CO_2 HEAT PUMP SYSTEM FOR HEATING AND COOLING OF NON-RESIDENTIAL BUILDINGS

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ABSTRACT

CO_2 heat pumps installed in non-residential buildings will achieve a high COP as long as the heat is rejected over a large temperature range and the return temperature in the heat distribution system is relatively low. In European non-residential buildings high-temperature radiators are commonly used to cover the space heating demand, and a CO_2 heat pump rejecting heat to only radiators will achieve a low COP due to the high return temperature. However, in many non-residential buildings the demand for heating of ventilation air after the heat recovery unit constitutes a relatively large share of the total heating demand of the building. Consequently, by making a serial connection of the radiator system and the ventilation heater batteries it is possible to obtain a relatively low return temperature and with that favourable operating conditions for a CO_2 heat pump. Preheating and reheating of hot water will lead to a further increase in the COP for CO_2 the heat pump system.

Computer simulations have demonstrated that a CO_2 heat pump system in non-residential buildings can achieve the same or higher seasonal performance factor (SPF) than heat pumps using conventional working fluids as long as the heat distribution system is designed for a low return temperature. The operational time of the ventilation system will have a major impact on the SPF of the CO_2 heat pump, since the return temperature in the heat distribution system is considerably lower when the ventilation system is switched on.

The construction of a prototype CO_2 heat pump system for heating and cooling of a 3,000–5,000 m^2 Norwegian non-residential building is now being planned. The nominal heating capacity of the heat pump will be in the range of 50 to 150 kW, and it will be designed as a single-stage unit using an inverter controlled reciprocating compressor.

1. INTRODUCTION

In Norway R407C, R134a and ammonia (R717, 25/40 bar systems) are the most commonly used working fluids in heat pumps for heating and cooling of non-residential buildings such as office buildings, commercial buildings, hotels, schools, nursing homes and hospitals. There are also a few heat pump installations using propane (R290). R407C and R134a are covered by the Kyoto Protocol due to their relatively high GWP values. Although the working fluid leakages from HFC heat pump plants are relatively small, it is regarded a better long-term solution to utilize working fluids that do not have any negative impact on the global environment, such as the non-synthetic (natural) working fluids ammonia, hydrocarbons and carbon dioxide (R744, CO_2).

CO_2 is one of the few non-toxic and non-flammable working fluids that neither contributes to ozone depletion nor global warming, and CO_2 therefore represents an interesting long-term alternative. CO_2 has excellent thermophysical properties, and by utilizing these properties by means of optimized component and system design for the heat pump unit and the heat distribution system, high energy efficiency may be achieved in heat pumps for non-residential buildings (Stene, Jakobsen, 2006).

2. HEAT PUMPS IN NON-RESIDENTIAL BUILDINGS – CONCEPT DESCRIPTION

I addition to the heat source temperature and the isentropic efficiency of the compressor, it is mainly the mean temperature during heat rejection that determines the COP of a heat pump. Due to the low critical temperature of CO_2 (31.1ºC), a CO_2 heat pump will reject heat by cooling of single-phase CO_2-gas at supercritical pressure in a gas cooler. Since the CO_2 outlet temperature from the compressor is relatively high (>90ºC), a CO_2 heat pump can easily meet high-temperature heating demands. However, in order to achieve a high COP for the heat pump, it is essential that *useful heat* is rejected over a large temperature range, resulting in a large enthalpy difference for the CO_2 in the gas cooler and a relatively low CO_2 temperature before throttling.

In European non-residential buildings high-temperature radiators are commonly used to cover the space heating demand, and CO_2 heat pumps rejecting heat to only radiators will achieve a relatively low COP due to the high return temperature. However, in many non-residential buildings the demand for heating of ventilation air after the heat recovery unit constitutes a relatively large share of the total heating demand of the building. Consequently, by making a serial connection of the radiator system and the ventilation heater batteries it is possible to obtain a relatively low return temperature and with that favourable operating conditions for a CO_2 heat pump. Preheating and reheating of hot water will lead to a further increase in the COP for the heat pump system. Figure 2.1 illustrates how a serial connection of heat loads with diminishing temperature requirements may lead to a relatively low CO_2 outlet temperature (t_C) from the gas cooler (Stene and Jakobsen, 2006).

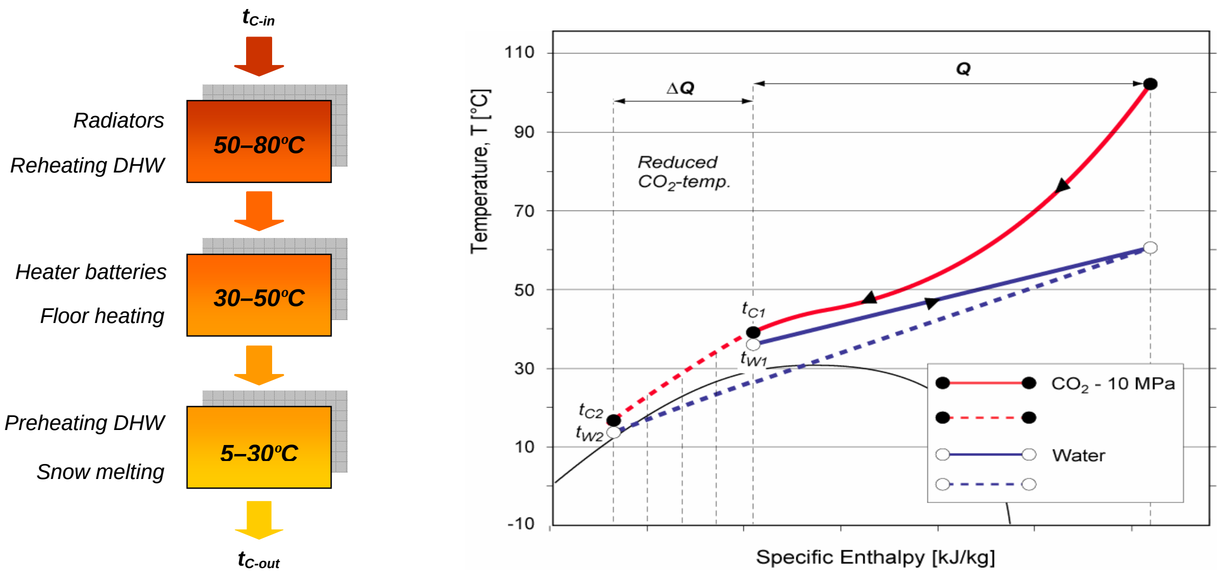

Figure 2.1 Illustration of how a serial connection of heat loads with diminishing temperature requirements will affect the CO_2 outlet temperature (t_C) from the gas cooler.

By employing a serial connection of the heat loads with diminishing temperature requirements, the return temperature in the heat distribution system is reduced from t_{W1} to t_{W2}. As a consequence, the CO_2 outlet temperature from the gas cooler drops from t_{C1} to t_{C2}, and the rejected heat from the heat pump increases by ΔQ. This in turn leads to a higher COP for the CO_2 heat pump.

The return temperature in the heat distribution system is determined by the heating effect and the temperature requirement for the different heating demands. In non-residential buildings with minimal hot water demands, the return temperature will be relatively high when the space heating demand is dominating. When there is a simultaneous demand for space heating and heating of ventilation air, the return temperature will be considerably lower due to the relatively low inlet temperature of the ventilation air. The ratio of the heating effects for space heating and heating of ventilation air is depending on the following factors:

- Heating effect – space heating:
  - Insulation standard (U-values) for the different parts of the building envelope, climate, indoor air temperature, period of use during day/week etc.
- Heating effect – reheating of ventilation air:
  - Air flow rates, efficiency of the heat recovery unit, set point temperature for the air, climate, type of ventilation system (CAV/VAV, mechanical/hybrid), period of use etc.
- Design of the hydronic system with radiators and ventilation heater batteries:
  - Temperature drop over radiators and heater batteries at different operating conditions

Figure 2.2 shows examples of measured heating effects (relative values) for space heating and heating of ventilation air for two Norwegian office buildings during a three day period (Thursday to Sunday) in February (Mathisen, 2006).

Bldg.	Year of construct.	Heated area	Ventilation system	Heating system
A	2002	850 m^2	Hybrid, VAV, no night stop	Hydronic, radiators
B	1996	24.000 m^2	Mechanical, VAV, night stop	Hydronic, radiators

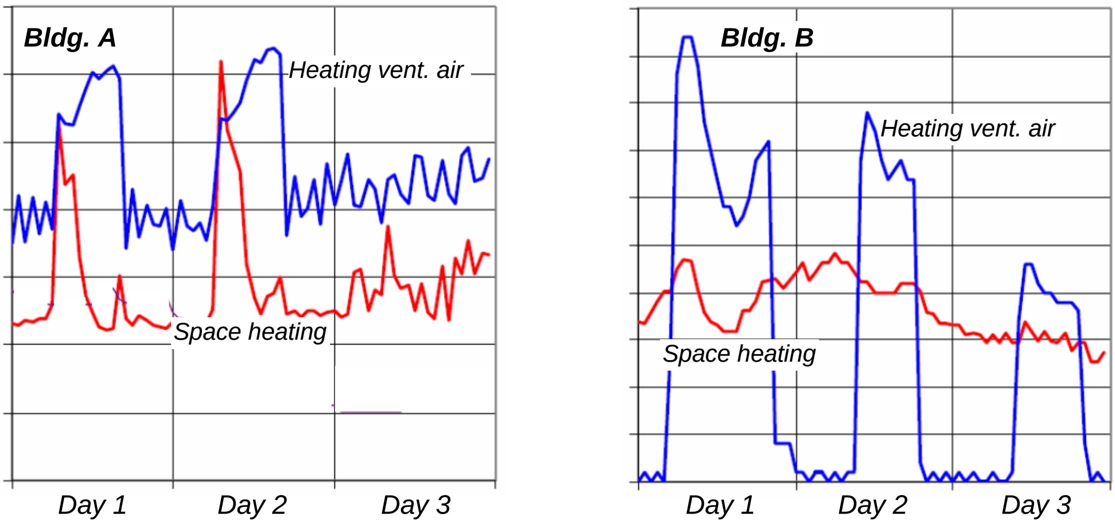

Figure 2.2 Measured heating effects (relative values) for space heating and heating of ventilation air for two Norwegian office buildings during a three day period in February, Thursday to Sunday (Mathisen, 2006).

The ratio of the heating effects for space heating and heating of ventilation air have been proved to be relatively constant during the year. As a consequence, the measured relative values during the three day periods shown in Figure 2.2 are representative for the entire year.

Figure 2.3 shows a principle example of a CO_2 heat pump unit supplying heat to a hydronic heat distribution system where the radiator circuits, heating batteries (ventilation), floor heating system, hot water system (preheating) and snow melting system are connected in series. By employing inverter controlled circulation pumps operating at constant differential pressure, i.e. volume flow control of the primary circuit in the heat distribution system, a low return temperature can be achieved at all operation conditions (Tengesdal, 2003).

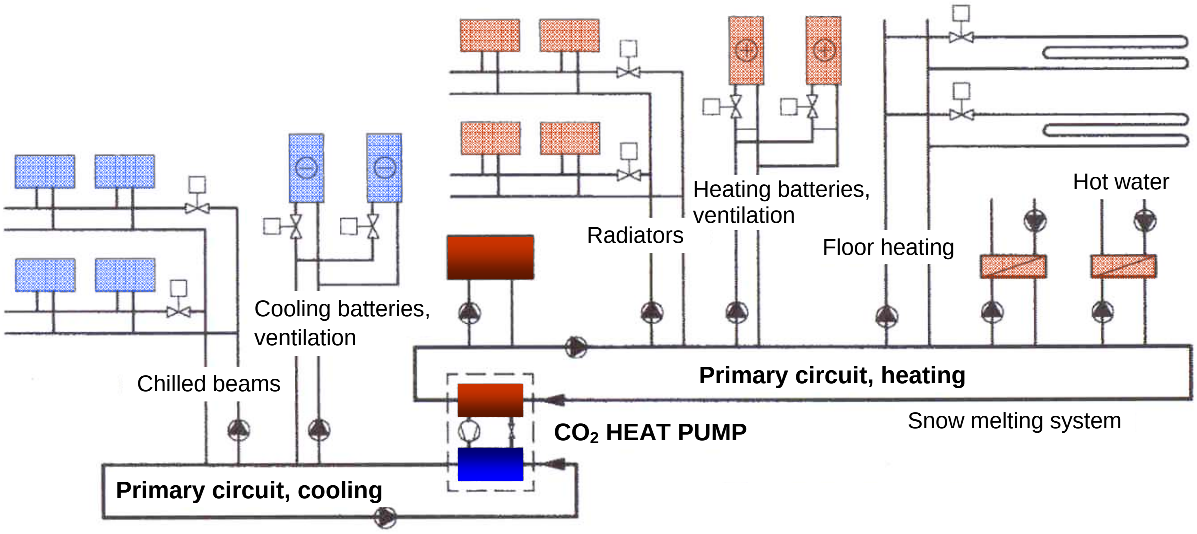

Figure 2.3 Principle example of a CO_2 heat pump system with serial connection of radiators, heating batteries, floor heating system and hot water system (Tengesdal, 2003).

3. CALCULATION OF ENERGY EFFICIENCY FOR CO_2 HEAT PUMPS

In order to evaluate the potential for CO_2 heat pumps for heating and cooling of non-residential buildings, the maximum achievable Coefficient of Performance (COP) as well as the Seasonal Performance Factor (SPF) of CO_2 heat pumps at different operating conditions have been calculated by means of in-house software developed at NTNU–SINTEF (Stene and Jakobsen, 2006).

Figure 3.1 shows the maximum achievable COP for a single-stage CO_2 heat pump unit as a function of the CO_2 outlet temperature from the gas cooler and the gas cooler pressure. In the calculations it has been assumed -5ºC evaporation temperature, 5 K superheating of the suction gas as well as 75% overall isentropic efficiency and 10% heat loss for the compressor. The design parameters are typical for heat pumps for heating and cooling of non-residential buildings, and where the heat pump utilizes sea water or energy wells in bedrock as heat source. The gas cooler pressure for CO_2 heat pump systems for heating of non-residential buildings typically range from 90 to 110 bar.

At 35ºC CO_2 outlet temperature from the gas cooler the calculated COP is approximately 3.0. At CO_2 outlet temperatures below 35ºC and high-side pressures between 9 and 11 MPa, the COP curves are practically linear, and the COP increases on average by 1% per degree Kelvin drop in the CO_2 outlet temperature. Moreover, at CO_2 outlet temperatures below 35ºC, the COP increases by roughly 2.5% per degree Kelvin rise in the evaporation temperature (Stene, 2004).

Since the CO_2 outlet temperature from the compressor is relatively high (>90ºC), a CO_2 heat pump can easily meet high-temperature heating demands. However, in order to achieve a high COP the temperature drop for the CO_2 must be large resulting in a considerable enthalpy difference in the gas cooler and a low CO_2 temperature before throttling.

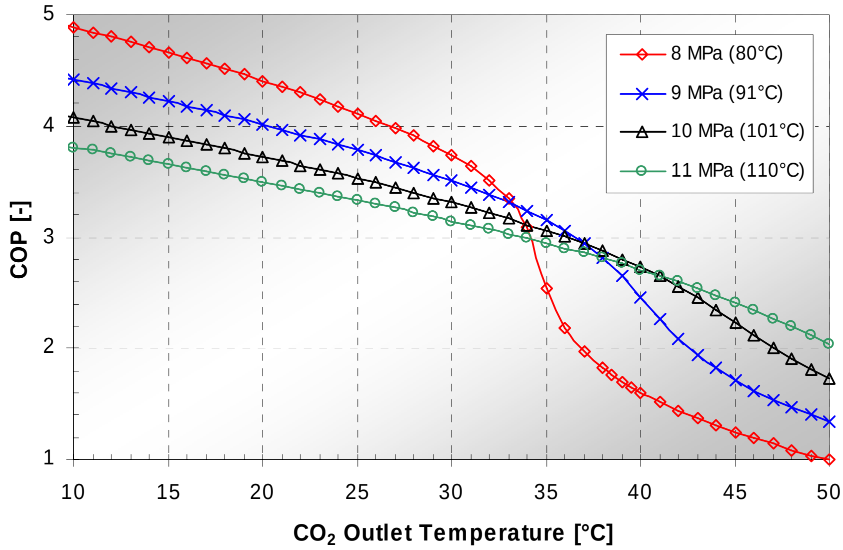

Figure 3.1 *Calculated COP for a single-stage CO_2 heat pump as a function of the CO_2 outlet temperature from the gas cooler and varying gas cooler pressure. The temperature at each gas cooler pressure represents the inlet CO_2 temperature in the gas cooler.*

Figure 3.2 shows the COP for a single-stage R134a heat pump unit as a function of the condensing temperature assuming -5ºC evaporation temperature, 5 K superheating of the suction gas as well as 70 and 80% overall isentropic efficiency and 10% heat loss for the compressor.

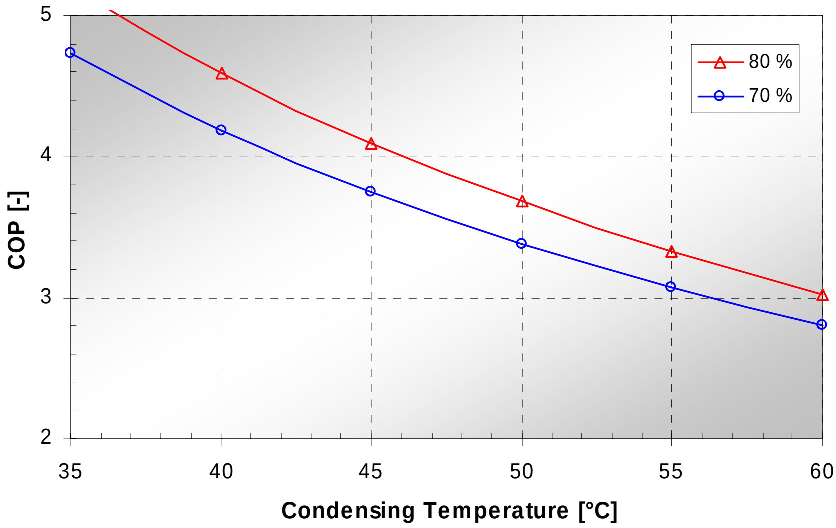

Figure 3.2 *Calculated COP for a single-stage R134a heat pump unit as a function of the condensing temperature. The overall isentropic compressor efficiency is 70 and 80%.*

With reference to Figure 3.1, the CO_2 heat pump achieves a COP of approx. 2.9 to 4.2 at CO_2 outlet temperatures in the range from 25 to 35ºC. For the R134a heat pump this corresponds to a supply water temperature from the condenser in the range from 39 to 56ºC at 5 K LMTD. In these calculations it has been assumed that the CO_2 compressor achieves 5 percentage points higher overall isentropic efficiency than the R134a compressor (Stene and Jakobsen, 2006).

The performance data from Figure 3.1 and 3.2 were used as a basis for calculating the Seasonal Performance Factor (SPF) for single-stage CO_2 and R134a heat pumps for heating and cooling of a 7,000 m^2 office building located in Oslo. The heat pump units were either operated in heating mode or cooling mode, and they supplied heating and cooling to hydronic distribution systems. The radiator circuits and the circuits for the ventilation heater batteries were connected in series, and the supply water temperature from the heat pump system was controlled according to an *outdoor temperature compensation curve*. Ambient air was used as heat source in heating mode and as heat sink in cooling mode. The efficiency of the heat recovery unit in the ventilation system was 60%. Table 3.1 shows the main design data for the building and the heat pump units.

Table 3.1 Main design data for the building, the heat recovery unit and the heat pumps.

Design load – space heating / heating of ventilation air	250 kW	300 kW
Design load – space cooling	400 kW	
	CO_2	**R134a**
Overall isentropic efficiency, compressor	75%	70%
LMTD – evaporator (constant)	8 K	8 K
Temperature approach – condenser/gas cooler (constant)	2 K	2 K
Efficiency – peak load unit (electro boiler)	100%	100%

The required swept volume for the compressors was designed in accordance with the maximum cooling demand. For both heat pump systems the necessary peak load in heating mode was covered by an electro boiler (bivalent heating system). For the CO_2 heat pump, auxiliary heating was only required at very high supply water temperatures due to the high inlet temperature in the gas cooler.

Figure 3.3 shows the calculated Seasonal Performance Factor (SPF) in heating mode for the heat pumps at different supply/return temperatures (design values) in the heat distribution system. Figure 3.4 shows the calculated SPF for both heating and cooling mode (Andresen and Stene, 2004).

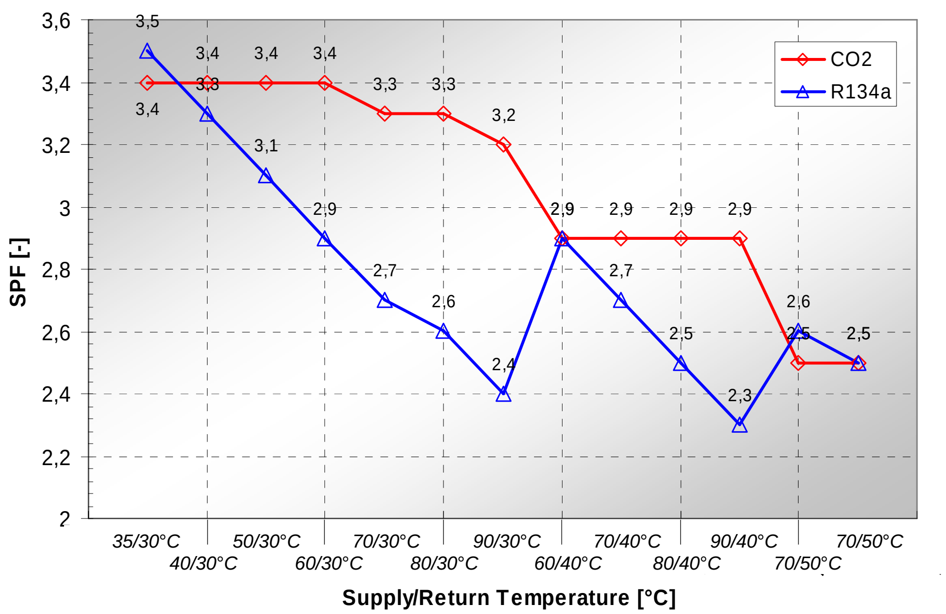

Figure 3.3 Calculated SPF in heating mode for the heat pump systems at different supply/return temp. (design conditions) for the heat distribution systems (Andresen, Stene, 2004).

In heating mode the CO_2 heat pump achieved the highest SPF at most operating conditions. The SPF of the CO_2 heat pump system was primarily influenced by the return temperature in the heat distribution system, while the SPF of the R134a heat pump system was mainly governed by the supply temperature (Andresen and Stene, 2004).

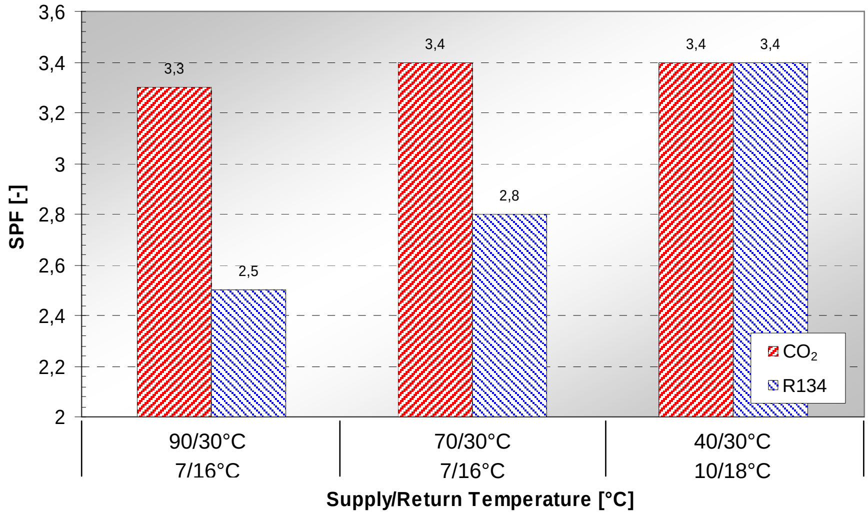

Figure 3.4 Calculated SPF in heating/cooling mode for the heat pumps at different supply/return temperatures (design conditions) for the distribution systems (Andresen, Stene, 2004).

The simulation results clearly indicated that a single-stage CO_2 heat pump for heating and cooling of an office building may achieve the same or higher SPF than a single-stage R134a heat pump as long as the temperature level in the heat distribution system is adapted to the characteristics of the CO_2 heat pump system. The CO_2 heat pump will achieve a higher SPF if the heating of ventilation air is dominating the total heating demand of the building, there is a considerable hot water demand or the CO_2 heat pump is equipped with an expander, an ejector or a heat exchanger that cools the CO_2 after the gas cooler and transfers the heat to the heat source (water or brine).

4. DESIGN OF A PROTOTYPE CO_2 HEAT PUMP SYSTEM

A prototype brine-to-water CO_2 heat pump system for heating and cooling of a 3,000–5,000 m^2 Norwegian non-residential building is now being planned (Stene and Jakobsen, 2006). The heating capacity of the heat pump will be in the range of 50 to 150 kW, and it will be designed as a single-stage unit using an inverter controlled reciprocating compressor. The construction of the prototype will be carried out by a Norwegian company, which has long-term experience in constructing high-quality refrigerating, air conditioning and heat pump systems for industrial applications.

High-quality components in the required capacity range are now commercially available:

- Compressor – Reciprocating CO_2 compressors for non-residential applications are available from several manufactures including Dorin, Bock, Mycom and Bitzer. The swept volume of the compressors range from 3 to 17 m^3/h, which corresponds to a heating capacity of approx. 15 to 60 kW at -10 to 0ºC evaporation temperature, 10 K suction gas superheating, 85% volumetric efficiency, 70% overall isentropic efficiency, 10% heat loss from the compressor shell, 90 bar high-side pressure and 30ºC outlet CO_2 temperature from the gas cooler.
- Evaporator – SWEP in Sweden has recently developed a brazed plate heat exchanger (PHE) with a maximum operating pressure of 64 bar, which corresponds to a CO_2 saturation temperature of approx. 25ºC. The maximum heat transfer area per unit is 8.8 m^2, which corresponds to an evaporator capacity of 110 kW when assuming an LMTD of 5 K and an U-value of 2,500 W/(m^{2}K). A tube-in-tube heat exchanger can also be used as an evaporator, but the overall heat transfer efficiency is lower than that of the plate heat exchanger.
- Gas cooler – SWEP has recently developed a brazed plate heat exchanger (PHE) with a maximum operating pressure of 140 bar. The maximum heat transfer area per unit is 4.04 m^2,

which corresponds to a heating capacity of 75 kW at 70/30ºC supply/return temperature in the space heating system, 75% overall isentropic efficiency and 10% heat loss for the compressor, 105 bar high-side pressure and 2 K temperature approach (Δt_A) at the gas cooler outlet. A tube-in-tube gas cooler unit is also a viable option, but this heat exchanger type has a lower heat transfer efficiency on the water side than a PHE.

- Other components – Expansion valves, low-pressure receivers as well as high-pressure control valves, safety valves, drying filers and tube fittings are commercially available.

Figure 4.1 shows a principle sketch of the prototype brine-to-water CO_2 heat pump system, where the high-side pressure is controlled by means of an LPR and a back-pressure valve (Shecco Cycle). The single-stage heat pump unit is equipped with an inverter controlled reciprocating compressor and two parallel gas cooler units for maximum performance at part load conditions, i.e. at reduced heating capacity and reduced water flow in the heat distribution system (Stene and Jakobsen, 2006).

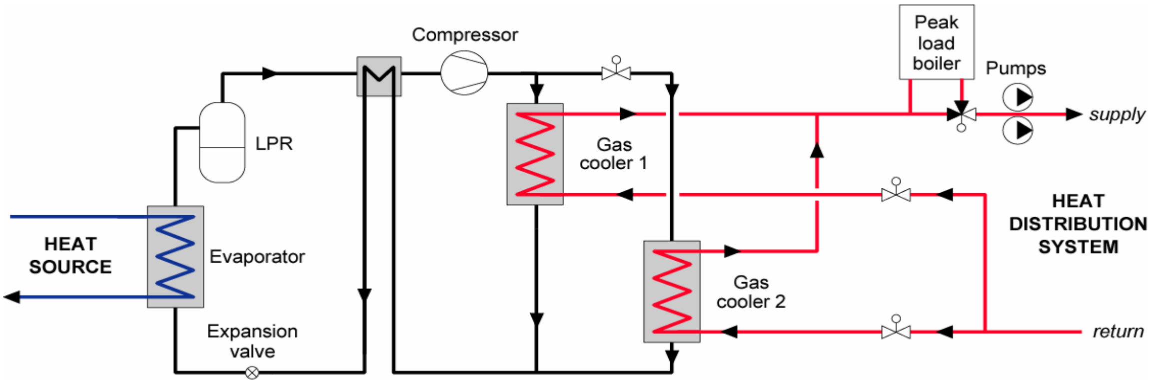

Figure 4.1 Principle sketch of the prototype brine-to-water CO_2 heat pump system.

5. CONCLUSIONS

Computer simulations have demonstrated that CO_2 heat pumps in non-residential buildings can achieve the same or higher seasonal performance factor (SPF) than heat pumps using conventional working fluids as long as the heat distribution system is designed for a low return temperature. The operational time of the ventilation system will have a major impact on the SPF of the CO_2 heat pump, since the return temperature in the heat distribution system is considerably lower when the ventilation systems is switched on. In contrary to conventional working fluids, CO_2 has no temperature limitations during heat rejection. Hence, in buildings with a dominating cooling demand, a CO_2 heat pump system will be able to cover the entire heating demand with high energy efficiency.

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