Residential Brine-to-Water Heat Pump System with CO_2 as Working Fluid

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ABSTRACT

Carbon dioxide (CO_2) is one of the very few non-toxic and non-flammable working fluids that neither contributes to ozone depletion nor global warming, and CO_2 is therefore regarded a promising long-term alternative to the CFCs, HCFCs and HFCs in many heat pumping applications. CO_2 has unique thermophysical properties, and by utilising these properties by means of optimised component and system design, high energy efficiency may be achieved.

Preliminary investigations have demonstrated that a residential CO_2 heat pump system is able to achieve the same COP as brine-to-water heat pumps using conventional working fluids. In order to demonstrate the potential of the technology, a project has been initiated at NTNU-SINTEF in Norway. In addition to modelling of the main components and system simulation, a full scale prototype brine-to-water CO_2 heat pump system will be designed and tested. The prototype will deliver heat to a low temperature floor heating system, and will cover the entire hot water demand in the house. The gas cooler will be divided into separate sections for preheating of city water, space heating and reheating of hot water, and the optimum supercritical pressure is estimated to be around 80 to 85 bar.

1. HEAT PUMPS BRING REDUCED GREENHOUSE GAS EMISSIONS

Since heat pumps upgrade low temperature heat from the environment (renewable energy) or from waste heat sources, they consume less primary energy than direct electric heating systems and oil and gas fired boilers. Depending on the climatic conditions and the quality of the heat pump system, residential heat pumps for space and hot water heating reduce the primary energy demand by typically 50 to 80%. *Hence, heat pumps have a considerable potential to reduce greenhouse gas emissions from heat production*.

Due to a number of reasons virtually all heat pumps are electrically driven. In most countries electricity is produced in different types of power plants using a mixture of fuels and other power sources such as coal, oil, natural gas, hydro power, nuclear energy as well as wind and solar energy. The potential for electric heat pumps to reduce CO_2 emissions is therefore depending on the specific CO_2 emissions from electricity generation, the Coefficient of Performance (COP) of the heat pump system, and the specific CO_2 emissions from competing heating systems such as oil and gas fired boilers.

When the electricity from combined heat and power (CHP) plants are used for heating purposes in heat pumps, the thermal energy production from these plants should be taken into account when calculating the specific CO_2 emission values. As a consequence, the specific CO_2 emissions from modern coal fired CHP plants will be reduced from 0.8 kg $CO_2/kWh_{electricity}$ to about 0.55 kg CO_2/kWh_{heat}. In Figure 1.1 the relative CO_2 emissions from various *residential heating systems* are compared, and the premises are as follows (Gilly 2000 and Stene 2000):

- Gas fired power plant: 0.35 kg CO_2/kWh_{el}
- UCPTE* 0.40 kg CO_2/kWh_{el}
- Coal fired CHP plant: 0.55 kg CO_2/kWh_{heat}
- Seasonal COP for the heat pumps: 2, 3 and 4
- Seasonal efficiency, gas boiler: 95% (LHV) – condensing boiler (propane/natural gas)
- Seasonal efficiency, oil boiler: 70% (LHV) – non-condensing boiler

**Union pour la co-ordination de la production et du transport de électricité (average European CO_2 emissions)*

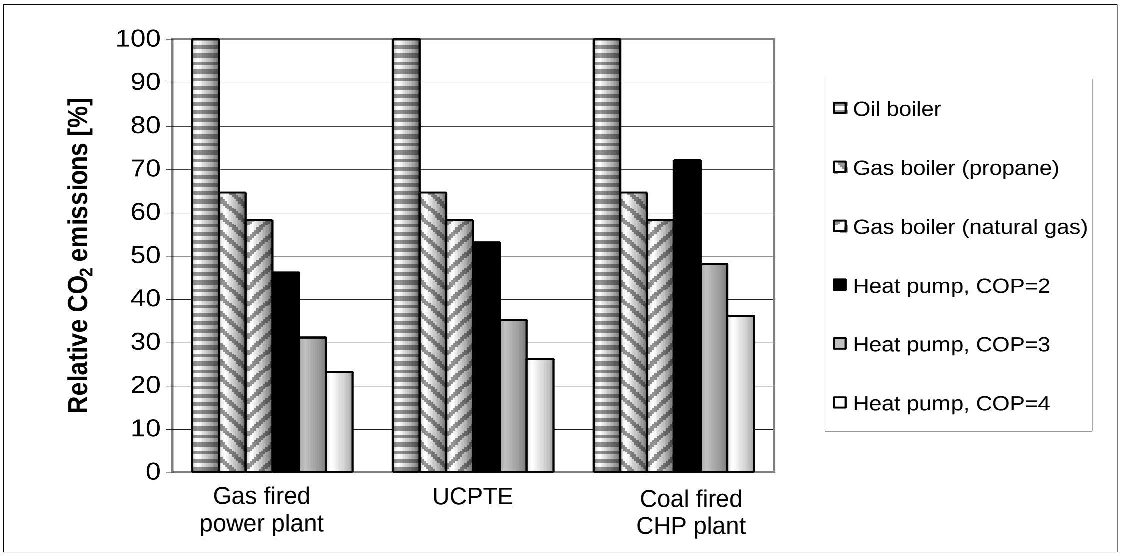

Figure 1.1 Relative CO_2 emissions of various heating systems, when the specific CO_2 emissions from power generation range from 0.35 to 0.55 kg CO_2/kWh_{el}. The seasonal energy efficiencies of the oil and gas fired boilers are 70% and 95%, respectively.

Electrically driven residential heat pumps offer substantial CO_2 reductions over conventional oil boilers. In comparison with high-efficiency gas fired boilers the reductions in CO_2 emissions are less pronounced, and when electricity is generated in coal fired CHP plants a relatively high COP is required in order to attain a net CO_2 reduction. Since there is a much higher potential for efficiency improvements in heat pumps than in oil and gas fired boilers, the main effort with regard to future residential heating systems should be concentrated on *developing high-efficiency heat pump systems*.

2. NATURAL WORKING FLUIDS – AN ENVIRONMENTALLY BENIGN ALTERNATIVE IN HEAT PUMP SYSTEMS

2.1 Long-Term Alternatives to the Ozone Depleting CFCs and HCFCs

During the last decade the most pressing research issue within the heat pump, air conditioning and refrigeration field has been the search for new and environmentally-acceptable working fluids that can replace the ozone-depleting ChloroFluoroCarbons (CFCs) and the HydroChloroFluoroCarbons (HCFCs). Most of the substances that have been considered are *new synthetic compounds*, namely the HydroFluoroCarbons (HFCs). Although these new compounds are extensively tested with regard to toxicity, flammability etc. they are foreign to nature. The main environmental drawback with the HFCs is the fact that these fluids are relatively strong greenhouse gases, and their greenhouse warming potential (GWP) factor is 1,300 to 3,500 times higher than that of CO_2. Widespread production and use of these fluids also include a risk of other unforeseen global environmental effects, which has already been experienced with the CFCs and HCFCs.

An alternative strategy is to apply naturally occurring and ecologically safe substances, so-called *natural working fluids*. These fluids include ammonia (NH_3), hydrocarbons (propane etc.), water, air and last but not least *carbon dioxide (CO_2).* As a result, all uncertainties and potential negative effects on the global environment are eliminated.

2.2 Characteristics of CO_2 as a Working Fluid

From an environmental point of view, carbon dioxide (CO_2, R-744) is almost the ideal working fluid since it is non-toxic, non-flammable and contributes neither to ozone depletion nor global warming[1]. In the late 1890's, CO_2 was preferred as a working fluid in various refrigerating and air conditioning systems. However, by the introduction of the CFCs in the 1930'ies and the HCFCs in the 1950'ies, the application of CO_2 was gradually reduced until it completely ceased during the sixties. After several decades of ignorance, CO_2 was "rediscovered" as a working fluid when several projects concerning CO_2 heat pumping systems were initiated at NTNU-SINTEF Energy Research in Norway in the early nineties. In recent years, the interest in CO_2 has been growing rapidly, and the fluid is now regarded as a promising long-term alternative in many heat pumping applications (Stene 1998).

Heat pump systems with CO_2 as working fluid will operate at much higher pressures than plants using HFCs, propane and ammonia. In other words, the evaporation pressures will be between 25 and 40 bar, and the heat rejection pressures in the range of 80 to 130 bar (Lorentzen 1994). Due to the high operating pressure and the relatively high specific enthalpy of evaporation of CO_2, the mass flow rate will be relatively low, and *the required compressor volume and dimensions of heat exchangers, piping, valves, and vessels will be considerably reduced compared with conventional systems*. As an example, the required swept volume for a reciprocating CO_2 compressor at 0°C suction gas temperature will be 70 to 85% lower than that of compressors designed for HFC working fluids (Stene 1998).

[1] *Since it is surplus CO_2 from industry that is being used as a working fluid, emissions of CO_2 from heat pumps due to unintentional leakages will not give a net contribution to the greenhouse effect (i.e. GWP=0).*

CO_2 has a very low critical temperature (31.1°C), and CO_2 heat pump systems will therefore have *transcritical operation* (Lorentzen 1994). This means that heat is absorbed from the heat source at subcritical pressure and constant temperature, and rejected above the critical point at supercritical pressure. The heat is not rejected by condensation of the fluid as with conventional working fluids but by cooling of single-phase high-pressure CO_2 in a *gas cooler*. Figure 2.1 displays the main difference between a conventional (sub-critical) heat pump cycle and the transcritical cycle by means of process lines in a pressure-enthalpy chart.

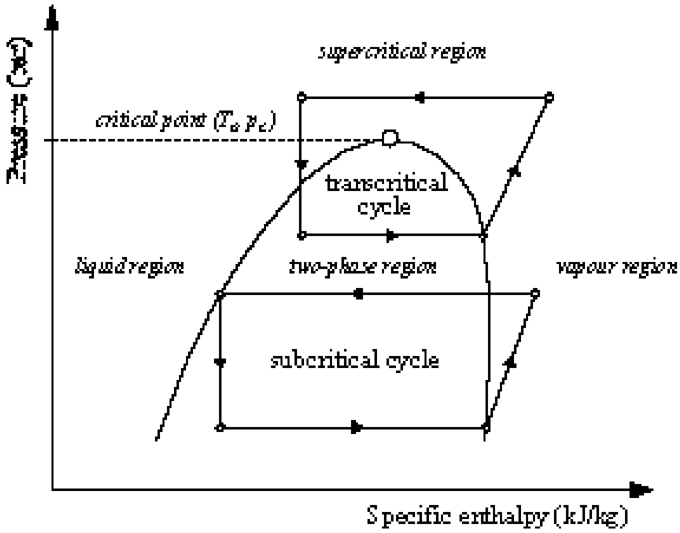

Figure 2.1 Principle comparison of subcritical and transcritical heat pumping cycles in a pressure-enthalpy chart (Stene 1998)

In contrary to conventional heat pump processes, the high-side pressure in a CO_2 process is independent from temperature, and the pressure is a result of the momentary CO_2 charge in the high-pressure side of the system. Moreover, both the heating capacity and the COP of the system are affected by the pressure level. Adequate pressure control can be achieved by adjusting the opening of the *expansion valve*, thereby transferring the CO_2 charge between the high-pressure and low-pressure side of the system (Lorentzen 1994).

The theoretical energy efficiency or COP of a simple transcritical CO_2 cycle is about 50% than that of heat pumping cycles using conventional working fluids, and this is mainly due to the large throttling loss of the fluid. However, in a practical CO_2 process, the energy efficiency may be competitive or even superior to the state-of-the-art HFC systems since:

- CO_2 has a lower pressure ratio than the HFCs, which results in relatively high isentropic and volumetric efficiencies for the compressor
- CO_2 has excellent heat transfer properties, and relatively large pressure losses can be tolerated in the system without a significant drop in saturation temperature. This may be utilised to achieve small temperature differences in the gas cooler and the evaporator
- Adequate system and component design may reduce the throttling loss considerably

3. DEVELOPMENT OF A RESIDENTIAL CO_2 HEAT PUMP SYSTEM FOR COMBINED SPACE AND HOT WATER HEATING

3.1 General Background

In recent years a number of universities, research institutions and companies have been evaluating and developing different types of residential heat pump systems using CO_2 as the working fluid. The applications include heat pump water heaters (Nekså et al 1998), heat pumps for retrofitting of boilers in high temperature radiator systems (Brandes and Kruse 2000), air-to-air systems for heating of ventilation air (Rieberer and Halozan 1998), and reversible air-to-air heat pumps / air-conditioners (Aarlien and Frivik 1997). None of these heat pump systems have yet been commercialised, but a 4.5 kW residential CO_2 heat pump water heater will be launched on the Japanese market in 2001 (Pettersen 2000).

In year 2000 NTNI-SINTEF Energy Research, Department of Refrigeration and Air Conditioning initialised a project where the aim is to develop and commercialise *a residential brine-to-water CO_2 heat pump for combined space and hot water heating*. The unique heat pump unit will be designed to cover the entire hot water demand, and it will be optimised for operating together with low temperature floor heating systems. The possibility of using the heat pump for reheating of ventilation air will also be evaluated in the project. The selection of the type of heat pump (heat pump concept) was made on the following basis:

- *High quality heat source:* In order to achieve a high energy efficiency and to maintain the heating capacity of the heat pump even at low ambient temperatures, rock, groundwater, or soil should be utilised as heat sources instead of ambient air. Owing to the operational reliability, most ground coupled heat pump systems are designed as indirect systems where a secondary fluid (brine) is circulating in a plastic coil or tube (ground heat exchanger).
- *Low temperature heat distribution system:* Hydronic floor heating systems installed in residences with adequate insulation standard require low distribution temperatures (30 to 40ºC), which is favourable for the COP of the heat pump. Another great advantage of a floor heating system is the superior thermal environment due the increased floor temperature and an almost ideal temperature profile in the room.
- *Focus on hot water production*: In recent years, transmission/infiltration losses from residences have been reduced by means of improved insulation, advanced windows etc. On the other hand there has been an increase in the use of hot tap water, which means that the energy demand for hot water heating constitutes an increasing share of the total heating demand in the residences. Consequently, residential heat pumps should preferably be designed to cover the entire hot water demand.

3.2 Characteristics of the CO_2 Heat Pump Unit

Since the CO_2 heat pump will have to operate in a transcritical cycle, heat will be rejected to the hot water and floor heating systems at a gliding temperature. In order to minimise the compression work as well as the exergy losses in the gas cooler, the high-side pressure should not exceed 80 to 85 bar (close to the critical pressure), and the gas cooler should be divided into three separate sections for: A) Preheating of hot water, B) Low temperature space heating (35ºC) and C) Reheating of hot water (60-65ºC). At this relatively low supercritical pressure and subsequent heat rejection to the floor and hot water systems, the CO_2 process will be rather similar to a conventional subcritical heat pump process with heat rejection from a subcooler, condenser and desuperheater. In Figure 3.1 the process lines for a supercritical CO_2 process at 85 bar high-side pressure as well as temperature profiles for the hot water and floor heating systems are drawn in a temperature-enthalpy chart. Heat is rejected from the CO_2 for reheating of hot water (3-4), space heating (2–3) and preheating of water (1-2).

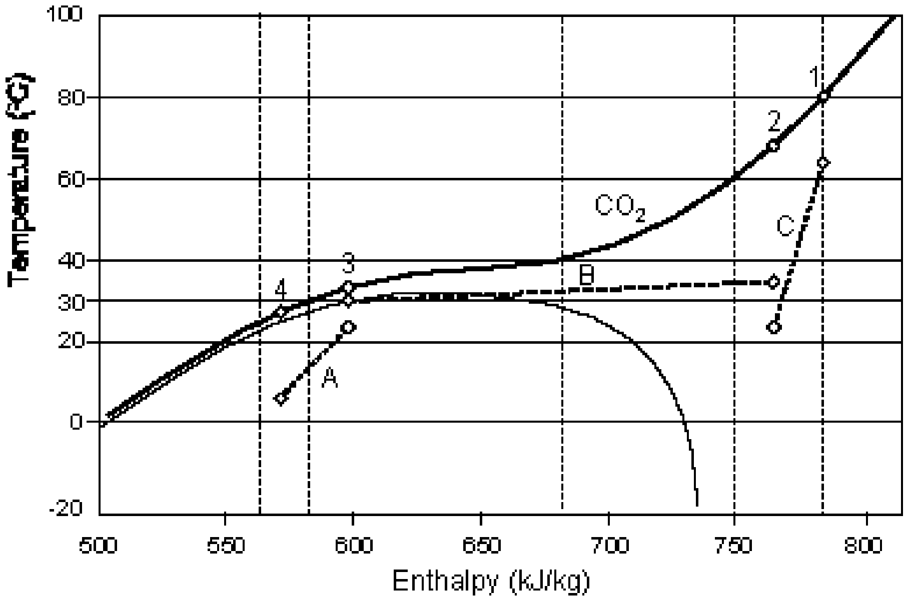

Figure 3.1 The CO_2 heat pump process for space and hot water heating illustrated in a temperature-enthalpy diagram. A) Preheating of hot water (3-4), B) Space heating (2-3), C) Reheating of hot water (1-2). The high-side CO_2 pressure is assumed to be constant at 85 bar (Stene 2001).

The slope of the CO_2 curve at constant pressure $(\partial T/\partial h)_p$ is the inverse of the specific heat capacity (C_p). Consequently, the specific heat capacity changes considerably during the heat rejection process, with a pronounced maximum around the critical point.

3.3 Calculations of COP and Temperature Profiles in the Gas Cooler

Modern residential brine-to-water heat pumps using HFCs and propane as working fluids typically achieve a *COP in the range of 2.5 to 3.7* at 0ºC inlet brine temperature and 45ºC supply water temperature (Van Dorn and Oosendorp 1997). Most of the units only preheats hot tap water, but there are also systems available that cover the entire hot water demand by means of a desuperheater.

In order to estimate the COP of the residential brine-to-water CO_2 heat pump unit for combined space and hot water heating, computer simulations have been carried out at different operating conditions. The premises and operating conditions were as follows (Stene 2001):

- *Hot water system:* 5-15°C city water temperature and 65°C hot water temperature.
- *Heat source:* Rock/groundwater well. +2/-1°C supply/return brine temperature
- *Floor heating system:* 30/35°C return/supply water temperature.
- *Evaporator:* -5°C evaporation temperature (30.5 bar). LMTD = 5 K. 5 K superheat.
- *Gas cooler:* 80 bar high-side pressure (85°C CO_2 exit temperature). Counter flow heat exchanger. 3-5 K temperature approach between CO_2 and water at the gas cooler outlet.
- *Compressor:* high-side pressure 80 bar. Pressure ratio 2.6. Isentropic efficiency 0.65 (motor efficiency included). 10% heat loss from the compressor.

Figure 3.2 shows the main results from the computer simulations. COP curves for a heat pump unit with isentropic compressor efficiencies of 70, 75 and 80% are also incl. (Stene 2001).

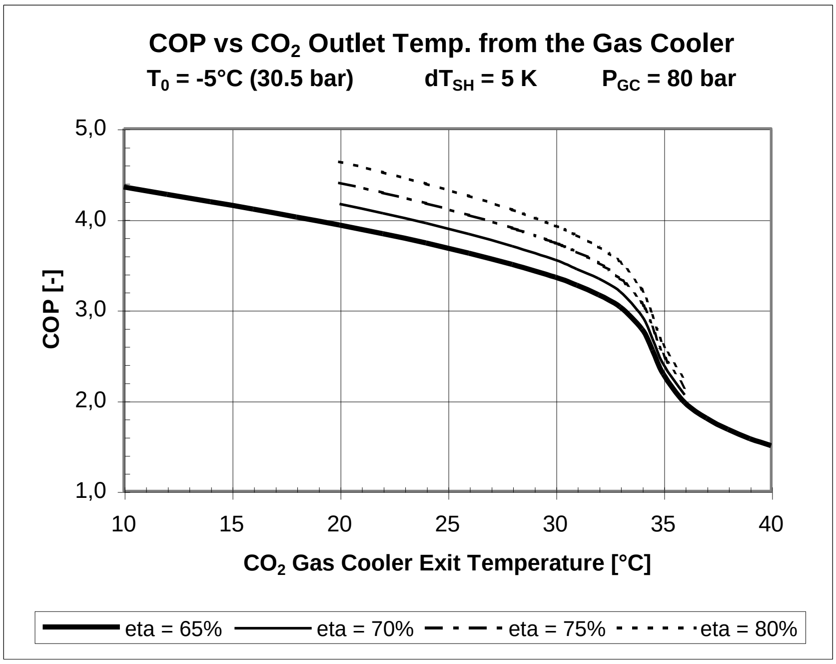

Figure 3.2 Calculated COP for a residential brine-to-water CO_2 heat pump for combined space and hot water heating at different gas cooler exit temperatures and isentropic efficiencies for the compressor (eta).

The COP increases when the CO_2 gas cooler exit temperature is reduced, since the specific enthalpy difference during heat rejection is increased whereas the specific compressor work is constant. At low hot water demands the COP will drop below 3 since the small city water flow is unable to cool the CO_2 more than a few tenth degrees. Moreover, when the heat pump provides hot water heating only, the sub-cooling is increased, and a COP close to 4 may be achieved. Consequently, *the share between the low temperature space heating demand and the high temperature hot water demand is a crucial parameter* when evaluating the CO_2 heat pump process, since the energy efficiency of the process is closely connected to the characteristics of heating demands and temperature levels.

A real heat pump installation will provide both space and hot water heating most of the year, and the ratio between the space heating demand and the hot water demand will depend on a number of factors such as the type/size/standard of house, ambient and indoor temperatures, solar radiation, number of residents as well as hot water consumption habits for the individual residents. In a Norwegian single family house, the hot water demand typically constitutes 15 to 20% of the total annual heating demand, whereas the share will be higher in warmer climates (e.g. Central Europe).

Figure 3.3 presents typical temperature curves for CO_2 and water in the tripartite counter flow heat exchanger during preheating (A) and reheating (C) of hot water as well as heating of water in the floor heating system (B). The heating demands for the floor heating system and the hot water system are 5.3 kW (83%) and 1.1 kW (17%), respectively, and the premises and operating conditions are the same as for the simulations in Figure 3.2. The normalised circuit length is defined as the ratio between the distance from the inlet to the actual location in the counter flow heat exchanger and the total length of the circuit.

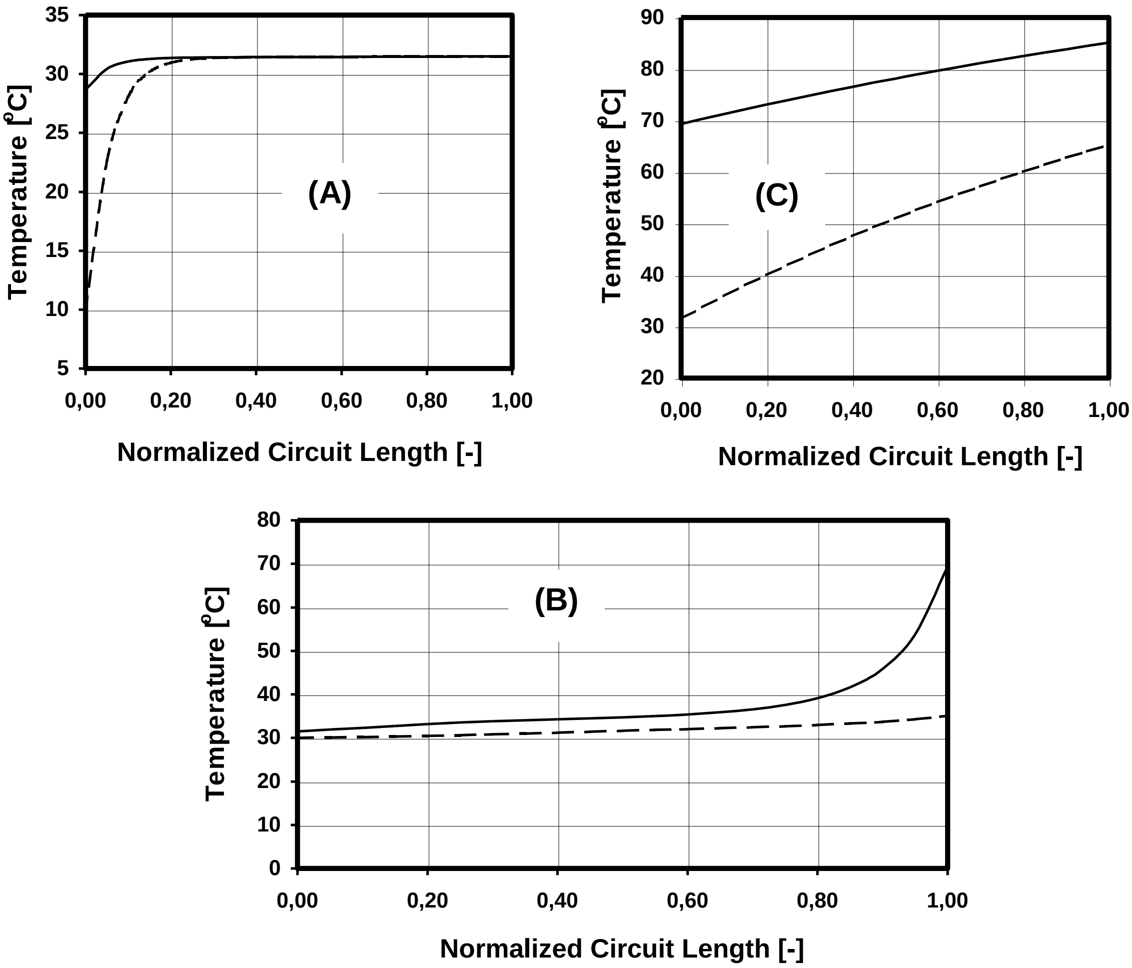

Figure 3.3 Example of temperature profiles for CO_2 and water in the counterflow heat exchanger: A) Preheating of hot water (10-30ºC), B) Space heating (30-35ºC), C) Reheating of hot water (30-85ºC). The heating demands for hot water and space heating are 1.1 kW and 5.3 kW, respectively (ratio 0.20).

3.4 The Prototype Heat Pump System

NTNU-SINTEF Energy Research will during 2002 carry out extensive testing of a full scale residential CO_2 heat pump unit installed in a pilot house in Norway. The heat source will be a brine circuit mounted in a rock well. The heat pump system will be designed as a bivalent heating system, which means that the 6.5 kW CO_2 heat pump unit will cover about 50% of the maximum space heating load at design outdoor temperature. The unit will in addition cover the entire hot water demand at 60 to 80ºC. Auxiliary heating will be provided by electric heaters. The heat distribution for space heating will be provided by a hydronic low temperature floor heating system (35/30ºC). Figure 3.4 shows a principle sketch of the heat pump unit.

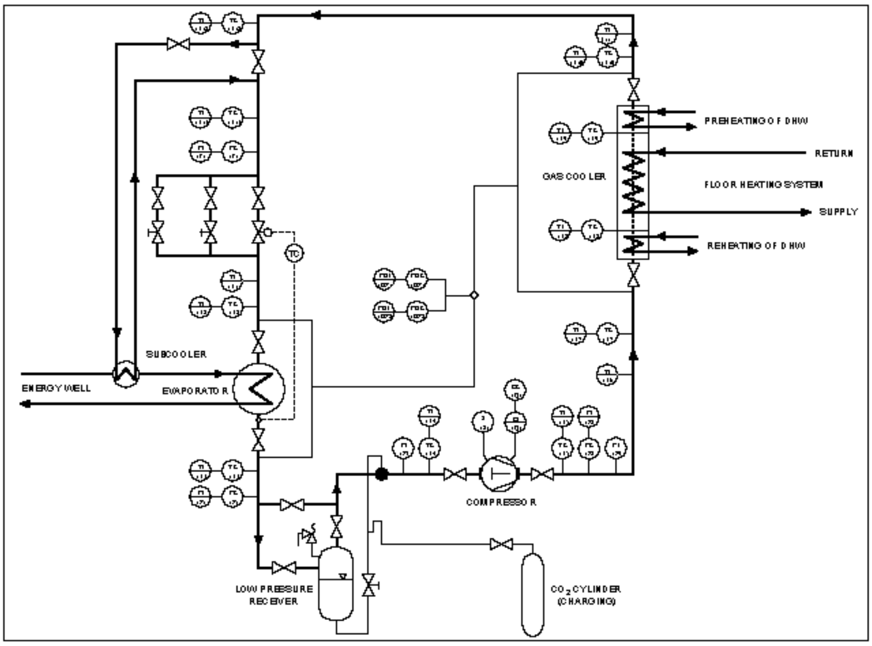

Figure 3.4 Principle sketch of the 6 kW prototype CO_2 heat pump unit

4. CONCLUSION

Preliminary investigations have shown that the CO_2 heat pump process is well suited for combined low temperature space heating and heating of hot water, and a ground coupled residential CO_2 heat pump may achieve the same energy efficiency as the state-of-the art propane and HFC heat pumps. However, in order to assess the real potential of this technology, more detailed studies are required with regard to system design, design and optimisation of heat exchangers as well as laboratory measurements.

5. ACKNOWLEDGEMENTS

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