Refrigerant Selection and Cycle Development for a High Temperature Vapor Compression Heat Pump

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Abstract

Different technological challenges have to be met in the course of the development of a high temperature vapor compression heat pump. In certain points of operation, high temperature refrigerants can show condensation during the compression which may lead to compressor damage. As a consequence, high suction gas superheat up to 20 K can be necessary. Furthermore high compressor outlet temperatures caused by high heat sink outlet temperatures (approx. 110 °C) and high pressure ratios can lead to problems with the compressor lubricant. In order to meet these challenges different refrigerant and cycle configurations have been investigated by means of simulation. Thermodynamic properties as well as legal and availability aspects have been considered for the refrigerant selection. The focus of the cycle configurations has been set on the realization of the required suction gas superheat. Therefore the possibility of an internal heat exchanger and a suction gas cooled compressor has been investigated. The simulation results showed a COP increase of up to +11 % due to the fact that the main part of the suction gas superheat has not been provided in the evaporator. Furthermore, the effect of increased subcooling has been investigated for a single stage cycle with internal heat exchanger. The results showed a COP of 3.4 with a subcooling of 25 K at a temperature lift of approximately 60 K for the refrigerant R600 (n-butane). Based on the findings of the simulations, a single stage high temperature heat pump cycle with the refrigerant R600 (n-butane) has been designed.

1. Introduction

High temperature applications of vapor compression heat pumps are facing certain challenges. For instance, an increasing temperature level has effects on the efficiency of the thermodynamic cycle in terms of temperature lift. High compressor outlet temperatures can cause problems with compressor lubrication. A central point for the development of a high temperature vapor compression heat pump (HTHP) marks the refrigerant selection. Certain refrigerants suitable for the high temperature application show a so called "overhanging" 2 phase region which can lead to condensation during compression and thus compressor damage ([1], [2], [3]). Further important aspects for refrigerant selection are thermodynamic properties as well as legal, environmental and safety issues. Calm [4] gives an overview of past and future refrigerant developments in this regard. Finally a

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thermodynamic cycle has to be designed in order to meet the requirements of heat pump efficiency and operational safety.

This work is based on the development of a high temperature vapor compression heat pump for the small and medium capacity range (up to 50 kW) with heat sink outlet temperatures of up to 110 °C. The technological problems have been considered and the consequences for the cycle design investigated. A refrigerant selection has been carried out based on a set of predefined criteria and aided by simulation where the most important aspects are pointed out in this work. For the cycle development, means of thermodynamic cycle simulation has been used.

Based on these findings a prototype of an R600-HTHP has been designed and is introduced in this work.

2. Challenges of high temperature application

High condensation temperatures have a general effect on the heat pump process which is from basic thermodynamic nature considering the Carnot process. Regarding real applications high condensation temperatures can lead to additional problems concerning heat compressor lubricant, compressor efficiency as well heat pump efficiency. Further aspects are refrigerant related consequences for the cycle design like overhanging 2 phase regions and the resulting required superheat. These effects shall be discussed in this section.

2.1. General effects of high condensation temperatures and temperature lift

Looking at the Carnot cycle it becomes clear that higher condensation temperatures, assuming constant evaporation temperatures, lead to lower theoretical cycle efficiencies which of course is valid for the real cycle as well. In the real cycle however, further effects of high condensation temperatures can be found mainly due to the refrigerants properties. The phase change enthalpy decreases with increasing condensation temperature. Keeping the evaporation temperature the same and thus the suction gas density, a lower heating capacity results for identical compressor swept volumes. Furthermore the specific compressor work increases with the overall isentropic efficiency decreasing at increasing pressure ratios. These two effects combined leading to a further decrease of heat pump efficiency. Depending on the refrigerant, a high condensation pressure can also become a topic. Heat losses are increasing because of a higher temperature level. Also refrigerant dependent are the resulting compressor outlet temperatures which may, due to an already high condensation temperature, exceed allowed compressor lubricant temperatures. In order to compensate at least the reduction of phase change enthalpy, it is possible to utilize larger values of subcooling. The possible degree of subcooling however depends on the heat sink temperature lift. If the heat sink temperature lift is not large, high values of subcooling can increase the condensation temperature and hence reducing efficiency. The evaporation temperature can be increased by avoiding high values of superheat in the evaporator and simultaneously realizing a small temperature difference on the heat source side. Since some high temperature refrigerants require a certain value of superheat, this task is of special interest when it comes to high temperature heat pumps and will be discussed in the following section.

2.2. Wet compression and minimum required superheat

Certain refrigerants suitable for high temperature application show an overhanging 2 phase region in the t/s-diagram as depicted in Fig. 1 for R600 (n-butane). It can be quantified by the slope of the saturated vapor curve (ds<0). Lai and Fischer [5] have discussed this characteristic in reference to Organic Rankine Cycles (ORC). Itard [6] and Reisner [1] have put it into context with high temperature vapor compression heat pumps. The consequence of the overhanging shape is the possibility of refrigerant condensation during compression ("wet compression") which can lead to compressor damage. Furthermore compressor heat losses could enhance the possibility of wet compression. A sufficient suction gas superheat can avoid entering the 2 phase region during the compression process. This required amount of superheat can be referred to as "minimum required superheat" (see [1]) for overhanging refrigerants which is highly dependent on the evaporation and condensation temperature as well as the refrigerant itself. Based on the definition of Minea [2] (isentropic compression, see also [3] and Fig. 1, left) and thermodynamic properties (obtained from [7]) curves for the minimum required superheat for different refrigerants can be found (see Fig. 1, right).
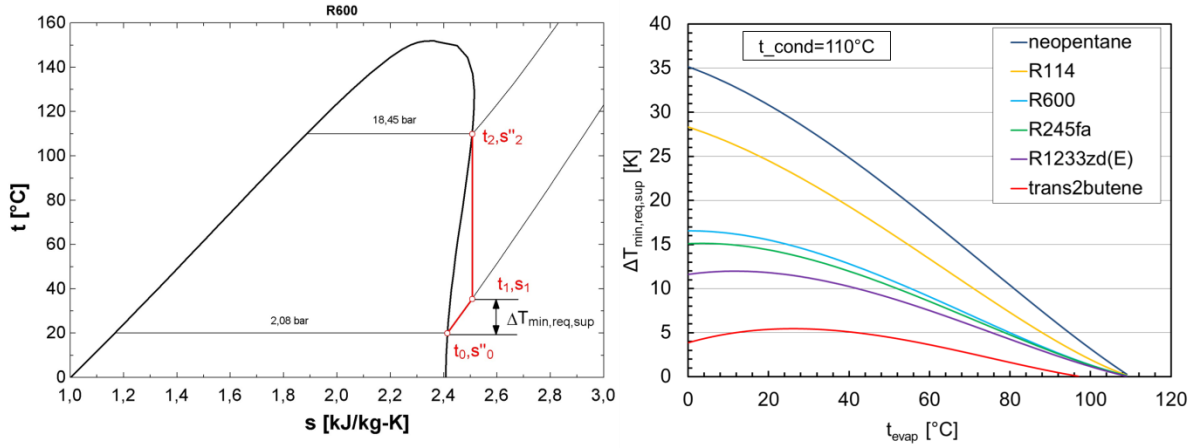
Fig. 1: Definition of the minimum required superheat ($\Delta T\_min, req, sup$) in the t/s-Diagram (left) and characteristics for the minimum required superheat as a function of evaporation temperature (t_evap) of different refrigerants for a condensation temperature of 110 °C

These curves are characteristic for each refrigerant in a given temperature range and can be used as a starting point for heat pump und control design. The characteristics show a maximum where the entropy (s) on the saturated vapor curve has a minimum. For the considered refrigerants, superheat values vary for an evaporation temperature of 40 °C and a condensation temperature of 110 °C between 5 K (Trans-2-butene) and 25 K (Neopentane). An important note to the minimum required superheat is that the definition is based on the assumption of an isentropic compression (see Fig. 1, $s_1 = s_2''$). Hence it is most likely that different values of superheat are necessary in real applications because of compressor irreversibilities to keep a safe distance from the 2 phase region. Realizing a high suction gas superheat within the evaporator will decrease the evaporation temperature and therefore the heat pump efficiency. Hence measures have to be taken in order to avoid high values of superheat in the evaporator e.g. by means of an internal heat exchangers (see [1]), the use of suction gas cooled compressors (see [3]) or other external heat sources. Therefore the preparation of suction gas superheat is essential for the design and the control strategy of high temperature heat pumps using overhanging working fluids.

3. Refrigerant selection

Several selection criteria have been set up in order find the most suitable refrigerant for the development of a high temperature heat pump in a capacity range up to 50 kW for a heat source temperature of approx. 60 °C and a heat sink outlet temperature of 110 °C. The selection process is briefly described in this section. The investigated group of refrigerants is based on the fluid databases of EES v. 9.901 [7] and additionally the currently developed (as in 2015) R1336mzz-Z [8].

3.1. Selection criteria

The main criterion for the refrigerant is the critical temperature (t_{crit}) if only, like in this case, a subcritical heat pump process is considered. Therefore the threshold of the critical temperature has been set to 120 °C in order to fulfill the desired outlet temperature of 110 °C. Another aspect is the vapor pressure of the refrigerant at ambient temperature (approx. 20 °C), which is important to stay above ambient pressure in order to avoid air infiltration into the refrigerant circuit at down times of the system. Since the Montreal Protocol [9] an ozone depletion potential (ODP) of zero is mandatory. Furthermore the EU [10] aims for phasing out refrigerants with a high global warming potential (GWP), therefore a limit of 150 has been considered. In order to find a certifiable heat pump concept according to EN378 [11], the refrigerant must be commercially available and listed in the mentioned standard. Finally the considered compressor limits the high and low pressure to 28 bar and 19 bar respectively. The applied criteria are summarized in Table 1.

Table 1: Defined criteria for the refrigerant selection

No.:	Characteristic	Criterion

1	Critical temperature	$t_{crit} \geq 120°C$
2	Vapor pressure at 20 °C	$p_{sat,v}(20°C) > 1{,}013 \; bar$
3	Ozone depletion potential (ODP)	0
4	Global warming potential (GWP)	<150
5	Commercially available	yes
6	Listed in Standard EN 378	yes
7	Threshold of high pressure	$p_{HD} \leq 28 \; bar$
8	Threshold of low pressure	$0{,}5 \; bar < p_{ND} < 19 \; bar$

The application of criterion 1 and 2 of Table 1 sorts out already the majority of refrigerants from the basic amount. For example R1336mzz-Z disqualifies because of a vapor pressure below 1,013 bar at 20 °C. For the remaining group of refrigerants the theoretical cycle performance and different parameters have been analyzed which is summarized in the following section.

3.2. Theoretical performance of high temperature refrigerants

The theoretical analysis on cycle performance have been carried out with the nonlinear equation solver EES v. 9.901. Therefore a basic single stage heat pump cycle has been defined (see Fig. 5a, Section 4.1). The refrigerant properties can be obtained from the EES v. 9.901 fluid databases. The utilized boundary conditions for the investigated cycle can be found in Table 2.

Table 2: Boundary conditions for the basic cycle analysis

Condensation temperature (t_{cond})	120°C
Evaporation temperature (t_{evap})	30°C/50°C/70°C
Isentropic efficiency (η_{is})	0,56
Superheat (ΔT_{sup})	10 K
Subcool (ΔT_{sub})	5 K

The used definition of the theoretical heat pump efficiency (COP_{th}) is given by equation (1). The definition of the isentropic compression efficiency (η_{is}) follows equation (2). The specific heating capacity (q_{vH}) can be obtained from equation (3).

$$COP_{th} = \frac{q_H}{w_{comp}} \qquad (1)$$

$$\eta_{is} = \frac{w_{comp,is}}{w_{comp}} \qquad (2)$$

$$q_{vH} = v'' \cdot (h''_{cond} - h'_{cond}) \qquad (3)$$

Where q_H describes the specific heating capacity and w_{comp} the specific compression work. $w_{comp,is}$ denotes the specific work of the isentropic compression. The specific suction gas density at saturated vapour state is defined by v'' at evaporation temperature. The enthalpies h''_{cond} and h'_{cond} denote the saturated vapor enthalpy and saturated liquid enthalpy respectively at condensation conditions. The remaining refrigerants have been evaluated according to the boundary conditions in Table 2 and equations (1) to (3). Fig. 2 shows the results for the theoretical efficiency (left) and the compression end temperatures (right). The highest efficiencies can be observed for the hydrocarbon (HC) Cis-2-butene, Sulfurdioxide as well as the hydro-flouro-olefin (HFO) R1234ze(Z) (isomer of R1234ze(E)) and the hydro-flouro-chloro-olefin (HCFO) R1233zd(E) with values close to 4 for a temperature lift of 50 K. In comparison, R600 and R600a show a COP_{th} of approximately 3.7 and 3.4 respectively. Another important aspect is the compression end temperature which is depicted in Fig. 2 on the right. The refrigerants R717 (Ammonia) and Sulfurdioxide display especially high compression end temperatures in range of 200 °C and 300 °C for the considered evaporation temperatures (30, 50 and 70 °C). These temperatures are way above maximum threshold temperatures for lubricants in heat pump application. Dimethylether and R1234ze(Z) show also rather high values within a range of 150 °C to 170 °C where special

lubricants might be necessary. Considering the assumed boundary conditions, the hydro-flouro-carbons (HFC) R236ea, R236fa, R245fa as well as the HCs R600, R600a show temperatures around 130 °C.
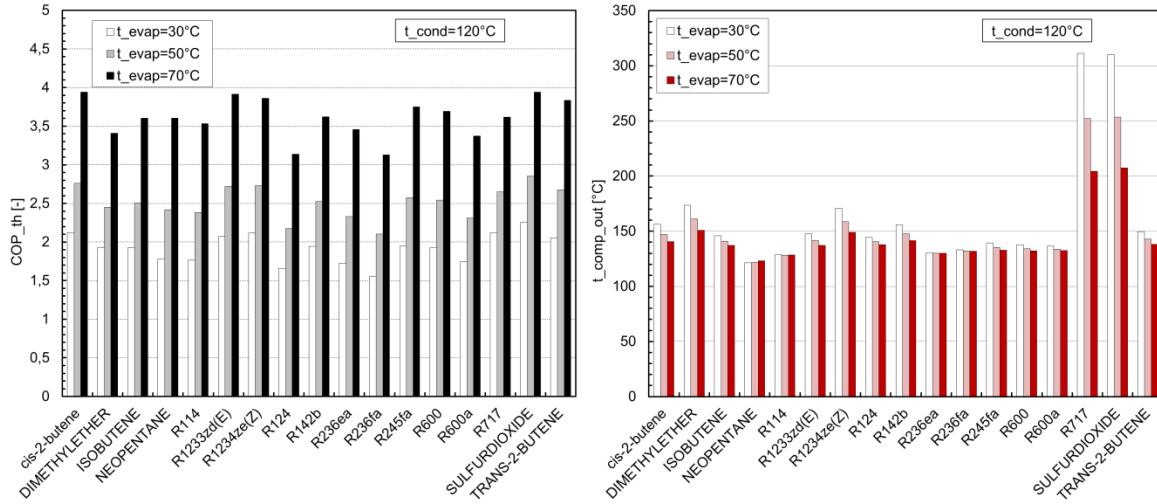

Fig. 2: Comparison of theoretical cycle efficiency (COP_th, left) and compression end temperature (t_comp, out, right) for a condensation temperature of 120 °C, different refrigerants and evaporation temperatures

The results for condensation pressure (p_{cond}, left) and the specific heating capacity (q_{vH}, right) are given in Fig. 3 for the remaining refrigerant selection according to the assumptions of Table 2. The maximum condensation pressure at a condensation temperature of 120 °C of approximately 90 bar is displayed by Ammonia followed by Dimethylether with approximately 47 bar. Both refrigerants are exceeding the pressure limit as well R124 or Sulfurdioxide for example. The specific heating capacity as a measure for compressor size shows similar values for R600a, R600 as well as R236ea, R114 or R1234ze(Z) of around 4000 kJ/m³ for an evaporation temperature of 70 °C. The maximum is achieved by ammonia which would be of course favourable with respect to the compressor size. The specific heating capacity is decreasing with decreasing evaporation temperature caused by decreasing suction gas density.
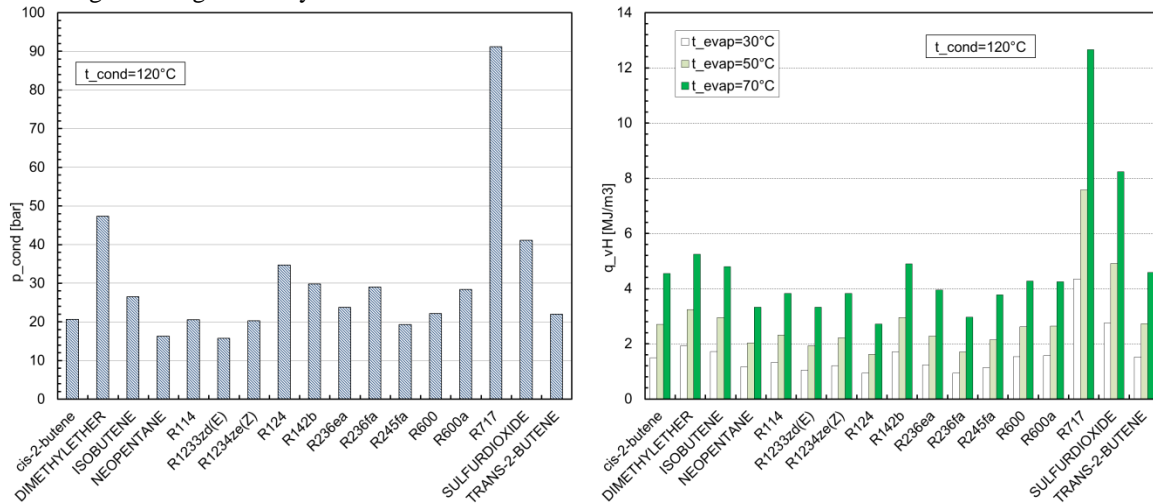

Fig. 3: Comparison of condensation pressure (p_cond, right) and specific heating capacity (q_vH, left) for different refrigerants at a condensation temperature of 120 °C

3.3. Results of the refrigerant selection

After applying criterion 1 and 2 of Table 2 on the basic list of fluids considered, 17 refrigerants remained in the shortlist. Simple cycle simulations showed that some refrigerants exceed the limits for condensation pressure which amongst other reasons disqualifies R717, Sulfurdioxide, Dimethylether. R114, R124 and R142b cannot be considered because of their ODP>0. Furthermore R236ea, R236fa and R245fa are exceeding the GWP limit of

150. In terms of HCs only R600 and R600a are listed in the EN378 standard which sorts out Isobutene, Cis-2-butene, Trans-2-butene and Neopentane. R1234ze(Z) is critical because of compression end temperatures up to 170 °C and the fact that R1234ze(Z) has not been commercially available (in contradiction to its isomer R1234(E)) at the time this work had been carried out and therefore neglected for further considerations. The vapor pressure of R1233zd(E) at 20 °C is 1,079 bar which is just slightly above the limit of Table 2. Hence slight ambient temperature fluctuations during heat pump downtimes could be unfavorable in terms infiltration. Further reasons for not selecting R1233zd(E) are that it is not listed in [11] and that it contains chlorine although its ODP is zero. The last remaining refrigerants are R600 and R600a. Since these refrigerants are isomers, their properties are very similar. R600 has a higher critical temperature (152 °C) compared to R600a (134.7 °C) which results in a higher theoretical efficiency (see Fig. 2). The condensation pressure level of R600 lies within the accepted limit, however R600a is critically close. Both refrigerants fulfill all the requirements of Table 2 but R600 has been finally selected as refrigerant for further heat pump development because of a more favorable pressure level and theoretical efficiency. However it has to be considered that R600 is flammable and therefore safety measures have to be taken for the prototype design. These investigations show also that the Butene group (cis-2-butene, trans-2-butene) is showing a good theoretical performance as well as a favorable pressure level. Furthermore they require a low minimum superheat (see also Fig. 1, trans-2-butene). However they are not commonly used in heat pump application but may be interesting for future developments.

4. Simulation and cycle design

In order to find an appropriate cycle configuration more detailed simulations have been carried out. The calculations have been again conducted with EES v. 9.901. The focus has lied especially on the source for superheat. The boundary conditions and results are summarized and briefly discussed in this section.

4.1. Method and boundary conditions

The simulation study has been carried out in two steps. The first step has been a set of design simulations in order to determine compressor swept volume and heat transmission coefficients (UA) of the condenser and evaporator for a basic single and a 2 stage heat pump cycle in a design point of operation. For the heat exchanger design, a pinch point temperature difference of 4 K has been assumed. The second step has been the application of the obtained parameter to the different cycle configurations. . For the simulation, heat losses as well as pressure drops due to fluid friction have been neglected. The compressor efficiency has been modelled as a function of the pressure ratio and obtained from manufacturer data. The same method has been applied to the compressors volumetric efficiency (λ_{vol}). The heat exchangers have been modelled with the obtained UA-values from the design simulations. Furthermore only steady state simulations have been conducted. The refrigerant is R600. The pressure ratios in the first and second stage of the 2 stage cycles have been assumed to be identical. The design parameters and boundary conditions are given in Table 3. Therein, the overall isentropic efficiency ($\eta_{is,ov}$) is defined as the ratio between compressor power with isentropic compression and the actual electrical consumption of the compressor.

Table 3: Parameters and boundary conditions for the cycle simulations

Refrigerant	R600
Design heating capacity ($\dot{Q}_{HD}$)	50 kW
Design pinch point (ΔT_{pp})	4 K
Design heat sink temperatures ($t_{in,sink,D}$/$t_{out,sink,D}$)	80 °C/110 °C
Design heat source temperatures ($t_{in,source,D}$/ $t_{out,source,D}$)	60 °C/50 °C
Overall isentropic efficiency ($\eta_{is,ov}$)	0.3 to 0.56
Volumetric efficiency (λ_{vol})	0.8 to 0.9
Superheat a compressor inlet for R600 (ΔT_{sup})	18.5 K
Subcool at expansion valve inlet (ΔT_{sub})	5 K

The investigated cycle configurations are mainly based on Cao et al [12] and include the most common heat pump cycle layouts. However since the suction gas superheat is a critical parameter for overhanging working fluids and has a main influence on heat pump efficiency (compare to section 2.1), different sources of superheat

preparation have been investigated. The six main cycle configurations are depicted in Fig. 4. The basic single stage cycle (1) displays the conventional heat pump cycle and serves as basis for further comparison. The advantages of the 2 stage cycle (2) are the reduced compressor power because of the staged compression as well as an increased condenser mass flow because of a higher suction gas density in the second stage.
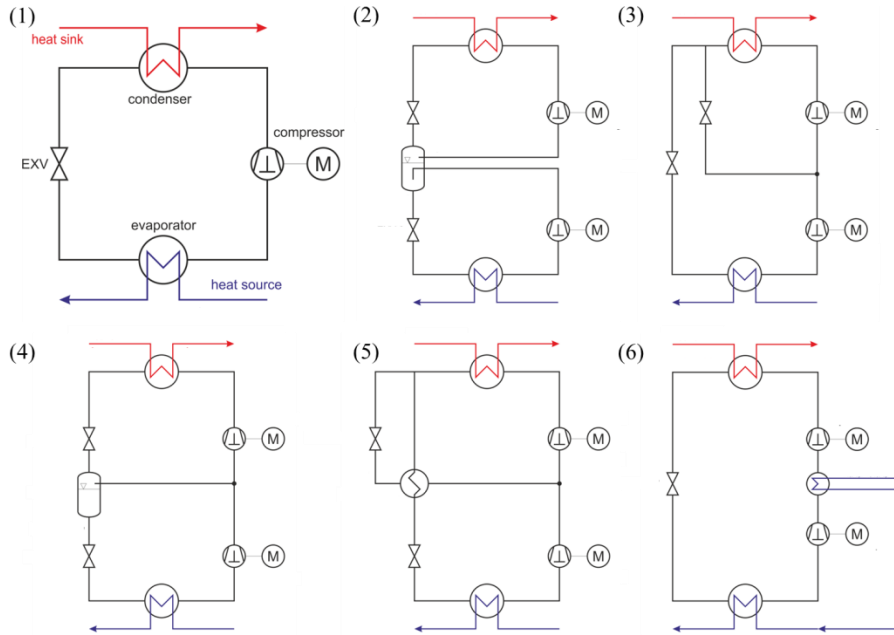

Fig. 4: Main group of cycle configurations for the simulation study based on [12], (1) basic, (2) 2 stage, (3) liquid injection (LI), (4) economizer with flash tank (ECO_FT), (5) economizer with heat exchanger (ECO_HX), (6) intercooler (IC),

The liquid injection (3, LI) should decrease compressor outlet temperatures and slightly increase condenser massflow and thus heating capacity. Furthermore compressor power should be decreased by staged compression. The economizer configurations (4, ECO_w_FT and 5, ECO_w_IHX) increase the condenser massflow and provide a multi stage compression. The intercooler provides (6, IC) again a staged compression as well as lower compressor outlet temperatures. All main cycle configurations of Fig. 4 had been extended with an internal heat exchanger (IHX) and a suction gas heat exchanger (SHX) for the cycle simulations. The SHX has been modelled basically as a separate heat source providing the suction gas superheat independently from the heat pump heat source. In this case the SHX is assumed to be a suction gas cooled compressor motor (see also [3]). Fig. 5 illustrates the different superheat configurations exemplary for the basic cycle.
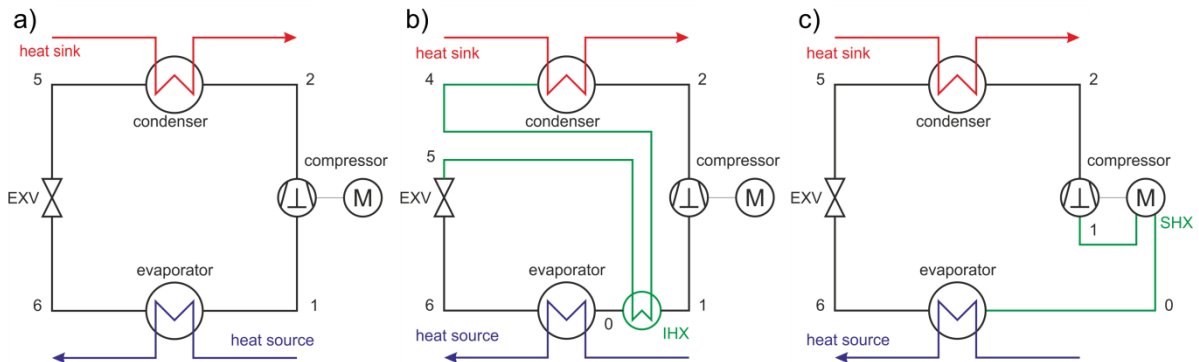

Fig. 5: Different cycle configurations in terms of suction gas superheat preparation, a) basic, b) internal heat exchanger (IHX), c) suction gas cooled compressor (SHX)

The combination of cycle layouts from Fig. 4 and Fig. 5 results in 20 variations of heat pump cycles. That means that there is a basic concept (Fig. 5a), a concept with internal heat exchanger (Fig. 5b, IHX) and a concept with suction gas heat exchanger (Fig. 5c, SHX). The 2 stage cycle (Fig. 4; 2) represents a special case. Since the

refrigerant state in the flash tank is assumed to be saturated, a sufficient superheat has to be ensured in order avoid a wet compression in the second compressor stage. Therefore it is necessary to utilize either an internal heat exchanger (DIHX) or a suction gas heat exchanger (DSHX) in the second stage. Both possibilities have been investigated. The same could be true for the remaining 2 stage concepts like the liquid injection or the ECO_FT. In real applications however, an appropriate superheated state at compressor inlet of the second stage has to be maintained by the intermediate by the EXV of the second stage. In the case of the intercooler the heat exchanger design has to be considered regarding sufficient superheat. The results of the simulation are summarized in the following section.

4.2. Simulation results and cycle selection

As stated before 20 cycle configurations have been investigated by means of simulation in order to find an appropriate heat pump cycle for high temperature applications. The boundary conditions have been applied according to Table 3. The results for the heating capacity ($\dot{Q}_H$, left) and the efficiency (COP_H, right) for two operation points and the refrigerant R600 are displayed in Fig. 6. It can be observed that the heating capacity for the design point (Fig. 6, green bars) of both Basic and 2_stage cycle, results in 50 kW as stated in section 4.1.

The influence of the IHX and SHX can be clearly observed for the heating capacity and the efficiency. Since the high required superheat has not been entirely provided in the evaporator, the evaporation temperature can increase, hence the suction gas density increases and therefore the heating capacity (Basic, Basic_IHX, Basic_SHX for example). The Basic_SHX cycle already shows an improvement of approx. 11 % in terms of COP and 27 % in heating capacity compared to the basic configuration. The efficiency increases with increasing evaporation temperature, although the refrigerant mass flow increases but the pressure ratio decreases. Hence the increase of compressor power is lower than the increase of heating capacity.
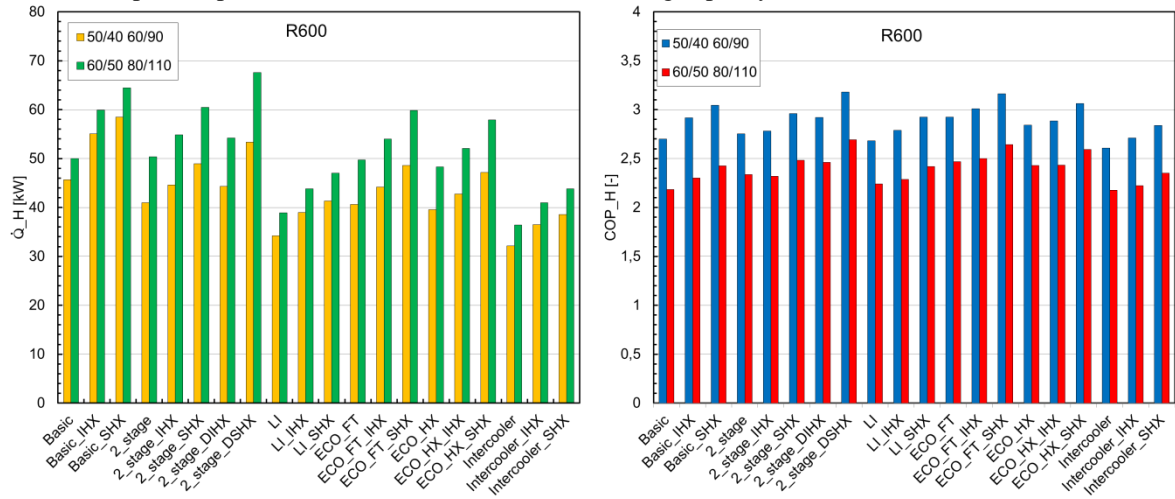

Fig. 6: Simulation results for heating capacity ($\dot{Q}_H$, left) and COP_H (right) for different cycle configurations and operation points (a) heat source 50/40 °C, heat sink 60/90 °C; b) heat source 60/50 °C , heat sink 80/110 °C)

The best performance is shown by the 2 stage cycle with SHX in both stages (2_stage_DSHX) for each operation point. For the design point (red bars), the heating capacity increases by approx. 34 % and for the COP_H by approx. 24 % compared to the basic cycle. The absolute efficiencies however are rather low with a maximum of 2.7 for the design point (80/110 °C) and the assumed compressor efficiencies. The economizer cycle with SHX and flash tank (ECO_FT_SHX) shows a similar COP improvement of approx. 21 %, but a lower improvement in the heating capacity by approx. 20 % compared to the basic cycle. This indicates a higher absolute compressor power demand compared to 2_stage_DSHX. In terms of performance the 2 stage cycle with DSHX should be selected for the heat pump prototype. However the higher system complexity in terms of hydraulics, components and control is increasing the investment costs significantly compared to the basic cycle. Another aspect is, if the 2_stage_DSHX is compared with the Basic_SHX cycle, the advantage decreases significantly with a remaining improvement for the heating capacity of 4 % and in terms of COP of 11 %. Therefore a final investigation of a basic cycle using IHX has been carried out. Based on the statements of section 2.1, the heat source temperature difference should be kept as small as possible to increase the evaporation temperature. Furthermore the increase of subcooling can increase the heating capacity if the source side

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temperature lift is high. Hence the source side temperatures have been changed to 60 C° (inlet) and 55 °C (outlet) and the useable subcooling to 25 K. The results showed a COP_H of 3.4 due to these measures and the application of an IHX, for sink temperatures of 80 °C (inlet) and 110° C (outlet) as well as condenser/evaporator pinch points of 4 K. Therefore the single stage with additional superheat preparation (IHX, SHX) and an extended use of refrigerant subcooling has been considered as final prototype cycle for the further development of the R600-HTHP because of the more simple system and acceptable efficiency values. The final prototype concept shall be introduced in the following section.

5. High temperature heat pump prototype

The conclusion of simulation results and the analysis of the different cycle concepts showed that the tradeoff between cycle complexity and efficiency is most favorable for the single stage cycle with superheat support. Since the suction gas superheat is such an important issue concerning current and future high temperature refrigerants, the heat pump test rig has been designed to be adaptable for different means of superheat preparation. As can be seen in Fig. 7, three different cycle modifications can be realized within the prototype. The first configuration is the basic heat pump cycle with a suction gas cooled compressor (and by pass of the IHX). The second possibility is a combination of IHX and SHX by closing the superheater valves. The last cycle is the Basic_SHX from section 4.1. Furthermore the subcooler can be by passed in order to investigate the interaction of condensation temperature, degree of subcooling and heat sink temperature lift. The heat pump prototype is designed for a maximum heating capacity of 40 kW at heat sink temperatures 80 °C/110 °C and heat source temperatures of 60 °C/55 °C. The utilized compressor technology is a modified inverter driven separating hood reciprocating compressor from the project partner Frigopol (see [12], [3]). The test rig enables to investigate and compare different means of superheat preparation as well as control strategies. The installed measurement equipment is designed to evaluate the heat pump cycle in t/s or t/h-diagrams as well as determining heat pump and compressor performance maps for model validation and further simulation studies. This prototype is currently under construction (as in October, 2016), thus no measurement data is available so far.
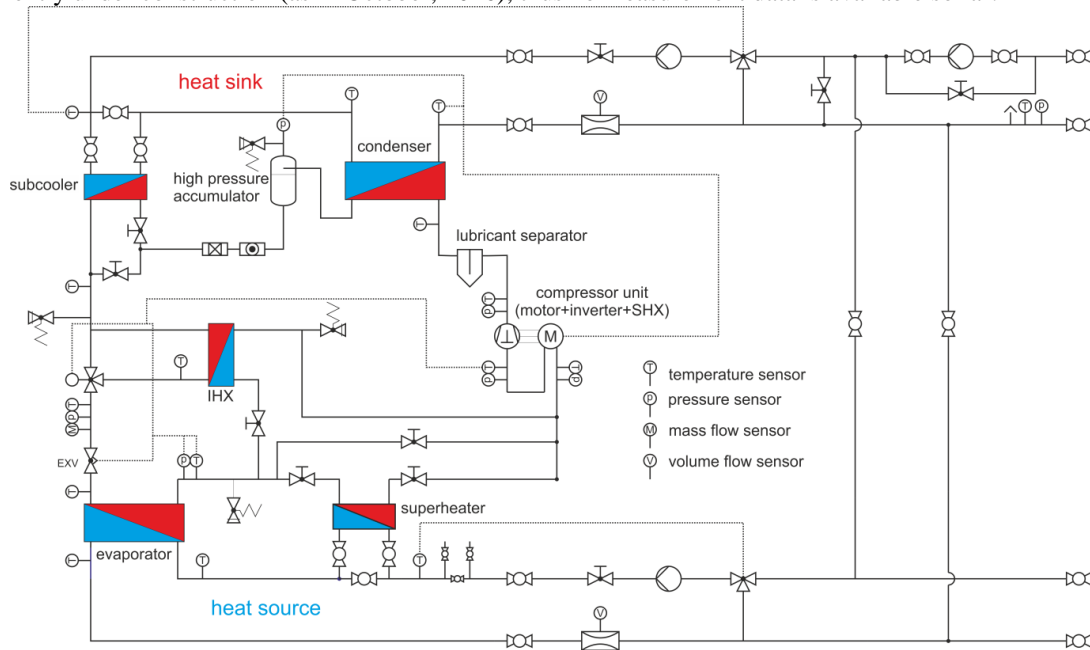

Fig. 7: Hydraulic layout for the high temperature heat pump test rig including sensor positions

6. Conclusion and outlook

The development process for a high temperature heat pump in the small and medium capacity range up to 50 kW and sink outlet (water) temperatures of 110 °C has been outlined in this work. Technological challenges coming with the high temperature application have been investigated. High compressor outlet temperatures as well as low heat pump efficiencies at high temperature lifts have to be met. The most interesting challenge has been found to be the overhanging nature of the 2 phase region for some high temperature working fluids. A so

called minimum required superheat to avoid wet compression can be determined and used for heat pump and control design. The required superheat however, can reach values up to 20 K. Hence a decoupling of the superheat preparation from the evaporator has to be ensured in order to increase evaporation temperature and thus efficiency. Furthermore, the chance of a wet compression during heat pump start up and dynamic operation can afford special start up and control strategies. The refrigerant selection process has been carried out by defining a set of criteria fitting the application boundary conditions. The basic amount of refrigerants has been obtained from [7]. A critical temperature limit of 120 °C and a normal boiling point below 20 °C have reduced the number of possible refrigerants down to 17. Considering availability, legal (environmental, certification) and theoretical performance R600 (n-butane) remained as selected refrigerant for further development. The cycle design has been supported by simulation. Therefore 20 different cycle configurations have been investigated with the focus on different means of superheat preparation (internal heat exchanger, suction gas heat exchanger). The results showed that the application of a suction gas heat exchanger can increase heat pump efficiency up to 11 % when a superheat of 18 K is required. Furthermore a 2 stage cycle with suction gas heat exchangers in the first and second stage showed the best performance under the assumed boundary conditions. However system complexity is rather high and the improvement potential compared to the Basic_SHX cycle decreases significantly. Therefore a single stage cycle with internal heat exchanger has been simulated. An extended use of subcooling (25 K) and a reduction of the heat source temperature drop resulted in an efficiency of 3.4 for a temperature lift of 60 K and a sink outlet temperature 110 °C. The results showed, that due to these rather simple measures, acceptable efficiency values for a single stage cycle can be achieved. Based on these findings a single stage high temperature heat pump with internal and suction gas heat exchanger using R600 as refrigerant has been designed and is currently under construction (as in October 2016). First results are expected in January 2017.

Acknowledgement

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