QUICK SELECTION OF HEAT PUMP TYPES AND OPTIMIZATION OF LOSS MECHANISMS

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Abstract: Making a rough estimation of performance for conventional vapor compression and vapor recompression heat pumps is straight forward: Dividing the Carnot efficiency by 2 results in a reasonable estimate. Still, actual performance of actual heat pumps could easily vary to a large extent.
With new and innovative heat pumps the discrepancies between the rough estimate and actual performance might be even larger as the Carnot efficiency is not the upper limit anymore due to the use of temperature glides. Lack of a simple method to determine the approximate performance of a heat pump will hinder the implementation of these novel types in process industry.
In this research it is shown that, for mechanical heat pumps, making use of the available temperature glide increases performance and reduces the payback period. While at low glides, heat driven absorption heat pumps show better pay back times, mechanical heat pumps with large glides show to be more effective even at high temperature lifts. Due to better performance, mechanical heat pumps are able to achieve better economical results over their technical life time although they can have a longer payback period.

Key words: heat pumps, industry, cost, temperature glide, temperature lift

1 INTRODUCTION

Heat pumps generally have a significant efficiency effect on the performance of processes in industry. Heat pumps are becoming more and more interesting for industry due to increasing energy cost and limitations on CO_2 emissions. With the arising of all new different heat pump types it is not always straight forward to select a certain type of heat pump for a specific industrial process application.
Allen and Hamilton (1983) created a steady state model for reciprocating water chillers to evaluate performance at full and part load. Hamilton and Miller (1990) developed a model by equation-fitting manufacturers catalog data of individual heat pump components along with thermodynamic relationships for the working fluid. It requires internal refrigerant pressures and temperatures.
This paper considers heat pump performance independent of refrigerant properties by starting from the fairly simple Carnot relation. This relation is worked out to include temperature glides (resulting in Lorentz COP), temperature driving forces and isentropic efficiency of the compressor.
Final goal of this paper is to present a performance map in which it becomes obvious which heat pump to select under what conditions. The focus is on compressor driven heat pumps. A comparison is made between vapor compression/recompression, compression-resorption and transcritical heat pumps. Absorption heat pumps and conventional boilers are included as reference.

2 HEAT PUMP CLASSIFICATION

Heat pumps can be divided in 3 types: mechanically driven, heat driven and heat transformers. Mechanically driven heat pumps can be found, among others, in the following types:

- Subcritical vapor compression heat pump
- Vapor recompression heat pump
- Transcritical vapor compression heat pump
- Compression-resorption heat pump
- HIDiC
- Thermoacoustic heat pump (linear motor driven)

The following heat driven heat pumps are most popular:

- Absorption heat pump
- Adsorption heat pump
- Thermoacoustic heat pump

The heat driven heat pumps can also be applied as heat transformers. More heat pump cycles exist than mentioned above, however, they are often in a very early research state or not yet accepted by industry.

2.1 Mechanically driven heat pumps

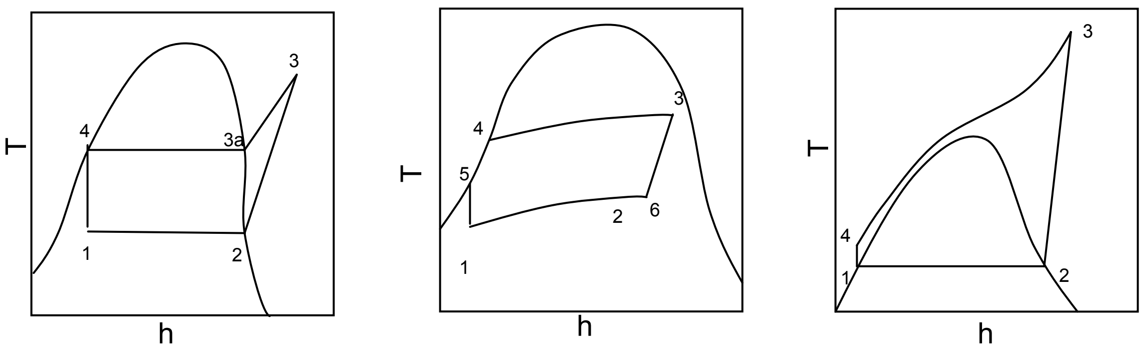

Figure 1: Typical temperature-enthalpy diagrams for a) subcritical vapor compression, b) compression-resorption and c) Transcritical heat pumps

2.1.1 Subcritical vapor compression heat pump

A vapor compression heat pump consists of 4 main parts: evaporator, compressor, condenser and expansion valve. For pure working fluids the subcritical vapor compression heat pump has no glide (temperature difference between inlet and outlet of a heat exchanger) over both condenser and evaporator. Superheat is required at state point 2 (see Figure 1a) to protect the compressor by making sure the fluid is fully evaporated. Best application is in areas where a low temperature lift is required and where the temperature glide of the heat source and heat sink is (almost) 0 K, which usually is the case in the reboiler and condenser at a distillation process of close boiling mixtures with very pure top and bottom products. Higher temperature lifts can be achieved when using multistage heat pump systems.

2.1.2 Vapor recompression heat pump

Vapor recompression heat pumps are open loop heat pumps. They are generally applied with an (almost) pure fluid; therefore the temperature glide over the condenser is almost 0 K. The overhead vapors are generally compressed and then condensed while heating the reboiler. In some cases the bottom flow is flashed, heat is picked up in the column condenser and then compressed to the pressure of the bottom stream of the column (bottom flash).

2.1.3 Compression-resorption heat pump

Compression-resorption heat pumps work with a binary mixture; most often ammonia/water. As a mixture generally has a boiling trajectory, there will always be a temperature glide over desorber and resorber (see Figure 1b). A pure vapor can be compressed, as well as a liquid/vapor mixture if the compressor is capable of compressing a wet mixture. Usually a vapor/liquid separator is placed at the end of the desorber; thus a saturated vapor is compressed and liquid is pumped up to higher pressure.

2.1.4 Transcritical vapor compression heat pumps

A transcritical vapor compression heat pump consists at least out of an evaporator, compressor, gas heat exchanger and expansion valve. A temperature profile exists over the gas heat exchanger, however, due to the supercritical nature of this part of the heat pump, no phase change takes place. The evaporator, where the fluid is subcritical, has zero temperature glide and a phase change occurs. Transcritical vapor compression heat pumps can achieve relatively high lifts with reasonable efficiency as long as temperature glides match the glides of the source and sink. Transcritical CO_2 heat pumps are commonly applied for space heating and hot water generation, as the critical pressure and temperature (31 °C) of CO_2 makes the heat pump suitable for the application. For the application in distillation systems, where the minimum source temperature is 90 °C or more, a fluid with a higher critical temperature has to be found as the source temperature is limited to the critical temperature of the fluid. The cycle is depicted in the temperature-enthalpy diagram in Figure 1c.

2.1.5 Heat integrated distillation columns (HIDiC)

In certain cases it is possible to split the process into two parts. An example is a distillation column where the rectifier and stripping section can be split from each other and exchange heat. In order to exchange heat the rectification section has to work at higher temperature and therefore higher pressure than the stripping section. This is reached by placing a compressor between the top of the stripping section and an expansion valve at the bottom of the rectification section. Possible advantage compared to compression-resorption heat pumps is the lack of one temperature driving force. The operating principle of a HIDiC is shown in Figure 2.

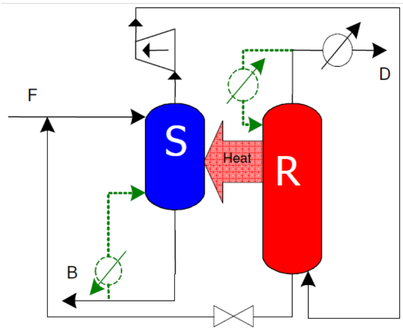

Figure 2: HIDiC working principle (de Rijke et al., 2007)

2.1.6 Linear motor driven Thermoacoustic heat pump

Thermoacoustic heat pumps are currently in the development stage. Since 2000, ECN is working on its development in the Netherlands. Except for the linear motor, the thermoacoustic heat pump has no moving parts and therefore it is expected that this type of heat pump will have lower maintenance costs compared to other mechanically driven heat pumps.

2.2 Thermally driven heat pumps

Among thermally driven heat pumps the absorption heat pump is most popular. This type of heat pump is able to reduce energy costs at high temperature lifts, but are less favorable than a mechanically driven heat pump. The heat driven heat pumps are characterized by a lower COP than mechanically driven heat pumps, but they practically don't require electric or mechanical energy to be driven, thereby making use of the lower prices for heat or fuel.

3 COMPARISON OF THE PERFORMANCE OF DIFFERENT CYCLES

While the sections above already indicate globally where it is best to apply a certain type of heat pump, it speaks in relative terms, i.e. low/high temperate lift, a certain amount of glide. The goal of this section is to provide a global "map" of where to apply a certain heat pump. This will be done on the basis of energy efficiency and on the basis of expected costs. Although this map will try to give a picture as good as possible, it is based on general assumptions and should only be used as a guideline, not as exact math.
All heat pumps perform best when working with low temperature lifts. The most simple and most developed heat pump types are vapor compression and vapor recompression heat pumps. Although they are the most developed, their temperature lifts are limited. Like vapor compression and vapor recompression heat pumps, HIDiCs and compression resorption heat pumps reach their highest efficiency at low temperature lifts. However, their relative advantages compared to the more conventional vapor compression and vapor recompression heat pumps become larger when temperature lift increases.

3.1 Comparison of advanced heat pump cycles with vapor compression heat pumps

Heat driven heat pumps can become effective when the COP of a compressor driven heat pump becomes small.

$$COP_H > COP_E \frac{\eta_E}{\eta_H} \tag{1}$$

Where η_E, η_H, COP_E and COP_H are the grid efficiency, boiler efficiency, electric COP and heating COP. The electrical COP_E can be defined as

$$COP_E = \eta_{Carnot} COP_{Carnot} \tag{2}$$

Where η_{Carnot} is the efficiency relative to the Carnot COP. When there is no temperature glide, i.e. for a vapor compression heat pump, COP_{Carnot} is given by

$$COP_{Carnot} = \frac{T_h}{T_h - T_l} \tag{3}$$

With T_h and T_l the sink and source temperatures. Determining the Carnot COP of a HIDIC, compression-resorption or transcritical heat pump is less straight forward and depends on the properties of the working fluid. For a transcritical heat pump the evaporator temperature is constant, while for the gas heat exchanger operating in the supercritical region the temperature changes along the heat exchanger. Starting from the Carnot relation again,

$$COP_{Carnot} = \frac{T_h}{T_h - T_l} = \frac{1}{1 - \frac{T_l}{T_h}} \tag{4}$$

The COP_{Carnot} for a compression resorption cycle is similar to the COP of the Lorentz cycle. Following the definitions for source and sink temperatures,

$$COP_{Lorentz} = \frac{\frac{\Delta T_{h,glide}}{ln\left(\frac{T_h}{T_h - \Delta T_{h,glide}}\right)}}{\frac{\Delta T_{h,glide}}{ln\left(\frac{T_h}{T_h - \Delta T_{h,glide}}\right)} - \frac{\Delta T_{l,glide}}{ln\left(\frac{T_l + \Delta T_{l,glide}}{T_l}\right)}} \quad (5)$$

With $\Delta T_{l,glide}$ and $\Delta T_{h,glide}$ the available glides in source and sink temperatures. Instead of using the definition above, Yilmaz (2003) suggests to use an approximate equation

$$COP_{Lorentz} = \frac{2T_h - \Delta T_{h,glide}}{2T_h - \Delta T_{h,glide} - 2T_l - \Delta T_{l,glide}} \quad (6)$$

Simplifying this equation by taking equal glides for both the low and high temperature side results in

$$COP_{Lorentz} = \frac{2T_h - \Delta T_{glide}}{2\Delta T_{lift} - 2\Delta T_{glide}} \quad (7)$$

With

$$\Delta T_{lift} = T_h - T_l \quad (8)$$

The Carnot and Lorentz COP's for different glides result in the graph shown in Figure 3.

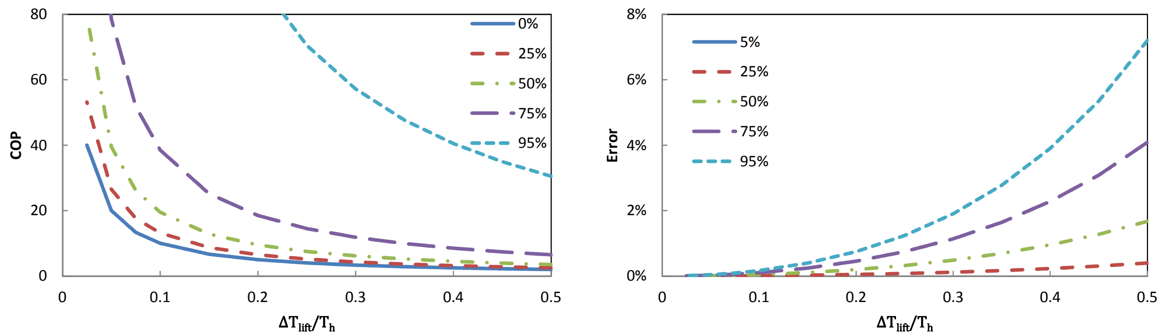

Figure 3: COP as function of dimensionless lift and dimensionless glide (left) and error made by implementation of Yilmas method for determining Lorentz COPs. The lines indicate the glide to lift ratio in percent.

The error introduced when eq. (7) is used instead of the exact formulation, eq. (5), is dependent on the inverse of Carnot and glide to lift ratio and is illustrated in the right side of Figure 3. The deviation between the method of Yilmaz (2003) and by using the definitions of source and sink temperature varies between 0 % (at 0 % glide and/or 0 % lift) and 7 % (at 95 % glide and 50 % lift). Although not 100 % accurate, the simplification by Yilmaz allows obtaining relatively simple relations between efficiencies due to the disappearance of the natural logarithm. Division of Eq. (7) by Eq. (3) simplifies to

$$\frac{COP_{Lorentz}}{COP_{Carnot}} = \left(1 - \frac{1}{2}\frac{\Delta T_{lift}}{T_h}\frac{\Delta T_{glide}}{\Delta T_{lift}}\right)\frac{1}{1 - \frac{\Delta T_{glide}}{\Delta T_{lift}}} \quad (9)$$

The results for different lift-to-sink temperature ratios are given in Figure 4a, illustrating that heat pumps that can make use of glides may lead to significant improvements in comparison to conventional heat pumps. Comparing Figure 4 also shows that the potential of transcritical heat pumps is significantly lower than the potential of compression-resorption heat pumps.

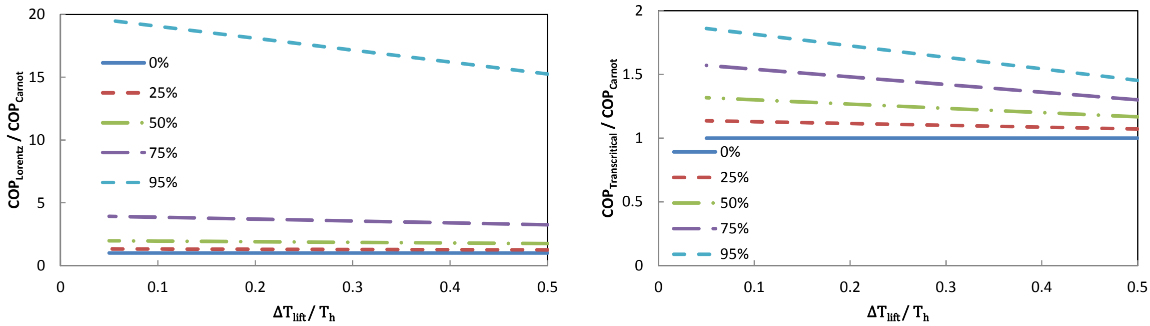

Figure 4: a) Ratio of Lorentz COP and Carnot COP and b) Ratio of Transcritical COP and Carnot COP, both as a function of dimensionless temperature lift with the glide to lift ratio (in percent) as parameter

3.1.2 Transcritical heat pumps

For Transcritical heat pumps, the source temperature glide is 0 K, thus equations (7) and (9) are replaced by eqs. (10) and (11). The behavior of eq. (11) is shown in Figure 4b.

$$COP_{transcritical} = \frac{2T_h - \Delta T_{h,glide}}{2T_h - \Delta T_{h,glide} - 2T_l} = \frac{2T_h - \Delta T_{glide}}{2\Delta T_{lift} - \Delta T_{glide}} \tag{10}$$

$$\frac{COP_{transcritical}}{COP_{Carnot}} = \left(1 - \frac{1}{2}\frac{\Delta T_{lift}}{T_h}\frac{\Delta T_{glide}}{\Delta T_{lift}}\right)\frac{1}{1 - \frac{\Delta T_{glide}}{2\Delta T_{lift}}} \tag{11}$$

3.1.3 Absorption heat pumps

The temperature glide in absorption heat pumps is close to 0 K. Following Thoser and James (1997), the ideal absorption heat pump COP is given by

$$COP_{ahp} = 1 + \frac{T_l}{T_h} \tag{12}$$

4 LOSS MECHANISMS

In heat pumps, there are several sources of efficiency losses:

- Temperature driving forces
- Compressor efficiency
- Pressure drop
- Superheat
- Throttling losses
- Mismatch between process and heat pump fluids

In this section an attempt is made to quantify the losses for the temperature driving forces and compressor efficiency.

4.1 Temperature driving forces

When heat transfer occurs, temperature driving forces are required. These driving forces don't allow a heat exchanger design to be 100 % effective. For instance, a hot stream entering a heat exchanger with temperature T_4 is being cooled down by a second flow with a temperature of T_1. While the hot stream will never reach T_1, the cold stream will never reach T_4, unless the heat exchanger is infinitely large. See Figure 5.

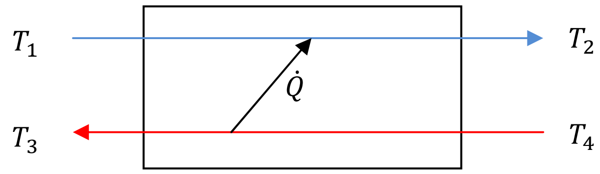

Figure 5: Temperature driving forces in heat exchangers

4.1.1 Vapor compression cycles

The temperature driving forces (TDF) affect the total efficiency of the heat pump. In eq. (13) the TDF are included in the COP.

$$COP_{TDF} = \frac{T_h + \Delta T_{h,driving}}{\left(T_h + \Delta T_{h,driving}\right) - \left(T_l - \Delta T_{l,driving}\right)} = \frac{1}{1 - \frac{T_l - \Delta T_{l,driving}}{T_h + \Delta T_{h,driving}}} \tag{13}$$

The efficiency loss due to the temperature driving forces can be determined. The equation gets simpler by assuming the driving forces for source and sink to be equal.

$$\eta_{Carnot}^{TDF} = \frac{COP_{TDF}}{COP_{Carnot}} = \frac{\frac{T_h + \Delta T_{driving}}{T_h + \Delta T_{driving} - T_l + \Delta T_{driving}}}{\frac{T_h}{T_h - T_l}} \tag{14}$$

Then rewriting, simplifying and back-substituting the Carnot COP results in eq. (15).

$$\eta_{Carnot}^{TDF} = \frac{COP_{Carnot} + \frac{\Delta T_{driving}}{\Delta T_{lift}}}{COP_{Carnot}\left(1 + \frac{2\Delta T_{driving}}{\Delta T_{lift}}\right)} \tag{15}$$

4.1.2 Vapor recompression cycle

The main difference between vapor compression and vapor recompression heat pumps is that vapor recompression have only one temperature driving force; usually on the high temperature side.

$$COP_{TDF} = \frac{T_h + \Delta T_{h,driving}}{\left(T_h + \Delta T_{h,driving}\right) - T_l} \tag{16}$$

With

$$\Delta T_{driving} = \Delta T_{h,driving} \tag{17}$$

and rewriting, simplifying and back-substituting the Carnot COP results in eq. (18)

$$\eta_{Carnot}^{TDF} = \frac{COP_{Carnot} + \frac{\Delta T_{driving}}{\Delta T_{lift}}}{COP_{Carnot}\left(1 + \frac{\Delta T_{driving}}{\Delta T_{lift}}\right)} \tag{18}$$

4.1.3 Compression-resorption cycle
In a similar fashion for the Lorentz efficiency,

$$COP_{TDF} = \frac{2T_h - \Delta T_{h,glide} + \Delta T_{h,driving}}{2T_h - \Delta T_{h,glide} + \Delta T_{h,driving} - 2T_l - \Delta T_{l,glide} + \Delta T_{l,driving}} \tag{19}$$

Again simplifying by assuming glide and driving forces for source and sink are equal,

$$\eta_{Lorentz}^{TDF} = \frac{COP_{Lorentz} + \frac{\Delta T_{driving}}{2\Delta T_{lift} - 2\Delta T_{glide}}}{COP_{Lorentz}\left(1 + \frac{2\Delta T_{driving}}{2\Delta T_{lift} - 2\Delta T_{glide}}\right)} \tag{20}$$

Notice that this equation reduces to the Carnot efficiency for glide = 0.

4.1.4 Absorption heat pump
Performance of the absorption heat pump when temperature driving forces are considered is given by

$$COP_{ahp} = 1 + \frac{T_l - \Delta T_{driving}}{T_l + \Delta T_{driving}} \tag{21}$$

4.2 Compressor efficiency

Carnot COP assumes isentropic work. The efficiency of the system is linearly dependent on the efficiency of the compressor.

$$COP = \frac{Q}{W}, \qquad W = \frac{W_{is}}{\eta_{is}} \rightarrow COP = \frac{\eta_{is} Q}{W_{is}} \tag{22}$$

5 ECONOMIC CALCULATION

For the economic calculation cost equations by Guthrie (1969) have been used. In this section the equations by Guthrie have been modified to European conditions by changing to SI units, to euro as valuta. Cost estimates have been done based on price data from 30-12-2010. Prices are € 65 / MWh for electricity and € 31.65 / MWh for gas. Minimum source and maximum sink temperatures are -11 2°C and 300 °C respectively. Only the investment is determined at fixed sink temperature (t_h = 150 °C) as the equipment has to be sized and temperature driving forces are a function of lift.
The following assumptions apply

- Heat pump temperature driving forces 5 % of lift, 10 % of lift, 25 % of lift.
- Compressor isentropic efficiency 60-80 %
- Boiler efficiency η_H = 85 %
- Wet Compression (no superheat losses)
- Equal temperature glide in hot and cold heat exchanger for CRHP
- Direct fired single stage absorption heat pump
- internal heat exchanger of the absorption heat pump 1.5 times larger than absorber

- Shell and tube heat exchangers with U-tubes and surface area between 19 to 465 m^2 for heat duty of 2790 kW. Design type is U-tube, design pressure is 27 bar. For absorption heat pumps stainless steel is chosen as construction material, due to corrosion problems when used with lithium bromide. This includes the generator which is built with design type is cylindrical and design pressure up to 34 bar. The other heat pump heat exchangers are built from carbon steel.
- Heat transfer coefficient: 1200 W/m^2/K
- Motor driven, centrifugal type gas compressor
- 8030 h/year of operation (92 % availability)

For a direct fired heater the installed cost is given by

$$Installed\ cost, € = 448Q^{0.85} \tag{23}$$

For shell and tube heat exchangers built of carbon steel for the mechanical heat pumps,

$$Installed\ cost, € = 6512.6A^{0.65} \tag{24}$$

For shell and tube heat exchangers built of stainless steel for the absorption heat pumps,

$$Installed\ cost, € = 10337.6A^{0.65} \tag{25}$$

And gas compressors,

$$Installed\ cost, € = 5121W^{0.82} \tag{26}$$

Cost savings for mechanical and heat driven heat pumps,

$$savings\ per\ MWh\ for\ Mech\ HP = \left(-\frac{C_{Gas}}{\eta_H} + \frac{C_E}{COP_{hp}}\right) \tag{27}$$

$$savings\ per\ MWh\ for\ heat\ driven\ HP = \left(-\frac{C_{Gas}}{\eta_H} + \frac{C_{Gas}}{COP_{ahp}}\right) \tag{28}$$

The amount to be invested per kW heating duty can be determined by selecting a simple pay-back time. The possible investment per kW heat duty increases rapidly with increasing COP up to a COP of 5. Increasing the COP further (by a factor of 4) only increases the possible investment by less than 40 %, as is shown in Figure 6a. Comparing against the possible investment cost per kW of installed compressor power, the increase is almost linear, even at high COP. This is shown in Figure 6b.

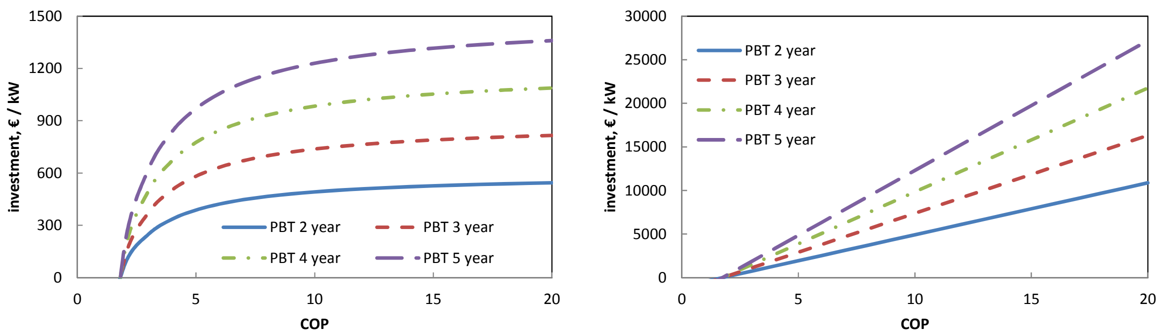

Figure 6: Maximum possible investment for mechanical heat pumps per kW of heating power (left) or compressor power (right) as function of COP. Payback time is shown as a parameter

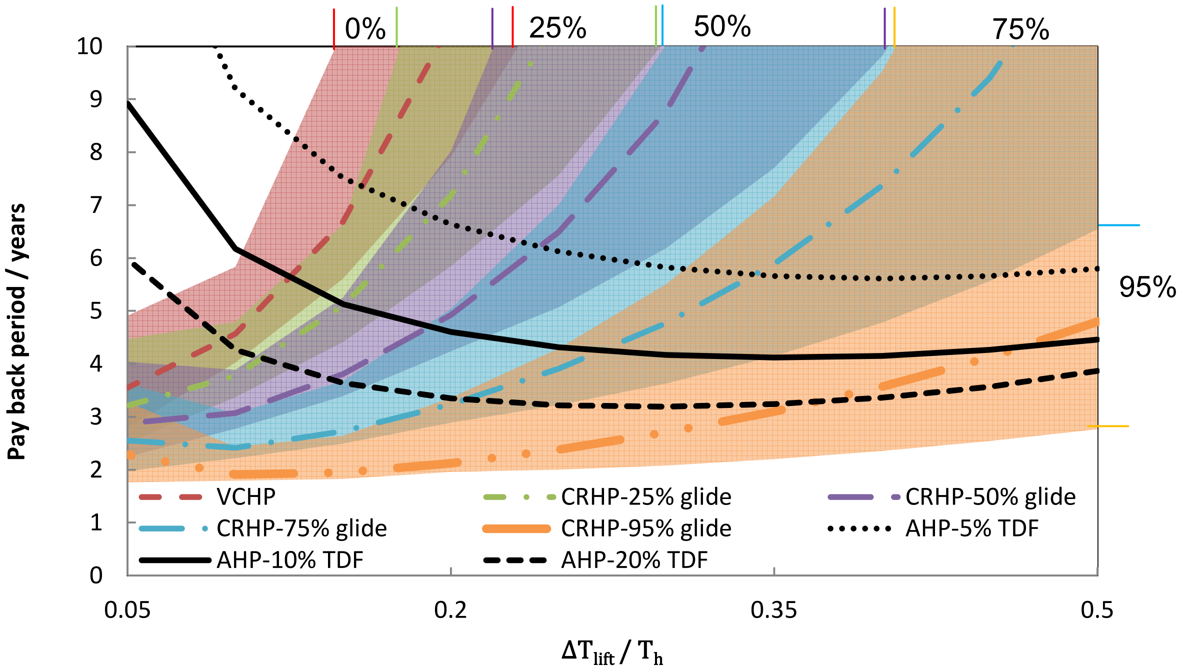

Figure 7: Payback period as function of dimensionless lift for different heat pump systems. Heat duties are 2.79 MW for all cases.

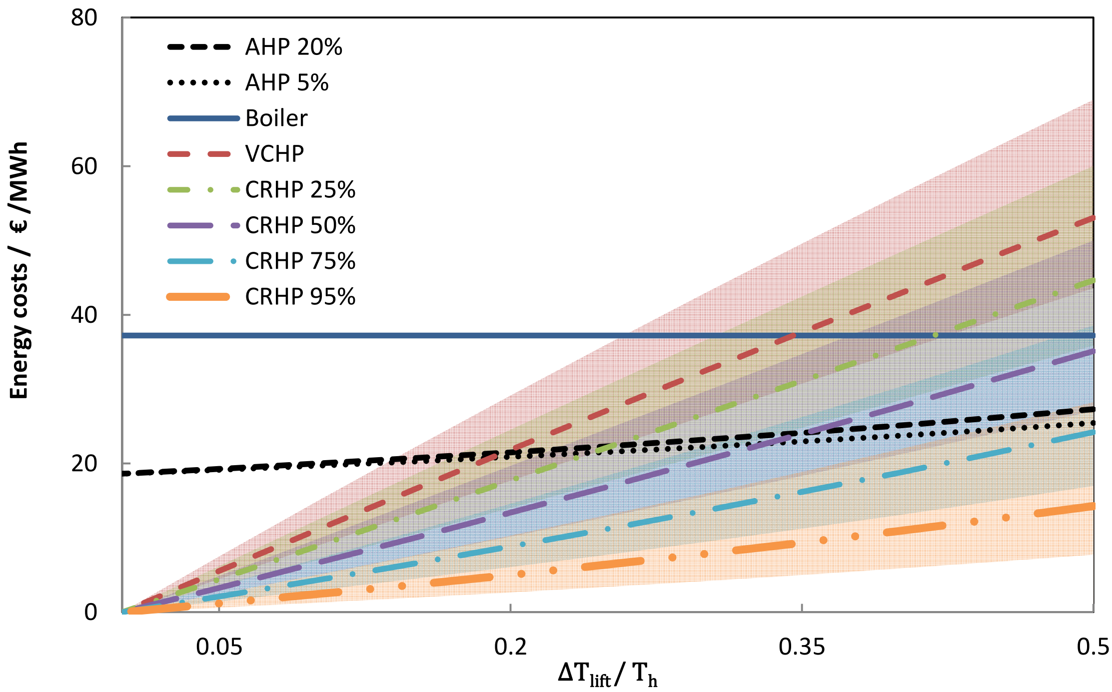

Figure 8: Energy costs per MWh heat as function of dimensionless lift for different heat pump systems. The heat duty is 2.79 MW for all cases

In Figure 7 the payback times based on a heat load of 2.79 MW (results in smaller heat exchangers for larger temperature driving forces) as a function of lift-to-sink temperature ratio are given. Figure 8 shows the operating (energy) costs for different systems as function of lift-to-sink temperature ratio. To determine the cost-effectiveness of applying a heat pump to a certain process, the lift-to-sink temperature ratio should be determined as well as the available glide in the process divided by the temperature lift of the process. These data should then be plotted in Figures 7 and 8 to obtain the payback period and energy costs savings. The ranges result from changes in compressor efficiency and the ratio between temperature driving forces and lift.

6 DISCUSSION

The vapor recompression heat pump is best applied in low temperature lift applications, as here the relative advantage of having only one temperature driving force compared to the two temperature driving forces in the vapor compression heat pump is the largest. In vapor compression heat pumps the refrigerant can be chosen in such a way that the vapor density at compressor inlet and the latent heat are large, therefore the required compressor size can be smaller and thus have lower cost. As the compressor in the vapor recompression heat pump is larger and more expensive, to come to a better pay back period the performance has to be significantly higher in terms of lower energy use.

At very low temperature lifts, the compression-resorption heat pump system can have an advantage over the vapor recompression heat pump as long as the temperature glide over both heat exchangers is larger than the temperature driving force of one of the heat exchangers. However, here the compression-resorption heat pump is also competing with the HIDiC, which has a glide over the heat exchanger as well. On the other hand, heat pumps like HIDiC have a temperature glide in such a way that the isobars are not close to being parallel, as opposed to compression resorption heat pumps. In wide boiling mixtures (high temperature lifts), where this non-parallelism is more pronounced, it might be better to apply a compression-resorption heat pump. In Figures 7 and 8, for instance CRHP-50 % glide indicates that on both reboiler and condenser there is a temperature difference between in- and outlet of 50 % of the ΔT_{lift} as defined in eq. (8).

Transcritical heat pumps are not yet applied as the heat pump fluid does not match the process conditions. Most research to transcritical heat pumps focus on CO_2, but a different fluid will have to be identified. Under the assumption that such a fluid is found, the best use may be there were the feed to a column is relatively pure already, but needs to be further separated. In such a column, the temperature glide above the feed is small compared to the available glide below the feed. For the gas heat exchanger, it should be considered to install the heat exchanger inside the column.

Comparing the heat pump systems with mechanical compressor against heat driven heat pumps like absorption heat pumps, it can be seen that absorption heat pumps are less sensitive to temperature lifts. While at lower temperature lifts, compressor driven heat pumps have the advantage, for larger temperature lifts the opposite is true. In all investigated cases, heat driven heat pumps are advantageous compared to conventional boilers. In Figures 7 and 8, for instance AHP-TDF 5 % indicates that the temperature driving force is 5 % of ΔT_{lift} as defined in eq. (8).

The equations used in this research are still very rough estimates, both for the heat pump efficiency and for the costs for compressors and heat exchangers. Predictions can be refined by including superheat and expansion losses. Furthermore, volume flow rates and operating pressures should be included to predict investment and payback time more accurately. However, this requires knowledge about the heat pump fluid to be used.

7 CONCLUSION

In this research an approach to predict the efficiencies of different types of heat pumps has been shown. These were used in combination with simple equations to determine the investment cost and cost savings to create a heat pump map. The heat pump map should be used as follows:

1. Determine the lift-to-sink temperature ratio for the process

2. Check for the available temperature glide and divide it over the lift
3. Put this data in Figures 7 and 8 to determine the possible economic performance

8 RECOMMENDATIONS

The performance map can be improved further by implementing losses for throttling, superheat and pressure drop.

9 ACKNOWLEDGEMENTS

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NOMENCLATURE

Variable			subscript	
A	Area	m^2	Carnot	Related to Carnot
C	Cost	€/MWh	CRHP	Compression-resorption heat pump
COP	Coefficient Of Performance	kW/kW	E	Electric
Q	Heat duty	kW	Gas	Gas
T_h	Upper sink temperature	K	H	Heating
T_l	Lower source temperature	K	Lorentz	Related to Lorentz
$\Delta T_{driving}$	Temperature driving forces	K	TDF	Temperature driving force
ΔT_{glide}	Temperature glide	K	VCHP	Vapor compression heat pump
ΔT_{lift}	Temperature lift	K	ahp	Absorption heat pump
W	Work	kW	hp	Mechanical heat pump
t	Celcius temperature	°C	is	Isentropic
η	efficiency			

REFERENCES

Allen J.J. and J.F. Hamilton. 1983. “Steady State Reciprocating Water Chiller Models”, ASHRAE transactions Vol. 89 Part 2A pp. 398-407.

De Rijke, A., Z. Olujic, and P.J. Jansens. 2007. “Development of a Concentric HIDiC with Heat Panels”, Symposium, DISTILLATION COLUMNS, Still in (growing) business

Guthrie K.M. 1969. “Data and Techniques for Preliminary Cost Estimating”, Chem. Eng., vol. 76, (6) pp. 114-142.

Hamilton J.F. and J.L. Miller. 1990. “A Simulation Program for Modeling an Air-Conditioning System”, ASHRAE Transactions, vol. 96 (1), pp. 213-221.

Tozer R.M and R.W. James. 1997. “Fundamental thermodynamics of ideal absorption cycles”, Int J. of Refrig., vol . 20, No. 2, pp. 120-135.

Yilmaz M. 2003. “Performance Analysis of a Vapor Compression Heat Pump Using Zeotropic Refrigerant Mixtures”, Energy Conversion and Management vol. 44, pp. 267-282.