CARBON DIOXIDE HEAT PUMP COMBINED WITH A SOLAR STRATIFIED STORAGE TANK

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Abstract: Different studies have shown that heat pump systems using the natural, environmentally friendly refrigerant carbon dioxide are able to achieve higher efficiencies than systems using conventional refrigerants when warming up heating water or potable hot water at a high temperature level (55°C-60°C). Instead of an isothermal condensation in conventional systems, the heat transfer always happens with continuously falling temperature in the gas cooler. In this paper it is shown, that this physical effect presents a higher optimizing potential of the high pressure side heat exchanger with regard to the efficiency of the cycle.
Furthermore, four different concepts of heat delivery are presented. Three concepts contain a direct heat delivery to the heating system from the gas cooler of the carbon dioxide cycle. In a further, fourth concept the heating system purchases heat out of the storage tank, which is charged by the heat pump. The concepts are discussed and evaluated under thermo dynamical, technical and economical aspects.

Key Words: ***Heat pump, carbon dioxide, gas cooler, low-energy-building***

1 INTRODUCTION

Within the scope of the project COHEPS which has been sponsored by the EU, several carbon dioxide heat pumps have been developed at different research facilities. Results have shown that warming up heating water or potable hot water at a high temperature level can be achieved more efficiently by a transcritical operating carbon dioxide heat pump in comparison to conventional heat pump systems (see Halozan et al. (1999), IEA (1999)). Thereby, new concepts of heat supply systems for low energy buildings have resulted.
Low energy buildings are characterized by a high level of heat insulation. The energy demand for heating water is thereby reduced. However, the amount of heat required to warm up potable hot water rises compared to the overall energy demand of the building. The heating system as a whole has to supply less heat, but considered relatively, it has to deliver heat more frequently at higher temperature demands. Furthermore, low energy buildings often have a thermal solar energy system which accompanies a stratified storage tank for peak demands (see e.g. Lemke et al. (1999), Kühl (2004), Schraps (2002), Wirth (2002)). Bühring (2001) has examined heat pump applications in low energy buildings. Hirsch (2000) has compared different systems for the heat supply of low energy buildings.
In this regard the carbon dioxide heat pump - with the ability to supply hot water efficiently and resulting in a lower mass flow rate - is an ideal extension to solar storage tanks. The combination of heat pump and thermal solar systems promises efficient usage of geothermal and solar energy.

The outlet temperature of a gas cooler has a great effect on the efficiency of carbon dioxide heat pump systems. In order to obtain a small approach temperature difference at the gas cooler, a counter-flow type heat exchanger is proposed. Carbon dioxide gas coolers were also examined e.g. by Chang et al. (2007) by means of simulation and by Hwang et al. (2005) experimentally. For further information, Nekså (2002) gives an overview of some of the activities in the CO_2 heat pump field.
This paper contains two parts. Firstly, the optimizing potential of a carbon dioxide gas cooler in comparison to a subcritical operating condenser is illustrated by means of simulation. In the second part the heat delivery at the heat sink to the space heating system is shown with different concepts. Each is discussed and evaluated under thermo dynamical, technical and economical aspects. A quantitative evaluation will not be considered.

2 OPTIMIZING POTENTIAL OF A GAS COOLER IN COMPARISON TO A CONDENSER

Due to the low critical temperature of carbon dioxide (31,1°C), heat rejection in a carbon dioxide heat pump normally takes place at a supercritical pressure. Subcritical operation, as known from conventional HFC heat pump systems, is only possible if the temperature level of the heat sink is rather low (about 20°C or lower). Higher heat sink temperatures lead to transcritical operation, where the carbon dioxide is cooled down continuously during heat rejection in a gas cooler (Rieberer 2005).
The high pressure side heat exchangers are the mayor difference between a carbon dioxide cycle and an HFC cycle. In this regard, the optimizing potential of the differently operating high pressure side heat exchangers is analyzed in this paper. Therefore, both heat exchangers are compared with equally assumed overall UA-values (in German called kA-value). To characterize the optimizing potential, only the overall UA-values will be enhanced step by step, i.e., each heat exchanger gets the same enhancement to achieve a neutral comparison.
For that purpose, a simulation model in the modelling language Modelica that was developed by Richter (2008) and based on Tegethoff (1999) is used here. In this model, the heat exchanger or heat exchange surface area is divided into a certain number of segments, which are geometrically identical. The larger the number of the segments is, the higher the level of discretization becomes. For the high pressure side heat exchangers it has become clear, that a number of 20 segments is adequate. Each segment consists of a refrigerant cell, a wall cell and a water cell. The UA-value of each segment is a fraction - in this case one twentieth - of the overall UA-value.
As opponent to the carbon dioxide (R744), the refrigerant propane (R290) as HFC is considered here. A carbon dioxide cycle as well as a propane cycle with equal boundary conditions at the heat sink and heat source was calculated by the simulation model of Richter (2008). To guarantee the comparability of the differently operating heat exchangers, the compressor efficiencies are held constant. The heat source on the evaporator side is ground water with a temperature of 10°C. The evaporation temperature of the carbon dioxide as well as of the propane is about 3,5°C. Both cycles contain an ideally operating refrigerant accumulator on the low pressure side at the outlet of the evaporator. Therefore, the suction state of the compressor is determined on the saturated vapor line.
Water is warmed up from 25°C to 60°C in the gas cooler / condenser. These temperatures are held constant with a control unit by controlling the water mass flow rate into the heat exchanger. Also the discharged heat flow of the gas cooler / condenser is held constant by means of another control unit through a theoretical adjustment of the compressor speed.
In this comparison only the optimizing potential is analysed and absolute values of COP are therefore not considered here. For this purpose, a detailed analysis of all boundary conditions especially the compressor efficiencies would be necessary. This, however, would bias the neutral comparison of optimizing potential between the gas cooler and the condenser.

Before going into the results, both differently operating heat exchangers are considered phenomenologically below.

2.1 Condenser

According to the changes of state of the refrigerant, the condenser can be divided into three parts (heat dissipation, condensation, sub cooling, see figure 1). In the temperature distribution curve of the refrigerant as a function of the enthalpy two sharp bends appear. One is at the saturated liquid line, the other at the saturated vapor line. The opposite flowing water exhibits no change in phase and is warmed up continuously. At each contact to the tube wall it has a lower temperature than the opposite flowing refrigerant. In a countercurrent

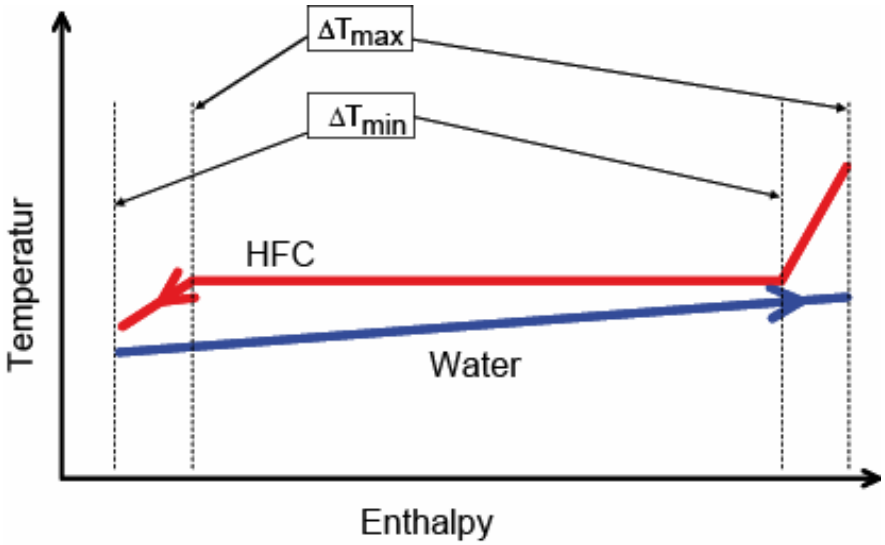


Figure 1: Indirect heat delivery into the storage tank for potable hot water supply, direct heat delivery for floor heating

heat exchanger the temperature difference between refrigerant and water becomes minimal at two points of the temperature distribution: one at the saturated vapor line when the refrigerant passes from a gaseous state into the two phase area (pinch point), the other at the water inlet / refrigerant outlet of the condenser. Furthermore, at two other points of the temperature distribution the temperature difference between refrigerant and water becomes maximal: once at the refrigerant inlet directly after compression (at the water outlet of the condenser), and again at the saturated liquid line where the refrigerant completes the condensation and gets sub cooled.

With a high temperature difference between both fluids, a relatively small heat transfer surface area is necessary. On the contrary, a low temperature difference requires a large heat transfer surface area. An improvement of the condenser by increasing this area (enlargement of the UA-value) causes the refrigerant outlet temperature to approach the water inlet temperature even further and a both sided approach at the pinch point on the saturated vapor line. However, the temperature differences at these points in the T,h diagram become so small that a further improvement of the UA-value exhibits only minimal further approach. Furthermore, the biggest temperature difference between refrigerant and water exists at the boiling point of the refrigerant at the end of the condensation. Despite an improvement of the condenser, the refrigerant cycle remains nearly at the same state points in the p,h diagram. An improvement above a certain quality grade of the heat exchanger has nearly no influence on the coefficient of performance of the refrigerant cycle.

The temperature curves as a function of the dimensionless length in figure 2 illustrate that this improvement actually is a waste of heat transfer surface area. At a UA-value above 1500 W/K more than the half of the heat exchanger operates at a temperature difference of close to 0 K.

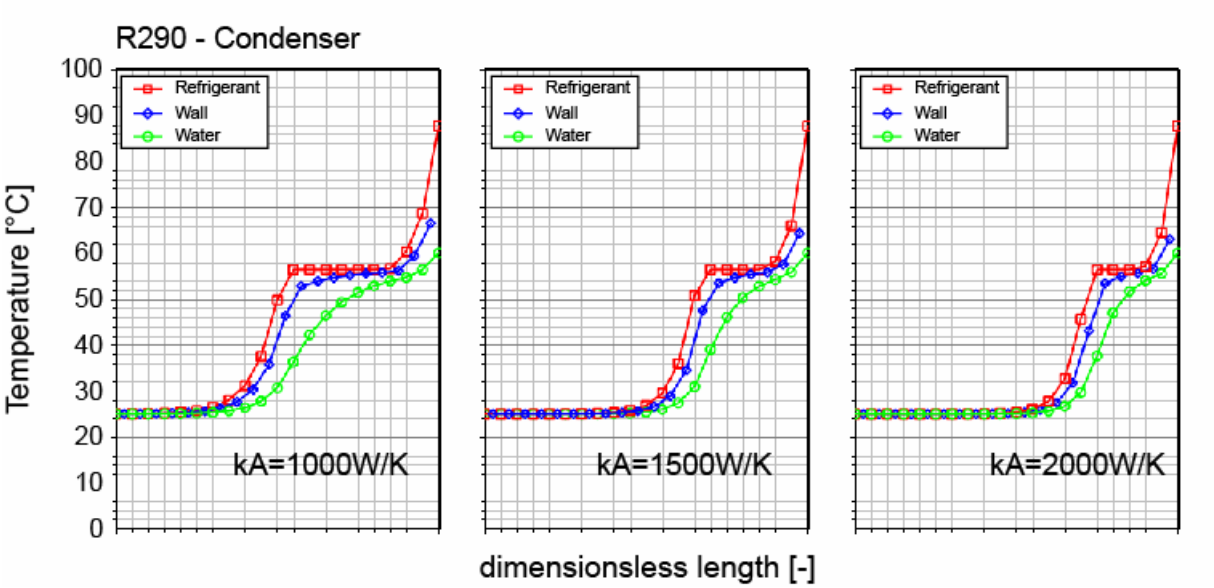


Figure 2: Temperature distribution in the condenser plotted over dimensionsless length for different UA values (german kA-values)

2.2 Gas cooler

Carbon dioxide as refrigerant does not have these distinctive sharp bends in the temperature distribution of the heat exchanger. Because of the absence of condensation due to the supercritical change of state, the heat exchanger is called gas cooler. The carbon dioxide exhibits sensible heat transfer and a continuous temperature drop through the heat exchanger. Over the whole length of the gas cooler, the temperature differences between refrigerant and water do not differ much in the several segments (cells). Therefore, an improvement of the heat exchanger (increasing the UA-value) is not limited by the abovementioned sharp bends but has a considerably higher influence on the COP. With an increase of the UA-value the temperature curves approach each other (see figure 3). Nevertheless, the temperature distribution exhibits an evenly spread temperature difference in each segment compared to the condenser in figure 2. The characteristic pinch point moves in such a manner that the high pressure can be lowered by which the cycle obtains a

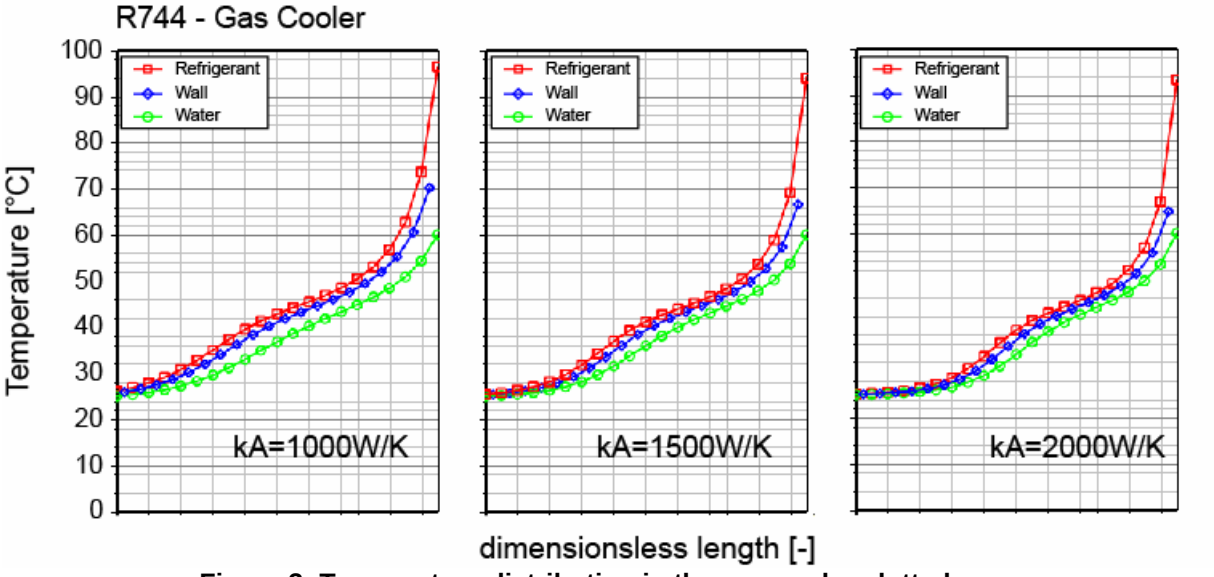


Figure 3: Temperature distribution in the gas cooler plotted over dimensionsless length for different UA-values (german kA-values)

better COP. The outlet temperature of the compressor can also be lowered. The effects of the pinch point are also discussed in Schiefelbein (2005).
Summarizing the above, the gas cooler of a carbon dioxide cycle responds considerably more sensitively to an increase of the UA-value. For this reason it is worthwhile improving the gas cooler exceedingly either by means of a larger overall heat transfer coefficient U or by a larger heat transfer surface area A while the sub critical operating condensers would have already reached the limit of improvement. In figure 4, the change of the COP is illustrated percentally by comparing the cycle with a UA-value of 500 W/K. A gas cooler improvement by increasing the UA-value from 500 W/K to 1500 W/K causes an increase of the COP of nearly 11% with this change. A condenser with a doubled increase of the UA-value already reaches the maximal improvement with an increase of the COP of nearly 3%.

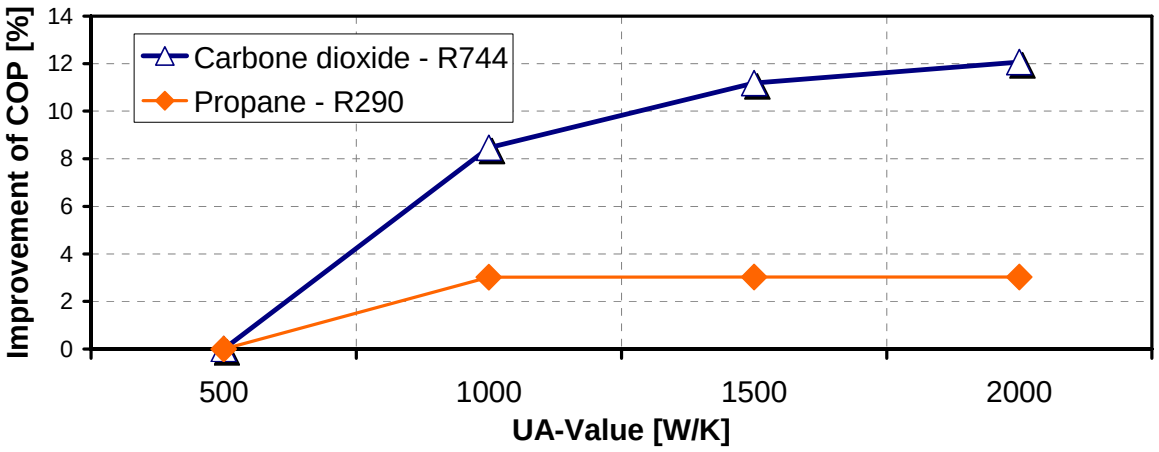


Figure 4: Improvement of COP due to the enhancement of the heat exchanger.

3 HEAT DELIVERY IN TRANSCRITICAL CARBON DIOXIDE CYCLE

In low energy buildings surface heating systems like floor heating are frequently applied. Beside ambient air, all earth-based systems are suitable as heat sources. Subsequently, different concepts of direct and indirect heat delivery with different flow arrangements of the gas cooler are considered. To guarantee a comparability of the concepts, the compressor and the evaporator are identical in each concept. The gas coolers however differ in number and arrangement, while the total length is kept constant for comparability reasons. To cover the peak demand, the supply of potable hot water must be available indirectly by means of a hot water storage tank. Hence, it should be distinguished between direct and indirect heat delivery for the heating system only and not for the hot water supply.

3.1 Heat delivery from the heat pump

3.1.1 Floor heating

Floor heating systems are generally operated at low temperature conditions. In the case of the direct heat supply, the gas cooler of the carbon dioxide cycle can be divided serially. For the potable hot water supply, water of lower temperature is taken from the lower part of the storage tank and is warmed up with a low volume flow in the first segment of the heat exchanger to a temperature level of 50-60°C and returns to the upper side of the storage tank. In the case of the floor heating a large heat exchange surface and a high water volume flow cause a considerably lower temperature rise of 28-35°C. This *low-valued* heat can be delivered by the second section of the gas cooler. In figure 5 the concept is shown schematically.

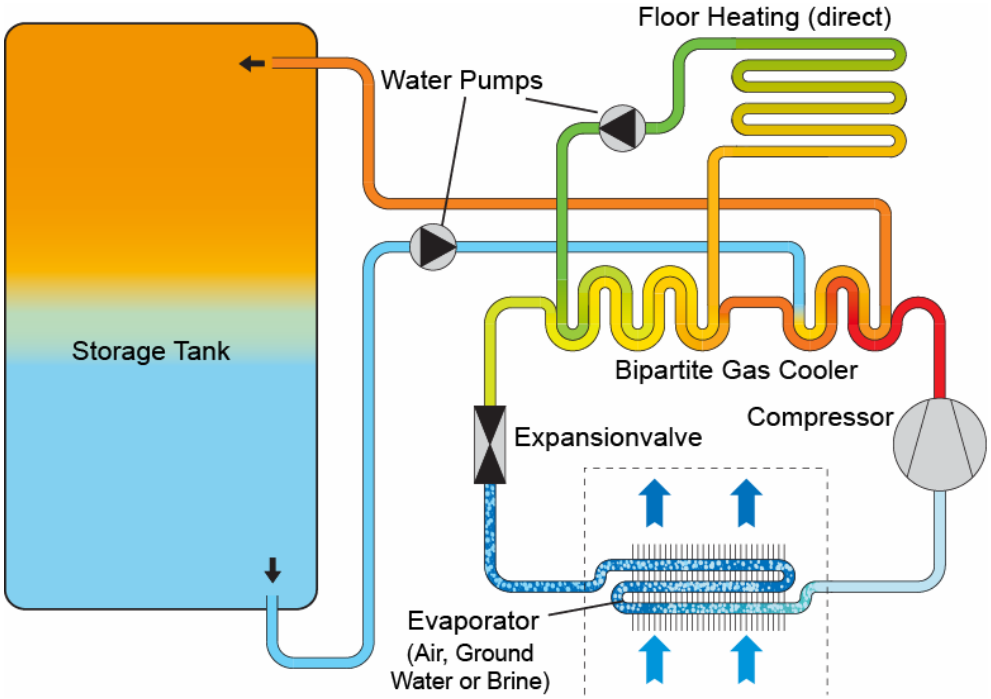


Figure 5: Indirect heat delivery into the storage tank for potable hot water supply, direct heat delivery for floor heating

In figure 6 an example is shown of the temperature curves of a gas cooler which is divided into two parts on the water side (simulation, high pressure 80 bar, length division 3:2). Due to the different gradients, the temperature of the water adapts itself to the temperature curve of the carbon dioxide in the T-h-diagram. The temperature of carbon dioxide at the refrigerant outlet of the gas cooler is limited by the temperature at the water inlet of the gas cooler, which returns from the floor heating. For a better efficiency of the carbon dioxide cycle, the water inlet temperature of the gas cooler has to be held low.

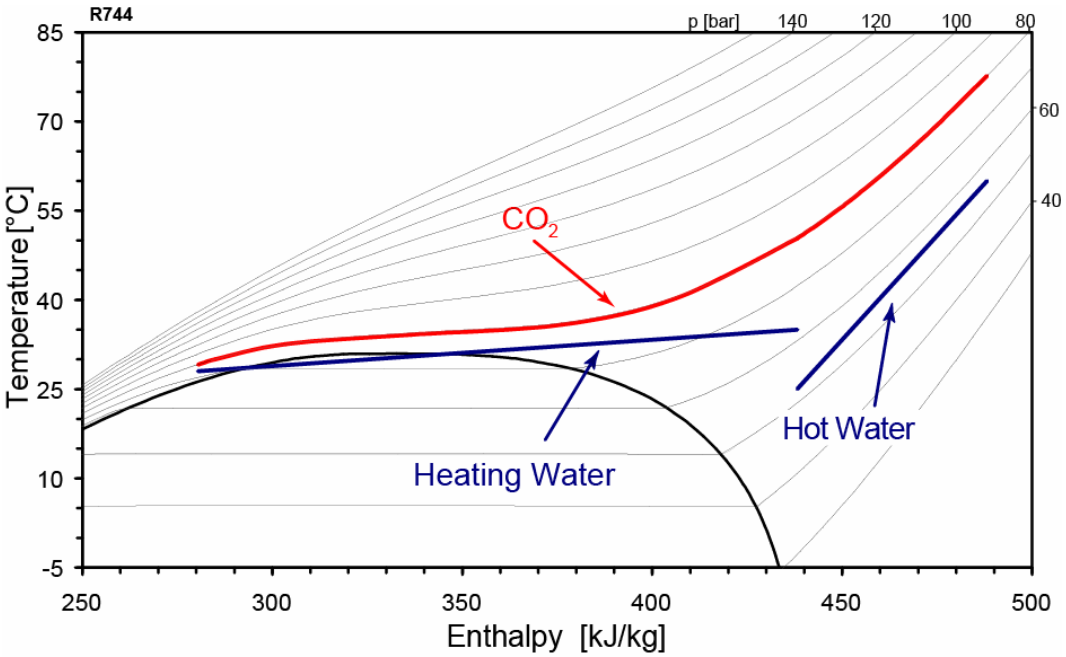


Figure 6: Temperature curves in the gas cooler in the T-h-diagram for the concept of figure 5 (simulation, high pressure 80 bar, division 3:2). Specific enthalpy of water is linear scaled to allow drawing into the carbon dioxide T-h-Diagram.

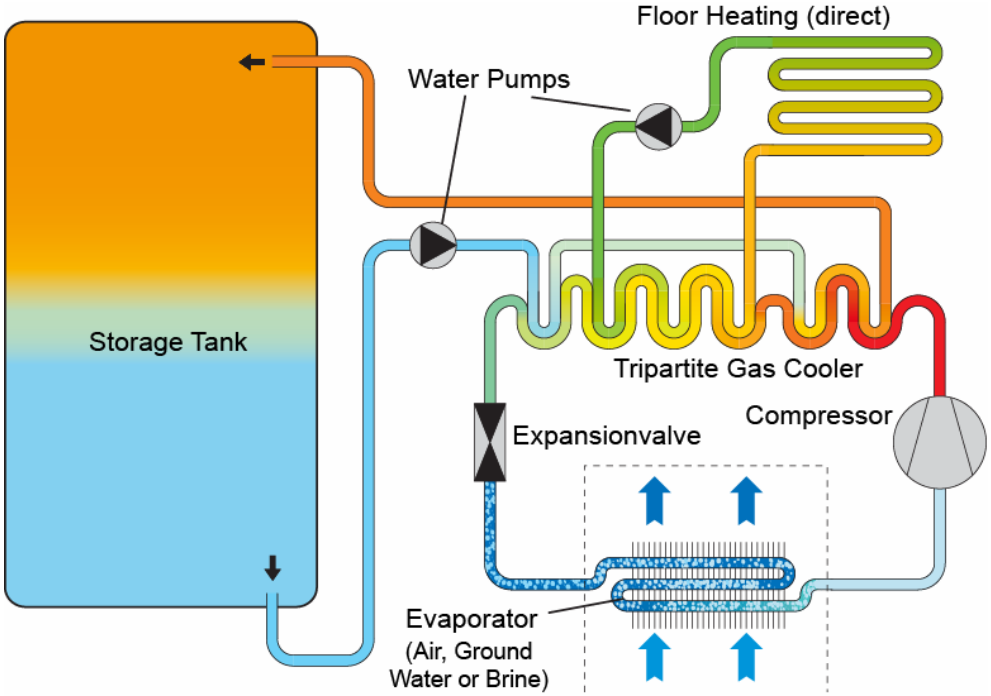


Figure 7: Indirect heat delivery into the storage tank for potable hot water with preheating to drop down the outlet temperature of carbon dioxide, direct heat delivery for floor heating

Another possibility to increase the efficiency of the heat pump is to drop the carbon dioxide gas cooler outlet temperature further. This can be achieved by a gas cooler divided into three parts on the water side (figure 7). A tripartite gas cooler has also been presented in Stene (2004). In comparison to the previous concept the refrigerant gets cooled down further after leaving the low temperature, middle segment of the gas cooler for the floor heating. This occurs when the cold water of the lower part of the storage tank is in the third segment of the gas cooler. The potential efficiency increase becomes bigger, the greater the difference is between the return temperature of the floor heating and the temperature in the lower part of

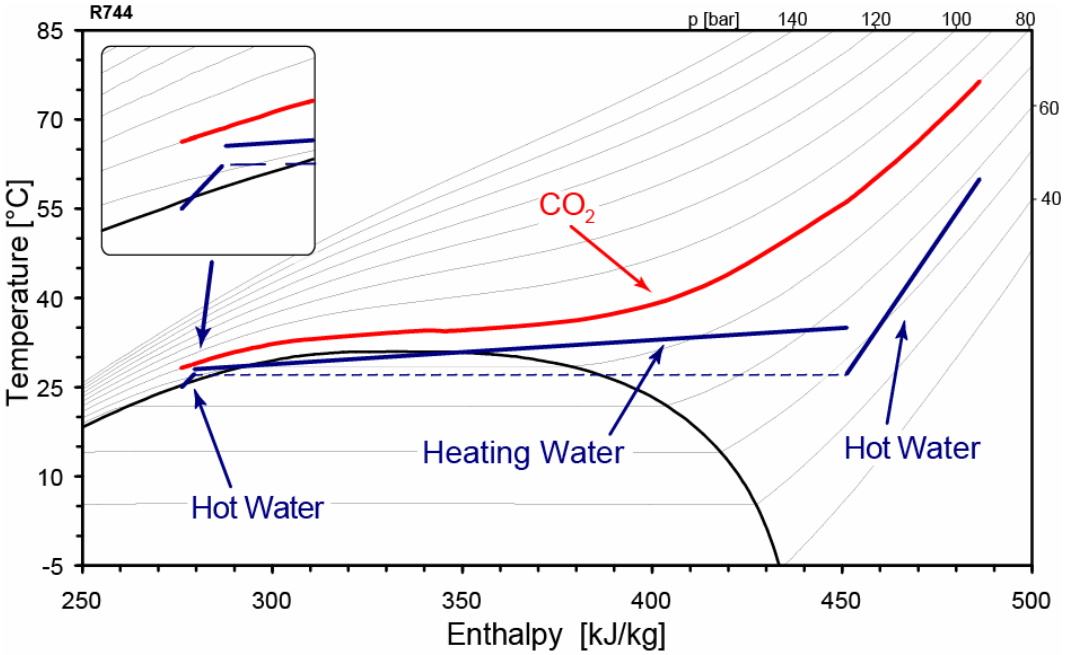


Figure 8: Temperature curves in the gas cooler in the T-h-diagram for the concept of figure 7 (simulation, high pressure 80 bar, division 5:8:2). Specific enthalpy of water is linear scaled to allow drawing into the carbon dioxide T-h-Diagram.

the storage tank. The temperature of the refrigerant cannot be cooled down to the inlet temperature of the water from the storage tank. This is not only for reasons of a finite heat transfer surface of the third gas cooler segment, but also because the water from the storage tank flows through the third and first segment of the heat exchanger (see figure 8). By heating up from 25°C to 60°C, the water already warms up a few degrees in the third segment of the gas cooler. Afterwards, it flows through the first segment (connection with dashed line) where it is heated completely up to 60°C. The water mass flow and the heat capacity flow are each identical in both segments. Both temperature curves of water have the same gradient. The heat capacity of the carbon dioxide changes considerably during this isobaric change of state. In the inflection point of the s-shaped curve it achieves the maximum. In the third segment of the gas cooler the heat capacity flow of the water of the storage tank is smaller than that of the refrigerant. The water is therefore subjected to a stronger temperature change than the refrigerant. The temperature of the refrigerant in the outlet of the gas cooler cannot be lowered to the entry temperature of the water. However, the water temperature at the outlet of the third segment approaches the temperature of carbon dioxide (small picture in figure 8).
In order to guarantee a lower temperature in the lower part of the storage tank than in the return of the floor heating, an adaptation of the plate heat exchanger to warm up potable hot water is required. This can be realized with direct as well as indirect heat delivery.
With regard to the complicated arrangement of the system with three gas coolers on the water side and their controllability, it is to be considered if the complexity justifies the relatively low gain. With the same material available for the gas cooler, the additional third segment causes a reduction of the heat exchange surface for other two segments. The desired increase of enthalpy difference is only moderate due to the abovementioned thermodynamic effects.

3.1.2 Air heating

In the case of direct heating of the entrance air without considering the air change rate of the low energy building, the same high temperature level is required as for potable hot water heating. Two gas coolers have to be arranged in parallel mode for the direct heat delivery

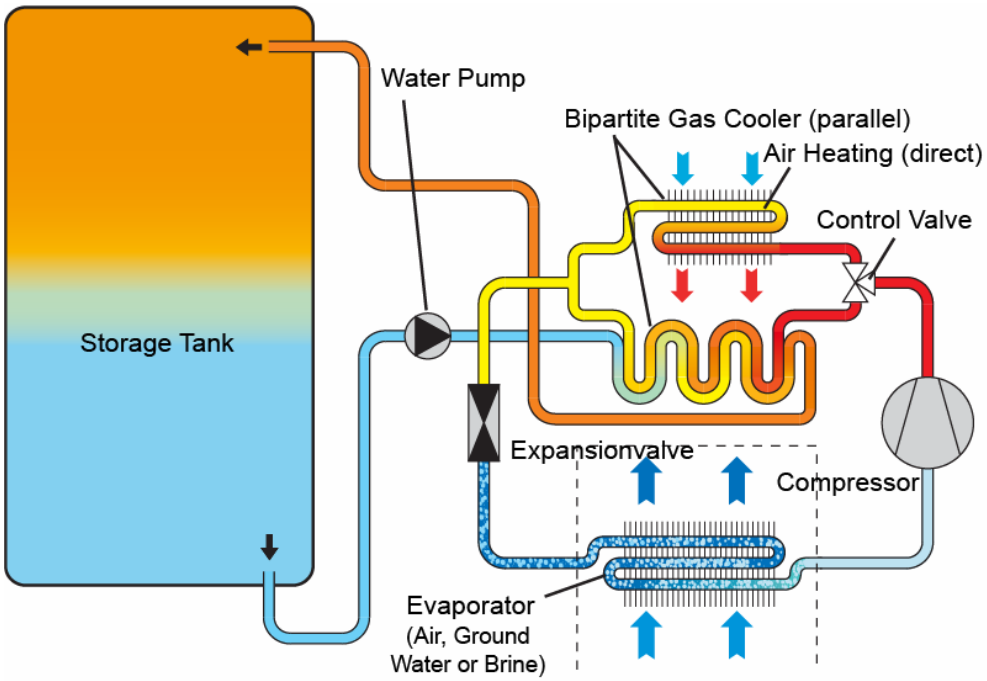


Figure 9: Indirect heat delivery into the storage tank, direct heat delivery by air heating in parallel arrangement of two gas coolers

into the air heating (see figure 9). According to the demands and the priority of potable hot water or heating requirement, both parallel heat exchangers can be controlled independently by a control valve with different refrigerant mass flow rates. It becomes clear that different concept scenarios appear which have to be optimized in each case. It is debatable whether the refrigerant cycle works without a refrigerant accumulator which manages the charge distribution for different operational modes.

3.2 Heat delivery out of the storage tank

Due to the different demands of potable hot water and heating water the operation time depends considerably on the operational mode. The charging of the storage tank to cover the hot water demand may be finished already, while the heating system must still be run. The inverse case is also possible, for example during the transitional months to summer time that a hot water demand exists but no heating water is needed. Due to the large number of operational modes with direct heat delivery, extensive control concepts are required. Each necessitates an efficient operation of the carbon dioxide heat pump. Furthermore the available heat transfer surface to meet a specific demand during combined operational modes is greatly reduced when the cycle is already employed to cover another demand with a different operating mode. Furthermore, if one of the demands is already fulfilled, only a considerable small heat exchange surface area is available for the fulfilment of the other demand in single operational mode.
The compressor should have adjustable power delivery to avoid an oversupply in heat capacity. It seems considerably easier to pursue the concept of the storage tank as the core of the heat supply system. The hot water supply as well as the heating system (floor heating or air heating) get access to the central storage tank (see figure 10). For this purpose a storage tank with sufficient water volume is necessary. With an installed coexistent thermal solar system which is usually common in low energy buildings, this storage tank is already available. The advantages are, on the one hand the simplification of the control engineering by reducing the number of possible operational modes. On the other hand, lower inlet water temperatures can be reached in the gas cooler of the carbon dioxide heat pump.

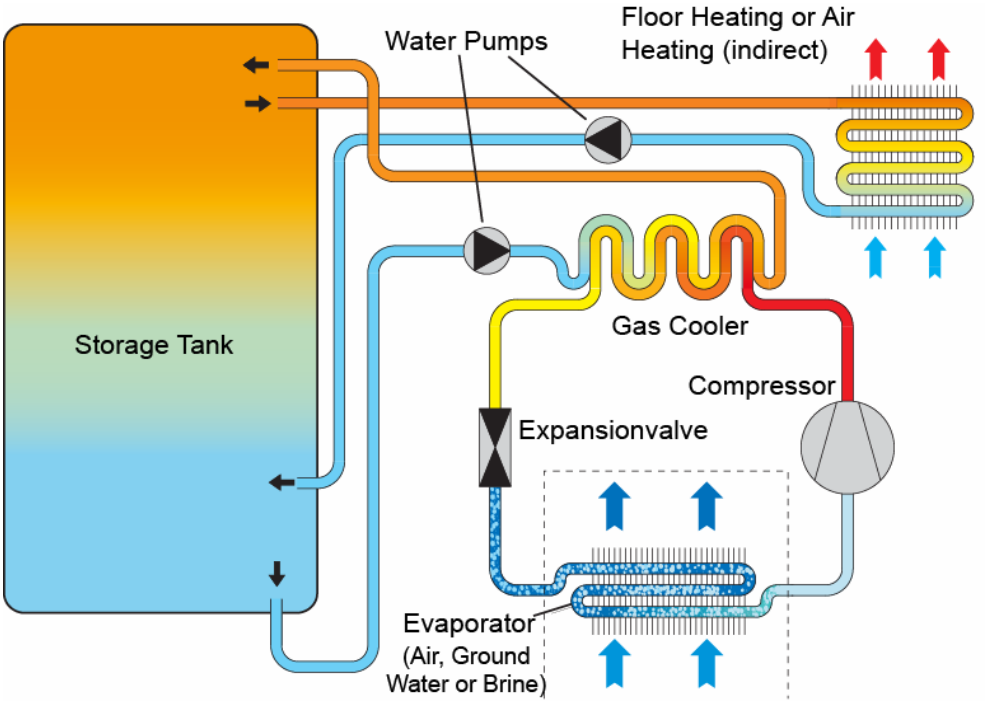


Figure 10: Indirect heat delivery into the storage tank for hot water and heating water supply

Furthermore a higher flexibility can be achieved with the heat supply system concerning the meeting of the hot water and heating demand. In this concept the whole heat exchange surface area for the heat delivery is available in one gas cooler. In contrast to direct heat delivery, a thermal solar system installed in the low energy building can also be used for the fulfilment of demand of the central heating system.
An overall energetic assessment of the presented concepts of heat delivery is theoretically possible with the annual coefficient of performance (COP). Indeed, it is very complicated to determine the optimum division relation of the serial as well as of the parallel switched gas coolers. The difficulty consists in identifying the optimum operation mode to cover the hot water and the heating water demand.
Concerning the heat pump the heat supply can occur for hot water and heating water either at the same time in a combined operational mode or in a separate operational mode if one of both demands is already fulfilled. This can be realized by an event regulation. Further information on this topic can be found in Kosowski et al. (2006).

4 CONCLUSION

The high level of heat insulation of low energy buildings reduces the energy demand of heating water. However, the amount of heat required to warm up potable hot water rises compared to the overall energy demand of the building. The heating system as a whole has to supply less heat, but considered relatively, it has to deliver heat more frequently at higher temperature demands (55°C-60°C). This operating mode advantages the application of a heat pump with carbon dioxide as refrigerant. Due to the transcritical process the carbon dioxide cycle is able to achieve higher efficiencies than systems using conventional refrigerants. In the gas cooler of the transcritical cycle the carbon dioxide is cooled down continuously with a non-constant temperature, whereas there the condenser of conventional systems shows a constant condensation temperature. Regarding to the efficiency of the whole cycle, this thermo-physical property offers a higher optimizing potential for the gas cooler by increasing the UA value than for the condenser.
In comparison to an isothermal operating condenser, the gas cooler does not have such a distinctive pinch point in the temperature distribution of the heat exchanger and responds

considerably more sensitively to an increase of the UA-value. Therefore, it is worthwhile improving the gas cooler exceedingly because of a considerably higher influence on the COP. Increasing the UA-value from 500 W/K to 1500 W/K causes an increase of the COP for the carbon dioxide cycle of nearly 11% while the condenser exhibits an increase of the COP of nearly 3% only without further improvement.
Furthermore, four different concepts of heat delivery have been presented and evaluated under thermo dynamical, technical and economical aspects. It turns out, to pursue the concept with the storage tank as the core of the total heat supply system. Both demands get access to the central storage tank. The simplification of the control engineering by reducing the number of possible operational modes and possibly lower inlet water temperatures in the gas cooler of the carbon dioxide heat pump are the advantages.
Furthermore, a higher flexibility can be achieved with the heat supply system concerning the meeting of the hot water and heating demand in buildings with improved thermal insulation. Because of the comparable low flow operation characteristic on water side, the carbon dioxide heat pump can be combined with low flow solar systems in one stratified storage device. Therefore, the thermal solar system installed in the low energy building can also be used for the remaining fulfilment of demand of the central heating system.

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