager Power Station Unit 3 was established in 2005. This has increased the potential operating range by extraction of additional steam from the turbine and utilization of it as driving heat for an absorption heat pump. This means that the plant at the same input fuel power can provide more heat by sacrificing additional power.

Two theses carried out during SVAF Phase 1 [Kaniadakis, 2014; Villegas, 2015], based on the model of Avedøreværket unit 1, analyzed how heat pumps based on, respectively, compression and absorption principle can extend the operating range of the plant by integration of geothermal energy in the system. In Figures 3 and 4 the configuration of the integration of the compression and absorption heat pumps with the plant are illustrated.

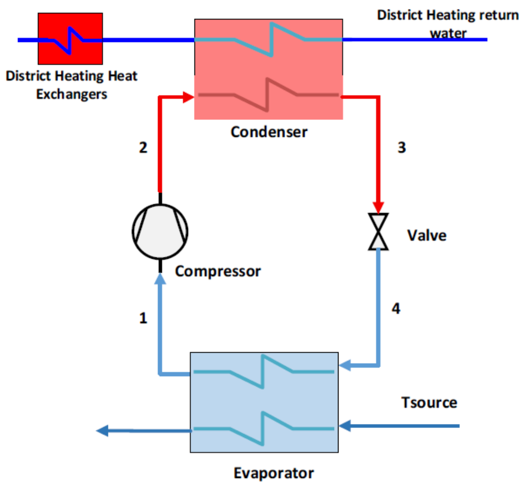


Figure 3 Integration of compression heat pump

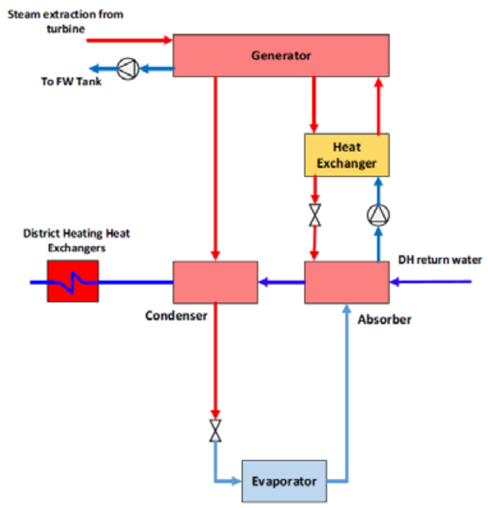


Figure 4 Integration of absorption heat pump

In either case, extraction steam from the low pressure part of the turbine is used for driving the heat pump. For the absorption heat pump the steam is used as driving heat for the heat pump generator. This can be compared to the compression heat pump, which does not directly use extraction steam, but is driven by electricity which is generated in the steam turbine. This is illustrated in Figure 5 which shows that the electric efficiency, defined as net power divided by fuel input, of the plant decreases due to the integration of a heat pump, which utilizes steam or power. In Figure 6 the energy utilization, district heat and power output divided by fuel input, is illustrated. It is clear that the compression heat pump obtain better utilization of the input fuel, than the absorption heat pump. Four different steam extraction options have been investigated for the absorption heat pump. No significant differences are observed between these.

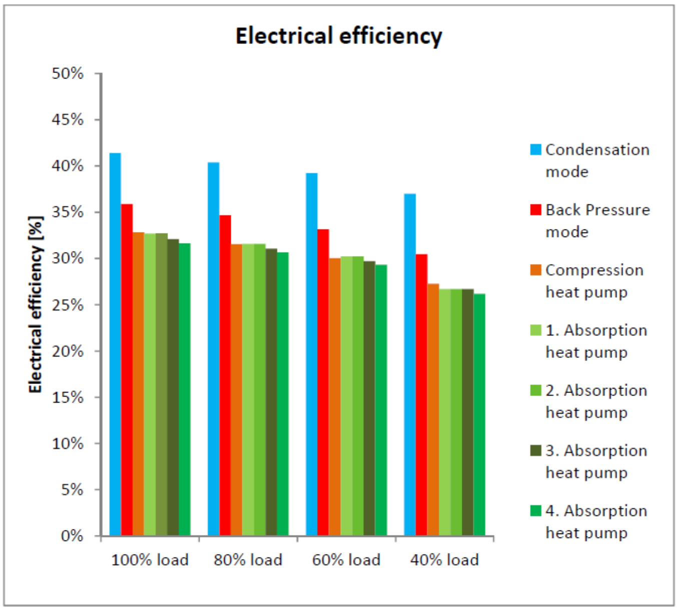


Figure 5 Electric efficiency

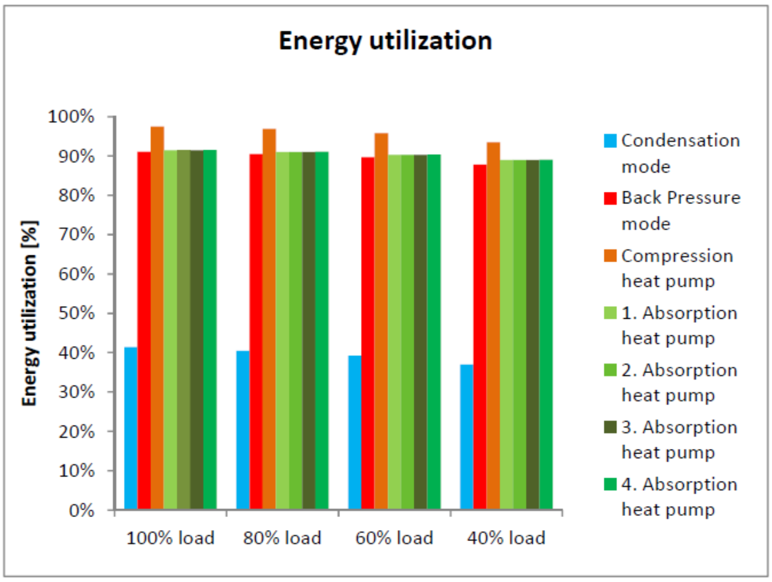
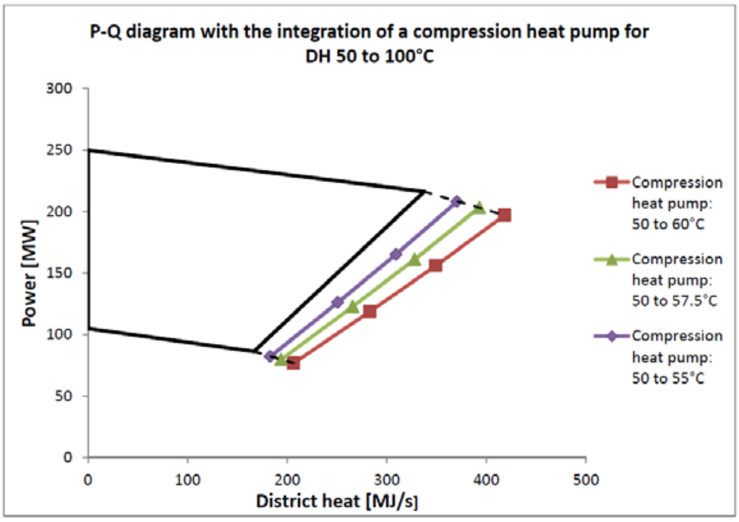


Figure 6 Energy utilization

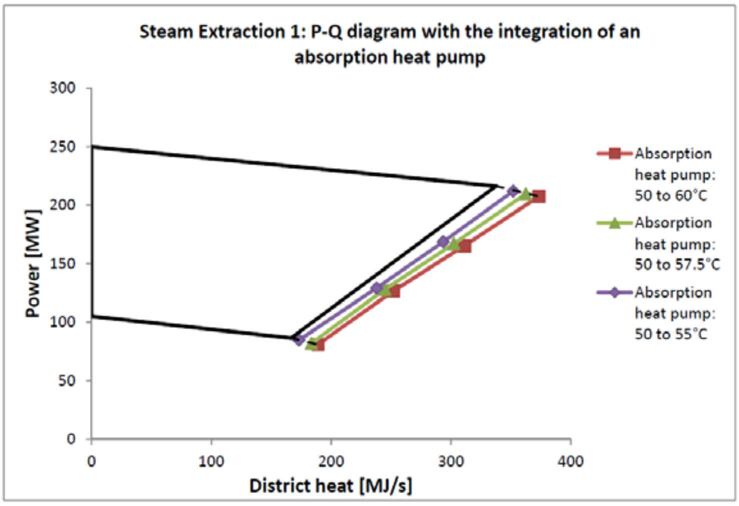
For the compression heat pump of in total 83 MJ/s Figure 7 illustrates how the system operation characteristic is expanded so that additional heat is produced by the consumption of additional power. The system can at full load now provide 197 MW electricity and 418 MJ/s heat. For the absorption system, the heat pump capacity can be 75 MJ/s. It is shown in Figure 8 that the plant can produce up to 208 MW of electricity and 378 MJ/s heat. The compression heat pump thus has a greater potential for increasing the operating range of the cogeneration facility.

An economic analysis based on [Energinet.dk, 2012] indicates a benefit of the absorption heat pump. Investment for the absorption system is 420 k€/MJ/s, while the compression heat pump will cost 570 k€/MJ/s. Accounting for the different capacities of the plants the absorption system will have a total investment of 230 million. kr., while the electric heat pump costs 370 million. kr.

Based on estimated average prices of fuel (96 DKK/GJ), heat (90 DKK/GJ) and electricity (240 DKK/MWh) and constant full load operation, the absorption heat pump has a payback period of about 10 years, whereas the compression heat pump is paid in about 17 years. An important factor in this result is electricity taxes, which will apply to the compression heat pump only. In addition the compression heat pump is in practice not limited to using electricity from the CHP plant, but may use wind power or other sources depending on the electricity prices, which improves the flexibility of the plant significantly.



Figur 7 P-Q-diagram kraftvarmeværk koblet med kompressionsvarmepumpe



Figur 8 P-Q-diagram kraftvarmeværk koblet med absorptionsvarmepumpe

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