# BRINE-TO-WATER $CO_2$ HEAT PUMP SYSTEM FOR SPACE HEATING AND HOT WATER HEATING IN RESIDENCES

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## ABSTRACT

Carbon dioxide ($CO_2$) represents an environmentally benign and safe alternative to conventional working fluids in residential brine-to-water heat pumps. A 6.5 kW prototype $CO_2$ heat pump for combined space heating and hot water heating has been extensively tested and analysed. The heat pump was equipped with a unique counter-flow tripartite gas cooler for preheating of domestic hot water (DHW), low-temperature space heating and reheating of DHW.

The $CO_2$ heat pump was tested for simultaneous space heating and DHW heating, DHW heating only and space heating only. The heat pump rejected heat to a floor heating system at supply/return temperatures 33/28, 35/30 or 40/35ºC, and the set-point temperature for the DHW was 60, 70 or 80ºC. The experimental results proved that a brine-to-water $CO_2$ heat pump system may achieve higher seasonal performance factor (SPF) than the most energy efficient brine-to-water heat pump systems on the market, provided that the heating demand for hot water production constitutes at least 25% of the total annual heating demand for the residence, the return temperature in the space heating system is relatively low (<30ºC) and the thermodynamic losses in the DHW tank can be reduced to a low level.

**Key Words:** ***Residential heat pump, space heating, hot water heating, carbon dioxide ($CO_2$).***

## 1 INTRODUCTION

During the last fifteen to twenty years, the most pressing research issue within the field of refrigeration, air-conditioning and heat pump systems has been the search for environmentally acceptable working fluids which can replace the ozone-depleting CFCs and HCFCs. Most of the substances evaluated and tested have been new synthetic compounds, namely HFCs. Although these new compounds have been extensively tested with regard to toxicity, flammability etc. they are foreign to nature. The main environmental drawback with the HFCs is the fact that they are relatively strong greenhouse gases, and their greenhouse warming potential (GWP) factor is 1300 to 3500 times higher than that of $CO_2$. Widespread production and use of HFCs also include a risk of other unforeseen global environmental effects, which has already been experienced with the CFCs and HCFCs.

An alternative to the HFCs is to apply naturally occurring and ecologically safe substances, so-called natural working fluids. The most important substances in this category are ammonia, hydrocarbons, carbon dioxide ($CO_2$), water and air. From an environmental point of view, $CO_2$ (R-744) can be regarded as an almost ideal working fluid, since it is non-toxic, non-flammable and neither contributes to ozone depletion nor global warming[1].

Since virtually all residential heat pump units are charged with HFCs, it has been highly relevant to examine whether $CO_2$ heat pumps can be successfully applied in the residential sector.

Previous work on residential $CO_2$ heat pumps has been dealing with systems for either space heating or hot water heating (Stene, 2004). It was therefore considered interesting to carry out a theoretical and experimental study of residential brine-to-water $CO_2$ heat pump systems for combined

[1] *The $CO_2$ which is used as a working fluid is a by-product from industrial processes.*

low-temperature space heating and hot water heating – so called integrated $CO_2$ heat pump systems. The main reasons for selecting this heat pump concept were as follows:

- *Increasing relative heating demand for DHW*
  Due to stricter building codes, the transmission and infiltration losses in houses have been reduced in recent years, whereas the ventilation losses and the domestic hot water (DHW) heating demand have become more significant. Hence, the annual heating demand for DHW in new houses constitutes an increasing part of the total heating demand (Breembroek and Dieleman, 2001).
- *High-efficiency ground-source heat pump systems*
  The average seasonal performance factor (SPF) of residential brine-to-water systems is typically 25% higher than that of air-to-water systems (Hubacher, 2004). They also maintain the heating capacity at low ambient temperatures, and have longer operational life-time due to relatively high and stable evaporation temperatures.
- *Low-temperature heat distribution systems*
  The lower the distribution temperature, the higher the SPF of the heat pump system. Residential low-temperature floor heating systems are now gaining an increasing market share in many European countries, while central low-temperature air heating systems are commonly used in the USA and Canada. (Breembroek and Dieleman, 2001).

## 2 MAIN CHARACTERISTICS OF INTEGRATED $CO_2$ HEAT PUMP SYSTEMS

Due to the low critical temperature of $CO_2$ (31.1ºC), an integrated $CO_2$ heat pump will reject heat by cooling of $CO_2$ at supercritical pressure in a gas cooler. In order to achieve a high coefficient of performance (COP), it is essential that useful heat is rejected over a large temperature range, resulting in a relatively low $CO_2$ outlet temperature from the gas cooler. A number of gas cooler configurations have been evaluated. It was found that a counter-flow tripartite gas cooler for preheating of domestic hot water (DHW), low-temperature space heating and reheating of DHW, would enable production of DHW in the required temperature range from 60 to 85ºC, and contribute to the highest possible COP for the heat pump unit (Stene, 2004). Figure 1 shows the principle of an integrated $CO_2$ heat pump system, where an expansion valve and a low-pressure liquid receiver (LPR) are used to control the supercritical (high-side) pressure in the tripartite gas cooler.

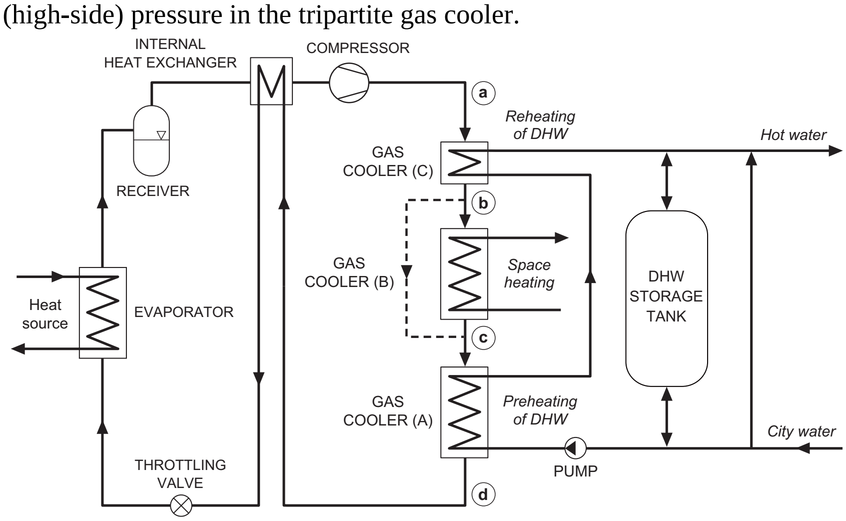


**Fig. 1. The principle of an integrated residential $CO_2$ heat pump system (Stene, 2004).**

Gas cooler units A and C are connected to an unvented single-shell DHW storage tank and an inverter controlled pump by means of a water loop. Gas cooler unit B is connected to a low-temperature hydronic heat distribution system with radiant floor heating, convectors or fan-coils.

An integrated $CO_2$ heat pump will be operating in three different modes: simultaneous space heating and DHW heating (Combined mode), hot water heating only (DHW mode) and space heating only (SH mode). During tapping of DHW, hot water from the top of the DHW tank is premixed with cold city water and delivered at the tapping site, while cold city water enters the bottom of the tank. During charging of the DHW tank in the Combined and DHW modes, the cold city water from the bottom of the DHW tank is pumped through gas cooler units A and C, heated to the desired temperature level, and returned at the top of the tank. During operation in the SH mode, there is no water circulation in the DHW circuit, and only gas cooler unit B is operative.

The heat rejection process in the three different operating modes is illustrated in temperature-enthalpy diagrams in Fig. 2. The supply/return temperatures for the floor heating system are 35/30ºC, while the city water temperature and the set-point for the DHW are 6.5 and 70ºC, respectively. In the Combined mode, the so-called DHW heating capacity ratio is about 45%, which means that 45% of the total heating capacity of the tripartite gas cooler is used for hot water heating.

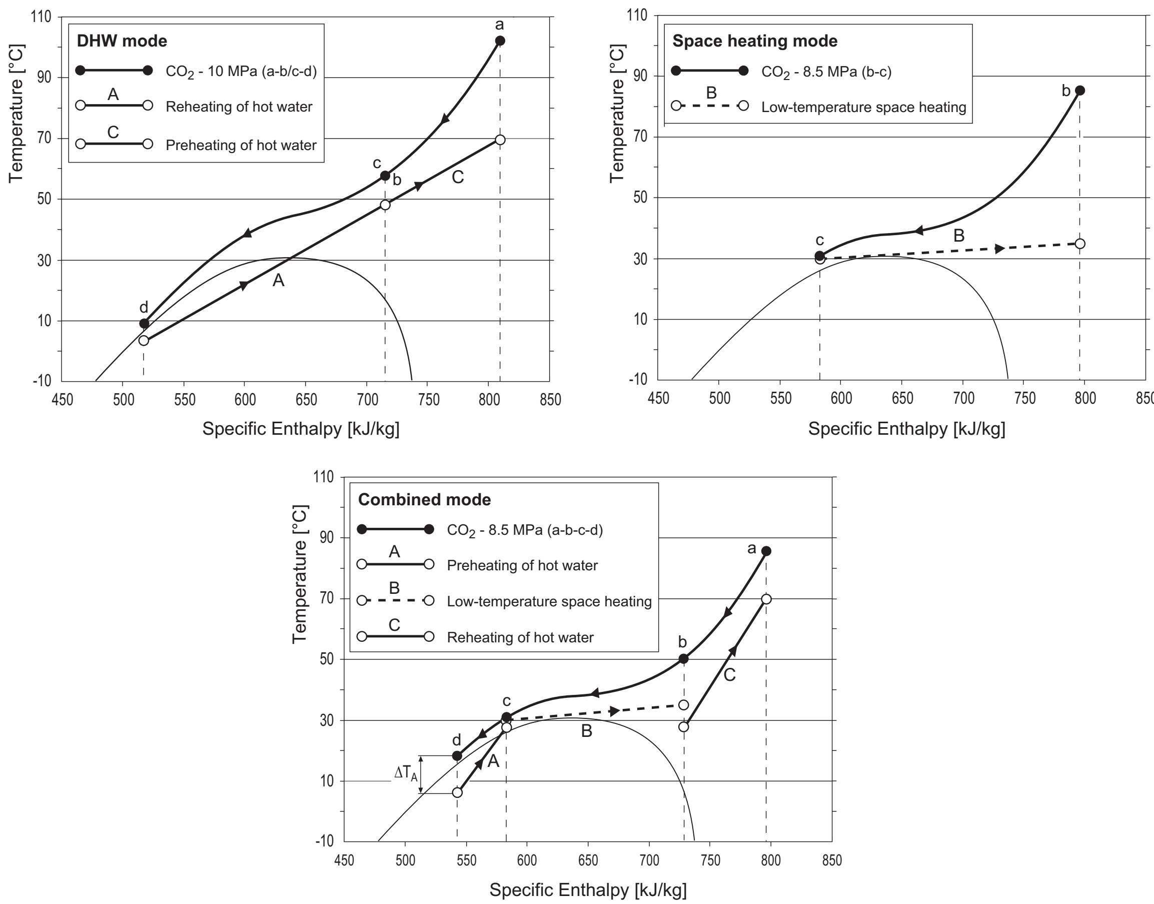


**Fig. 2. Illustration of the heat rejection process for a counter-flow tripartite $CO_2$ gas cooler operating in the DHW mode, space heating (SH) mode and combined mode (Stene, 2004).**

In the Combined mode, the counter-flow tripartite gas cooler enables a small temperature approach ($\Delta T_A$) at a moderate high-side pressure. As a consequence, the COP may be even higher than that of the DHW mode, where the optimum high-side pressure is typically 1 to 1.5 MPa higher. The COP in the SH mode will be much lower than that of the Combined mode and the DHW mode. This is a result of the bad temperature fit between the $CO_2$ and the water, and the fact that the $CO_2$ outlet temperature from the gas cooler is limited by the return temperature in the space heating system. For the same reason the COP in the SH mode will be lower than that of a conventional heat pump cycle, where the heat is rejected by condensation of the working fluid (Kerherve and Clodic, 2002).

## 3 TESTING AND MODELLING OF A PROTOTYPE $CO_2$ HEAT PUMP UNIT

A prototype brine-to-water $CO_2$ heat pump unit for combined space heating and hot water heating was constructed in order to document the performance and to study component and system behaviour over a wide range of operating conditions (Stene, 2004). The specifications for the $CO_2$ heat pump unit are presented in Table 1.

**Table 1. Specifications for the prototype $CO_2$ heat pump unit (Stene, 2004).**

| | |
|---|---|
| Compressor | • Hermetic two-stage rolling piston – operated as a single-stage unit<br>• Swept volume – 1.19 $m^3$/h at 6000 rpm<br>• Lubricant – polyalkylene glycol (PAG) |
| Evaporator | • Counter-flow single-pass tube-in-tube HX – stainless steel<br>• Surfaces $CO_2$/brine side: 0.30 / 0.38 $m^2$ |
| Tripartite gas cooler | • Counter-flow single-pass tube-in-tube HX – stainless steel<br>• Surfaces $CO_2$ side: 0.26 / 0.28 / 0.07 $m^2$ – total 0.61 $m^2$<br>• Surfaces water side: 0.35 / 0.38 / 0.09 $m^2$ – total 0.82 $m^2$ |
| Suction gas heat exchanger | • Counter-flow single-pass tube-in-tube HX – stainless steel<br>• Surfaces: 0.06 / 0.07 $m^2$ |
| High-side pressure control | • LPR and manually operated back-pressure valve |

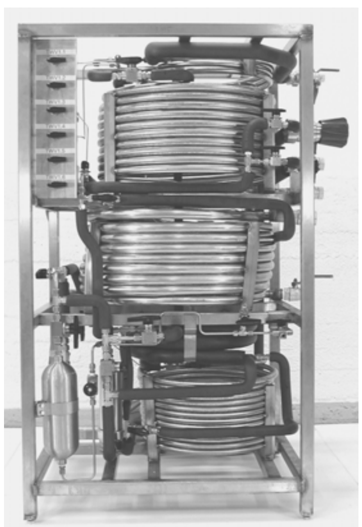

**Fig. 3. The prototype $CO_2$ heat pump unit for combined space heating and hot water heating.**

The prototype heat pump was designed for a total heating capacity of 6.5 kW in the Combined mode at -5ºC evaporation temperature, 5 K suction gas superheat, 8.5 MPa high-side pressure, 60ºC DHW temperature and 35/30ºC supply/return temperatures for the space heating system.

The prototype was tested in the Combined mode, DHW mode and SH mode. The supply/return temperatures for the space heating system were 33/28, 35/30 or 40/35ºC, and the DHW temperature was 60, 70 or 80ºC. The average city water temperature was 6.5ºC, and most tests were carried out at an evaporation temperature of -5ºC. At each temperature programme, the gas cooler pressure was varied in order to find the optimum high-side pressure that corresponded to the maximum attainable COP. The relative uncertainty for the measured COPs was less than 3% (95% confidence level).

A steady-state computer model for an integrated $CO_2$ heat pump unit using a tripartite single-pass counter-flow tube-in-tube gas cooler was also developed, in order to analyse and supplement the measurements from the prototype $CO_2$ heat pump (Stene, 2004). The model was established in Microsoft Excel/Visual Basic, and the Gnielinsky correlation (VDI Heat Atlas, 1993) was used to estimate the convective heat transfer coefficients for the single-phase $CO_2$ and the water. The simulation model was tested and verified by means of experimental data from the prototype heat pump unit. The model predicted the outlet temperatures and the heating capacities for the gas cooler units within ±0.3ºC and ±0.6%, respectively.

## 4 EXPERIMENTAL RESULTS

Figure 4 shows the measured maximum COP at optimum high-side pressure as a function of the DHW heating capacity ratio during operation in the combined mode. The average city water temperature was 6.5ºC, and the evaporation temperature was -5ºC.

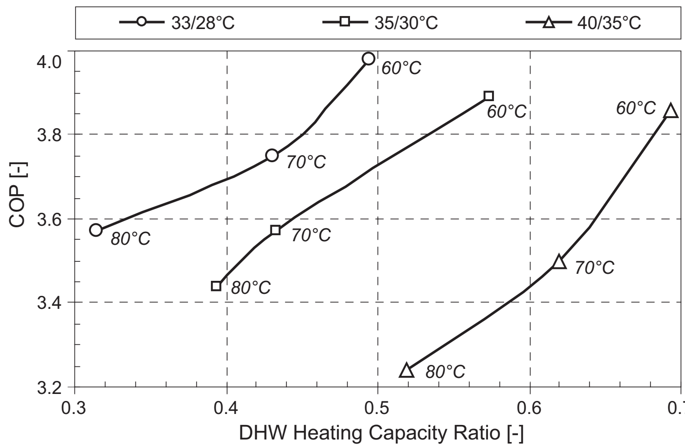


**Fig. 4. The measured relationship between the maximum COP and the DHW heating capacity ratio during testing in the Combined mode (Stene, 2004).**

The temperature levels in the space heating and DHW systems had a considerable impact on the DHW heating capacity ratio for the prototype heat pump unit. The higher the temperature level for the space heating system, and the lower the set-point temperature for the DHW system, the larger the DHW heating capacity ratio. At 40/35ºC supply/return temperature for the space heating system and 60ºC DHW temperature, the DHW heating capacity ratio was almost 70%, and the heat pump was practically operating as a heat pump water heater. On the other hand, at 33/28ºC supply/return temperatures for the space heating system and 80ºC DHW temperature, the DHW heating capacity ratio was about 30%, and more than 2/3 of the heat was given off to the space heating system.

Figure 5 shows the measured maximum COP at 60ºC DHW temperature and various supply/-return temperatures for the space heating system. The numbers at the bottom of the Combined mode bars display the DHW heating capacity ratio.

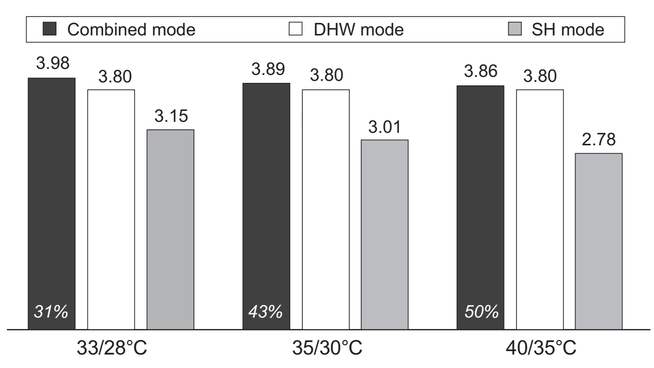


**Fig. 5. The measured maximum COP at 60ºC DHW temperature and various supply/return temperatures for the space heating (SH) system (Stene, 2004).**

While the COPs during testing in the Combined mode and DHW mode were almost identical, the COP in the SH mode was roughly 20 to 30% lower than that of the Combined mode. The largest COP difference was measured at the highest temperature level in the space heating system. By increasing the DHW temperature to 80ºC, the COP in the Combined mode and the DHW mode dropped by 15%, and the average COP difference between the Combined mode and the SH mode was reduced from about 25% to 15%. The latter was a result of the reduced water flow rate in the DHW circuit.

The measured overall isentropic efficiency for the prototype rolling piston compressor ranged from about 0.52 to 0.55 at 6000 rpm. An exergy analysis of the prototype $CO_2$ heat pump system, showed that the relative exergy loss for the compressor constituted about 40% in the SH mode and about 55% in the Combined mode and DHW mode (Stene, 2004). This is considerably higher than that of conventional brine-to-water heat pump systems. Consequently, for integrated $CO_2$ heat pumps it is of particular importance to use a high-efficiency compressor.

## 5 SIMULATION RESULTS

Figure 6 shows the simulated relative COP for the prototype heat pump in the Combined mode at varying inlet water temperature for the DHW preheating gas cooler unit (ref. Fig.1, Gas cooler unit A). The supply/return temperatures for the space heating system were 35/30 or 40/35°C, the DHW temperature was 60 or 80°C, and the high-side pressure for each temperature program was kept constant at the measured optimum value. The reference inlet water temperature was 5ºC.

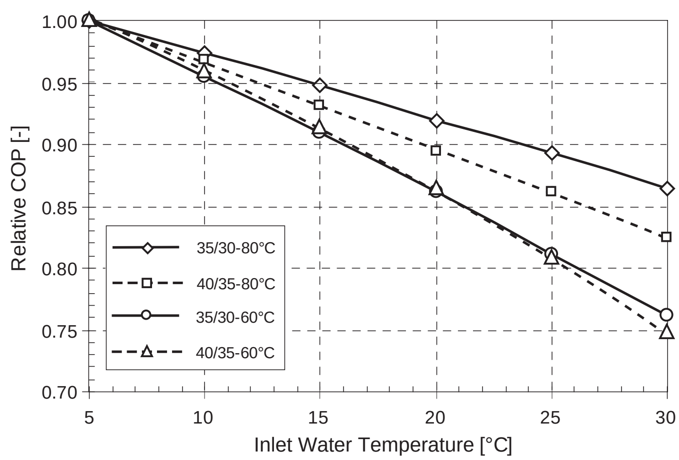


**Fig. 6. The simulated relative COP in the Combined mode as a function of the inlet water temperature and varying set-point temperatures for the space heating and DHW systems (Stene, 2004).**

The inlet water temperature had a significant impact on the COP for the $CO_2$ heat pump unit during operation in the Combined mode, since it governed the maximum possible cool-down of the $CO_2$ in the DHW preheating gas cooler unit. As an example – by increasing the inlet water temperature from 5 to 20ºC, the COP at 35/30-60ºC dropped by approximately 15%. In the DHW mode, the COP was even more sensitive to variations in the inlet water temperature, since the entire heating capacity for the heat pump was used for DHW production.

Measurements on a standard-sized DHW tank (Stene, 2004), proved that conductive heat transfer between the DHW and the cold city water in the tank during the tapping and charging periods may result in a considerable increase in the average inlet water temperature for the DHW preheating gas cooler unit. Inevitable mixing of hot and cold water in the tank will lead to a further increase in the thermodynamic losses. In order to achieve a high COP for integrated $CO_2$ heat pump systems, it is therefore essential to develop tailor-made DHW tanks that minimizes the thermodynamic losses.

## 6 COMPARISON OF SEASONAL PERFORMANCE FACTORS (SPFs)

The seasonal performance factor (SPF) for the prototype $CO_2$ heat pump and a high-efficiency residential brine-to-water R410A heat pump was estimated, assuming constant inlet brine temperature for the evaporator (0ºC) and constant temperature levels in the space heating system (35/30ºC) and the DHW system (10/60ºC) (Stene, 2004). The COPs for the R410A system was gathered from TNO-MEP in the Netherlands (Oostendorp and Traversari, 2001) and WPZ Töss ,Heat Pump Test Centre in Switzerland (www.wpz.ch). An improved $CO_2$ heat pump system with 10% higher COP than the prototype system was also investigated in order to demonstrate the future potential of the $CO_2$ system. Higher COP can be achieved by using a more energy efficient compressor, optimizing the tripartite gas cooler or replacing the throttling valve by an ejector. The latter is capable of increasing the COP of the $CO_2$ heat pump by as much as 10 to 20% (Ozaki et al., 2004).

For the $CO_2$ heat pump systems, the thermodynamic losses in the DHW tank (i.e. mixing and internal conductive heat transfer) were not included. The R410-system used a shuttle valve for prioritized DHW heating, and electric immersion elements were used for reheating of the DHW from 53 to 60ºC. Table 2 shows the COPs for the heat pump systems at the selected operating conditions.

**Table 2. The COPs for the $CO_2$ and R410A heat pump systems (Stene, 2004).**

| | |
|---|---|
| Prototype $CO_2$ heat pump | • COP = 3.0 – SH mode at 35/30ºC<br>• COP = 3.8 – DHW mode at 10/60ºC<br>• COP = 3.9 – Combined mode at 35/30ºC and 10/60ºC |
| Improved $CO_2$ heat pump | • COP = 3.3 – SH mode at 35/30ºC<br>• COP = 4.2 – DHW mode at 10/60ºC<br>• COP = 4.3 – Combined mode at 35/30ºC and 10/60ºC |
| R410A heat pump | • COP = 4.6 – SH mode at 35/30ºC<br>• COP = 3.2 – DHW mode at 10/55ºC |

Table 2 demonstrates that the $CO_2$ heat pumps and the R410A heat pump have reversed COP characteristics; i.e., the $CO_2$ units achieve the highest COP during operation in the Combined mode and DHW mode, whereas the R410A unit achieves the highest COP in the SH mode. In Fig. 7, the estimated SPFs for the three heat pump systems during monovalent operation are presented as a function of the seasonal DHW heating capacity ratio.

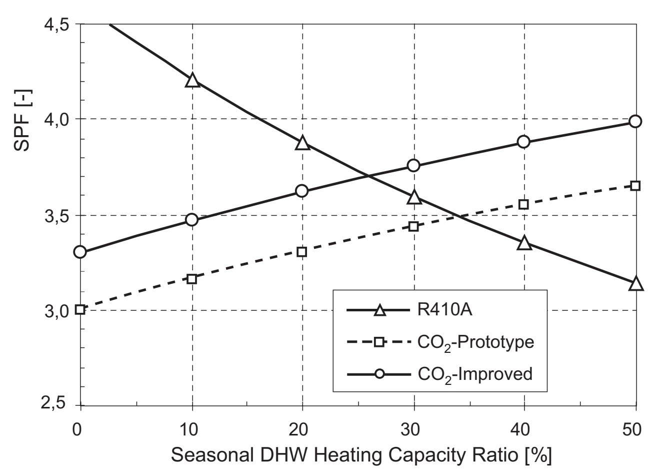


**Fig. 7. The estimated SPF at monovalent operation for the R410A heat pump system, the prototype $CO_2$ heat pump system and the improved $CO_2$ heat pump system (Stene, 2004).**

At low seasonal DHW heating capacity ratio, the R410A heat pump system was more energy efficient than the $CO_2$ systems due to their poor COP during operation in the SH mode. At increasing seasonal DHW heating capacity ratio, the SPF of the $CO_2$ systems were gradually improved, since an increasing part of the heating demand was covered by operation in the Combined mode and DHW mode. On the other hand, the SPF for the R410A heat pump system dropped quite rapidly at increasing seasonal DHW heating capacity ratio, since the COP during operation in the DHW mode was about 30% lower than that of the SH mode. At the actual operating conditions, the break-even for the prototype $CO_2$ system occurred at a seasonal DHW heating capacity ratio around 35%, whereas the break-even for the improved $CO_2$ system was about 10 percentage points lower. The break-even for the $CO_2$-systems during bivalent operation was virtually the same as during monovalent operation.

In conclusion, in existing houses where the DHW ratio typically ranges from 10 to 15%, an R410A heat pump system will be more energy efficient than an integrated $CO_2$ heat pump system. However, in well-insulated houses or in low-energy houses, where the DHW ratio typically ranges from 20 to 45%, an optimised $CO_2$ heat pump system may achieve the highest SPF.

## 7 CONCLUSION

An integrated residential brine-to-water $CO_2$ heat pump system may achieve higher seasonal performance factor (SPF) than the most energy efficient brine-to-water heat pump systems provided that: 1) The heating demand for hot water production constitutes at least 25% of the total annual heating demand for the residence, 2) The return temperature in the space heating system is relatively low (<30ºC) and 3) The thermodynamic losses in the DHW tank can be reduced to a low level. The latter requires a special tank design in order to minimize mixing and conductive heat transfer between the hot and cold water during the tapping and charging periods.

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