

SPACE HEATING AND HOT WATER SUPPLY SYSTEM WITH A TRANSCRITICAL CO₂-HEAT PUMP

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ABSTRACT

A transcritical CO₂ heat pump prototype for a combined space and hot water heating system for residential European-style low-energy-houses has been designed, built and tested. The goal of the project was to verify the potential of the CO₂-heat pump process in such an application and to gain experience with the CO₂-heat pump for later commercial realisation.

In this report the heat pump performance data are presented. The heat pump was implemented in a typical heating system of a European-style low-energy house which consists of a hot water supply, a heating system for floor and/or radiator heating system and a room ventilation facility. The heating system was modified to account for the characteristic performance losses of the transcritical heat pump at heat sink inlet temperatures above approximately 25-30°C. Design aspects of the heating system and overall system performance data are presented. The CO₂ heat pump was found to be capable to significantly improve the overall performance of the heating system compared to similar systems with subcritical HFC or propane heat pumps. Based on the experimental results a preproduction prototype of a house heating system is built and tested. Field tests with the prototypes are going to be carried out to gain more experience on the system characteristics.

1 INTRODUCTION

Since the discovery of the impact of CFCs and HCFCs on ozone layer destruction and global warming, new fluorocarbon chemicals have been developed and used for refrigeration. However, a real and enduring alternative are natural fluids, among which carbon dioxide has received much attention since its revival about a decade ago (Lorentzen 1990, Lorentzen and Pettersen 1993, Lorentzen 1994). Due to its thermodynamic properties, carbon dioxide has the potential to outperform other working fluids, especially when operating with large temperature differences between heat source and heat sink combined with low temperatures of the working fluid after heat rejection. These conditions are given when hot water with temperatures above 60 °C is generated from relatively cool water having temperatures of less than 20 °C. Therefore the use of carbon dioxide heat pumps is especially interesting in low-energy houses, where the generation of hot water accounts for approximately 50 % of heat consumption.

In the current project, a transcritical CO₂ air to water heat pump is developed as a commercial product for a low-energy single family home in central Europe. According to the standard Minergie-P, which is well established in Switzerland and in Germany is referred to as "Passivhaus", these homes make use of passive energy sources and require an additional heating power of 10 W/m² at the design point of minus 11 °C outside temperature. It is the target of the current development to provide 10 W/m² heating power for an area of 200 m², which results in a total heat output of 2 kW. The heat pump also generates hot water, where its advantages above heat pumps with conventional working fluids is commercially important.

As a first step towards a commercial product, first knowledge of operating a heat pump with carbon dioxide is gained in a thermodynamic analysis. A prototype heat pump allowing accurate measurements of efficiency in a large range of operating conditions is tested. Data important for efficiently operating

and controlling the heat pump is gained and presented in this study. It is also suggested how to incorporate the heat pump in the heating system of a low-energy building to ensure its efficient operation.

2 THERMODYNAMIC ANALYSIS

The coefficients of performance of the heat pump COP_{HP} in this paper is used according to the following definition.

$$COP_{HP} = \frac{\phi_{HP}}{P_{el,Comp}} \quad (1)$$

From various published flow circuits for CO₂ heat pumps the circuit proposed by Lorentzen (Lorentzen 1990, Lorentzen and Pettersen 1993, Lorentzen 1994) was chosen for the current investigation because it requires less hardware and less control valves in comparison to other circuits. This makes it suitable to be used in a commercial product, which is the end goal of the current project. The flow circuit considered in this work is shown in Fig. 1. The quantities Φ_{HP} and $P_{el,comp}$ used to calculate the COP_{HP} according to equation (1) are also indicated in the figure. In the original proposition of the flow circuit, the compressor inlet is preheated by an internal heat exchanger. In this work, an external heat source like the compressor's wasted heat is also considered to serve for preheating the compressor suction line slightly above saturation. Therefore the system under investigation is equipped either with an external heat exchanger or with an internal heat exchanger. Both arrangements were analysed. In the circuit shown in Fig. 1, the receiver separates liquid CO₂ from gaseous CO₂ and should always contain both phases. It also serves to supply or absorb liquid when the required amount of CO₂ in the circuit changes due to changes in the operating conditions.

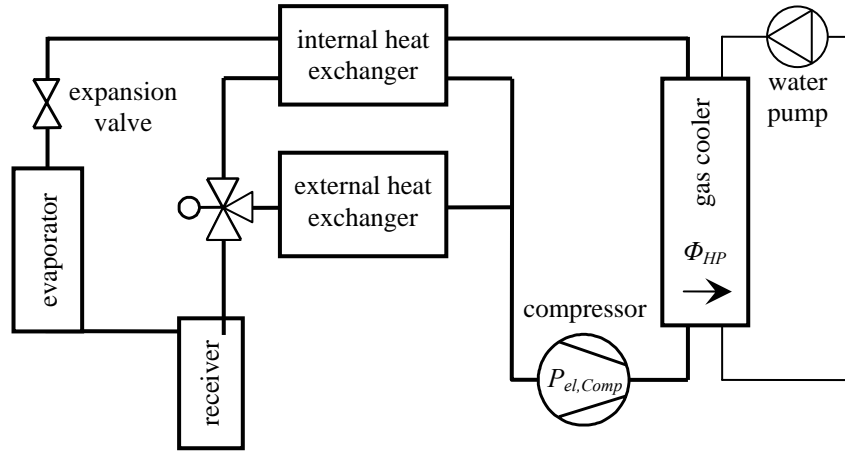


Fig. 1. CO₂ circuit considered in this study. The heat pump operates either with an internal heat exchanger or with an external heat exchanger.

An extensive thermodynamic analysis of the flow circuit and its possible integration in a heating system is performed in using a theoretical model implemented in a computer program. The thermodynamic properties of CO₂ are calculated with subroutines from the University in Essen programmed by Schrey at the Institute für Angewandte Thermodynamik und Klimatechnik in 2002. The heat transfer properties of CO₂ close to the critical point are modelled with modified Fewster equations (Kim et al. 2001). The compressor performance is calculated according to published data of a reciprocating compressor from Dorin type CD 4.027 S (Neksa et al. 1999, Hubacher and Groll 2003).

The temperatures required at the heat pump's water outlet are between 35 °C for floor heating and 70 °C for hot water supply. Since these temperatures are above the critical point of CO₂, the heat pump has to be operated in a transcritical mode and only transcritical operation is investigated in this study. It is known that a transcritical heat pump has an optimal compressor discharge pressure, where the heat pump's performance COP_{HP} has a maximum. In the circuit from Fig. 1 the compressor discharge pressure is controlled by the throttling valve.

The thermodynamic analysis shows that the heat pump's performance COP_{HP} is lower for water inlet temperatures above about 25 °C. This is a consequence of the thermodynamic properties of carbon dioxide in the vicinity of the critical point. This finding is important when integrating the CO₂ heat pump in a heating system, since conventional heating systems typically have water coming back from the heating system in excess of 25 °C. Additionally, it is found that using the internal heat exchanger does not improve the heat pump's coefficient of performance COP_{HP} . This is in agreement with published data gained with a transcritical CO₂ circuit (White et al. 1997). In case an internal heat exchanger is used, the evaporation temperature has to be limited in order to ensure preheating of the compressor suction line above saturation also when running with low water inlet temperatures and at the same time high temperatures of the heat source. This can be achieved in limiting the amount of CO₂ in the circuit. The analysis has also shown, that with an internal heat exchanger, the COP_{HP} is less depending on the high pressure level in comparison to an external heat exchanger.

3 PROTOTYPE

A laboratory prototype of the CO₂ circuit from Fig. 1 was built as shown in the schematic of **Error! Reference source not found.**. The scale of the prototype is the same as of the commercial product, i.e. 2000 W heating power Φ_{HS} . Purchasing the components turned out to be difficult, since components for conventional heat pumps are only rated to about 25 bar. The reciprocating compressor from Dorin type CD 4.027 S used for the thermodynamic analysis was not available. A prototype of a Danfoss reciprocating CO₂ compressor type TN1416 is used instead. Lubrication is ensured by the Polyester (POE) oil type Reniso C85E, which is recommended by the compressor manufacturer. It dissolves very well in CO₂ above the critical point, which is the high pressure side in the present prototype. Therefore no oil film reduces the heat transfer in the gas cooler. For the low pressure side, a comparison of the oil's density with the one of liquid CO₂ shows that the oil is denser at temperatures above -20 °C, i.e. in the range of the present application.

The two heat sources of the evaporator and of the external heat exchanger were implemented with electrical heating. The heating power of the evaporator is controlled such as to achieve a preset evaporation temperature. The heating power of the external heat exchanger is controlled such as to achieve a preset amount of temperature increase above saturation. The implementation with electric heating allows very accurate control of the operating conditions when experimentally determining the system's behaviour. It also allows to run through a large range of operating conditions.

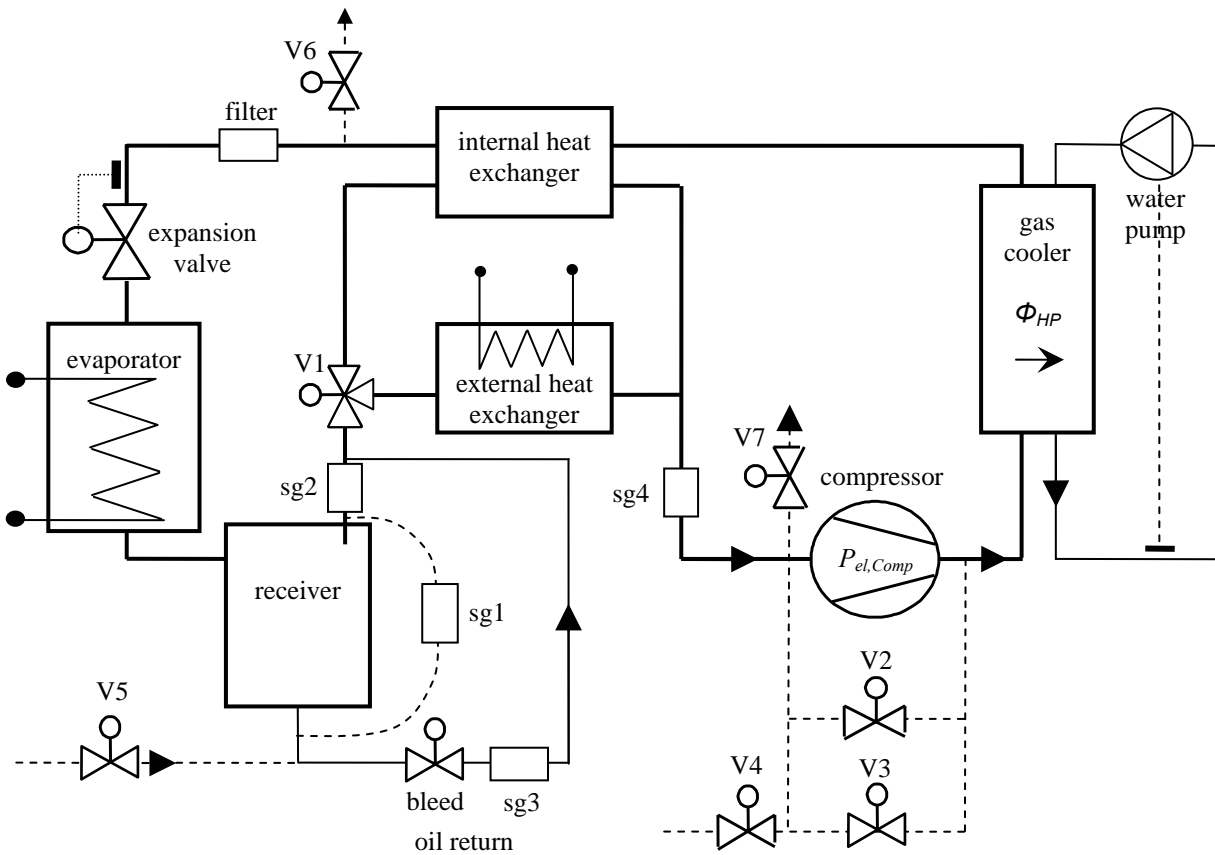


Fig. 2. Schematic of laboratory prototype of CO₂-heat pump. sg: sight glass, V: manual valves.

The receiver is equipped with a sight glass sg1 between two flexible hoses that can be moved vertically. This allows to check the liquid level in the receiver. Another sight glass sg2 indicates whether liquid drops are carried over with the gaseous CO₂. A manual bleeding valve at the bottom of the receiver allows to feed the oil from the bottom of the receiver back into the CO₂ circuit with visual access through another sight glass sg3. With the manual valve V1 it is determined whether the external heat exchanger or the internal heat exchanger is active. The phase composition of the fluid entering the suction side of the compressor can be observed through sight glass sg4. The remaining valves V2 to V7 from **Error! Reference source not found.** serve filling and emptying the circuit and fulfil safety functions. All equipment is thermally insulated in order to reduce heat transfer with the environment. The heat loss through the insulation is measured to distort the power balance by up to 2%. The prototype is equipped with appropriate sensors to measure pressure and temperature of the working medium and of the water as well as to measure the water flow rate. Additionally the electrical power consumption of the evaporator, the external heat exchanger and the compressor are measured.

The mass flow of the working fluid CO₂ is not measured directly. It is calculated from power balances across the boundaries of the gas cooler, across the expansion valve and evaporator as well as across the internal or the external heat exchanger depending on which unit is active. The power balance of the expansion valve plus evaporator can be done separately from the power balance of one of the two heat exchangers under the condition that there is no liquid CO₂ leaving the receiver, which is checked visually. In this case, the state of the fluid in the receiver is on the saturation line. The expansion valve has to be included in the power balance of the evaporator, because the enthalpy of the two-phase fluid entering the evaporator can not be determined from measurements of pressure and temperature.

4 MEASUREMENT RESULTS

The heat pump's coefficient of performance COP_{HP} is measured at different operating conditions in steady state operation. Operating conditions to be tested are closely related to how and where the commercial product is intended to be used. In hot water generations for example, the inlet and outlet temperatures of the water are 17 °C and 70 °C respectively. The evaporation temperature is approximately 5 °C below the temperature of the heat source. Evaporation temperatures ranging from minus 20 °C to 10 °C are assessed in this investigation. The pressure in the evaporator, i.e. on the low pressure side of the circuit is the saturation pressure of carbon dioxide at the respective evaporation temperature. The pressure on the high pressure side, i.e. the compressor discharge pressure is controlled by the throttling valve and is varied in order to find the level with maximum performance. As an example, coefficients of performance COP_{HP} calculated according to equation (1) from measurement data are shown in Fig. 2. The external heat exchanger was used to heat the working fluid 7 °C above saturation. Measurement data is only taken, when the state of carbon dioxide in the receiver is at saturation. This is the case when temperature at the evaporator inlet is the same as at the evaporator outlet and when the exit of the receiver is gaseous with no liquid drops.

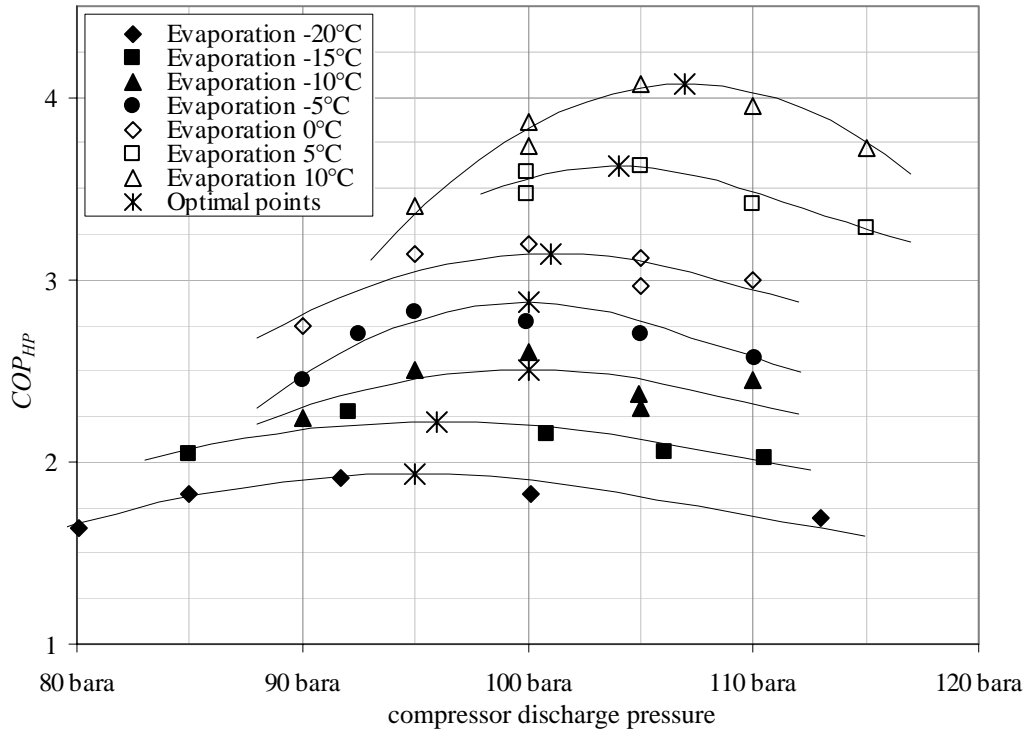


Fig. 2. Performance data COP_{HP} of prototype heat pump for hot water generation from 17 °C to 70 °C depending on discharge pressure for various evaporating temperatures using the external heat exchanger to heat 7 °C above saturation. Solid lines are least squares fits. Their maxima are indicated with asterisk.

It can be seen that for every evaporating temperature there exists an optimum discharge pressure, where the heat pump's performance COP_{HP} has a maximum. This discharge pressure is referred to as the optimum pressure. Similar behaviour at one specific operating condition has been published in literature (e.g. Ma et al. 2002 figure 2). In order to determine the optimum pressure and the maximum efficiency $COP_{HP,max}$, a least squares fitting procedure is used to place a curve through the data of each evaporating temperature. The maxima of these curves are taken to be the points of maximum heat pump efficiency

$COP_{HP,max}$. They are indicated with an asterisk in Fig. 2. The optimum discharge pressures as well as the maximum heat pump performance $COP_{HP,max}$ depending on evaporation temperature are shown in Fig. 3 and Fig. 4 respectively. Similar results to the ones shown in Fig. 4 either calculated or measured have been published previously (e.g. Rieberer et al. 1999 figure 2, Strmmen et al. 1999 figure 5, Aarliien and Neks 2002 figure 1, Ma et al. 2002 figure 7).

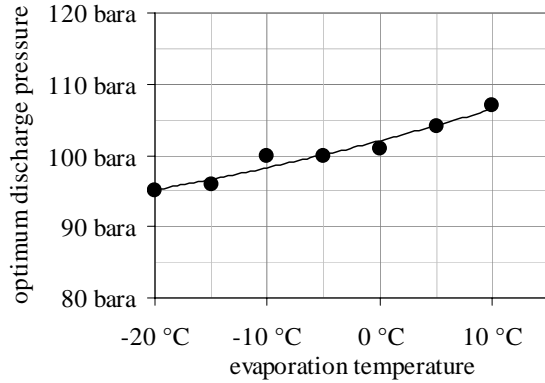


Fig. 3. Discharge pressure for maximum performance of heat pump for hot water generation from 17 °C to 70 °C depending on evaporation temperature.

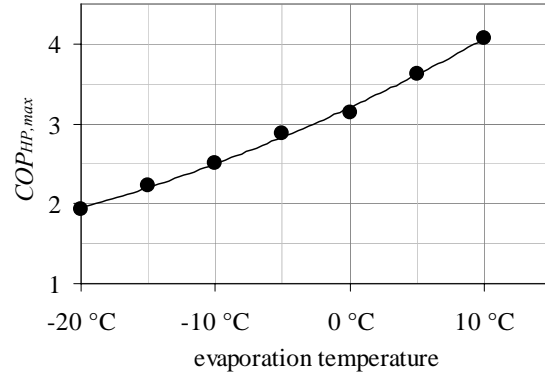


Fig. 4. Maximum coefficient of performance COP_{HP} for hot water generation from 17 °C to 70 °C depending on evaporation temperature.

The behaviour shown in Fig. 2 to Fig. 4 depends on the temperatures of the water inlet and water outlet and on the characteristics of the compressor and of the gas cooler. Therefore this mapping is required for every heat pump application in order to obtain a control algorithm for the throttling valve. In our example of hot water generation, the throttling valve needs to control the discharge pressure along the line shown in Fig. 3 to obtain a maximum heat pump efficiency.

5 IMPLEMENTATION IN A HEATING SYSTEM

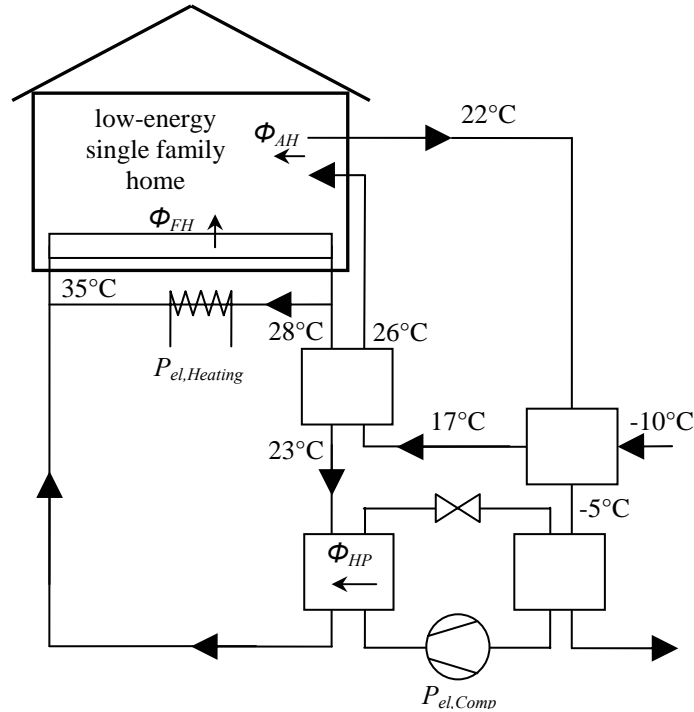
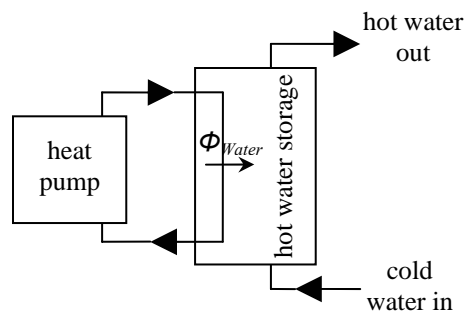


Fig. 5. Heating system with combined preheated ventilation and floor heating for a low energy home ensuring a water inlet temperature to the heat pump of less than 25°C. Figures are examples for minus 10 °C outside temperature and air exchange rate of 0.8/hour.

The heating system under development is intended to be used in a low-energy single family home in central Europe. A total area of 200 m² and a room height of 2.5 m is considered. According to the standard Minergie-P, which is well established in Switzerland, these homes require a heating power of 10 W/m². This results in a total required heating power of 2 kW. Fig. 5 suggests how to incorporate the heat pump in a heating system allowing the heat pump to operate efficiently. As seen in the thermodynamic analysis, this is only possible if the water inlet temperature is below 25 °C. In exchanging heat between the water exiting the floor heating before entering the heat pump and the fresh air used for ventilation, the temperature of the water inlet to the heat pump can be kept below 25 °C. An additional electrical heating can provide more than 2-kW heating power during the few days of the year when the outside temperature drops below the one considered as the design point, i.e. minus 11 °C. Due to the simplicity of the installation, the heat pump should be integrated in hot water generation as shown in Fig. 6.

Fig. 6. Simple integration of heat pump for hot water generation.



The coefficient of performance of the heating system COP_{HS} and of the hot water generation arrangement COP_{HW} used here are according to the definitions given in equations (2) and (3). In COP_{HS} the heat introduced through air heating Φ_{AH} and through floor heating Φ_{FH} are considered as well as the electrical power used by the compressor $P_{el,Comp}$ and by the extra electrical floor heating $P_{el,Heating}$ during cold days.

$$COP_{HS} = \frac{\phi_{Heating}}{P_{el,Comp} + P_{el,Heating}} = \frac{\phi_{AH} + \phi_{FH}}{P_{el,Comp} + P_{el,Heating}} \quad (2)$$

$$COP_{HW} = \frac{\phi_{Water}}{P_{el,Comp}} \quad (3)$$

Under the assumption that half of the heat consumption is by hot water and half of the consumption is by heating, the total efficiency COP_{total} is given by

$$COP_{total} = \frac{COP_{HS} + COP_{HW}}{2} \quad (4)$$

The combination of the heating system and hot water generation arrangement from Fig. 5 and Fig. 6 are analysed with regards to their coefficient of performance defined in equations (2) to (4). Using the published data for compressor efficiencies of the Dorin compressor as described above, the COPs shown in Fig. 7 are computed. It can be seen that a total efficiency of up to 4.5 can be achieved with existing technology. However, as described above, the Dorin compressor was not available and a compressor from Danfoss was used instead. Its efficiencies measured during the present investigations cause the total efficiency of the system to be somewhat lower than anticipated in figure 8.

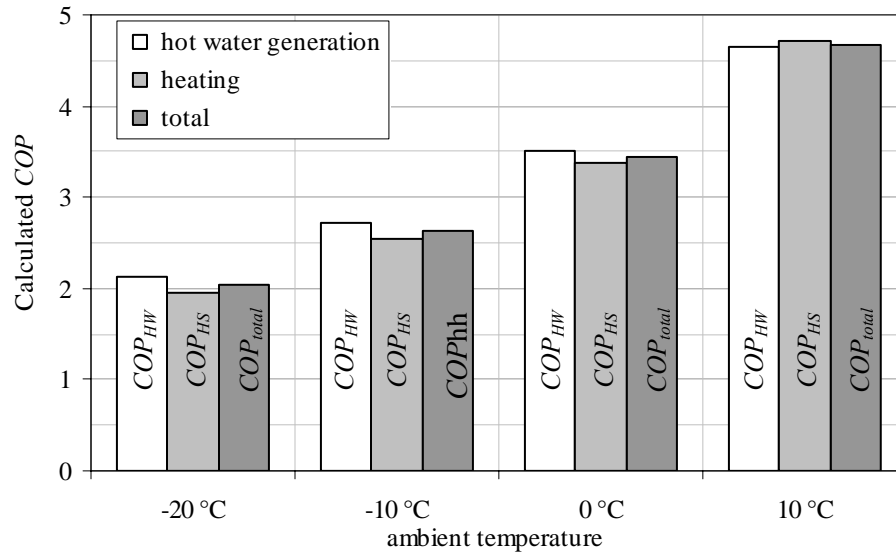


Fig. 7. Coefficients of performance for heating according to Figure 6 and for hot water generation according to Figure 7 as well as total efficiency at different ambient temperatures. Published data for compressor efficiency of a Dorin compressor is used. Air exchange rate is 0.4/hour.

6 CONCLUSIONS

Transcritical CO₂ heat pumps outperform heat pumps with other working fluids when generating hot water. Therefore this technology is selected to develop a commercial heat pump for low-energy one family homes in central Europe, where hot water generation accounts for half of the required heat consumption. The circuit proposed by Lorentzen is chosen because of its simplicity. A theoretical thermodynamic analysis gives insight in how to operate the heat pump and indicates the achievable efficiencies. A prototype is built and the heat pump's characteristics is assessed at different operating conditions. Optimal discharge pressures for maximum efficiencies are established to obtain a control algorithm for the throttling valve. The heat pump's integration in the heating system and for hot water generation is proposed in a way that the temperature of the water inlet to the heat pump remains below 25 °C. This is important to allow the heat pump to operate efficiently. An analysis of the overall system showed that COPs of up to 4.5 can be achieved with existing technology. However available compressor technology restricts the efficiency to somewhat lower values.

NOMENCLATURE

COP_{HP}		Coefficient of performance of the heat pump
$COP_{HP,max}$	–	Maximum coefficient of performance at the optimum discharge pressure
COP_{HS}		Coefficient of performance of the heating system
COP_{HW}		Coefficient of performance of the hot water generation arrangement
COP_{total}	–	Coefficient of performance of the total system
$P_{el,Comp}$	W	Electrical power used by the compressor
$P_{el,Heating}$	W	Electrical power used by the additional electrical heating
Φ_{AH}	W	Heat delivered through air heating
Φ_{FH}	W	Heat delivered through floor heating
$\Phi_{Heating}$	W	Heat delivered by the heating system
Φ_{HP}	W	Heat delivered by the heat pump
Φ_{Water}	W	Heat delivered into the hot water

REFERENCES

- Aarlién R. and Neksa P. 2002. "CO₂ as working fluid in heat pumping equipment" *7th International Energy Agency Heat Pump Conference*, Beijing, Paper A5-06, 19-22 May 2002
- Hubacher B. and Groll E.A. 2003. "Performance measurements of a hermetic, two-stage carbon dioxide compressor" *International Congress of Refrigeration 2003*, Washington DC (USA), Paper 029.

- Kim J.H., Yoon H.S., and Kim M.S. 2001. "An experimental investigation of heat transfer characteristics during in-tube gas cooling process of carbon dioxide" *Proceedings of the Conference of the International Institute of Refrigeration Commission B2*. Paderborn, Germany, pp. 526-533.
- Lorentzen G. 1990. "Trans-critical vapour compression cycle device" International Patent Publication WO9007683, 12 July.
- Lorentzen G. and Pettersen J. 1993. "A new, efficient and environmentally benign system for car air-conditioning" *International Journal of Refrigeration*, Volume 16, Number 1, pp. 4-12.
- Lorentzen G. 1994. "Revival of carbon dioxide as a refrigerant" *International Journal of Refrigeration*, Volume 17, Number 5, pp. 292-301.
- Ma Y., Zha S., Wang J., Li M. and Lu W. 2002. "The study of CO₂ transcritical cycle water source heat pump and the analysis of the application" *7th International Energy Agency Heat Pump Conference*, Beijing, Paper B3-06, 19-22 May.
- Nekså P., Dorin F., Rekstad H., Bredesen A., and Serbisse A. 1999. "Development of semi-hermetic CO₂-compressors" *20th International Congress of Refrigeration*, IIR/IIF, Sydney, Volume III, Paper 424, 19-24 September.
- Nekså, P. (1999) "CO₂ heat pump systems" *6th International Energy Agency Heat Pump Conference*, Berlin, 31 May-2 June.
- Rieberer R., Nekså P. and Schiefloe P.A. 1999. "CO₂ heat pumps for space heating and tap water heating" *20th International Congress of Refrigeration*, IIR/IIF, Sydney, Volume III, Paper 305, 19-24 September.
- Strømmen L., Bredesen A.M., Eikevik T., Nekså P., Pettersen J. and Aarli R. 1999. "Heat pumping systems for the next century" *20th International Congress of Refrigeration*, IIR/IIF, Sydney, Volume III, Keynot Address, Paper 734, 19-24 September.
- White S.D., Cleland D.J., Cotter, S.D., Stephenson R.A., Kallo R.D.S., and Fleming, A.K. 1997. "A heat pump for simultaneous refrigeration and water heating" *IPENZ Transactions*, Vol. 24, No.1/EMCh.

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