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Comparison of configurations of high-temperature heat pumps for biogas-upgrading

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ABSTRACT

High temperature heat pumps in industrial processes can enable an increase of their energy efficiency. The temperature swing adsorption process (TSA) enables an efficient separation of carbon dioxide. An essential part of the TSA process is a heat distribution concept, for which high temperature heat pumps are a good choice, since heat must be removed at low temperature (evaporation 35-50°C) and supplied at high temperature (condensation 110-125°C). With this aim different heat pump configurations have been numerically investigated using Dymola/Modelica™. A number of heat pump configurations, varying pre-selected refrigerants, circuit configurations, etc., have been simulated in order to compare the results by compressor size and an internal suction gas superheater (IHX).
The results show the relative displacement volume of each heat pump configuration and give recommendations of the use of the IHX as basis for investment costs and economical solutions.
Keywords: High-Temperature Heat Pump, Industrial Application, Heat Recovery.

1. INTRODUCTION

The use of high temperature heat pumps in industrial processes has been accelerated for several years. Use temperatures of 90°C is state of the art. As development progressed, both the usage temperatures and the thermal performance of the heat pumps were increased. Despite a growing interest in heat pump technology, knowledge about it and its possible applications is not widespread in industry. The present paper shows technical aspects of heat pump development when used in a so-called temperature swing adsorption process (TSA process for short). Of course, the results can also be applied to other industrial processes in order to implement energy efficiency measures in industrial processes.
At the TSA process, carbon dioxide is adsorbed on an adsorbent at a low temperature (about 45 to 60°C) and desorbed at a higher temperature (about 95 to 115°C), regenerating the adsorbent. The appropriate temperature levels and the corresponding separation efficiency between adsorption and desorption depend essentially on the affinity for adsorption of carbon dioxide. Similarly high affinities should be achieved in the TSA process as in amine washing, the energy requirement should be considerably lower. The TSA process has been successfully researched in climate protection research for several years. (Xu, Song, Andresen, Miller, & Scaroni, 2002) and (Sonnleitner, Schöny, & Hofbauer, 2017)
Preliminary investigations have shown that there is great potential for the TSA process, provided that suitable continuous process control with separate adsorption and desorption zones can be implemented. It must be continuously cooled (adsorption zone in the adsorber) and heated (desorption

zone in the desorber). High temperature heat pumps are very well suited for this purpose. (Vogtenhuber, et al., 2017)
When designing heat pumps for use in the TSA process, source temperatures between 35°C and 55°C and sink temperatures between 105°C and 115°C must be assumed. The selection and design of high temperature heat pumps raises a number of questions. The high operating temperatures at the desorber compared to products available on the market, the resulting temperatures in the heat pump condenser (condenser for short), the operating temperatures at the adsorber and the resulting temperature differences between heat pump evaporators (evaporators for short) are the basis for analyzing possible high temperature heat pumps and comparing them with each other.

2. GENERAL DESIGN ASPECTS

In order to design high temperature heat pumps, the know-how of cooling systems and heat pumps for room heating and hot water preparation is used. The main differences to heat pumps in the range up to 80°C utilization temperature lie in the refrigerants used and the requirements placed on the single components in the refrigeration cycle. For the application of high temperature heat pumps in industrial processes, there are different refrigerants that are better or less suitable for each area of application. Tailored to the aforementioned application of the separation of carbon dioxide (Drexler-Schmid, Lauermann, Baumhakel, & Helminger, 2018) and (Helminger, Lauermann, Baumhakel, & Drexler-Schmid, 2018) describe the selection of potential refrigerants, the selection of refrigeration cycle configurations and aspects of the heating and cooling capacities of different configurations. The selection of potential refrigerants was based on technical, economic and ecological aspects. Refrigeration cycle configurations were modelled, configured and simulated for the intended operating conditions. The mentioned documents focus on the presentation of different configurations of high temperature heat pumps and expected operating parameters, which provide the basis for economic evaluations of the operation. In the present paper, aspects of design and thus investment costs are also dealt with. Each refrigeration cycle consists of the main components evaporator, compressor, condenser and expansion valve. In some refrigeration cycle configurations, additional components such as economizers or ejectors can be found. In this article, the costs of the compressors and the use of an internal suction gas superheater (IHX) are examined in more detail.
The thermal performance in the evaporator and condenser and the electrical power consumption of the high temperature heat pump are determined by the mass flow of the refrigerant in the refrigeration cycle and depend on the displacement volume of the compressor. The displacement volume and thus the compressor size is of essential importance for the costs of a high temperature heat pump.
Depending on their thermodynamic properties, refrigerants suitable for high temperature heat pumps carry the risk that, in the event of insufficient suction gas superheating, compression into the wet steam area could occur, which would damage the compressor. The required suction gas superheating is low at low temperatures and increases as the temperature difference between evaporator and condenser increases. As a rule, a suction gas superheating of about 5 K prevails in the evaporator. In order to ensure additional suction gas superheating, internal suction gas superheaters (IHX) are used, which cause additional costs. It must therefore be decided at the design process of the high temperature heat pump, depending on the operating conditions, whether this additional component is to be installed or not.

3. OPERATING CONDITIONS, MODELLING, AND SIMULATIONS

The operating parameters for the present investigation are determined by the mode of operation of the TSA process. It is essential that both the required cooling capacity at the adsorber and the heating capacity at the desorber are provided by the high-temperature heat pump. For the high-temperature heat pump, the source and lower inlet and outlet temperatures are decisive in the first step of the investigation. The thermal performance is mainly determined by the delivery volume of the compressor(s) and can be determined by suitable selection of the compressor(s) or influenced by its operating frequency. For further investigation it is assumed that suitable compressors are available and the focus is on operating temperatures. The operating temperatures are shown in Fig. 1 (Drexler-Schmid, Lauermann, Baumhakel, & Helminger, 2018). This results in a total of 180 operating points, which are defined as follows:

- Source in temperature (evaporator) of 35, 40, 45, 50 and 55°C,
- Source temperature difference (evaporator) of 3, 5, 7.5 and 10 K,
- Sink in temperatures (condenser) of 95, 97.5 and 100°C,
- Sink out temperatures (condenser) of 105, 110 and 110°C and
- Heating capacity at condenser of 140 kW at source in temperature of 50°C, source temperature difference of 7.5K, Sink in temperature of 100°C and Sink out temperature of 115°C

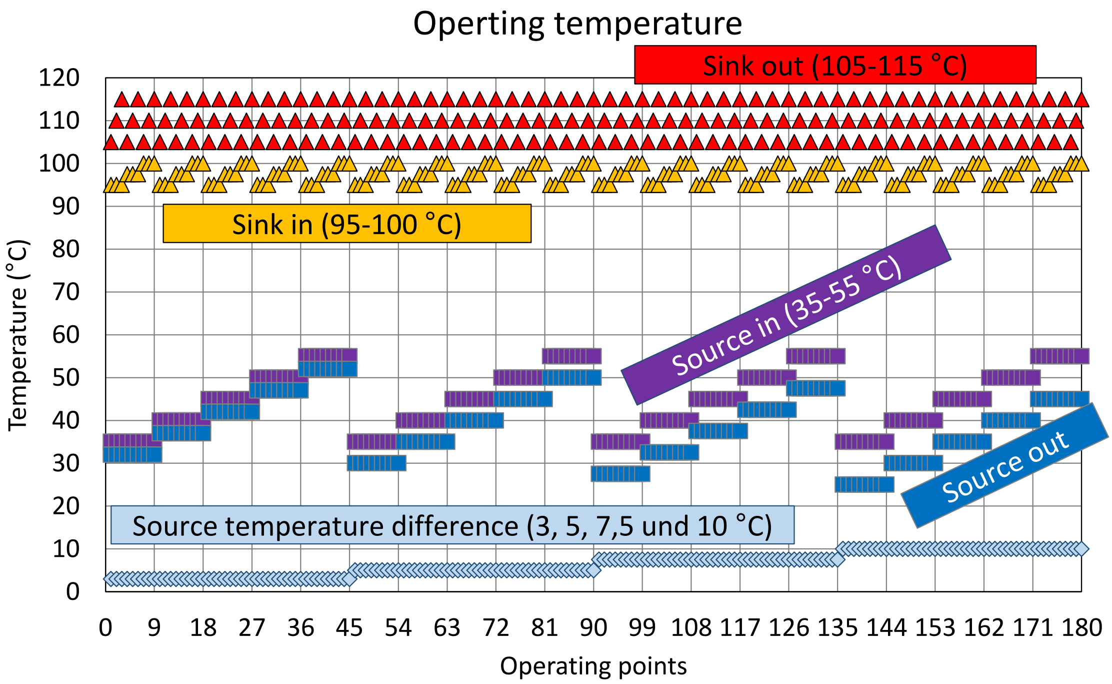

Figure 1: Expected operating range and operating conditions for heat source and heat sink (Drexler-Schmid, Lauermann, Baumhakel, & Helminger, 2018)

The following diagrams in Figure 2 show the investigated refrigeration cycle configurations. The main components of the refrigeration cycles are given at the refrigeration cycle “Single stage” and are the compressor (Co), the condenser (C), the expansion valve (ExV) and the evaporator (E). In the "EVI" refrigeration cycle (marked with b) in Figure 2), an economizer is installed which provides a vaporous refrigerant for injection into the compressor. The refrigeration cycle configuration “Two cycle” (marked with c) in Figure 2) consists in principle of two refrigeration cycles “Single stage”, in which each compressor circulates half the refrigerant. In this way, one of the two refrigerant circuits does not have to overcome the entire temperature difference between the heat source and the heat sink. The use of an ejector is a method of increasing the efficiency and reducing losses. In contrast to the refrigeration cycle configuration "Single stage", the compressor does not have to overcome the entire pressure ratio between condenser and evaporator. “Cascades” (marked with d) in Figure 2) are divided into a low temperature refrigeration cycle (low temperature cycle for short) and a high temperature refrigeration cycle (high temperature cycle for short). In the “Cascade”, each individual refrigeration cycle has to overcome a lower pressure ratio than in the "Single stage" configuration. This means that the compressor in each refrigeration cycle can be operated closer to its design point and thus a higher compression efficiency can be achieved. Both refrigeration cycles are connected by a heat exchanger (so-called evaporator condenser) in which the refrigerant of the low temperature cycle condenses and the refrigerant of the high temperature cycle evaporates simultaneously. In the refrigeration cycle "COS-ejector" (marked with e) in Figure 2), the refrigerant evaporates in two evaporators marked with E1 and E2. The cycles are described in more detail in (Drexler-Schmid, Lauermann, Baumhakel, & Helminger, 2018) and (Helminger, Lauermann, Baumhakel, & Drexler-Schmid, 2018).

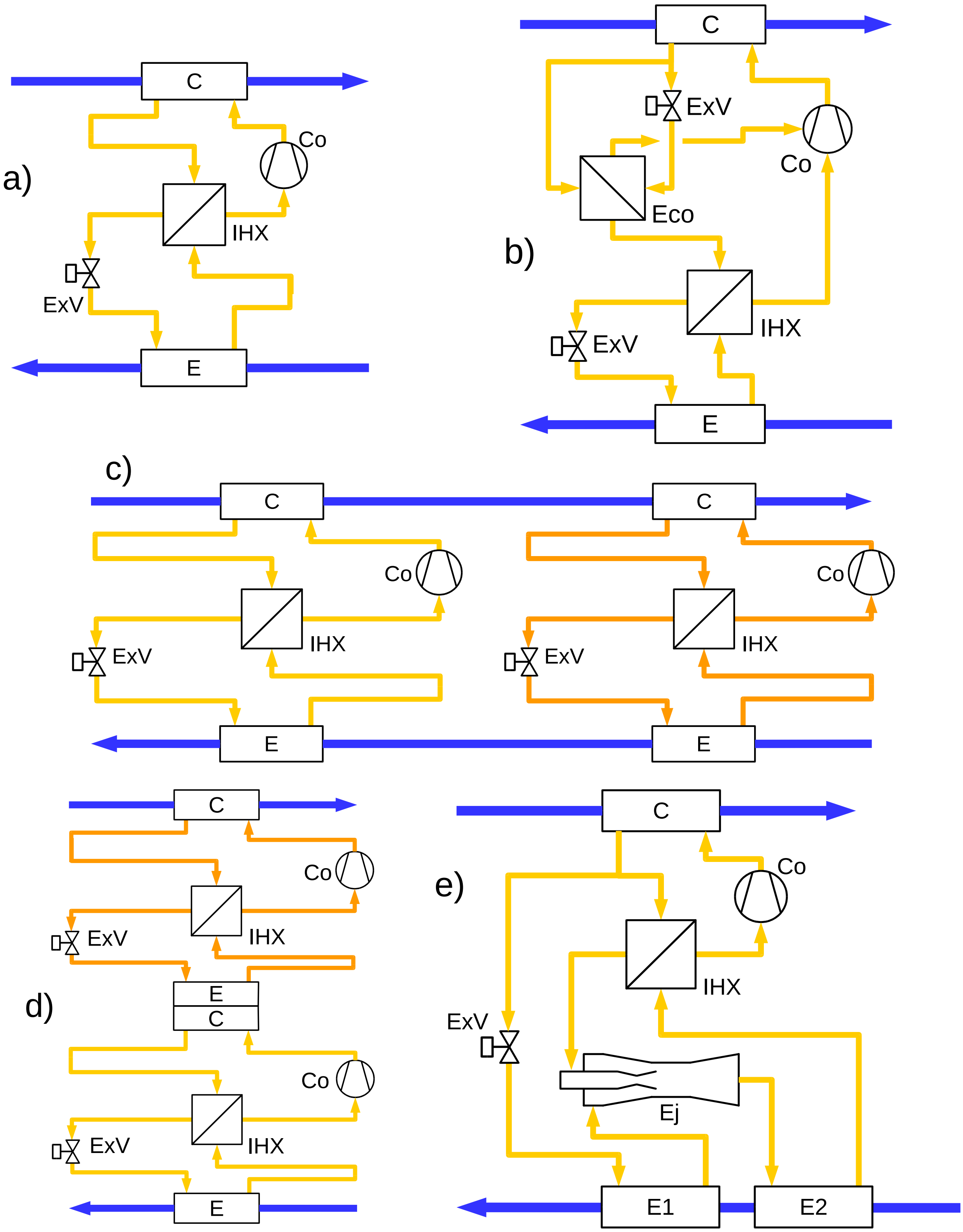

Figure 2: Schematic sketch of refrigeration cycles a)„Single stage“, b)„EVI“, c)„Two cycle“, d)„Cascade“ and e)“COS-Ejector“ (Drexler-Schmid, Lauermann, Baumhakel, & Helminger, 2018)

In all refrigeration cycle configurations optional internal suction gas superheaters (IHX) were provided. In the simulation, the function of the IHX is only activated if this is necessary for operation. Preselected refrigerants are assigned to the investigated refrigeration cycle configurations, resulting in 14 configurations, which are summarized in Table 1. The modelling and simulation was done with models in Modelica™. The TIL-library was used for the thermal modelling, which is widely used in refrigeration engineering. (Helminger, Lauermann, Baumhakel, & Drexler-Schmid, 2018)

Table 1: Refrigerants and regrigeration cycles given by (Drexler-Schmid, Lauermann, Baumhakel, & Helminger, 2018) and (Helminger, Lauermann, Baumhakel, & Drexler-Schmid, 2018)

Refrigeration cycle	Refrigerants
Single stage	R600 R1234ze(Z)
EVI (enhanced vapour injection)	
Two cycle	
COS-Ejector	
Cascade Low temperature cycle/high temperature cycle	R717/R1234ze(Z) R1234ZE(E)/R1234ze(Z) R717/R600 R600a/R600 R600a/R1234ze(Z) R717/R1336mzz-Z

4. RESULTS REGARDING COMPRESSOR SIZE AND IHX

In this section the different compressor sizes are compared based on the displacement volume. The displacement volume of the refrigerant circuit "Single stage" with the refrigerant R600 is used as a reference and the other 13 variants are compared. The required heating capacity in the TSA process is 140 kW at 50°C evaporator inlet temperature (source in) and 115°C condenser outlet temperature (sink out, see chapter 3). Figure 3 shows the relative displacement volume of each configuration to the reference configuration (circle with white filling). For all configurations (except the reference configuration) the overall displacement volume is shown as a black diamond. In the "Two cycle" configurations with R600 and R1234ZE(Z) the displacement volume of each refrigerant cycle is shown as a diamond with a white filling. For the cascades, the displacement volume of the low temperature cycle is shown as a square and the displacement volume of the high temperature cycle as a triangle.

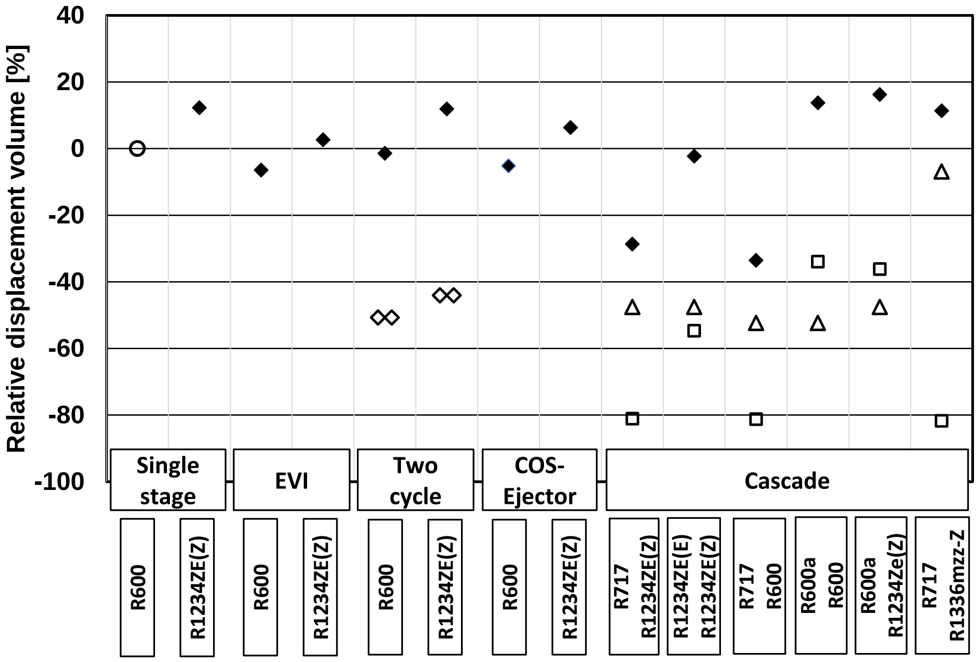

Figure 3: Relative displacement volume to configuration "Single stage" with R600

The use of the IHX in each configuration is analyzed on the basis of the calculated suction gas superheating. Figure 4 shows how the suction gas superheating behaves in the configuration “Single stage” with R600 at each operating point. It is shown how the calculation results are distributed depending on the operating temperatures.

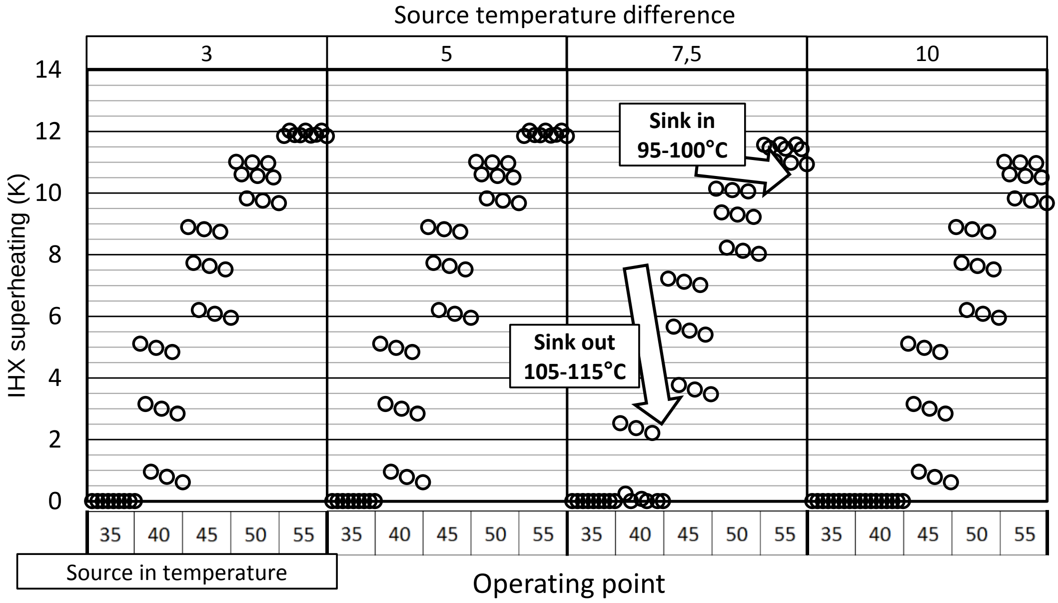

Figure 4: Suction gas superheating in the IHX of the configurations "Single stage" with R600

Figure 5 shows the suction gas superheating for each operating point of the configurations "Single stage", "EVI", "Two cycle" and "COS-Ejector" with R600 and R1234ZE(Z), respectively. Operating points where no suction gas superheating is required are not shown.

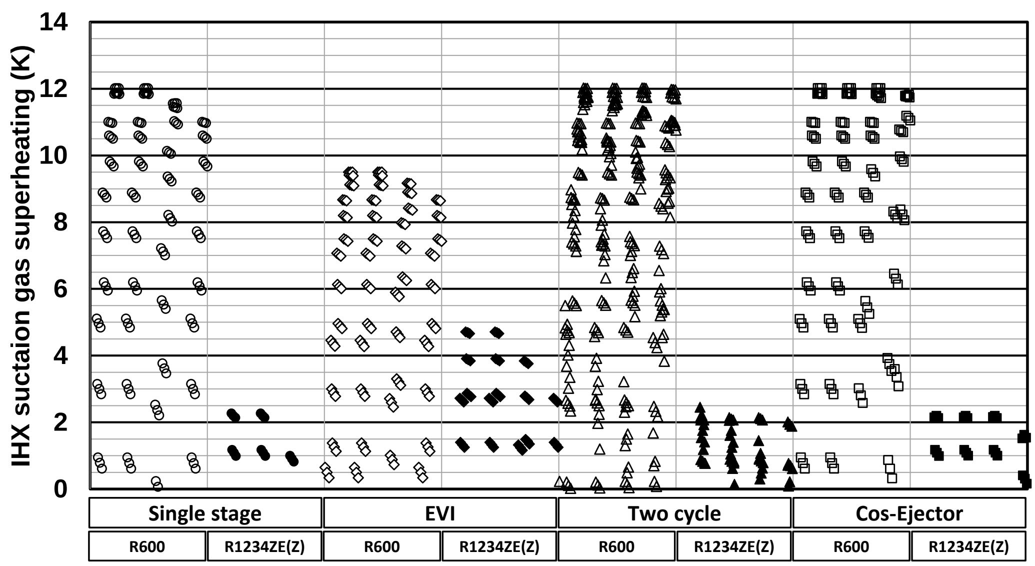

Figure 5: Suction gas superheating in the IHX of the configurations „Single stage“, „EVI“, „Two cycle“ and „COS-Ejector“ with R600 and R1234ZE(Z)

The calculation results of the required suction gas superheating in each low temperature and high temperature cycles of the cascades are summarized in Figure 6. The operating points that do not require suction gas superheating are also shown. For refrigerant circuits with R717, this is expected to be the case at all operating points.

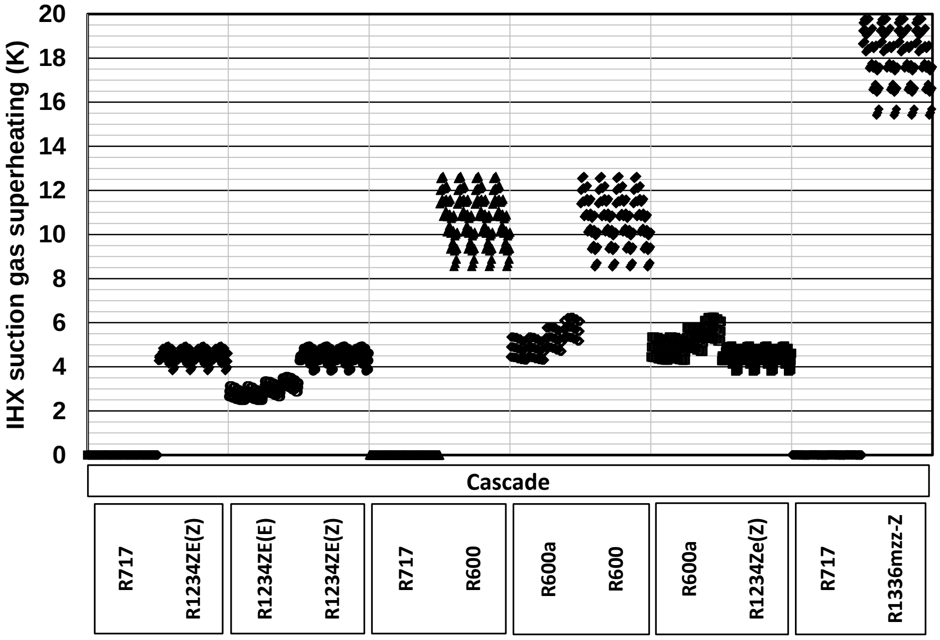

Figure 6: Suction gas superheating in the IHX in "Cascade" configurations

5. CONCLUSIONS

As expected, the evaluation of the displacement volume in Figure 3 of the compressors shows that the refrigeration cycle configurations "Single stage", "EVI", "Two cycle" and "COS-Ejector" with R1234ZE(Z) require a slightly larger displacement volume than refrigeration cycle configurations with R600. A lower displacement volume is required for the "EVI" and "COS-Ejector" refrigeration cycle configurations. In the case of cascades, the total displacement volume can be significantly lower in some cases compared to the reference by using R717 in the lower temperature cycle. Only the cascade R717/R1336mzz-Z requires a slightly higher overall displacement volume, which is due to the displacement volume of the high temperature cycle with R1336mzz-Z. The cascade with R1234ZE(E)/R1234ZE(Z) results in approximately the same total displacement volume. The cascades R600a/R600 and R600a/R1234ZE(Z) require a slightly higher total displacement volume.

When evaluating the suction gas superheating, IHX for refrigeration cycles with a required suction gas superheating below 3K can be omitted ("Single stage", "Two cycle" and "COS-Ejector" with R1234ZE(Z)). If the suction gas superheating is between 3 and 6K ("EVI" with R1234ZE(Z)), it should be checked if the operating range can be restricted and whether an IHX is still required. For suction gas superheating above 6K (R600), an IHX is recommended unless a significantly restricted operating range indicates a low required suction gas superheating. As expected, no IHX is required for cascades with R717 in the low temperature cycle. For cascades with suction gas superheating of about 5 K, either the low temperature cycle or the high temperature cycle could be equipped with an IHX and the IHX in the other cycle could be omitted. The high temperature cycles with R600 or R1336mzz-Z in the respective cascades should be equipped with an IHX.

The numerical assessment of the heat pump configurations in this work, the influence of the different heat pump cycles and different refrigerants is shown. The results provide information on the necessary compressor capacity and the question whether or not a suction gas superheater is needed for the specific application of biogas upgrading. Both of these components play a significant role in investment costs.

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NOMENCLATURE

C	Condenser	Co	Compressor
COS	Condenser outlet split	E	Evaporator
Eco	Economizer	Ej	Ejector
EVI	Enhanced vapour injection	ExV	Expansion valve
IHX	Internal suction gas superheater	TSA	Temperature swing adsorption

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