SIMULATION OF INTEGRATED HEAT PUMP SYSTEMS FOR HEATING AND COOLING OF LOW-HEATING-ENERGY BUILDINGS

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Abstract: In low-heating-energy buildings heat pumps have become an attractive heating system, due to a notable reduction of the space heating requirements and the introduction of low temperature heating systems, which enable high efficiencies. Comfort cooling becomes also more interesting in the residential sector, especially in lightweight buildings with large south-oriented windows, which partially tend to overheat in summer.

A heat pump has the unique feature that heating and cooling can be provided with the same system and also at the same time (domestic hot water and cooling). Thus an integrated heat pump system with the functions heating, cooling and preparation of domestic hot water is an interesting solution for the energy supply of low-heating-energy buildings.

Three different system concepts for integrated heat pump systems have been analyzed in detail by means of simulations. The heat pump cycle was modelled including all heat exchangers and was analyzed concerning the efficiency and possible improvements using different refrigerants (R134a, R290, R744). For the further investigations, which will include the construction of a prototype and a detailed dynamic simulation of the system, a brine/water system with the refrigerant R744 will be used.

Key Words: ***System Analysis, Comparison of Refrigerants, Multifunctional Heat Pump***

1 INTRODUCTION

The energy demand of buildings has decreased drastically in the last 25 years. This is due to a rapid development of building materials and construction techniques. In Austria today's buildings have only about 25% of the space heating demand (40-50 kWh/(m².a)) of the average building that was built 30 years ago (200 kWh/(m².a)). With low energy buildings and passive houses the following problems arise:

- Conventional heat generators (oil, gas or biomass boilers) are hardly available with capacities lower than 10 kW.
- A connection to the gas grid or to a district heating system is - due to economical reasons - often not possible for buildings with such low heat demands.
- The heat load necessary for the preparation of DHW is often higher than the heat load for space heating.
- Lightweight buildings with large south oriented window areas have a risk of overheating, if shading devices are not available or not properly used.

A possibility to solve these problems is a multifunctional heat pump that can cover space heating and cooling, both in combination with the preparation of DHW. While the latter is an additional load during the heating season, the condenser heat can be used for the DHW preparation in summer, when space cooling is necessary.

Within a project funded by the Austrian Ministry of Traffic, Innovation and Technology, that is carried out in the framework of Annex 32 within the Heat Pump Programme (HPP) of the International Energy Agency (IEA) such an integrated heat pump system is currently developed at the Institute of Thermal Engineering. In the first project phase three different system concepts for an integrated heat pump system have been studied by means of simulations. The objective of the simulations was to find out, which efficiencies can be achieved with different refrigerants in the different systems, in order to enable a decision for the refrigerant and the system to use for the further work. The basic concepts of the systems are presented in the following section.

2 SYSTEM CONCEPTS FOR INTEGRATED HEAT PUMPS

3 different system concepts are investigated for an Austrian climate (design ambient temperature: -12°C, heating degree days $HDD_{20/12}$: 3200 Kd/a).

2.1 System 1: Reversible Air/Air Heat Pump, Air Heating System

This system is a fresh air heating system with controlled ventilation. The heating and active cooling of the building is done solely via the hygienically necessary air exchange (0.4 h^{-1}). Therefore the heating and cooling capacity is limited to about 10-12 W/m². Thus this system is suitable only for buildings with a passive house standard (ultra low energy buildings with a heating demand of <15 kWh/(m^2.a)). If higher heating or cooling capacities are required, the air flow rate and thus the air exchange rate would have to be increased, as otherwise unacceptable high (or low) air supply temperatures would be required. Due to the increased air exchange the ventilation losses of the building would be increased. Additionally the increased air exchange would cause a very dry indoor air at low ambient temperatures due to the low absolute humidity of the supply air.

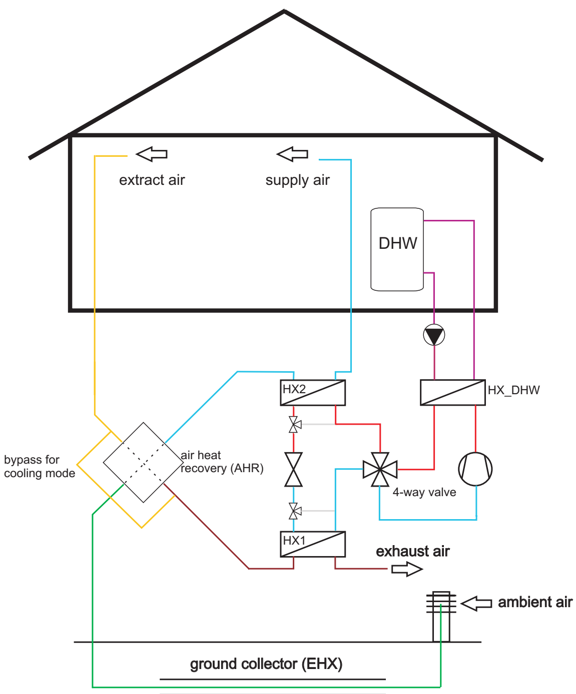

Figure 1: Schematic of System 1 (simplified)

A schematic of the system design is shown in Figure 1. The ambient air is pre-heated via a ground collector (EHX), which consists of pipes, buried in the ground in a depth of about 1.5 m. The EHX ensures an air temperature higher than 0°C at the outlet, in order to prevent a freezing of condensate on the exhaust air side of the air heat recovery heat exchanger (AHR). In the summer the ambient air is pre-cooled in the EHX.

The AHR is a cross flow heat exchanger, in which heat is transferred from the extract air from the rooms to the fresh air coming from the EHX. The AHR is assumed to achieve a heat recovery rate of 70%.

The heating and cooling of the supply air and the preparation of domestic hot water (DHW) is done via a "reversible" heat pump, i.e. the flow direction of the refrigerant in a part of the cycle can be reversed via a 4-way-valve (compare Figure 1). The following operation modes of the heat pump are possible:

<u>Heating mode:</u> The ambient air is preheated in the EHX and the AHR and reheated to the temperature necessary to cover the heating demand via the condenser of the heat pump situated in the supply air duct (HX_2). Alternatively the reheating can be done via an electrical

heater (not shown in Figure 1), if the heat pump is charging the DHW tank. The extract air flow from the rooms enters the AHR, thereby transfers a part of its heat to the fresh air, and then enters the evaporator situated in the exhaust air duct (HX_1), where it is used as the heat source for the heat pump. The second condenser for the preparation of DHW (HX_{DHW}) is only flown through by fresh water in the DHW mode.

DHW mode: The heat exchanger HX_{DHW} acts as the condenser in the DHW mode during charging the DHW tank. HX_2 is bypassed and the re-heating of the supply air is done via the electrical heater, which is installed in the supply air duct. HX_1 acts as the evaporator of the heat pump. A combined operation mode with simultaneous heating and DHW preparation is possible, if HX_{DHW} is used as a de-superheater and HX_2 as the condenser of the heat pump.

Cooling mode: In the cooling mode the air heat recovery is bypassed, as the air coming from the EHX is normally colder than the extract air. The heat pump process is reversed via the 4-way-valve. The heat exchanger installed in the supply air duct (HX_2) acts as the evaporator of the heat pump, cooling the air coming from the EHX to the temperature necessary to cover the cooling demand. The heat exchanger in the exhaust air duct (HX_1) acts as the condenser. In the “reverse mode” the refrigerant flows through these heat exchangers in the reverse direction, therefore they are operated in parallel-flow instead of counter-flow. With regard to the condenser this is a disadvantage, as the process has to be operated with higher condensation temperatures.

In case of a simultaneous demand for DHW (if the tank is not fully charged) the condensation of the refrigerant is done in the second condenser for the preparation of DHW (HX_{DHW}) and HX_1 is bypassed.

2.2 System 2: Brine/Water Heat Pump, Hydronic Heating System

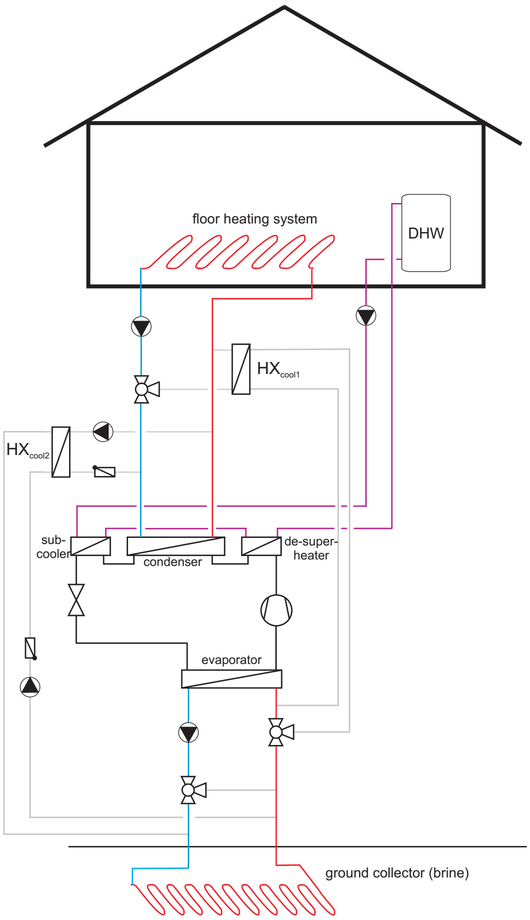

Figure 2: Schematic of System 2 (simplified)

In this system a hydronic floor heating system is used both to heat and to cool the building. Controlled ventilation is not used, the air exchange is done solely via window ventilation assuming an air exchange rate of 0.4 h^{-1}. Using a hydronic heating system much higher heating and cooling capacities can be achieved compared to an air heating system.

A schematic of the system design is shown in Figure 2. The heat pump unit consists of a non-reversible refrigerant cycle. A ground/brine-collector is used as the heat source of the heat pump and partially also as the heat sink in the cooling mode.

Heating mode: In the heating mode the condenser of the heat pump is used to heat the water flowing through the floor heating system to the supply temperature necessary to cover the heating demand. The evaporator of the heat pump uses the brine cycle of the ground collector as the heat source.

DHW mode: For the preparation of hot water a DHW tank is used, which is charged by the heat pump. The DHW is pre-heated in a subcooler and then re-heated in a de-superheater, which is connected to the subcooler in series on the water side. The evaporator of the heat pump uses the brine cycle of the ground collector as the heat source.

Cooling mode: In the cooling mode the floor heating system is used to cool the living space. With this system three different cooling modes are possible:

a) Active cooling with simultaneous preparation of DHW:
The heat collected in the floor heating system is transferred to the brine cycle (and further to the evaporator) via the heat exchanger HX_{cool1}. The condenser heat is used to charge the DHW tank via the subcooler and the de-superheater. In this mode both the warm and the cold side of the heat pump are utilized.

b) Active cooling:
If the DHW tank is already fully charged, the condenser heat can alternatively be transferred to the brine cycle via the additional heat exchanger HX_{cool2}. This also requires an additional pump in the brine cycle. For buildings with a small cooling demand, like it is usually the case in residential buildings in Austria, it is thinkable to omit active cooling (without simultaneous DHW preparation), i.e. no HX_{cool2} and second connection to the brine cycle are required.

c) Passive cooling:
As the brine loop is connected to the floor heating system via HX_{cool1}, it is possible to perform passive cooling with the ground heat exchanger acting as the heat sink, i.e. without operation of the heat pump.

2.3 System 3: Reversible Brine/Water Heat Pump, Hydronic Heating System

System 3 is very similar to System 2. The main difference is that a reversible heat pump cycle is used. In comparison to System 2 this has the advantage that HX_{cool1}, HX_{cool2} and the additional pump in the brine cycle (compare Figure 2 and Figure 3) are not necessary. This saves costs and shall increase the efficiency of the heat pump process, as the temperature differences occurring in HX_{cool1} and HX_{cool2} can be omitted. On the other hand the heat pump cycle is more complex and passive cooling is not possible.

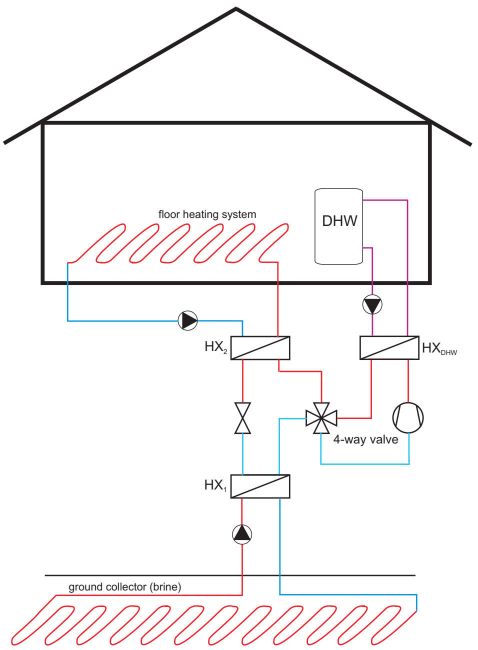

Figure 3: Schematic of System 3 (simplified)

Heating mode: In the heating mode the condenser of the heat pump is used to heat the water flowing through the floor heating system to the supply temperature necessary to cover the heating demand. The evaporator of the heat pump extracts heat from the brine cycle of the ground collector.

DHW mode: For the preparation of hot water a DHW tank is used, which is charged via a second condenser (HX_{DHW}). A combined operation mode with simultaneous heating and DHW preparation is possible, if HX_{DHW} is used as a de-superheater and HX_2 as the condenser of the heat pump.

Cooling mode: In the cooling mode the heat pump cycle is reversed via the 4-way-valve. Thus both HX_2, which is used as the evaporator in this mode, and HX_1 are operated in parallel-flow, as the flow direction of the refrigerant is reversed. This has the same effects as in System 1 (compare section 2.1). Two different cooling modes can be used:

a) Active cooling with simultaneous DHW preparation:
HX_{DHW} is used as the condenser in order to charge the DHW tank. Thus both the warm and the cold side of the heat pump are utilized.

b) Active cooling:
If the DHW tank is already fully charged, the condenser heat is discharged into the brine cycle of the ground collector via HX_1.

Passive cooling is not possible with this system.

2.4 System Choice

All three system concepts have been analyzed in detail by means of simulations of the refrigerant cycle. The further investigations within the project will be carried out with only one system. The choice was made for System 2 mainly due to the following reasons:

- The system to be developed shall be suitable for a broad range of low energy buildings with different heating demands. System 1 (fresh air heating system) can only be applied in buildings that have a heating demand of < 15 kWh/(m².a).
- Air heating systems are hardly used in residential buildings in Austria, where mainly hydronic heating systems are installed. Therefore it is expected, that the user acceptance for an integrated heat pump in combination with a hydronic heating system will be far higher in the Austrian market.
- In general the cooling demand of residential buildings in the Austrian climate can be kept at a minimum. If only small cooling loads occur, it should be possible to cover these with passive cooling, without using the heat pump. A reversible refrigerant cycle, as it is used in System 3, is therefore not considered as necessary. However, if passive cooling is not sufficient, System 2 also offers the possibility of active cooling.

3 CYCLE SIMULATIONS

Within the scope of this article only a part of the work that was performed concerning the simulation of different system concepts can be presented in detail. As System 2 was chosen for the further investigations, this article focuses on the simulations of this system. The calculations are performed for a HFC refrigerant (R134a) and two “natural” refrigerants (R744=CO_2, R290=propane).

For the simulations of the refrigerant cycles of the different systems a steady state model, which was established in the program EES (Klein, 2007), is used. The approach used for the modelling is briefly described in section 3.3.

3.1 Boundary Conditions Concerning the Building and the Heating System

In order to define the boundary conditions concerning the heating and cooling capacities and the supply temperatures, that have to be provided by the heat pump system, as well as the return temperatures coming from the heating system, a reference low-energy-building was defined. The assumptions for the building are based on simplified considerations concerning the transmission and ventilation losses and the internal and external (solar) gains. The heating or the cooling capacity, which has to be provided by the heat pump system at a certain ambient temperature, is determined based on the values given in Table 1.

The internal and external heat sources are assumed to be constant, independent of the ambient temperature. The transmission and ventilation losses are considered to be linearly depending on the ambient temperature T_{amb}. The according heating capacity $\dot{Q}_{heat}$ and cooling capacity $\dot{Q}_{cool}$ as a function of the ambient temperature are shown in Figure 4 together with the transmission and ventilation loads and the internal and external gains.

Table 1: Assumptions concerning the building

Description	Value	Unit
Heated floor area	140	m²
Heated volume	392	m³
Transmission heat losses (incl. infiltration)*	0.068	kW/K
Air exchange rate	0.4	h^{-1}
Internal heat sources	3	W/m²
External heat sources	5	W/m²

*transmission heat loss per K temperature difference between the indoor air and the ambient air

The loads shown in Figure 4 are based on a room temperature of 20°C during the heating season. In the cooling mode the room temperature is chosen depending on the ambient temperature in order to fulfil the requirements of the German Standard DIN 1946-2 (1994). In the cooling mode the air exchange rate is assumed to be increased to 2 h^{-1} at ambient temperatures lower than 26°C (lower than the assumed room temperature). This measure is assumed to be achieved by opening the windows and it reduces the cooling demand (compare Figure 4: reduction from $\dot{Q}_{cool*}$ to $\dot{Q}_{cool}$).

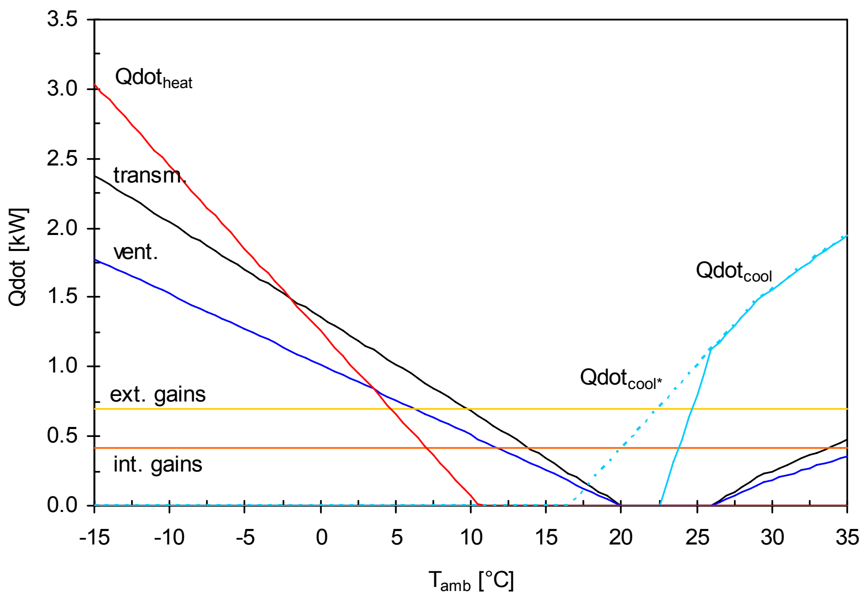

Figure 4: Loads as a function of the ambient temperature T_{amb}, based on the assumptions given in Table 1

Using these assumptions results in an annual heating demand of 37 kWh/(m².a) in an average climate (Meteonorm, 2005) of Graz. The annual cooling demand is 2.5 kWh/(m².a).

Both heating and cooling are assumed to be done via a hydronic floor heating system. The supply and return temperatures at the design ambient temperature of -12°C are chosen with 35 and 30°C respectively. The resulting supply and return temperatures as a function of the ambient temperature are calculated assuming a radiator exponent of 1.1. In the cooling mode a constant supply temperature into the floor heating system of 20°C is used. The cooling capacity is controlled via the flow rate through the floor heating system.

A brine/ground-collector is used both as the heat source and as the heat sink in the cooling mode (if the DHW tank is fully charged). As a simplification the brine temperature at the outlet of the ground heat exchanger is assumed to be only depending on the ambient temperature (see the brine inlet temperature into the evaporator in Figure 8).

3.2 Layout of the Refrigerant Cycle

Figure 5 shows the layout of the investigated refrigerant cycle. The compressor lifts the refrigerant from the evaporation pressure to the condensation pressure (1-2). Then the refrigerant enters the de-superheater (2-2a), which is used for the re-heating of DHW (W3-W4). In the condenser the refrigerant is condensed by rejecting heat from the refrigerant to the hydronic heating system (2a-2b). In the subcooler the refrigerant is subcooled (2b-3), preheating the DHW flowing on the other side (W1-W2). In the internal heat exchanger (IHX) heat is transferred from the high pressure side (3-4) to the low pressure side (6-1) of the refrigerant. An electronic expansion valve throttles the refrigerant pressure to the evaporation pressure (4-5). In the evaporator heat is absorbed by the refrigerant from the heat source at constant temperature (5-6). A low pressure accumulator is used to compensate the different amounts of refrigerant in the cycle in different operating modes. The refrigerant state at the outlet of the accumulator is assumed to be at the upper saturation line (x=1). In section 3.5 illustrations of example processes in the T-h diagram are shown together with the results of the simulations.

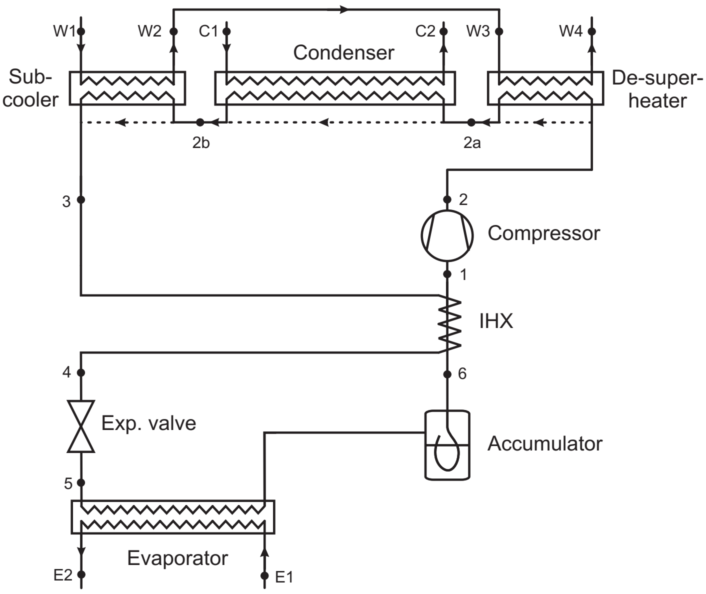

Figure 5: Layout of the investigated heat pump cycle

The same type of control is assumed for all refrigerants. An electronic expansion valve is used to adjust the high-side pressure according to the current operating mode. For every operating point an optimum high-side pressure exists. For every operating mode the advantage of using the optimum high-side pressure in comparison to a fixed high-side pressure in the individual modes is investigated in the simulations.

3.3 System Components and Assumptions

<u>*Heat exchangers:*</u> All heat exchangers are modelled as counter-flow heat exchangers using the UA-LMTD method (Incropera and DeWitt, 2002). As the geometry of the heat exchangers is not considered, a detailed calculation of the heat transfer coefficients and the pressure drop is not possible. Thus the pressure drop and the associated temperature drop in the heat exchangers is not considered. All heat exchangers are calculated using a certain product of the heat exchange surface and the average overall heat transfer coefficient (UA). A reference UA is defined for each heat exchanger and each refrigerant at a certain design point, assuming a minimum temperature difference (ΔT_{min}) of 5 K between the two fluid sides. For the IHX ΔT_{min} is chosen with 1 K. The UA of the condenser (gas cooler for space heating) for CO_2 is also chosen in order to reach a ΔT_{min} of 1K. This is done, as the convective heat transfer coefficient is assumed to be higher for CO_2 than for the other refrigerants (compare Rieberer, 1998).

For operation conditions different from the design point the UA of the heat exchanger is adapted according to simplified assumptions depending on the mass flow rate on both fluid sides. In case of a fluid on one or both sides that has a varying c_p over the heat exchanger length, the heat exchanger is subdivided into several elements, in which c_p is approximately constant.

Compressor: For all simulations it is assumed that a speed controlled hermetic compressor is used. Concerning the compressor efficiency the overall isentropic efficiency η_{is} and the volumetric efficiency η_{vol} are of importance. The definitions are:

$$\eta_{is} = \frac{P_{is}}{P_{mot}} = \frac{(h_{ref,2,is} - h_{ref,1}) \cdot \dot{m}_{ref}}{P_{mot}} \tag{1}$$

and

$$\eta_{vol} = \frac{\dot{m}_{ref} \cdot v_{ref,1}}{\dot{V}_{swept}} \tag{2}$$

The overall isentropic efficiency reflects the real power requirement (motor power, P_{mot}) for the compression compared to an isentropic compression of the refrigerant from the pressure at the compressor shell inlet (p_{suc}) to the pressure at the compressor shell outlet (p_{dis}), including all losses in the shell. The volumetric efficiency is the ratio of the refrigerant mass flow rate ($\dot{m}_{ref}$) to the theoretical mass flow rate given by the swept volume flow rate ($\dot{V}_{swept}$) and the specific refrigerant volume at the suction port ($v_{ref,1}$). The thermodynamic states of the refrigerant during the compression are shown schematically in the log p-h diagram in Figure 6.

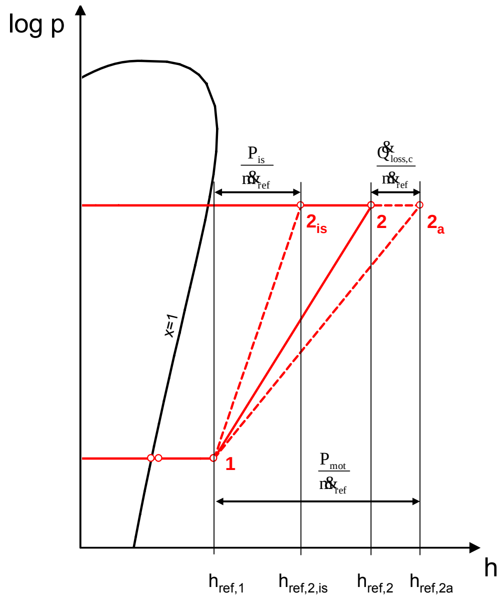

Figure 6: Illustration of the compression process in the log p-h diagram

For the isentropic and the volumetric efficiency data is used, which was derived from performance data sheets of compressors with a suitable capacity. Both efficiencies are considered in dependence of the pressure ratio ($\pi = p_{dis} / p_{suc}$) and the compressor speed (n). Exemplary the assumed efficiencies of the CO_2-compressor are shown in Figure 7.

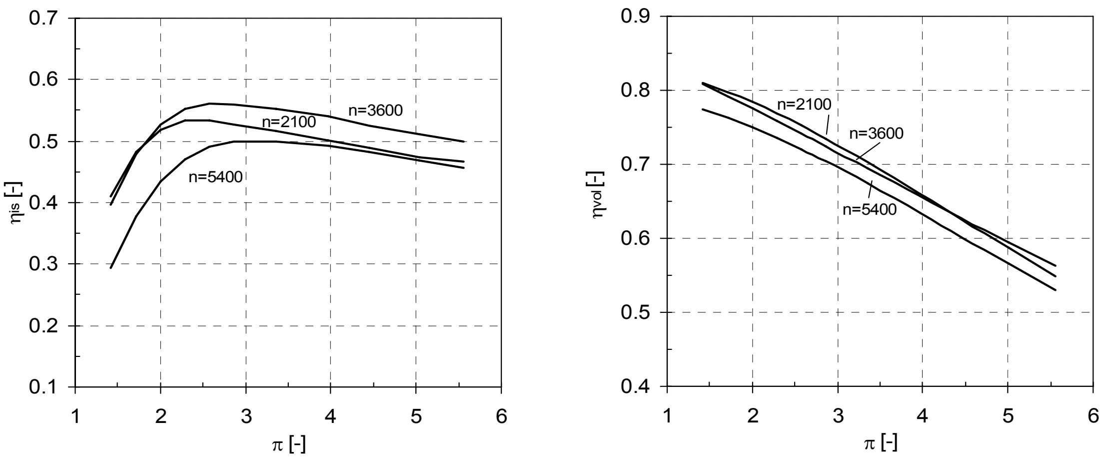

Figure 7: Overall isentropic (left) and volumetric (right) efficiency of the CO_2-compressor in dependence of the pressure ratio (π) and the compressor speed (n)

The heat losses of the compressor ($\dot{Q}_{loss,c}$) are calculated according to equation (3), where T_{sur} is the temperature of the air surrounding the heat pump, which is assumed with 15°C. The factor $f_{loss,c}$ is assumed with $3x10^{-3}$ [W/(W.K)]. With these assumptions the heat losses of the compressor are in the range of 10-20% of P_{mot} depending on the temperature difference.

$$\dot{Q}_{loss,c} = P_{mot} \cdot f_{loss,c} \cdot \left(\frac{T_{ref,2a} + T_{ref,1}}{2} - T_{sur} \right) \tag{3}$$

3.4 Calculation of the COP and the SPF

<u>COP:</u> The Coefficient of Performance (COP) of the refrigerant cycle is calculated as the useful capacity output (depending on the mode of operation) divided by the electrical power requirement of the compressor. For the calculation of the COP the electricity consumption of the circulation pumps (hydronic and brine system) is disregarded.

<u>SPF:</u> The seasonal performance factors (SPF) of System 2 are determined using the calculated COPs of the different refrigerants in the heating mode, the combined heating & DHW mode and the DHW mode (assuming an optimum high-side pressure in each case). Cooling is disregarded, as it accounts only for a small part of the total energy demand. The distribution of the heating hours is used assuming an average climate of Graz. The annual DHW demand is assumed to be 21 kWh/(m².a) (DHW temperatures according to section 3.5.1).

3.5 Results of the Simulations

3.5.1 DHW mode

The resulting COPs as a function of the ambient temperature for the different refrigerants (R134a, R290, R744) in the DHW mode are shown in Figure 8. The preparation of DHW is done with a constant water flow rate of 50 l/h, a water inlet temperature of 15°C and an outlet temperature of 60°C. This results in a thermal capacity of 2.6 kW, which has to be provided by the condenser. The capacity of the heat pump is adjusted via the compressor speed in order to meet this requirement.

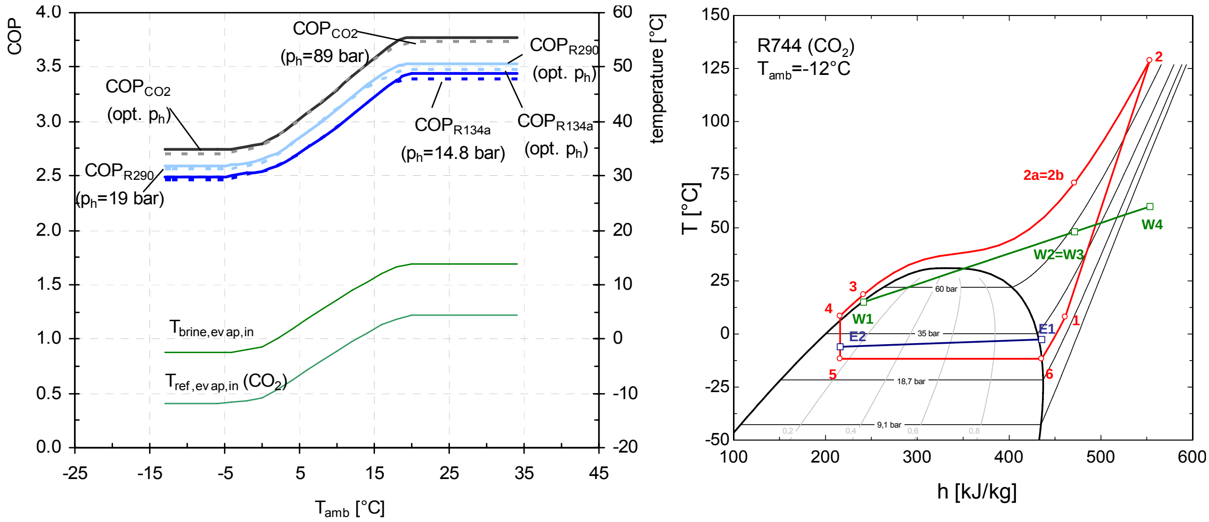

Figure 8: Left: COP in the DHW mode as a function of the ambient temperature T_{amb} for different refrigerants; brine inlet temperature into the evaporator ($T_{brine,evap,in}$), evaporation temperature ($T_{ref,evap,in}$, for CO_2); Right: illustration of an example process in the T-h diagram (refrigerant: CO_2)

For all refrigerants the differences between the results with an optimum high-side pressure (p_h) and the results with a constant p_h are quite small, as the optimum high-side pressure does not vary strongly with the ambient temperature due to the constant conditions on the water side of the condenser. Due to the advantageous thermal properties of CO_2 (temperature glide during the supercritical heat rejection), the best results are achieved with this refrigerant.

3.5.2 Heating mode

In the heating mode the compressor speed is adjusted in order to achieve the necessary heating capacity and the necessary supply temperature at a given return temperature. The swept volume of the compressor is chosen in order to meet the heat load of the building at the design ambient temperature (-12°C) with a compressor speed of 6000 min^{-1} (see Fig. 4: transm. + vent.). If the minimum compressor speed of 1000 min^{-1} is reached, the mass flow rate through the floor heating system is increased, in order not to exceed the necessary supply temperature. With these assumptions the compressor can be operated with a variable speed up to an ambient temperature of 2°C with all refrigerants (compare Figure 9). At higher ambient temperatures the heat pump is operated on/off. The losses caused by this kind of operation are disregarded, as they are assumed to be quite similar for all refrigerants.

The COP is increasing with an increasing ambient temperature for all refrigerants due to the decreasing supply temperature, the decreasing heating capacity and the increasing brine temperature and therefore decreasing condensing pressure and increasing evaporation pressure (see Figure 9).

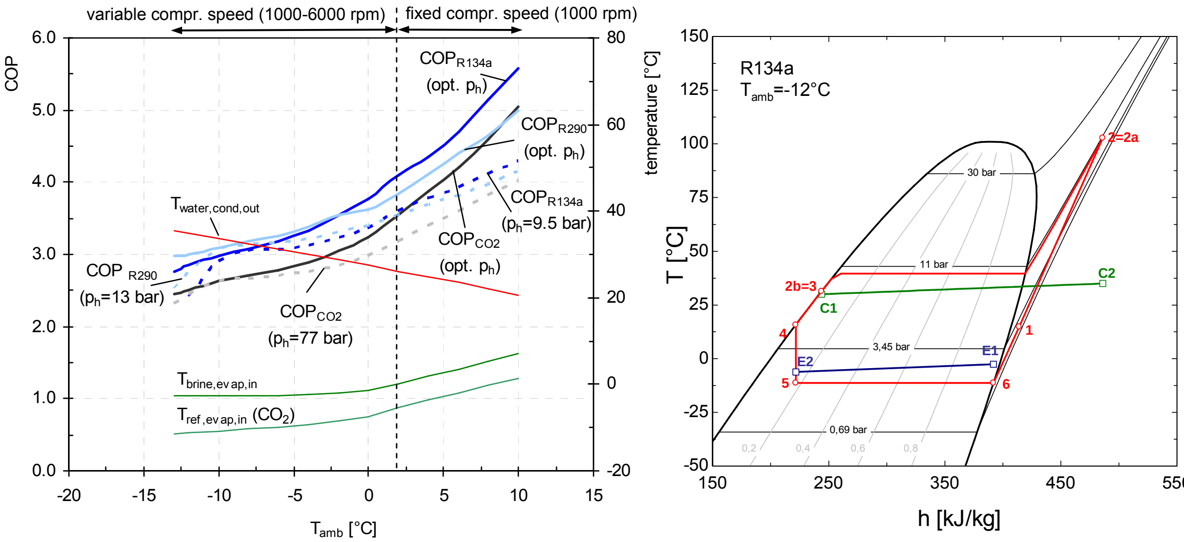

Figure 9: Left: COP in the heating mode as a function of the ambient temperature T_{amb} for different refrigerants; brine inlet temperature into the evaporator ($T_{brine,evap,in}$), evaporation temperature ($T_{ref,evap,in}$, for CO_2); Right: illustration of an example process in the T-h diagram (refrigerant: R134a)

With R290 and R134a higher efficiencies are achieved than with CO_2. The main reasons are the relatively high return temperatures from the heating system and the temperature glide in the gas cooler in the transcritical operation, which is disadvantageous for the low temperature lift in the condenser in the heating mode, and the resulting bad temperature fit between the CO_2 and the water. At higher ambient temperatures, at which the cycle can be operated with a high-side pressure below the critical pressure, the difference between the efficiency of CO_2 and the other refrigerants decreases.

The difference between the results assuming an optimum high-side pressure (p_h) and the results with a constant p_h show, that it is advantageous to adapt p_h according to the ambient temperature, as the optimum p_h varies strongly with the ambient temperature.

3.5.3 Combined Heating and DHW Mode

In the combined heating and DHW mode the compressor speed is adapted in order to achieve the necessary heating capacity at the condenser. The DHW enters the subcooler with a temperature of 15°C, the DHW flow rate is adapted in order to achieve a temperature of 60°C at the outlet of the de-superheater. If the minimum compressor speed is reached, the heat pump is controlled on/off.

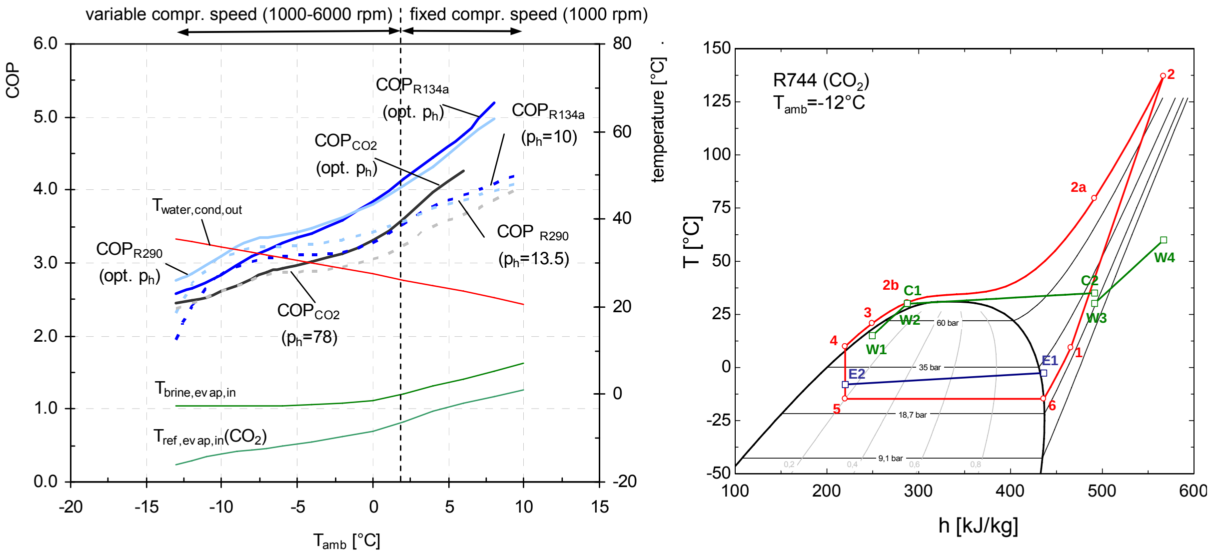

Figure 10: Left: COP in the combined Heating and DHW mode as a function of the ambient temperature T_{amb} for different refrigerants; brine inlet temperature into the evaporator ($T_{brine,evap,in}$), evaporation temperature ($T_{ref,evap,in}$, for CO_2), DHW heating capacity ($Qdot_{DHW}$, for CO_2); Right: illustration of an example process in the T-h diagram (refrigerant: CO_2)

The resulting COP as a function of the ambient temperature for the different refrigerants is shown in Figure 10. Due to the lower water inlet temperatures into the heat pump (cold water temperature entering the subcooler) the COPs are slightly higher than in the heating mode in most of the operating points for all refrigerants. Also in this mode of operation it is advantageous to adapt p_h according to the ambient temperature, as the optimum p_h varies strongly with the different supply temperatures at different ambient temperatures.

3.5.4 Combined Cooling and DHW Mode

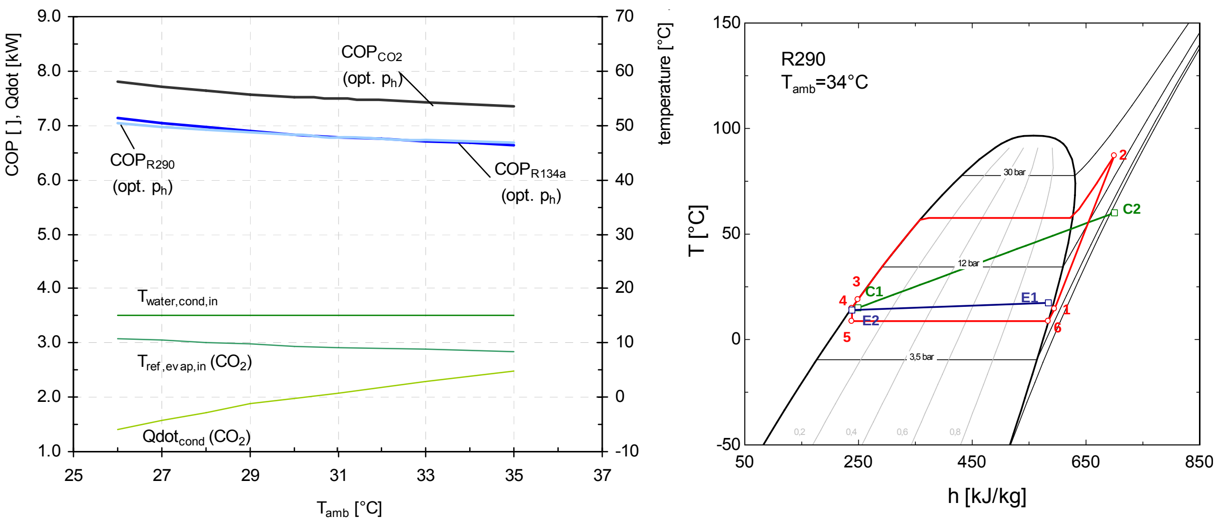

Figure 11: COP in the combined cooling and DHW mode as a function of the ambient temperature T_{amb} for different refrigerants; evaporation temperature ($T_{ref,evap,in}$, for CO_2), water inlet temperature into the condenser ($T_{water,cond,in}$), heating capacity of the condenser ($Qdot_{cond}$, for CO_2); Right: illustration of an example process in the T-h diagram (refrigerant: R290)

In this mode the evaporator is used for space cooling and the condenser heat for the preparation of DHW. The process is controlled via the compressor speed in order to achieve the necessary cooling capacity and supply temperature at the outlet of the heat exchanger HX_{cool1} (compare Figure 2). As the condenser capacity varies with the cooling capacity and therefore with the ambient temperature, the flow rate of the DHW, which enters the subcooler with a temperature of 15°C, is adapted in a way that the DHW reaches a temperature of 60°C at the outlet of the de-superheater.

The resulting COPs as a function of the ambient temperature for the different refrigerants in this operation mode are shown in Figure 11. As the optimum high-side pressure p_h is almost constant in the regarded range of the ambient temperature, the results for the operation with a constant p_h are not plotted in Figure 11. Like in the DHW mode the best efficiencies are achieved with the refrigerant CO_2.

4 CONCLUSIONS AND OUTLOOK

Three different system concepts for integrated heat pumps have been investigated. Mainly because of reasons concerning the applicability and the user acceptance a system in combination with a hydronic floor heating system was chosen for the further work. The results of the simulations - obtained with the described assumptions - lead to following conclusions concerning the refrigerant choice (System 2):

- With the refrigerant R290 the highest seasonal performance factor (SPF=3.6) can be expected with the used assumptions. However, as R290 is flammable, there are relatively strict safety regulations concerning the refrigerant content in the cycle and/or the necessary air volume in the room of installation (prEN 378, 2007).
- With CO_2 and R134a approximately the same SPF can be achieved (SPF=3.3). While CO_2 enables higher efficiencies for the preparation of DHW, R134a is advantageous in the heating mode.
- With regard to the decreasing heating energy demands of low energy buildings and the therefore rising fraction of the energy demand for DHW and due to the fact that CO_2 is a natural refrigerant, CO_2 will be used for the further investigations in the project.

The further work will include the construction of a prototype and a more detailed simulation of the refrigerant cycle, as well as a simulation of the transient interaction of the integrated heat pump with the heating system and the building.

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