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# TECHNICAL AND ECONOMIC WORKING DOMAINS OF INDUSTRIAL HEAT PUMPS: PART 1 - 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 is carried out considering the constraints of available refrigeration equipment and a requirement of a positive Net Present Value of the investment. The considered sink outlet temperature range is from 40 °C to 140 °C, but for the heat pumps considered in this paper, the upper limit is 100 °C. Five heat pumps are studied. For each set of heat sink and source temperatures the optimal solution is determined. At low sink temperature glide R717 heat pumps show best performance, while at higher sink glide transcritical R744 may become important. In a second paper, the results of the VCHP are compared to a similar study considering the ammonia-water absorption-compression heat pump.

**Key Words** Industrial heat pumps, working domain, economic evaluation, natural working fluids

## 1. INTRODUCTION

Large vapour compression heat pumps (VCHP) are increasingly used in industry, and for heat production in district heating networks. The type and configuration of the installed heat pumps depend on the local legislation as well as the layout of the sink/source process streams. For industrial applications the range of sink temperatures are typically between 50 °C and 90 °C. This sink temperature span is likely due to limitations for the heat pump technology, 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. The main part of the investigated working fluids, are banned today (or will be shortly) throughout most of Europe, when considering large industrial scale systems. Of the natural alternatives equipment limitations are significant. 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).

This study seeks to reveal the possible operation areas for several natural working fluids when considering temperature lift and sink temperatures, using the one stage vapour compression heat pump cycle. Four heat pumps utilising natural working fluids will be compared in terms of technical, thermodynamic and economic constraints, in order to include the heat pump performance and investment into the consideration. One HFC working fluid (R134a) is included in the study for comparison of the feasibility of natural working fluids. The considered sink temperature range is from 40 °C to 140 °C. In Jensen et al. (2014), the ammonia-water absorption-compression heat pump is studied and compared to the results of VCHP of this paper.

## 2. METHOD

Examination of the working domain is carried out for five single stage VCHP. The comparison is carried out without considering possibilities for improved performance by e.g. internal heat exchangers. A model of each heat pump is implemented in Engineering Equation Solver (EES, 1992).

The five heat pumps are compared using both economical and technical constraints. The technical constraints considered are typically 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 five 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 is considered.

## 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 is assumed to be pure water, which is pressurized to avoid phase change. Moving boundary models with pinch point temperature difference are used to model heat exchange with both sink and source media (Nellis and Klein, 2009). A principle sketch of a VCHP, and a temperature - heat load diagram for an azeotropic working fluid, are presented in Fig. 1. In the condenser, the working fluid is sub-cooled until it reaches the pinch temperature difference with the sink entrance.

The performance of the VCHP is calculated using constant efficiencies for compressor and electrical motor, as well as 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 Neksa et al. (1998) and further discussed in Cecchinato et al. (2010). For all the considered temperature configurations of transcritical heat pumps, the heat rejection pressure allowing the highest COP is determined and used.

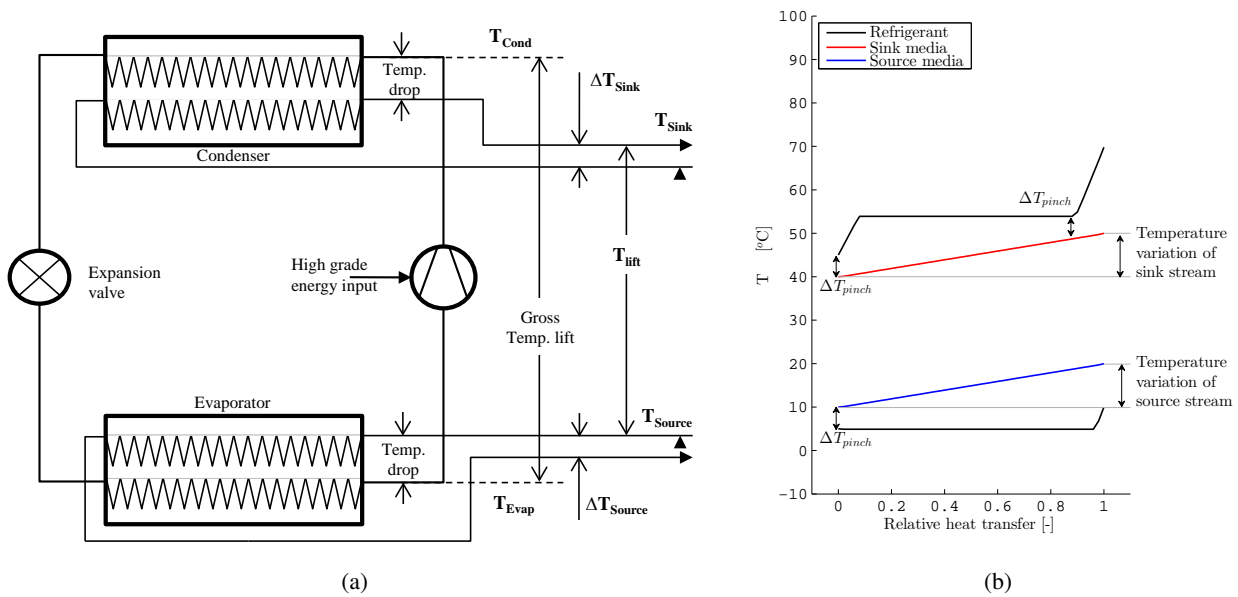


Figure 1. (a) Principle sketch of the vapour compression HP, (b) temperature - heat load diagram for condenser and evaporator where temperature variation of both sink and source is 10 K

Table 1. Operating point and performance of VCHP 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

## 2.2. Estimation of plate heat exchanger area and pressure drop

Heat exchange processes are important in any VCHP 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 are implemented and used in moving boundary models of the heat exchangers.

Chevron type plate heat exchangers are considered, as they appear to be cost efficient and typically used in such systems today. The correlations used for the analysis are presented in tabel 2, where as the dimensions are for a fixed type (SWEP International AB, 2013).

## 2.3. Compressors and operating conditions

In order to achieve the considered temperatures using vapour compression heat pumps, specially designed com-

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

Component	Media		Heat transfer	Pressure drop
Condenser	H <sub>2</sub> O		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 <sub>2</sub> O		Martin (1996)	Martin (1996)
Evaporator	Rxxx	two-phase:	Yan and Lin (1999)	Yan and Lin (1999)
		vapour only:	Martin (1996)	Martin (1996)

pressors should be 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 some 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, the compressor discharge temperatures are limited to 180 °C.

Compressors from large international manufactures were investigated, where both price and operation limits were available. Five different types have been identified according to different working fluid properties, flammability and availability. The data for these are presented in Table 3. Compressor types 1 - 3 are similar,

Table 3. Available compressor technology with current operating limits

Working fluid	Type	Pressure limit	Lubrication max. temp.	Capacity (1500 RPM)
	–	Bar	°C	m <sup>3</sup> /h
R134a	1	28	180	5-280
R290	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

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 has not been constrained to individual compressor technologies, but due to data availability considering the compressor cost, the below results represent reciprocating piston compressors.

## 2.4. Economic evaluation

The economic evaluation of the heat pumps is 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 the components. As discussed in section 2.3, coherent and comprehensive data for specific components are not straightforward to uncover. To aggregate from the data at hand, it is assumed that:

- Purchased Equipment Cost (PEC) for an open type compressor is 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 is dependent on the shaft power.
- PEC for a heat exchanger is a function of the heat exchange area.
- PEC of an expansion valve is neglected.
- Total capital investment of a component is 4.16 higher than PEC of the component. This is done to account for additional cost related to new investment (Bejan et al., 1996) at an existing facility.
- Electricity and natural gas prices correspond to the market cost for industrial consumers according to Danish Energy Agency (2013).
- For the comparison with natural gas burner investment costs are neglected, as the heat pump may replace

existing installations.

- Source heat is readily available as a process stream.
- Interest and inflation rate is fixed (7% and 2% respectively) and the plant is assumed to operate for a technical lifetime of 15 years.
- Cost correlations are assumed to be valid for heat pump capacities between 100 kW and 2 MW. Restrictions are due to data availability for compressor and heat exchanger correlations.

The component prices are based on prices from intermediate Danish trade business such as H. Jessen Jürgensen A/S (2013) and individual manufactures, eg. Johnson Controls (2013) for R717 HP equipment.

### 3. RESULTS

The working domains of the individual heat pumps are calculated for a given amount of operating hours and heat pump capacity. A parametric variation of the two quantities is presented in Fig. 2a. Contours of two

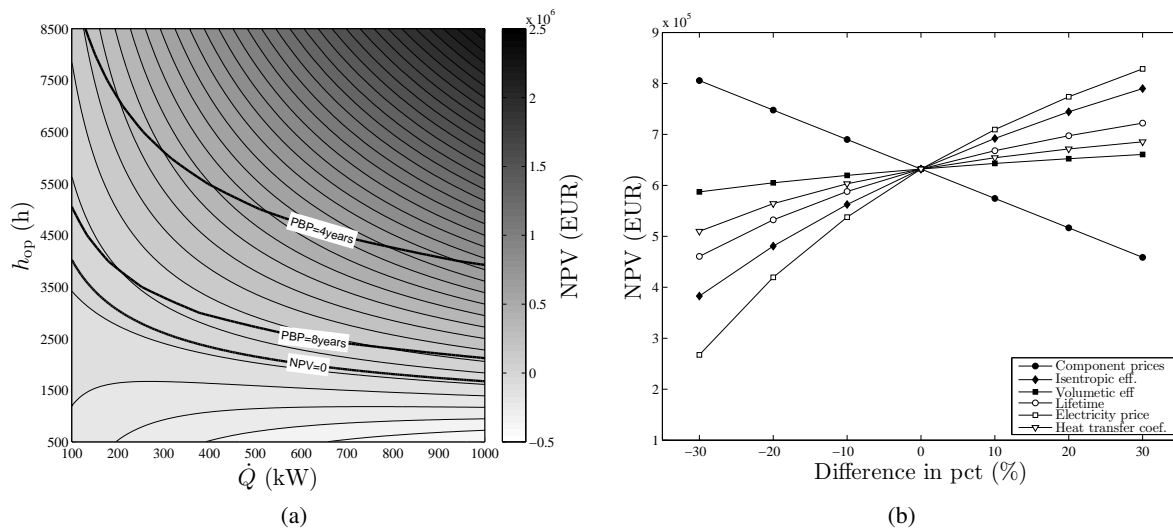


Figure 2. Key parameters influence to cost of heat - R717-HP heat pump

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 plant. All three economic constraints, NPV=0 as well as PBT=4 years and PBT=8 years will be printed in each plot, allowing the reader to decide the appropriate economic constraint.

It is worth noticing, that all solutions in Fig. 2a represent COP=5.1 and VHC=5.9 MJ/m<sup>3</sup>, which are within the recommended levels defined by Brunin et al. (1997). Although within the limits, a large part of the solutions are 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 detail.

Based on the results of Fig. 2a it is decided to investigate the heat pump working domains, by considering a case of 3500 operating hours yearly, and  $\dot{Q}_{HP} = 1000$  kW. Fig. 2b shows a parametric variation of other significant assumptions at this operation point. The assumptions for electricity price and isentropic efficiency are the largest causes for changed heat cost and NPV. The observed sensitivity is assumed not to limit the validity of the comparisons between the technologies presented below.

The working domain of the five heat pumps are presented in Fig. 3 to 5 for three different sink and source temperature glides. Fig. 3 presents the case:  $\Delta T_{source}=10$  K /  $\Delta T_{sink}=20$  K. Fig. 4 is with increased sink glide:  $\Delta T_{source}=10$  K /  $\Delta T_{sink}=40$  K, whereas Fig. 5 is with increased source glide:  $\Delta T_{source}=20$  K /  $\Delta T_{sink}=20$  K. The hatched areas of the individual plots are areas where the considered technology is not appropriate. If  $T_{source,out}$  is below 0 °C another source media should be considered. If  $T_{sink,in} < T_{source,in}$  heat should be exchanged directly between the two streams before a heat pump solution is considered.

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

Each individual plot shows the constraints for the specific working fluid, components and economic restrictions. Fig. 3 shows that a combination of the four natural working fluids allow a large working domain. The NPV

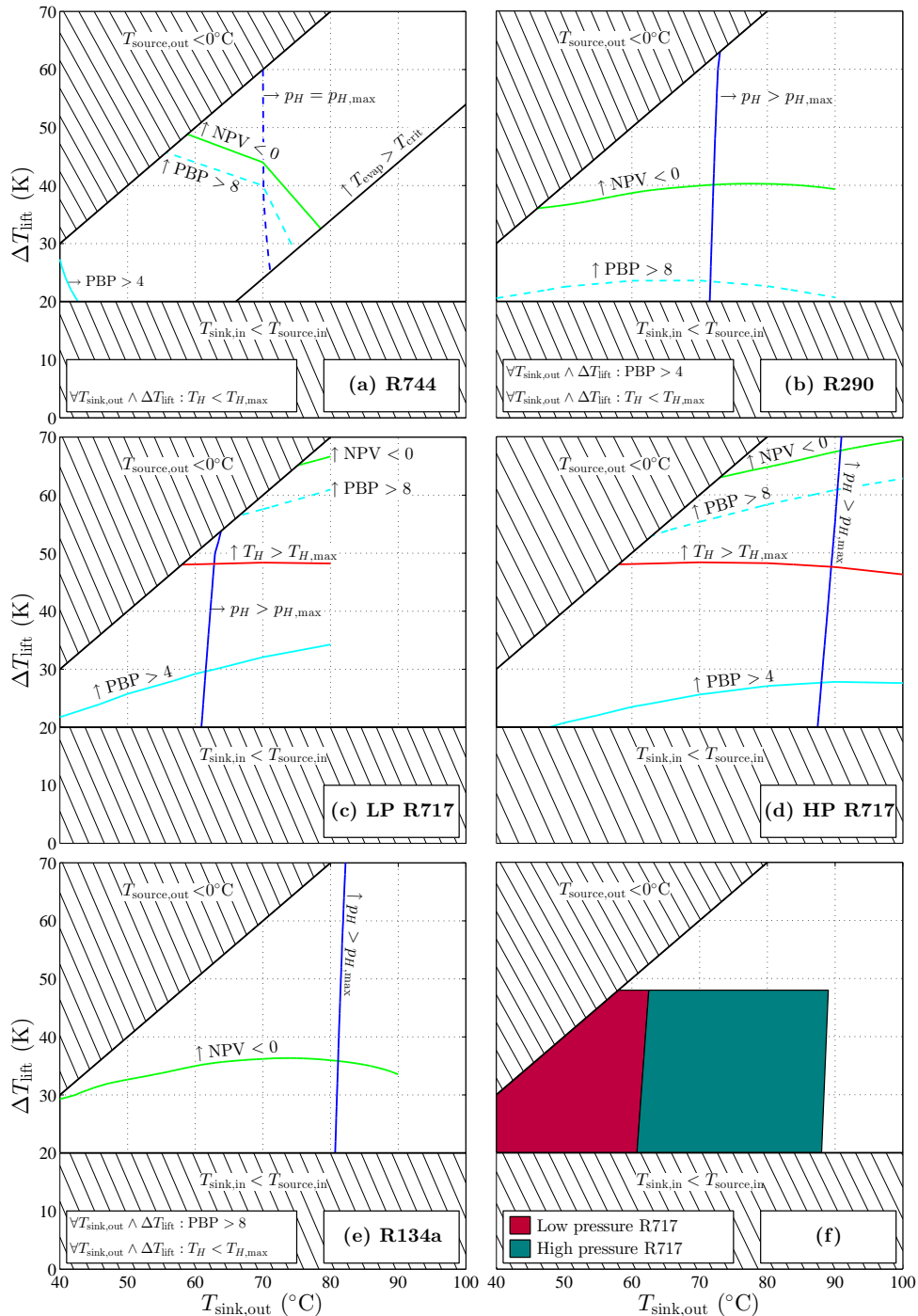


Figure 3. Working domains for five different heat pumps -  $\Delta T_{\text{source}}=10 \text{ K} / \Delta T_{\text{sink}}=20 \text{ K}$

constraints of R717 LP and R717 HP include almost all of the considered temperature span, but the technology is restricted by high compressor discharge temperatures, as well as the allowable high pressure. For the three remaining heat pumps the discharge temperatures are below the limit in all cases. Specifically for Fig. 3a the critical pressure of the working fluid limits the working domain, as two-phase evaporation is not possible in some of the considered combinations. 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.

Fig. 3f highlights the economically best solutions in the domain. For the sink and source glide considered in Fig. 3 it is shown that R717 LP and R717 HP are the optimal solutions in all of the possible sink and lift combinations. All solutions presented in Fig. 3f are with a positive net present value.

For Fig. 4 and 5 similar plots are presented. At higher sink temperature glide (Fig. 4), the high discharge temperature for R717 decreases the area of feasible solutions. At the same time, transcritical R744 is economically

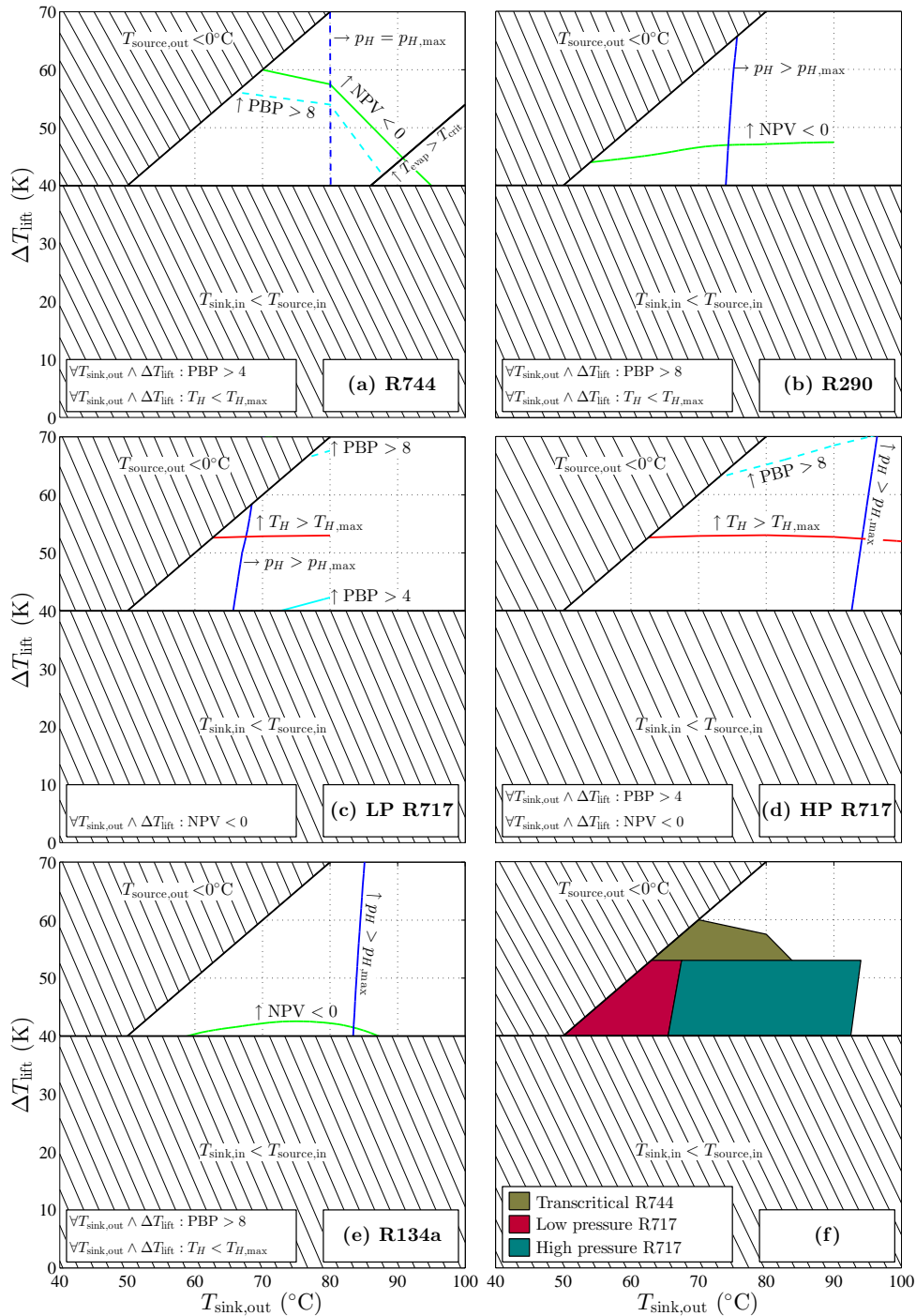


Figure 4. Working domains for five different heat pumps -  $\Delta T_{\text{source}}=10 \text{ K} / \Delta T_{\text{sink}}=40 \text{ K}$

feasible at even higher temperature lifts. From considering Fig. 4f it is found that R744 is the optimal solution, in the areas where R717 is constrained by discharge temperature. As can be seen from Fig. 4a - 4e none of the considered heat pumps allow a PBT below 4 years.

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. From Fig. 5 it is clear that the working domains of the heat pumps are closely related to those of Fig. 3. Because of the increased pressure ratio between evaporator and condenser, the R717 discharge temperature constraint is moved below a temperature lift of 40 K. Similarly, the economic constraints are moved towards a lower temperature lift.

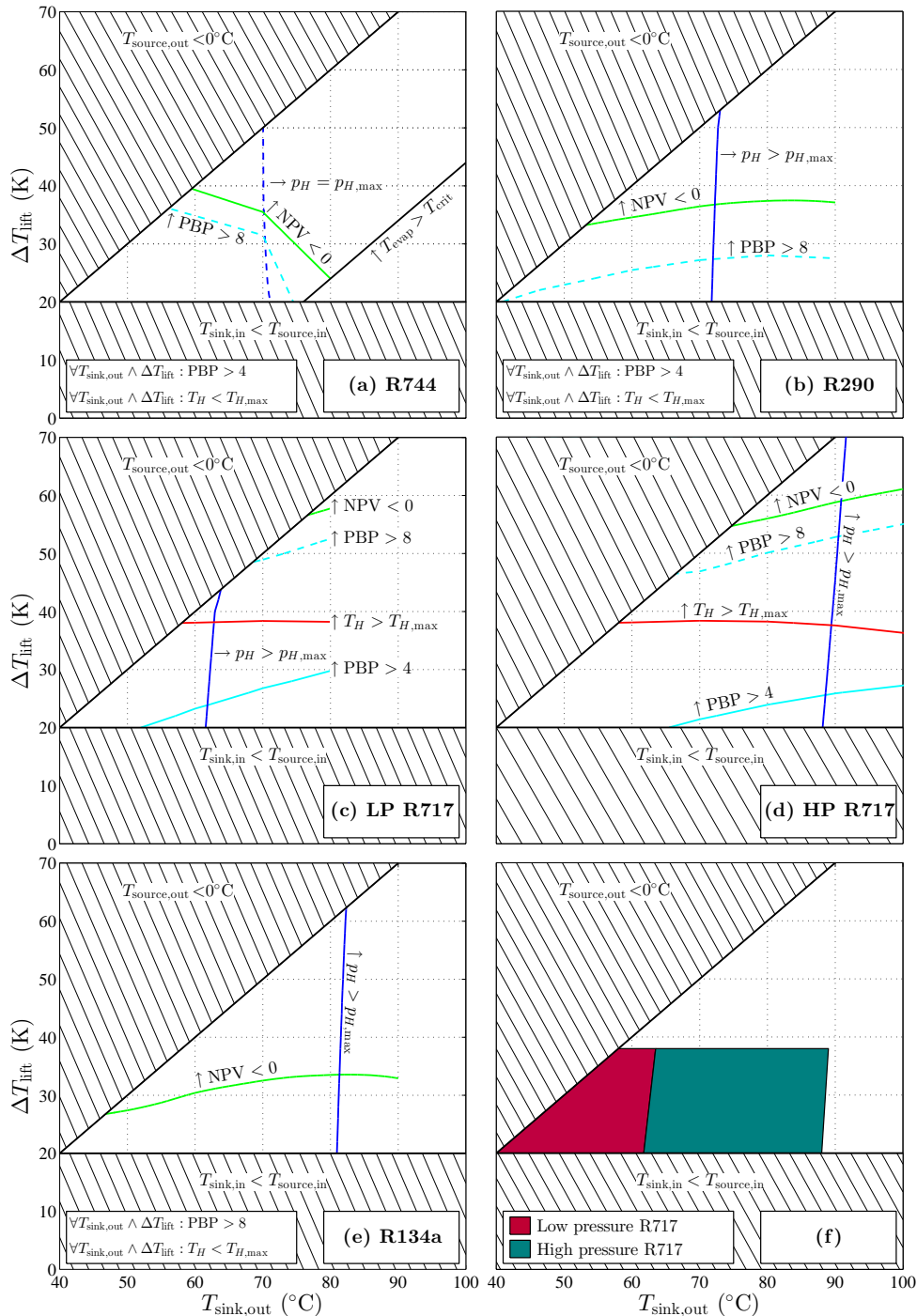


Figure 5. Working domains for five different heat pumps -  $\Delta T_{\text{source}}=20\text{ K} / \Delta T_{\text{sink}}=20\text{ K}$

#### 4. DISCUSSION

The economic correlations used in the investigation are based directly on data from a Danish supplier. Reasonable price correlations are not easily found. Producers and intermediate trade businesses seem reluctant to reveal the true prices. Unlisted component cost or sealed price agreements result in what appears to be uncompetitive heat pump systems, as the heat cost is increased. From Fig. 2b a 25 % increased NPV is found for a 30 % reduction in component cost. Such a change would significantly increase the feasible area for all 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 between the types, but only between heat pumps and natural gas installations.

The results presented here stem solely from simulation results by verified models but are not validated with experimental results. 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.

## 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 show higher detail in the feasibility of investigated solutions than the use of semi-economic parameters such as COP and VHC.

All of the five considered heat pumps show working domains where net present value is positive, when compared to the fuel cost of a natural gas burner. Three of the five heat pumps are advantageous when considering all of the presented constraints. The three heat pumps are LP R717, HP R717 and transcritical 744.

## ACKNOWLEDGEMENTS

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## NOMENCLATURE

### Abbreviations

COP coefficient of performance  
HP high pressure  
LP low pressure  
NPV net present value  
PBP pay back period  
VCHP vapour compression heat pump  
VHC volumetric heating capacity

### Symbols

$p$  pressure (Bar)  
 $\dot{Q}$  heat flow rate (kW)  
 $T$  temperature (K)

### Greek symbols

$\Delta$  difference

### Subscripts & Superscripts

H high  
max maximum

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