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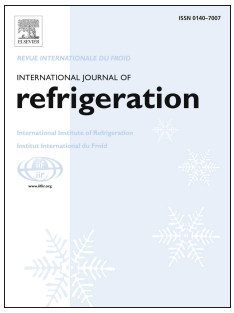

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- The working domain of 6 single stage vapor compression heat pumps are investigated
- Technical constraints based on commercial components are applied
- Economic viability is determined based on net present value
- R717 propose the best available technology at low and medium sink temperature
- For other temperature configurations R600a and R744 systems may be advantageous

Technical and economic working domains of industrial heat pumps: Part 1 - single stage vapour compression heat pumps

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Abstract

A large amount of operational and economic constraints limit the applicability of heat pumps operated with natural working fluids. The limitations are highly dependent on the integration of heat source and sink streams. An evaluation of feasible operating conditions was carried out considering the constraints of available refrigeration equipment and a requirement of a positive net present value of the investment. Six heat pump systems were considered, corresponding to an upper limit of the sink temperature of 120 °C. For each set of heat sink and source temperatures the best available technology was determined. The results showed that four different heat pump systems propose the best available technology at different parts of the complete domain. Ammonia systems presented the best available technology at low sink outlet temperature. At high temperature difference between sink in- and outlet, the transcritical R744 expands the working domain for low sink outlet temperatures.

Keywords: Industrial heat pumps, working domain, economic evaluation, natural working fluids, pinch design method

1. Introduction

Large vapour compression heat pumps (HP) may be used for process optimisation of industrial plants (Townsend and Linnhoff, 1983) or for utility supply in urban areas with heat networks (Ommen et al., 2014). For each individual process type (and temperature range) a wide span of design parameters and configuration options exists, which must be evaluated in order to calculate the best available heat pump technology for the assignment. The best available technology typically depends on the performance and investment of the heat pump 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. Many systems have been built with sink temperatures between 50 °C and 90 °C for industrial applications, but the sink temperature span may likely be due to limitations of the heat pump technology or limited cost effectiveness, rather than a limited demand at higher temperatures (Annex-21, 1995). Brunin et al. (1997) investigate the working domain of many working fluids in the temperature range up to 200 °C. A large fraction of the investigated working fluids are banned today (or will be shortly) throughout most of Europe (UN Ozone Secretariat, 2000; European Parliament, 2014) for large industrial scale systems. The study does not consider economics as such, but employs two physical constraints to represent economic feasibility. The constraints are coefficient of performance (COP) and volumetric heating capacity (VHC). The temperature variations of the two secondary streams (sink and source) are fixed at 10 K though out the investigation.

In the case of integration of heat pumps within tem-

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Nomenclature

Abbreviations

HP high pressure

LP low pressure

NDA Non-disclosure agreement

Symbols

A area (m^2)

AC annual cost (€)

c cost factor (€kWh^{-1})

CRF capital-recovery factor (-)

COP coefficient of performance (-)

FC annual cost of fuel consumption (€)

H annual operating hours (h)

h specific enthalpy (kJ kg^{-1})

i rate of return (%)

$\dot{m}$ mass flow rate (kg s^{-1})

N number of plates (-)

n lifetime in years (years)

NPV net present value (€)

OMC operation and maintenance cost (€)

p pressure (bar)

PBP pay back period (year)

PV present value (€)

PEC purchased equipment cost (€)

TCI total capital investment (€)

VHC volumetric heating capacity (MJ m^{-3})

$\dot{Q}$ heat flow rate (kW)

s specific entropy (kJ kg^{-1} K^{-1})

T temperature (K)

U heat transfer coefficient (kW m^{-2} K^{-1})

$\dot{W}$ work rate (kW)

v specific volume (m^3 kg^{-1})

$\dot{V}$ volume flow rate (m^3 s^{-1})

X size or capacity of component

Greek symbols

α cost function exponent (-)

Δ difference

η efficiency

Subscripts & Superscripts

b boiler

eff effective

el electrical

H high

HP heat pump

j denomination of technical project

L inflation

max maximum

NG natural gas

W equipment item with known cost

WF working fluid

Y equipment item with calculated cost

perature reservoirs of limited heat capacity, such as in pinch analysis (Linnhoff and Hindmarsh, 1983) or district heating networks (Ommen et al., 2014), the difference in temperature between inlet and outlet of a stream influences the performance of the investigated solution. Different temperature variations for sink or source, but for similar sink temperature and temperature lifts, may influence the choice of best available technology as well as the economic feasibility of such a modification.

The focus of this study was to reveal the possible working domains for several natural working fluids when considering variations in temperature lift and sink temperatures, and a range of different sink and source temperature variations. The investigation was limited to include only the single

stage vapour compression heat pump cycle. Five heat pump systems utilising natural working fluids were compared in terms of technical, thermodynamic and economic constraints, in order to include the heat pump performance and investment into the consideration. The considered natural working fluids were: R290, R600a, R744 and R717, where the latter utilises both low and high pressure components. One HFC working fluid (R134a) was included in the study for comparison of the feasibility of natural working fluids. By comparing the net present value or pay back period of the heat pump integration, it was possible to determine which of the considered systems proposes the best available technology.

In part 2 of this paper (Jensen et al., 2015), the ammonia-water hybrid absorption-compression heat pump is studied and compared to the best available vapour compression HP technology determined in this paper.

2. Method

Examination of the working domain has been carried out for six single stage vapour compression HP systems. Various possibilities for improved performance by e.g. internal heat exchangers have been disregarded for the analysis. A model of each heat pump was implemented in Engineering Equation Solver (F-Chart, 1992).

The heat pumps were compared using both economic and technical constraints. The considered technical constraints may be caused by the behaviour of the working fluid or by limitations in the development of suitable components. This is further discussed in section 2.3.

A few effects have been neglected as they are assumed of similar magnitude between the investigated heat pumps. Such effects include pressure drop in pipes, the extent of non-useful superheat and subcooling and compressor heat losses. Only full load steady state operation has been considered for the economic analysis.

2.1. Vapour compression heat pump

For industrial processes heat is often transferred by heat transfer fluids, which may be oil or water based. In this study it was assumed to be pure water, which was pressurized to avoid vaporisation of the secondary working fluid at elevated temperatures. Pinch point temperature difference was used to model heat exchange with both sink and source media (Nellis and Klein, 2009). A principle sketch of a vapour compression HP, and a temperature - heat load diagram for an azeotropic working fluid, are presented in Fig. 1. In the condenser, the working fluid was assumed sub-cooled until it reaches the pinch temperature difference at the sink entrance.

The performance of the HP was calculated using constant efficiencies for compressor and electrical motor, as well as fixed temperature differences in the heat exchangers. The used values are presented in Table 1.

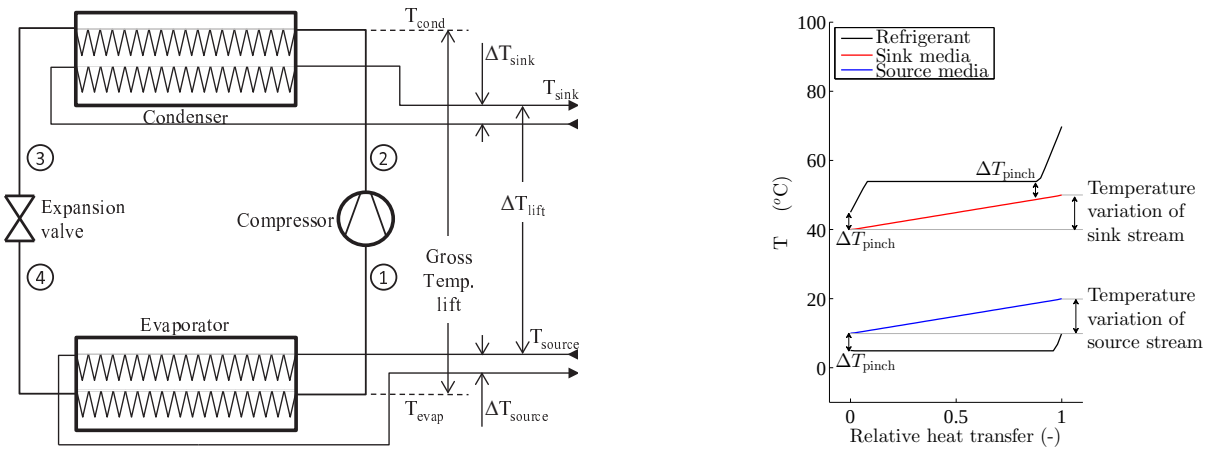
For the case of R744, where heat rejection from the working fluid is at supercritical pressure, the heat pump performance is affected by the gas cooler pressure as presented in Nekså et al. (1998) and further discussed in Cecchinato et al. (2010). By changing the heat exchange process of the gas cooler, also the investment of the heat pump system is affected. For all the considered temperature configurations of transcritical heat pumps, the heat rejection pressure allowing the optimal net present value (further explained in section 2.4.1) was determined and used. This methodology is proposed in order to obtain a fair comparison with all configurations, where additional degrees of freedom allow optimisation of the system design.

2.2. Estimation of plate heat exchanger area and pressure drop

Heat exchange processes are important in any HP and a significant part of the physical system, with high influence to the investment and the derived heat cost. Detailed heat transfer correlations for both evaporators and condensers were implemented and used in moving boundary models of the heat exchangers.

Chevron type plate heat exchangers were considered, as they appear to be cost efficient and are typically used in such systems today. The correlations used for the analysis are presented in tabel 2, while the plate dimensions are for a fixed type corresponding to pressure level and working fluid constraints (SWEP International AB, 2013). In this way an increment in heat exchange area will result in increased amount of plates.

For all of the considered heat pump systems, the calculated investment cost includes liquid receivers at an intermediate pressure for both subcritical and supercritical systems (Corberan, 2011; Kim et al., 2004). By using this option, it is possible to obtain



(a) Principle sketch of the vapour compression HP

(b) Temperature - heat load diagram for condenser and evaporator. Temperature variation of both sink and source is 10 K

Figure 1: Principle sketch and temperature - heat load diagram of the vapour compression HP

Table 1: Operating point and performance of HP and natural gas boiler

Type of data	Value	Unit	Designation
Efficiency	0.8	-	Compressor isentropic efficiency
	0.8	-	Compressor volumetric efficiency
	0.95	-	Compressor electric motor efficiency
	0.9	-	Natural gas burner efficiency
Temperature	5	K	Pinch point temperature difference in condensers, gascoolers and evaporators
	5	K	Compressor suction superheat

Table 2: Applied heat transfer and pressure drop correlations for the evaporator, condenser and gascooler

Component	Media	Zone	Heat transfer	Pressure drop
Condenser	H_2O		Martin (1996)	Martin (1996)
Condenser	Rxxx	vapour only:	Martin (1996)	Martin (1996)
		two-phase:	Yan et al. (1999)	Yan et al. (1999)
		transcritical:	Martin (1996)	Martin (1996)
		liquid only:	Martin (1996)	Martin (1996)
Evaporator	H_2O		Martin (1996)	Martin (1996)
Evaporator	Rxxx	two-phase:	Yan and Lin (1999)	Yan and Lin (1999)
		vapour only:	Martin (1996)	Martin (1996)

the desired subcooling in the condenser, and operate the system freely at start-up and shut-down procedures.

2.3. Compressors and operating conditions

In order to achieve the considered temperatures using vapour compression heat pumps, specially designed compressors are used. The pressure limit of the high pressure side may in many cases dictate the achievable sink temperatures, but also the suction pressure may pose limitations. Similarly for transcritical processes, where the pressure is not directly dependent on the sink temperature, the pressure limit has an effect on the heat pump performance. The condensing temperature and pressure may in rare cases be lower than that of the sink stream leaving the condenser. This is possible in the case where a high fraction of heat dissipation is from superheated vapour.

Oil degradation may pose limitations due to high temperatures in the compressor. To reduce wear and excess degradation, the compressor discharge temperatures are limited to 180 °C (Nekså et al., 1998).

Compressors from large international manufactures were investigated, where both price and operation limits were available. Five different types were identified according to different working fluid properties, flammability and availability. The limiting operation data for these are presented in Table 3.

Compressor types 1 - 3 are similar, where type 1 is applicable for HFC working fluids, type 2 is prepared for flammable environments and type 3 is equipped for R717 specifically. Additionally two high pressure compressors are included, where type 4 is for R717 and type 5 is for transcritical R744 processes. The investigation was not constrained to individual compressor technologies, but due to data availability considering the compressor cost, the below results represent reciprocating piston compressors. For type 1 - 3 price data was aggregated from several manufacturers with similar operating limits and performance.

2.4. Economic evaluation

The economic evaluation of the heat pumps was based on the economic method presented by Bejan et al. (1996), where individual component costs are used to account for the overall collected system. The method requires detailed cost data for components presented in a process flow diagram.

In order to obtain coherent and comprehensive data for specific components, the aggregation of data required several assumptions. It was assumed that:

- Purchased Equipment Cost (PEC) for an open type compressor was solely dependent on the type (specified in Table 3) and the swept volume of the compressor.
- PEC for an electrical motor with a fixed efficiency was dependent on the shaft power.
- PEC for a heat exchanger was a function of the heat exchange area and pressure limit.
- PEC of an intermediate pressure receiver was a function of volume and pressure limit.
- The PECs of expansion valve and oil separator were neglected.
- Total Capital Investment (TCI) of a component was calculated as 4.16 higher than PEC of the component (Bejan et al., 1996). This was done to account for additional cost related to new investment at an existing facility. The costs include installation, piping, instrumentation, electrical equipment, engineering and supervision, as well as startup and working capital etc.
- Electricity and natural gas prices correspond to the market cost for industrial consumers in the year 2012 according to Danish Energy Agency (2013a).
- The investment cost of already installed natural gas burners were neglected. This is the case if the heat pump replaces an existing installation.
- Source heat was assumed readily available as a process stream.
- Interest and inflation rate were fixed rates (7% and 2% respectively (Danish Energy Agency, 2011)).
- The technical lifetime of the plant was assumed to be 15 years (Fazelpour and Morosuk, 2014). This assumption may be seen as a conservative estimate, as higher technical lifetimes (20 years) are reported by other sources (Blarke, 2012; Danish Energy Agency, 2013b).
- Cost correlations are assumed to be valid for heat pump capacities between 100 kW and 2 MW. Restrictions are due to data availability for component correlations.

Purchased equipment cost (PEC) functions have been developed based on prices from intermediate Danish trade business and individual manufactures. The cost functions, Eq. 1, were constructed as pro-

Table 3: Available compressor technology with current operating limits

Working fluid	Type	Pressure limmit	Lubrication max. temp.	Capacity (1500 RPM)
	–	Bar	°C	m^3/h
R134a	1	28	180	5-280
R290	2	28	180	5-280
R600a	2	28	180	5-280
R717-LP	3	28	180	5-180
R717-HP	4	50	180	90-200
R744	5	140	180	6-25

posed by Bejan et al. (1996) where the purchase cost of an equipment item PEC_Y at a size or capacity X_Y can be calculated based on knowledge of the cost PEC_W at a different size or capacity X_W by use of a scaling exponent α.

$$\mathrm{PEC}_Y = \mathrm{PEC}_W \left(\frac{X_Y}{X_W}\right)^{\alpha} \tag{1}$$

Data for the component cost correlations and references are listed in table 4 for each of the investigated HP systems. In many cases, the data corresponds to several brands with similar characteristics in terms of economy and applicability. For some of the investigated components, the cost correlations are not presented due to non-disclosure agreements (NDA) with the manufacturers.
When comparing technical solutions the best available economically solution is preferred. Two different profitability criteria have been used for the analysis, the (simple) payback period (PBP) and the detailed net present value (NPV). The method for determination of the two factors is explained in sections 2.4.1 and 2.4.2.

2.4.1. Net present value

The profitability evaluation determines the net present value between two alternative systems, by considering the time value of cash flow streams during the lifetime of the investment. The present value of the proposed heat pump system was compared to the present value of the existing solution. The capital-recovery factor (CRF), Eq. (2) (Bejan et al., 1996), was used to calculate the levelized cost over the lifetime of the plant n:

$$\mathrm{CRF} = \frac{i^{\mathrm{eff}}(1+i^{\mathrm{eff}})^n}{(1+i^{\mathrm{eff}})^n - 1} \tag{2}$$

Where i^{eff} is the effective interest rate over the life time of the system, calculated as seen in Eq. (3) (Bejan et al., 1996), where i is the interest rate, and i_L is the inflation rate.

$$i^{\mathrm{eff}} = \frac{1+i}{1+i_L} - 1 \tag{3}$$

The annual fuel cost of the plant was calculated according to Eq. (4) for a heat pump or Eq. (5) for natural gas burner (Bejan et al., 1996).

$$\mathrm{FC}_{\mathrm{HP}} = \frac{\dot{Q}}{\mathrm{COP}} \cdot c_{\mathrm{el}} \cdot H \tag{4}$$

$$\mathrm{FC}_{\mathrm{NG}} = \frac{\dot{Q}}{\eta_{\mathrm{NG}}} \cdot c_{\mathrm{NG}} \cdot H \tag{5}$$

where $\dot{Q}$ is the desired heat rate of the demand. COP is the coefficient of performance of the heat pump, and the natural gas burner efficiency is denoted η_{ng}. c denotes the fuel price for either electricity or natural gas and H is the annual amount of operating hours.
The present value of the expenses associated with the project j was calculated as presented in Eq. (6) (Bejan et al., 1996).

$$\mathrm{PV}_j = \mathrm{TCI}_j + \frac{\mathrm{FC}_j}{\mathrm{CRF}} + \mathrm{OMC}_j \tag{6}$$

where TCI_j is the total capital investment of all required components for system j. OMC_j is the operation and maintenance cost (OMC) (not including fuel cost) for system j, which is assumed to be a fixed factor (20 %) of TCI_j (Bejan et al., 1996). For natural gas burners the capital investments are considered as sunk cost ($\mathrm{TCI}_{\mathrm{NG}} = 0$) and thus the OMC costs are similarly neglected.
The NPV was calculated as presented in Eq. (7) for the case where a heat pump is proposed as an alternative for a natural gas burner in a facility.

$$\mathrm{NPV}_{\mathrm{HP}} = \mathrm{PV}_{\mathrm{NG}} - \mathrm{PV}_{\mathrm{HP}} \tag{7}$$

Table 4: Used cost correlations for component types divided by application limits

Component	type	PEC_W (€)	X_W	$\alpha(-)$	Source
Compressor	R134a	10631	178.4 (m^3 h^{-1})	0.79	trade business [1 2 3]
	R290, R600a	19850	279.8 (m^3 h^{-1})	0.73	trade business [1 2]
	R717-LP	11914	178.4 (m^3 h^{-1})	0.66	trade business [1 2]
	R717-HP	NDA	NDA	NDA	manufacturer [4]
	R744	12109	25.6 (m^3 h^{-1})	0.42	trade business [1 2]
Electrical motor	R134a, R717-LP, R717-HP	10710	250 (kW)	0.65	trade business [1]
	R290, R600a	0	0	0	incl. in compressor [1 2]
Receiver	R134a, R290, R600a, R717-LP	1444	0.089 (m^3)	0.63	trade business [1]
	R717-HP	1934	0.089 (m^3)	0.66	trade business [1]
	R744	2744	0.050 (m^3)	1.12	trade business [1]
Plate heat exchanger	R134a, R290, R600a, R717-LP	15526	42 (m^2)	0.8	trade business [1 2 3]
	R717-HP	NDA	NDA	NDA	manufacturer [5]
	R744	NDA	NDA	NDA	manufacturer [5]

[1]H. Jessen Jörgensen A/S (2013)
[2]FK Teknik A/S (2013)
[3]Ahlsell Danmark ApS (2013)
[4] Johnson Controls, Inc. (2013)
[5]SWEP International AB (2013)

If the NPV is positive, the proposed heat pump is a cost effective alternative to the already installed burner.

2.4.2. Pay back period

The PBP determines the length of time required for the proposed installation to fully recover the TCI by reduction in FC and OMC compared to the existing solution. PBP is often used as a measure for feasibility of technical projects, but the factor does not include the time value of money or include cash flows that occur after the calculated period. The PBP was calculated as presented in Eq. 8 (Bejan et al., 1996).

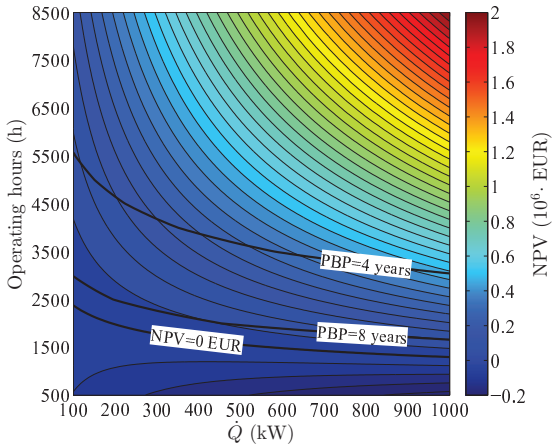
$$PBP_{HP} = \frac{TCI_{HP}}{(FC_{NG} - FC_{HP}) + (OMC_{NG} - OMC_{HP}) \cdot CRF} \quad (8)$$

3. Results

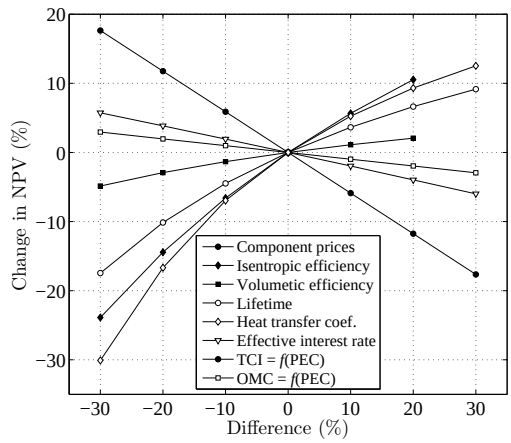
Detailed results of the thermodynamic state points, heat transfer and costs are presented in appendix Appendix A for an R717-LP heat pump system operating at $T_{sink,out}$ = 60 °C, T_{lift} = 20 K, ΔT_{sink}=20 K, ΔT_{source}=10 K. For this analysis the heat pump operating hours and capacity was assumed to be 3500 h and 1000 kW respectively. Similar analysis was performed for the remaining systems.

For the analysis of the working domains, the operating hours and heat pump capacity have been kept constant. The operating point corresponds to that presented in appendix Appendix A. The impact of the two parameters were determined by a parametric variation, which is presented in Fig. 2a. Contours of two different simple pay back periods (PBP) and the net present value of zero (meaning that both solutions are equally feasible for the technical lifetime of the heat pump system) are plotted. Similar variations of economic constraints, NPV=0 as well as PBP=4 years and PBP=8 years have been included in each working domain, allowing the reader to decide the appropriate economic constraint.

It is worth noticing, that all solutions in Fig. 2a represent COP=6.5 and VHC=7.7 MJ/m^3 (according to calculation in Appendix Appendix A), which are within the recommended levels defined by Brunin et al. (1997). Although within the limits, a significant part of the solutions were found to be infeasible compared to the fuel cost of a natural gas boiler, which indicate that the use of semi-economical constraints may not include the required level of detail. Based on the results of Fig. 2a the heat pump working domains were investigated for a case of 3500 operating hours yearly, and $\dot{Q}_{HP}$ = 1000 kW. Fig. 2b presents a parametric variation of significant assumptions for calculation of the NPV at this operation point. The NPV is most sensitive

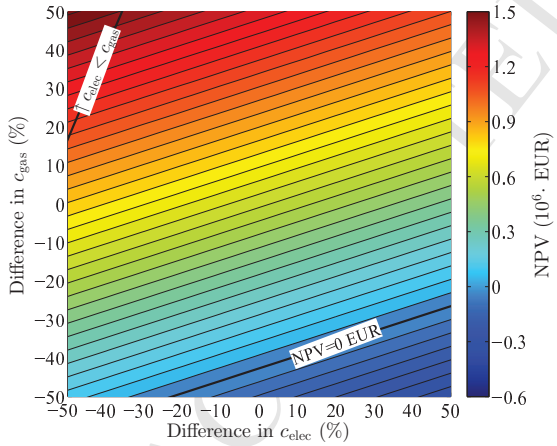


(a) NPV of HP system with variations in size and yearly operation hours

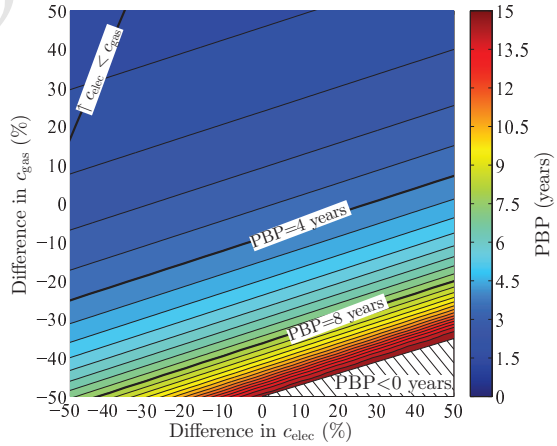


(b) Parametric variation of key economic assumptions

Figure 2: Influence of key economic assumptions on NPV
R717-HP heat pump operating at $T_{\text{sink,out}}$ = 60°C, T_{lift} = 20°C, ΔT_{sink}=20 K, ΔT_{source}=10 K



(a) NPV of HP system with variations in fuel cost



(b) PBP of HP system with variations in fuel cost

Figure 3: Impact of natural gas and electricity prices on NPV and PBP
R717-HP heat pump operating at $T_{\text{sink,out}}$ = 60°C, T_{lift} = 20°C, ΔT_{sink}=20 K, ΔT_{source}=10 K

to the assumptions of heat transfer coefficients and isentropic efficiency. The observed sensitivity is assumed not to limit the validity of the economic comparisons between the technologies, as offsets would result in similar changes for all the considered systems.

Fig. 3a - 3b presents a variation of natural gas and electricity prices for the two used economic criteria. Both natural gas and electricity prices were varied by ± 50 % for the investigated heat pump solution. Based on the slope of constant NPV in Fig. 3a it is found that the system is more sensitive of the price of natural gas than of electricity, and that the gradients of fixed electricity or natural gas prices are close to constant. At high gas prices and low electricity prices an area is indicated, where the price of electricity is lower than natural gas. At low gas prices and high electricity prices, the heat pump NPV can become negative.

For the PBP in Fig. 3b similar areas are indicated where the simple PBP becomes negative, or where the natural gas price exceeds the electricity price. Opposite to NPV, the gradients of fixed electricity or natural gas prices are far from constant throughout the investigated price span. By comparison of the two economic criteria, it is clear that the NPV is less sensitive to variations in fuel cost. Another benefit of the use of NPV as the economic criterion, is that the NPV evaluation returns a monotonic result.

The technical and economic limitations for heat pump operation were investigated for different sink and source temperature glides. The presented temperature glides are: (sink/source): 10K/10K, 20K/10K, 20K/20K and 40K/10K. In the study the working domains were established by considering the operational boundaries for each considered heat pump type for 10K/10K (Fig. 4) and 40K/10K (Fig. 5). The remaining temperature glides (20K/10K and 20K/20K) are presented in appendix Appendix B as Fig. B.1 and Fig. B.2 respectively.

The hatched areas of the individual plots are areas where the considered technology is not appropriate. One area is caused if $T_{\text{source,out}}$ is below 0 °C, which requires another source media to be considered, and thus influences the heat transfer characteristics. This could be feasible if a brine was applied, but it would imply cooling the heat source below ambient temperatures, which may only be relevant if a cooling demand is satisfied. The second area corresponds to a case where $T_{\text{sink,in}} < T_{\text{source,in}}$. In such a heat transfer processes, heat should be exchanged directly between the two streams before a heat pump is integrated in the process (Townsend and Linnhoff, 1983; Annex-21, 1995).

The economic constraints are presented as green lines (NPV) and turquoise / dashed turquoise lines for PBP. Red lines indicates high discharge temperatures, whereas blue lines show the pressure constraints for the considered systems.

Each individual plot shows the constraints for the specific working fluid, showing both the components and the economic restrictions. By examination of Fig. 4 it can be seen, that combination of the four natural working fluids allows for a large working domain. The NPV constraints for R717-LP and R717-HP include a large fraction of the considered temperature span, but the technology is restricted by high compressor discharge temperatures, as well as the allowable high pressure. For the four remaining heat pumps, the discharge temperatures are below the limit in all cases, but at the same time it may be seen that the NPV constraint is at a lower ΔT_{lift} compared to the two ammonia configurations.

Specifically for all R744 configurations (e.g. Fig. 4a) the critical pressure of the working fluid limits the working domain at high $T_{\text{sink,out}}$, as two-phase evaporation is not possible in some of the considered combinations. Furthermore, the dashed pressure constraint indicates where optimal pressure is above the maximum allowable for the compressor. Beyond this line the high pressure side is kept at the maximum allowable pressure, which results in decreased performance and changed slope of the economic constraints.

For 40K/10K (Fig. 5) similar plots are presented. At higher sink temperature glide, the high discharge temperature for R717 decreases the area of feasible solutions. At the same time, transcritical R744 is economically and technically feasible at even higher temperature lifts, than for the case 10K/10K. It can be seen that an area exist where R744 increases the allowable ΔT_{lift}, in the areas where R717 is constrained by discharge temperature. None of the considered heat pumps for sink/source glide of 40K/10K in Fig. 5a - 5f allow a PBT below 4 years. From investigation of Fig. 4 and 5 it is found that the trends for NPV are similar for all subcritical systems, although at different magnitude for ΔT_{lift}. In order to further understand the mechanisms, an investigation of the thermodynamic performance and investment has been performed for one case

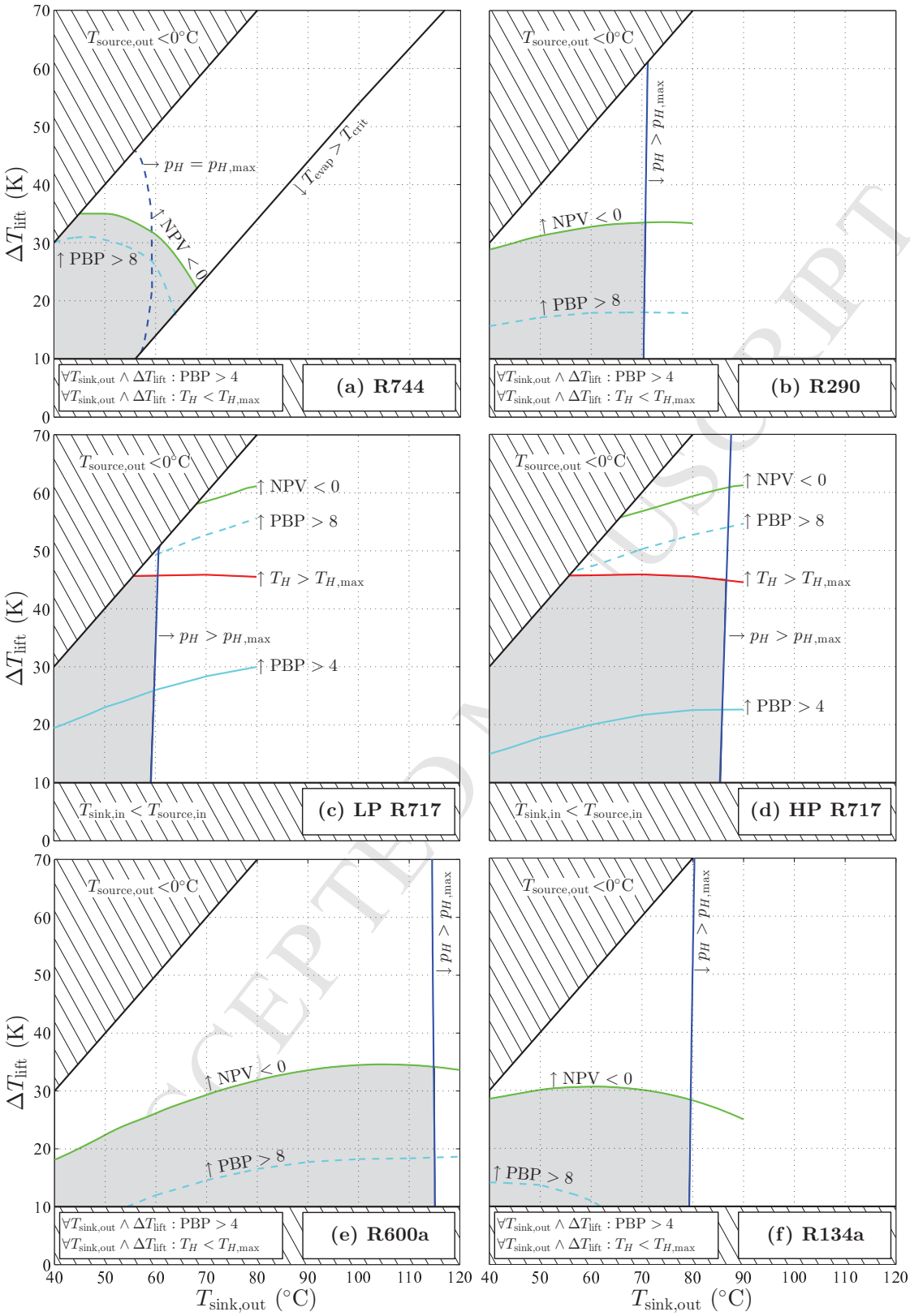


Figure 4: Working domains for five different heat pumps - ΔT_{sink}=10 K / ΔT_{source}=10 K

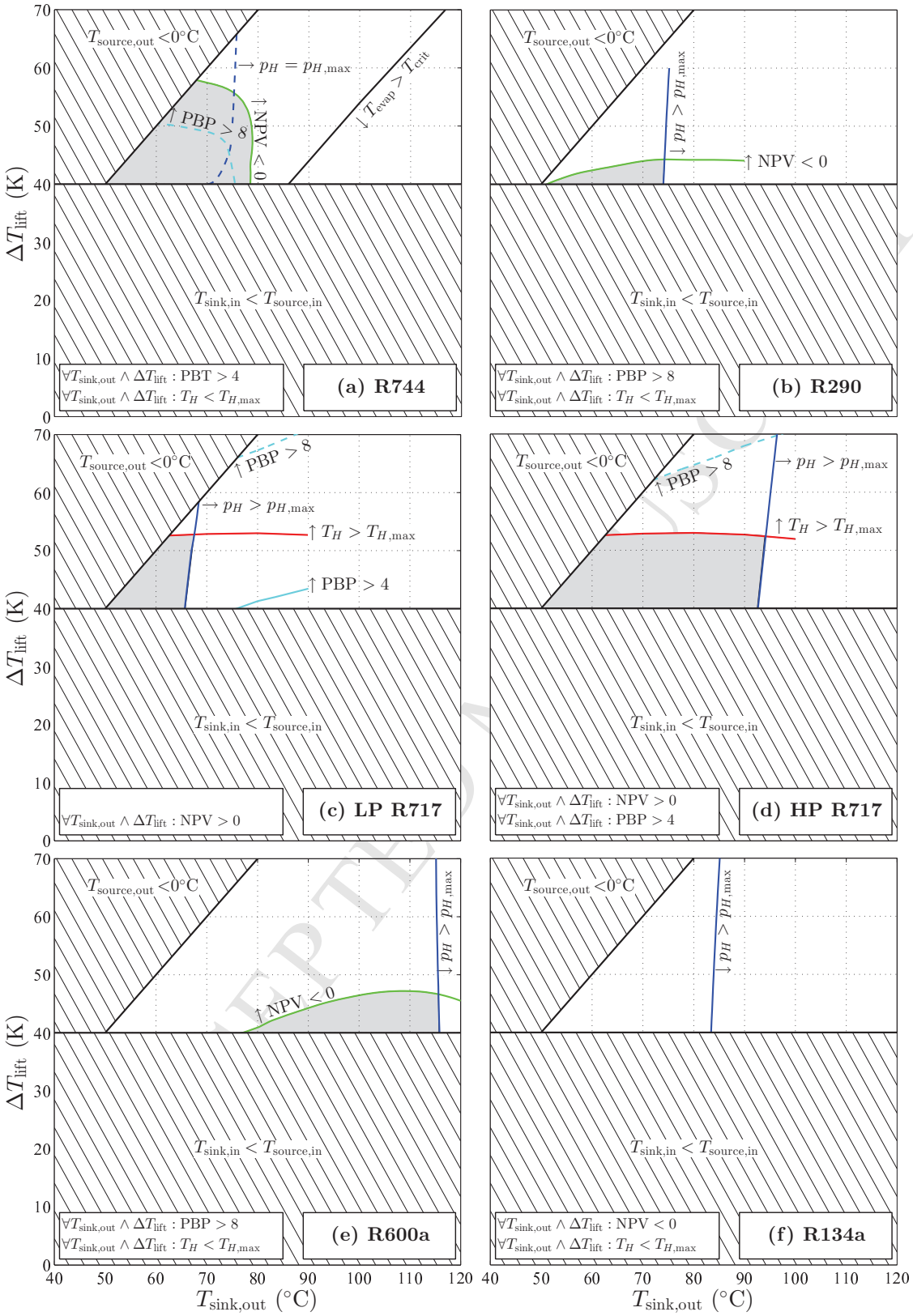


Figure 5: Working domains for five different heat pumps - ΔT_{sink}=40 K / ΔT_{source}=10 K

(corresponding to R600a at 10K/10K sink/source glide in Fig. 4e). The trends for COP and investment are presented in Fig. 6a and 6b respectively. High COP is obtained for low temperature lifts resulting in low FC_{HP}. The investment TCI_{HP} is lowest at high sink outlet temperatures and medium (e.g. 30-50 K) temperature lift. At lower lift the investment required for the condenser is increased, while at low sink outlet temperature and high temperature lift the PEC of the compressor is significantly increased, as the suction volume of the heat pump is high.

High temperature glide in the source is typically used in the cases where the capacity of the source at low temperature glide does not fully match the heat demand. Optimal heat production may then be considered as a tradeoff between capacity and performance. By inspection of 20K/20K (Fig. B.2), it is clear that the working domains of the heat pumps are closely related to those of 20K/10K (Fig. B.1), although the economic constraints are moved towards a lower temperature lift. Because of the increased pressure ratio between evaporator and condenser, the R717 discharge temperature constraint is moved below a temperature lift of 40 K.

A full comparison of working domains for all heat pump systems is presented in Fig. 7 and Fig. 8 for all of the considered sink and source glides. Fig. 7a presents the combination of boundaries for each individual working fluid for sink/source glide of 10K/10K and Fig. 7b highlights the economically best (NPV) solutions in the entire working domain. Similar plots are presented for the alternative glides. Fig. 7c-d correspond to 20K/20K glide. Fig. 8a-b and Fig. 8c-d are for 20K/10K and 40K/10K respectively.

For all of the sink/source glides, it is found that R717-LP configurations are optimal within the established technical and economical boundaries for the system. For increasing sink temperatures, R717-HP is cost optimal in the area constrained by discharge temperatures and high pressure limit. For further increase in sink temperatures, R600a configurations are the only feasible option until the limitations on pressure at approximately 115°C. The R600a configurations are furthermore limited by an economic constraint at ΔT_{lift} between 30 to 40 K for most glides, and 40 to 50 K for the 40K/10K glide (Fig. 8d).

Transcritical R744 is the optimal configuration at high ΔT_{lift} if the sink temperature glide is large. For the glide 40K/10K a significant area exsist, where the technology provides the only solutions which are technically and economically feasible. For the remaining cases with sink temperature glide at 20 K or below, the maximal ΔT_{lift} for R744 is below the constraints of R717-LP.

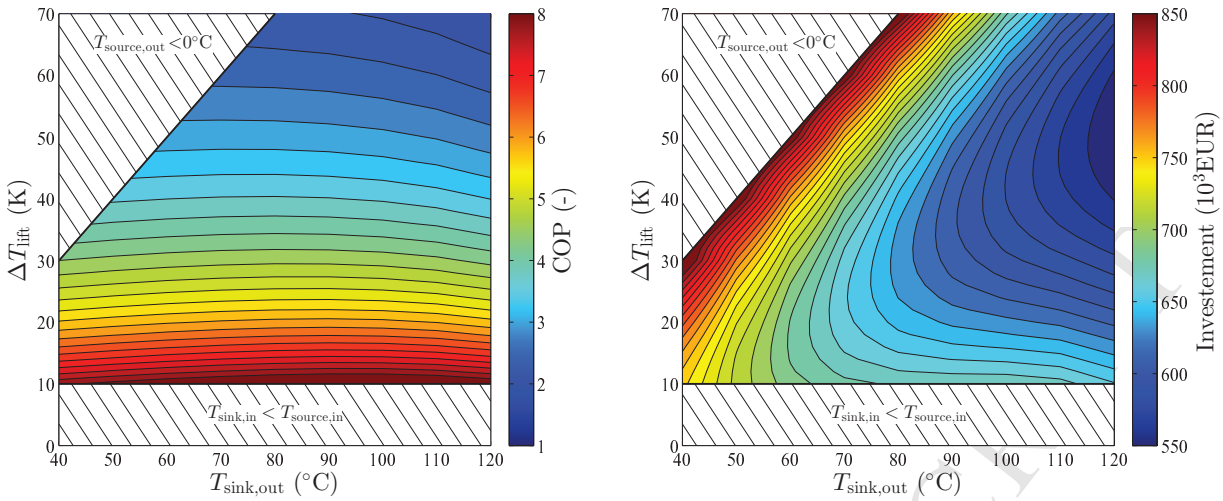
4. Discussion

The economic correlations used in the investigation are based directly on data from Danish suppliers or international manufacturers. List prices may be seen as a conservative estimate on investment cost, as sealed price agreements may change the actual component investment cost. From Fig. 2b a 18 % increased NPV is found for a 30 % decrease in component cost. Such a change would significantly increase the feasible area for many of the considered heat pump types. It is expected that the cost reductions are similar for the investigated technologies, and as such the analysis would not change the relationship significantly between the types, but only between heat pumps and natural gas installations.

The results presented here stem from simulation results by verified models. Both economic correlations and heat transfer correlations are validated. Unfortunately, it is not possible to validate the investment cost of complete installations, due to lack of data availability in the open literature. However, the same approach has been applied for all investigations and thus it is assumed that if any fault occurs it will mainly effect the absolute values, and not the relation between the considered systems.

The heat transfer correlations used for condensation and evaporation are derived from experiments with R134a, but used in the evaluation for all working fluids, assuming that the individual transport properties of the fluids allow the required detail for calculating the correct heat transfer coefficient. An updated experimental investigation of the correlation at the correct temperature levels is being pursued at the time of writing. The used correlations have been compared to data from a few manufactures with promising results for several working fluids. Investigation of the parametric variations in Fig. 2b shows that the effect of changed heat transfer coefficient by ± 30% is between -30% to 12% for the total heat transfer coefficient on NPV.

Several steps to increase the working domain of individual heat pump systems may be considered such as: two stage compression, liquid injection in suction line, internal heat exchangers, heat pumps connected in series or cascade solutions etc. The en-



(a) COP (b) Investment

Figure 6: Trend of COP and investment for $T_{\text{source,out}}$ and ΔT_{lift} - R600a heat pump for 10K/10K sink/source glide

ergetic or technical improvements of NPV are typically reduced by increased investment in terms of additional components for the thermodynamic cycle. Such improved working domains by energetic or technical improvements have not been considered in this study, but may be the focus of subsequent analysis. At certain temperature configurations, where the best available technology may shift to a different system due to an expanded working domain (ie. from R600a to R717-HP) the gain in NPV may be significant. If a shift in best available technology is not possible, the maximal improvement due to cycle improvements (not including heat pumps in series) is typically below 10 % on COP (Granryd, 2005). With additional investment according to the changed PI-diagram the gain in NPV will be lower than this figure.

5. Conclusion

From a variation of operation hours and heat production of a heat pump, it is found that the net present value or pay back period shows higher detail in the feasibility of investigated solutions than the use of semi-economic parameters such as COP and VHC.

All of the six considered heat pumps show working domains where net present value is positive, when compared to the fuel cost of a natural gas burner. Four of the heat pumps are advantageous when considering all of the presented constraints. These heat pumps are R717-LP, R717-HP, R600a and transcritical R744. In areas covered by the working domain of several configurations, the R717-LP and R717-HP allow the optimal NPV.

By investigation of the four sink/source working domains for vapour compression heat pumps it is found, that sink temperatures of up to 115 °C and temperature lifts up to 40 K can be achieved by four common heat pump systems. For both R717-LP and R717-HP heat pump systems, the discharge temperature is a significant limitation for expansion of the working domains, where as for the remaining technologies the economic constraints restrict feasible solutions.

Acknowledgements

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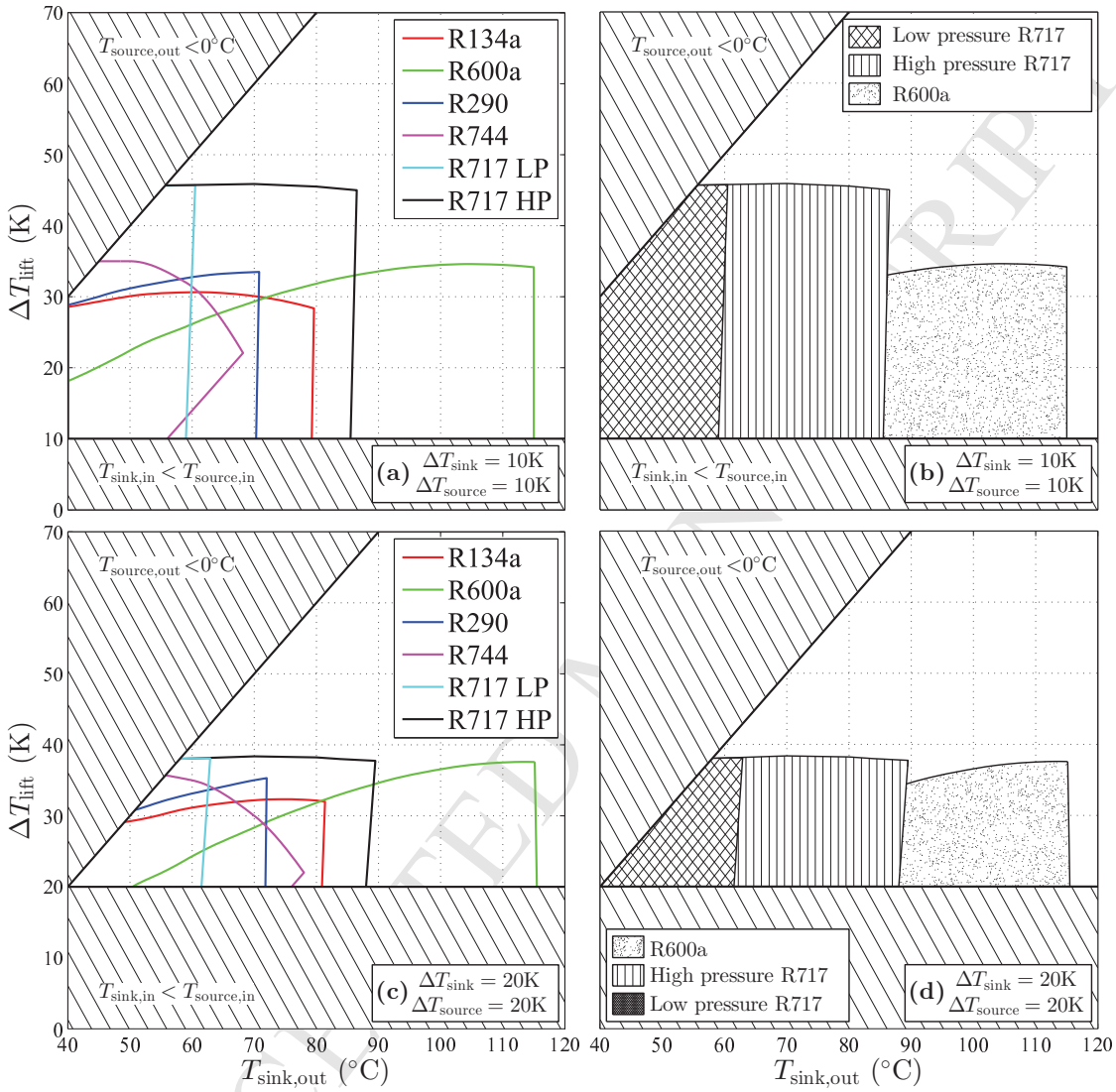

Figure 7: Compilation of working domains (a & c) and hatched areas with best available technology based on NPV (b & d) for selected heat pumps -
(a & b) ΔT_{sink}=10 K / ΔT_{source}=10 K
(c & d) ΔT_{sink}=20 K / ΔT_{source}=20 K

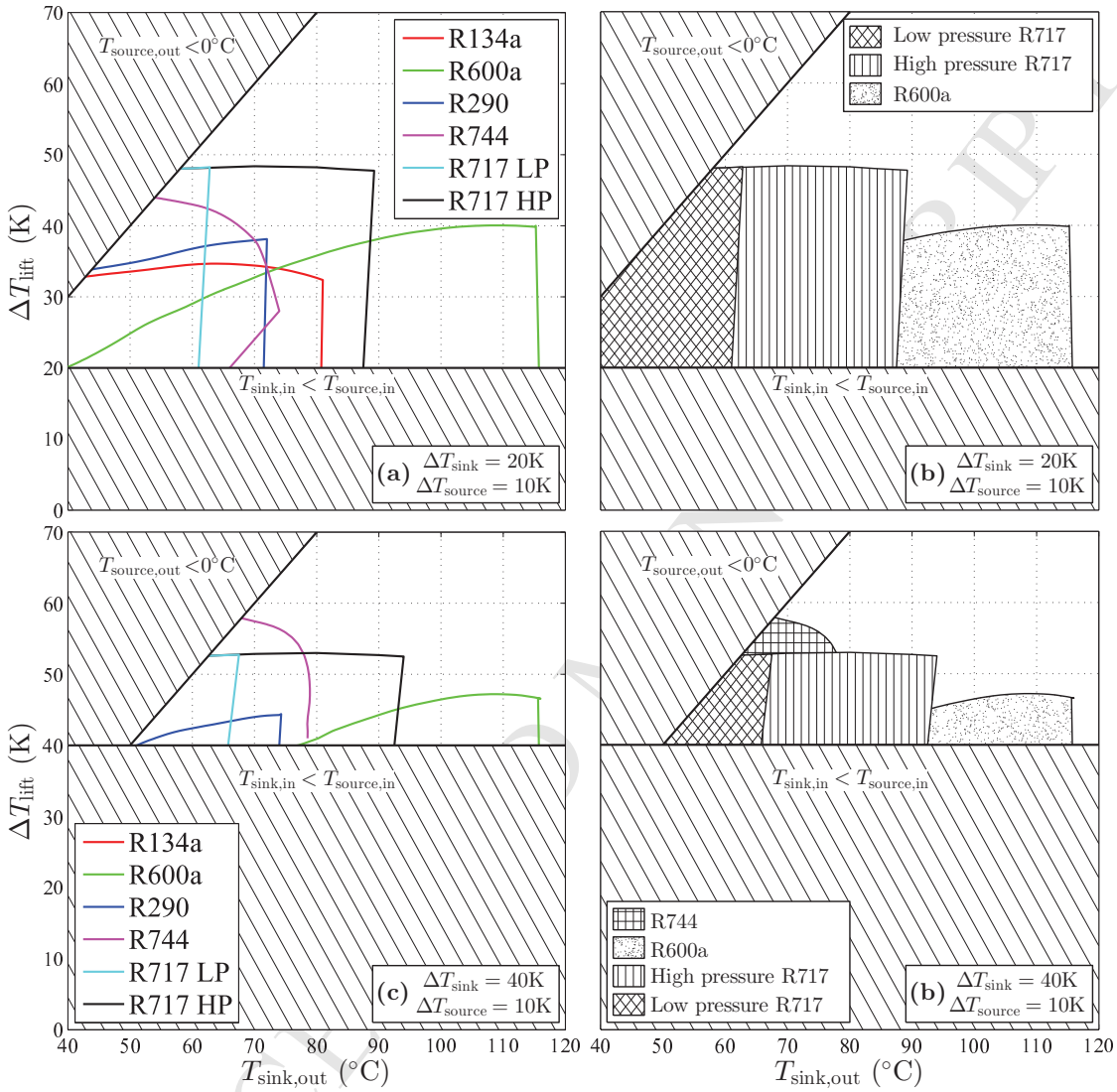

Figure 8: Compilation of working domains (a & c) and hatched areas with best available technology based on NPV (b & d) for selected heat pumps -
(a & b) ΔT_{sink}=20 K / ΔT_{source}=10 K
(c & d) ΔT_{sink}=40 K / ΔT_{source}=10 K

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Appendix A. Detailed results of the thermodynamic, heat transfer and economic model

Table A.1: Thermodynamic state points

j^{th} stream	media	$\dot{m}_j$ (kgs^{-1})	T_j (°C)	p_j (bar)	h_j (kJkg^{-1})	s_j (kJkg^{-1}K^{-1})
(1)	R717-LP	0.79	30	10.03	1499	5.37
(2)	R717-LP	0.79	120.3	27.39	1684	5.47
(3)	R717-LP	0.79	45	27.39	415	1.72
(4)	R717-LP	0.79	25	10.03	415	1.74
Sink,out	Water	11.96	60	9.96	252	0.83
Sink,in	Water	11.96	40	10.00	168	0.57
Source,in	Water	20.43	40	10.00	168	0.57
Source,out	Water	20.43	30	9.96	126	0.43

Table A.2: Heat and work load

$\dot{Q}_{\text{Evaporator}}$	855	kW	COP	6.5	-
$\dot{Q}_{\text{Condenser}}$	1000	kW	$\dot{V}_1$	467	m^{3}h^{-1}
$\dot{W}_{\text{Comp.}}$	146	kW	V_{reciever}	0.062	m^3
$\dot{W}_{\text{Comp.,el}}$	153	kW			

Table A.3: Heat exchanger dimensions, heat transfer and pressure drop coefficients and working fluid charge estimation

Evaporator:			Condenser:		
ΔP_{WF}	0.2	kPa	ΔP_{WF}	0.2	kPa
ΔP_{water}	3.8	kPa	ΔP_{water}	3.7	kPa
$U_{\text{evaporation}}$	1.483	kWm^{-2}K^{-1}	$U_{\text{superheat}}$	0.466	kWm^{-2}K^{-1}
$U_{\text{superheat}}$	0.380	kWm^{-2}K^{-1}	$U_{\text{condensation}}$	3.696	kWm^{-2}K^{-1}
			$U_{\text{subcooling}}$	1.494	kWm^{-2}K^{-1}
N	597	-	N	345	-
A	73.2	m^2	A	44.1	m^2

Table A.4: PEC, TCI, total investment and operational cost

Break down of TCI costs				Break down of FC, OMC, NPV and PBP		
Component	X_Y	PEC (€)	TCI (€)		PV (€)	AC (€)
Compressor	467 m^3h^{-1}	22481	93521	TCI_{HP}	300228	-
Elec. motor	146 kW	7533	31337	TCI_{NG}	0	-
Evaporator	73.2 m^2	24919	103663	FC_{HP}	462937	44303
Condenser	44.1 m^2	16092	66942	FC_{NG}	1435810	137407
Receiver	0.062 m^3	1146	4767	OMC_{HP}	60052	5747
				OMC_{NG}	0	0
				PV_{HP}	823217	-
				PV_{NG}	1435810	-
Total HP investment (€)			300228	NPV (€)	612593	-
				PBP (year)	-	3.437

Appendix B. Technical and economical operating boundaries for 20/10 and 20/20

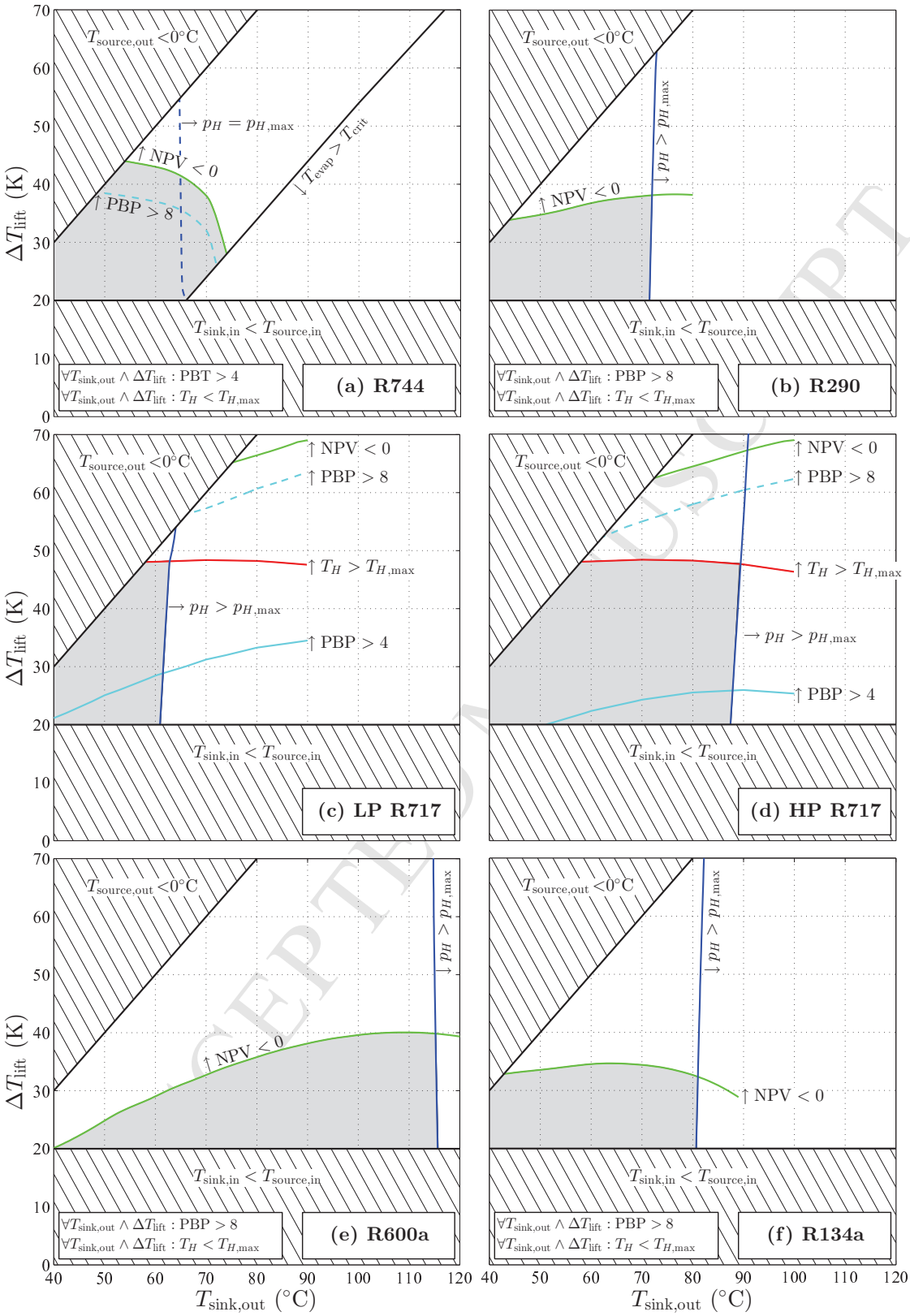

Figure B.1: Working domains for five different heat pumps - ΔT_{sink}=20 K / ΔT_{source}=10 K

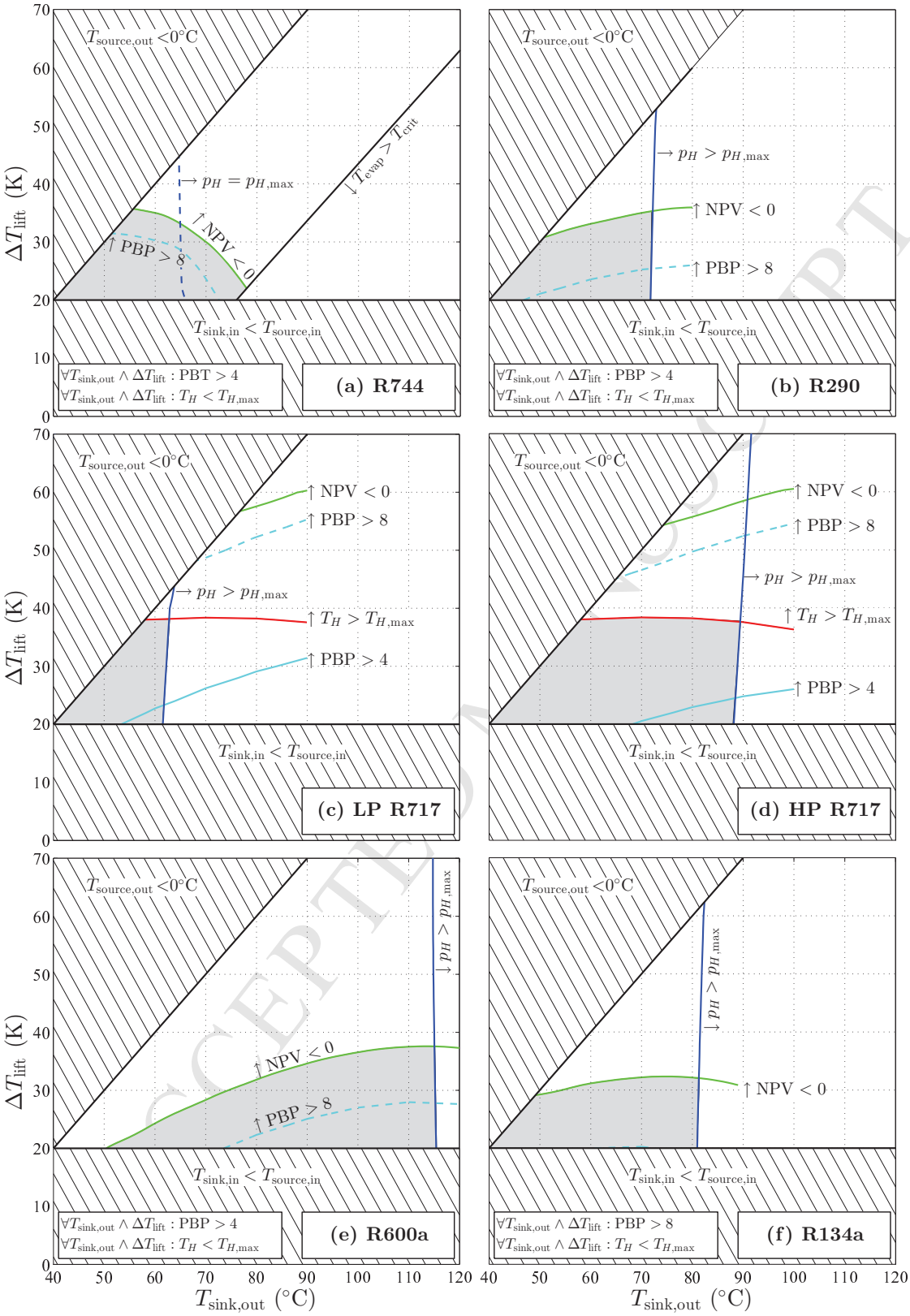

Figure B.2: Working domains for five different heat pumps - ΔT_{sink}=20 K / ΔT_{source}=20 K