HIGH-EFFICIENCY CO_2 HEAT PUMP WATER HEATER SYSTEMS FOR RESIDENTIAL AND NON-RESIDENTIAL BUILDINGS

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In hotels, hospitals, nursing homes, sport centres, block of flats and apartments buildings the annual energy demand for heating of domestic hot water (DHW) typically range from 25 to 60% of the total annual heating demand of the building. Due to the considerable DHW heat load, a centralized DHW system that covers the entire demand can be a very profitable installation. Possible heating systems include electric immersion heaters, solar collectors in combination with electric immersion heaters for supplementary heating, gas-fired boilers, pellet-fired boilers and heat pump water heaters (HPWH) utilizing e.g. ambient air, exhaust ventilation air, groundwater, boreholes in crystalline rock or grey water as heat source.

This papers shows that HPWH systems using carbon dioxide (CO_2, R744) as working fluid will typically achieve 20% higher COP than the most energy-efficient HPWH systems on the market using HFC or propane as working fluid. In terms of energy efficiency and utilization of renewable heat it even outperforms state-of-the-art solar heating systems. Air-to-water and water-to-water CO_2 HPWHs in the capacity range from about 5 to 60 kW have now become commercially available in Japan and Europe from a number of Japanese manufacturers.

1 Evaluation of Different Heat Pump Water Heater Systems

1.1 Heat Exchanger Configuration and System Design

The COP for a heat pump unit is heavily depending on the condensation temperature/pressure, and the COP typically increases by 3-4% per K reduction in the condensation temperature. In order to enable production of domestic hot water (DHW) in the required temperature range (60-80ºC) and still achieve a relatively high COP for a heat pump, state-of-the-art heat pump water heater (HPWH) systems are normally equipped with a desuperheater and possibly a subcooler. A *desuperheater* is a heat exchanger that cools down the hot exhaust gas from the compressor for reheating of DHW, while a *subcooler* is a heat exchanger that cools down the working fluid after the condenser (condensate) for preheating of DHW. Many HPWH systems are also using a combination of a desuperheater and a *suction gas heat exchanger*. The latter heat exchanger transfers heat from the condensate to the suction gas at the compressor inlet, and increases the exhaust gas temperature and the superheating enthalpy for the HPWH.

Figure 1 on the following page shows, as an example, a schematic diagram of a HPWH system equipped with a subcooler (*SC*), condenser and desuperheater *(DSH)*, while Figure 2 shows the cooling curve of the working fluid and the heating curve of the water in a Temperature-Enthalpy diagram when the city water (*CW*) is being heated from 5/15 to 70ºC.

Heat pump systems using carbon dioxide (CO_2, R744) as working fluid represent a new and promising technology, e.g. for HPWH systems. CO_2 is a non-flammable and non-toxic fluid that does not contribute to global warming as the HFC working fluids, i.e. GWP=0. Due to the unique thermophysical properties of CO_2, high energy efficiency can be achieved if the heat pump system is correctly designed and operated in order to utilize the properties of the fluid.
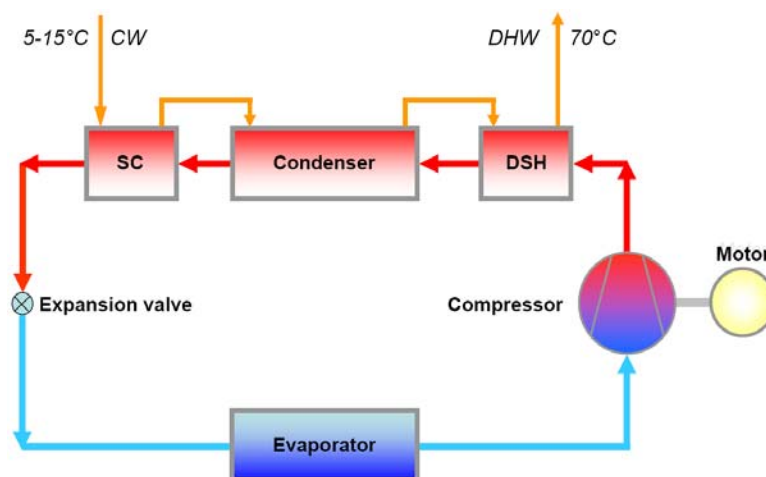
Figure 1   Heat pump water heater equipped with three separate heat exchangers in series.
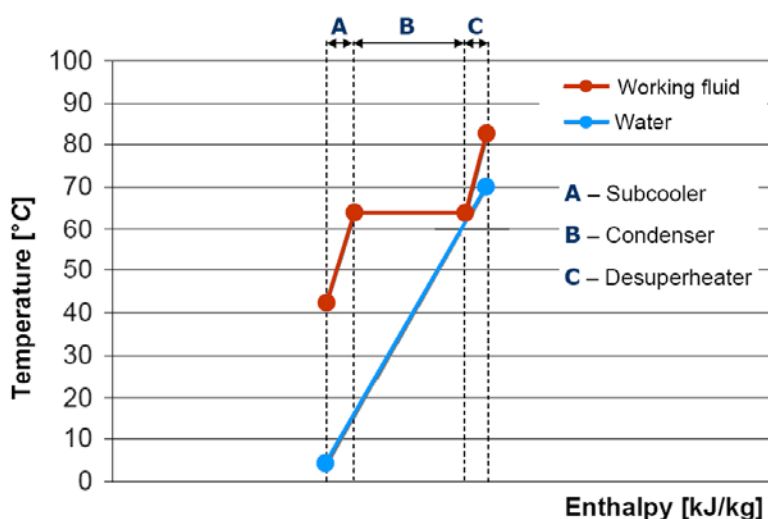

Figure 2   Heat pump water heater with heat rejection in a subcooler (A), condenser (B) and desuperheater (C) for heating of city water from 5 to 70ºC.

Due to the low critical temperature of CO_2 (31.1ºC), a CO_2 HPWH will be operating in a so-called transcritical heat pump cycle were heat is rejected by cooling of CO_2 vapour at supercritical pressure in a counter-flow gas cooler. Figure 3 on the following page shows an example of temperature profiles for CO_2 and water in a CO_2 gas cooler for DHW heating.

1.2   COP Comparison for Different HPWH Systems

Four different HPWH designs have been analyzed in order to identify the system with the highest COP at varying evaporation temperature (-10 to +10ºC), varying inlet city water temperature (5 to 30ºC) and varying outlet water temperature (60 to 85ºC). The HPWH systems were as follows (Hjerkinn, 2007):

- **System 1** – Heat pump with condenser and desuperheater
- **System 2** – Heat pump with subcooler, condenser and desuperheater
- **System 3** – Heat pump with suction gas heat exchanger, condenser and desuperheater
- **System 4** – *CO_2 (R744)* heat pump with a single counter-flow gas cooler unit
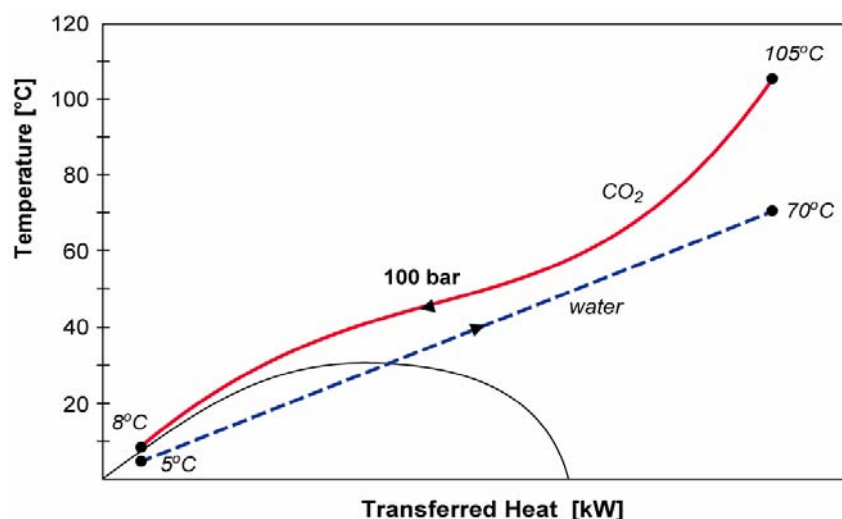
Figure 3   Example of temperature profiles in a CO_2 gas cooler for DHW heating.

HPWH systems no. 1, 2 and 3 were simulated with both R134a and R290 (propane), since these working fluids have a sufficiently high condensation temperature (60-70ºC) when using standard components with 25 bar pressure rating.

In order to attain equal boundary conditions for the four different heat pump units, the various heat exchanger combinations were simulated with equal UA-values, which limited the size and heat transfer efficiency for the heat exchangers. The UA-value ranged from *1,800 to 2,400 W/K*, and the higher the UA-value the lower the condensation temperature.

Figure 4 shows the calculated COP for the different HPWH systems as a function of the evaporation temperature, t_0 (Hjerkinn, 2007). In the calculations it was assumed a total UA-value of 2,100 W/K for both the condenser/heat exchanger combinations and the CO_2 gas cooler, 5 K superheated vapour from the evaporator, 5ºC inlet city water temperature and 70ºC DHW temperature. The overall isentropic efficiencies for the compressors were calculated on the basis on typical efficiency curves from laboratory measurements.
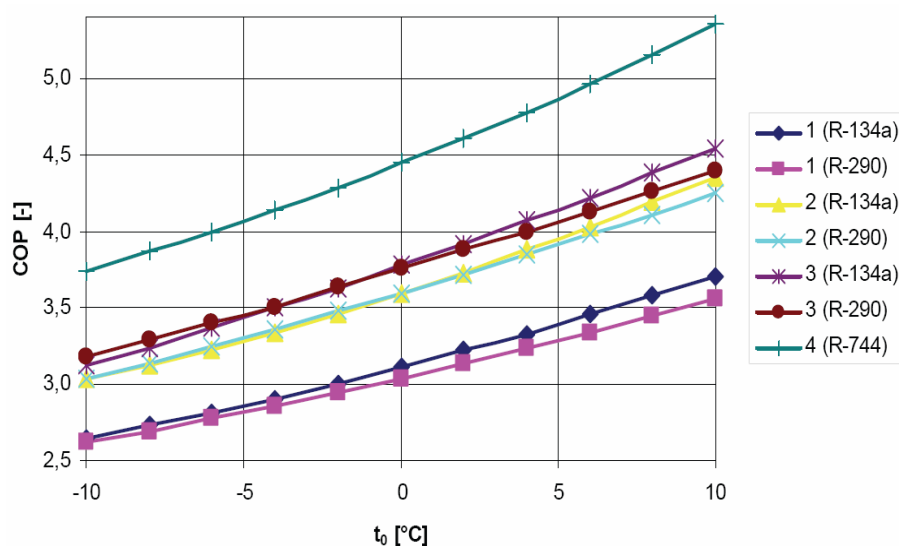

Figure 4   Calculated COP as a function of the evaporation temperature t_0 (Hjerkinn, 2007).

For the state-of-the-art HPWH systems with R134a or R290 as working fluid, *System 2* (sub-cooler, condenser and desuperheater) and *System 3* (suction gas heat exchanger, condenser and desuperheater) achieved more or less the same COP at varying operating conditions. *System 1* (condenser and desuperheater) achieved roughly 15% lower COP than System 2 and 3. The main reason for the lower COP was that System 1 operated at a higher condensation temperature due to poorer temperature fit between the water and the working fluid.

The most energy efficient HPWH was *System 4*, the CO_2 system, which in average achieved about 20% higher COP than System 2 and 3. This was mainly due to higher compressor efficiency and the excellent temperature fit in the gas cooler between the CO_2 and the water, which minimized the average CO_2 temperature during heat rejection. In addition to the high energy efficiency for the CO_2 HPWH, CO_2 has the advantage of being a non-flammable, non-toxic and environmentally friendly fluid with zero GWP factor when used as a working fluid.

2   Case Study – Design and Evaluation of a CO_2 HPWH

"Bergen Housing Cooperative" is about to construct a number of apartment buildings of passive house standard at Damsgårdssundet in Bergen, Norway with about 300 flats. An in-depth analysis has been carried out for one of the buildings with 40 flats in order to develop a high-efficiency centralized CO_2 HPWH system (Hjerkinn, 2007).

2.1   Design of the HPHW and the Storage Tanks

The energy demand for DHW heating in the apartment building was estimated to be approx. 170,000 kWh/year, i.e. about 4,250 kWh/year for each flat. This also included DHW to washing machines and dish-washing machines (600 kWh/year). The CO_2 HPWH was designed for a heating capacity of 26 kW at 12ºC inlet city water temperature, 70ºC DHW temperature and 3 K temperature difference between the outlet CO_2 and the inlet city water in the gas cooler. The isentropic and volumetric efficiency of the compressor was 0.70 and 0.75, respectively. The HPWH heating capacity was calculated assuming 18 hours operating period per 24 hours for the heat pump unit and the application of four 1,000 litres DHW storage tanks to cover large momentary DHW demands (Hjerkinn, 2007). Figure 5 shows the estimated 24 hour DHW consumption diagram for the apartment building, with the maximum DHW demand between 16-19 o'clock in the evening (approx. 3 hours maximum).
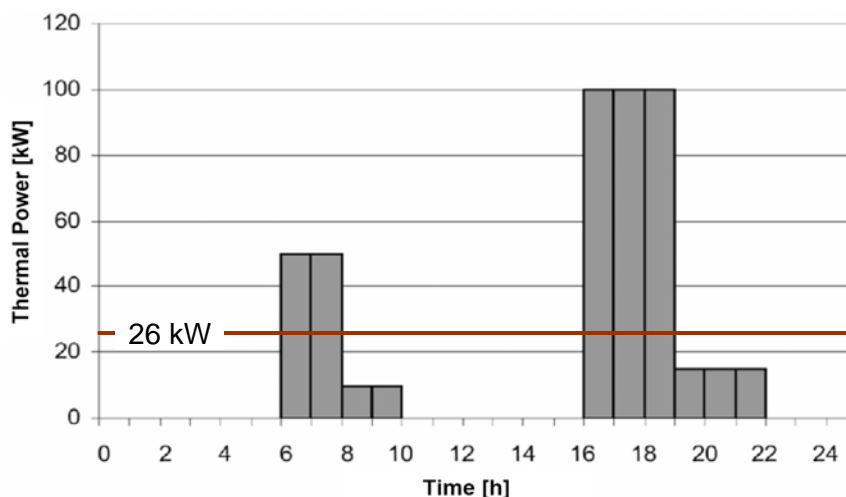

Figure 5   24 hours DHW consumption diagram for the apartment building (Hjerkinn, 2007).

2.2   Component and System Design

The 26 kW CO_2 HPWH system was simulated in *CSIM* (Skaugen, 2002), which is an advanced computer programme developed at NTNU-SINTEF for optimization of CO_2 heat pump systems. CSIM calculates, among other things, real evaporator, gas cooler, suction gas heat exchanger and compressor performance based on laboratory measurements.

The CO_2 HPWH was equipped with a 60 bar plate heat exchanger as evaporator (*SWEP*), a reciprocating compressor with 150 bar pressure rating (*Bitzer*), a 140 bar counter-flow plate heat exchanger as gas cooler (*SWEP*), a counter-flow tube-in-tube suction gas heat exchanger, an automatic expansion valve (back-pressure valve) and a low-pressure receiver (LPR). The expansion valve and the LPR were used for optimization of the pressure in the gas cooler at varying operating conditions in order to maximize the COP.

The inlet water temperature in the gas cooler has a considerable impact on the COP of a CO_2 HPWH, and the lower the temperature the higher the COP. A low inlet water temperature leads to a low CO_2 outlet temperature from the gas cooler and with that a large enthalpy difference during heat rejection. Figure 6 shows simulated relative COP values for a CO_2 HPWH unit at varying inlet water temperature and 60 and 80°C DHW temperature. Example – by increasing the inlet city water temperature from 5°C to 15°C at 60°C DHW temperature, the COP is reduced by approximately 10% (Stene, 2004).

From Figure 6 it can be concluded that the DHW storage system for a CO_2 HPWH should use relatively small diameter storage tanks connected in series in order to minimize conductive heat transfer between the DHW and the city water in the tanks (exergy loss). Efficient diffusers at the tank inlets should also be installed in order to minimize the water velocity and consequent mixing of water at different temperature levels (exergy loss).
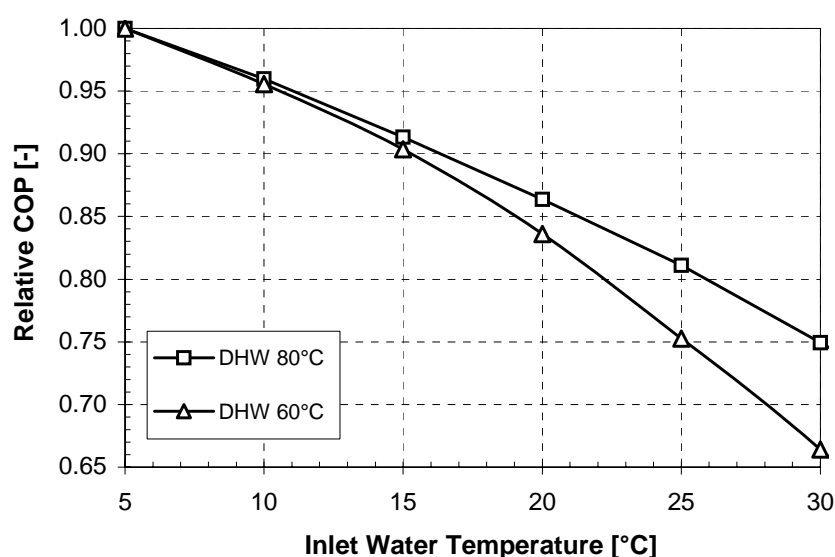

Figure 6   Simulated relative COPs for a CO_2 HPWH as a function of the inlet water temp. to the gas cooler at 60 and 80ºC set-point temperature for the DHW (Stene, 2004).

Figure 7 on the following page shows a schematic diagram of the CO_2 HPWH system including four 1,000 litre single-shell storage tanks connected in series and an inverter controlled pump, *Pump 1* (Hjerkinn, 2007).
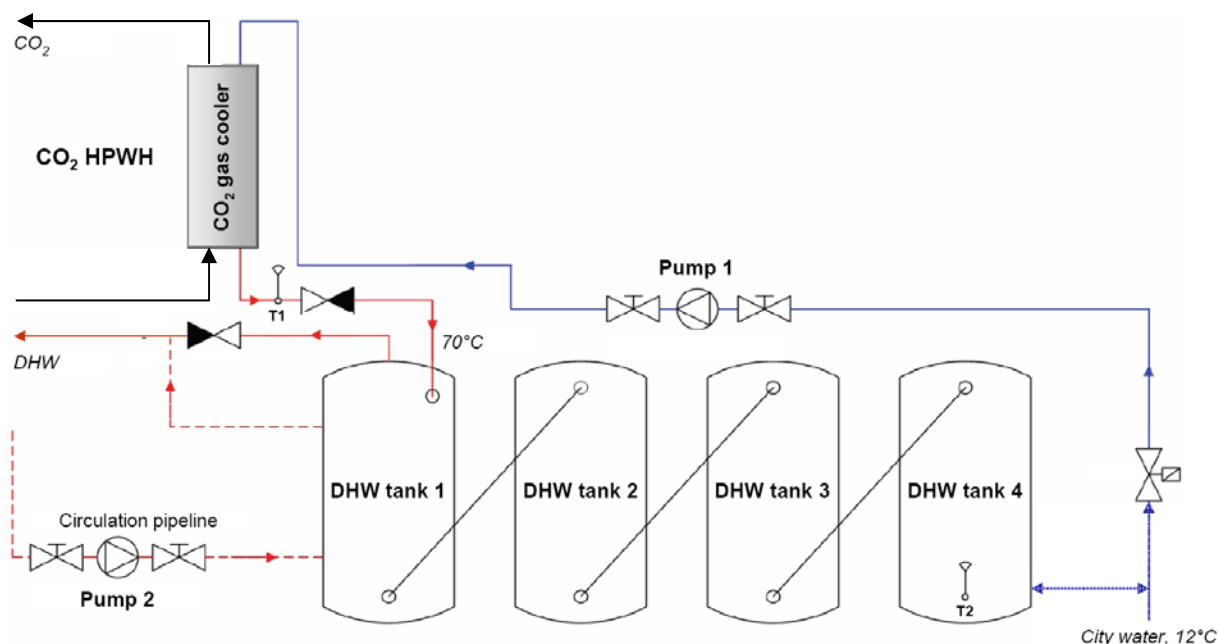
Figure 7   Schematic diagram of the 4000 litre DHW storage tank system connected to the single counter-flow CO_2 gas cooler unit (Hjerkinn, 2007).

2.3   Control Strategy

The DHW system is designed as a closed unvented (pressurized) system, where DHW tank 4 is connected to the city water supply and DHW tank 1 is connected to the tapping sites.

During the tapping periods, cold city water will enter the bottom of DHW tank 4 whereas the same amount of hot water will flow from the top of DHW tank 1 to the tapping sites. The CO_2 HPWH will normally run during the tapping periods. An inverter controlled pump (*Pump 1*) circulates the cold city water to the single CO_2 gas cooler unit, where the water is being heated to the set-point temperature (*T1*) before it enters DHW tank 1. When the tapping period has ceased, the CO_2 heat pump will be running as long as the water temperature (*T2*) at the bottom of DHW tank 4 is lower than the set-point temperature (e.g. 70ºC). The gas cooler pressure for the CO_2 heat pump unit is optimized in order to maximize the COP for the heat pump at varying operating condition.

2.4   Simulation Results – COP and Profitability

When using groundwater at 7ºC as the heat source, the calculated seasonal performance factor (SPF) for the CO_2 HPWH was approximately 3.8 (Hjerkinn, 2008). This corresponds to an annual energy saving of about 70-75% compared to a conventional DHW system with electric immersion heaters, and is 20-35 percentage points higher that that of Scandinavian DHW systems based on solar collectors and electric immersion heaters for supplementary heating. Figure 8 shows the energy saving (i.e. the utilization of renewable heat – green bars) for the CO_2 HPWH compared to electric immersion heaters and a solar heating system.

The maximum (permissible) investment cost for the CO_2 HPWH system was as high as €125,000 (4,800 €/kW) when using the following boundary conditions: 170,000 kWh/year annual heating demand, 3.5 average heat pump COP (conservative value), 6% real interest rate, 15 years economic lifetime, 0.1 €/kWh (0.80 NOK/kWh) electricity price and electric immersion heaters as reference DHW system.
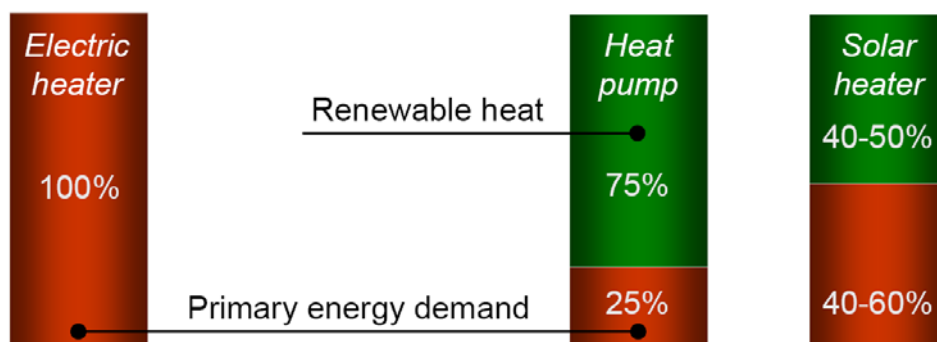
Figure 8   Primary energy demand and utilization of renewable heat for different hot water heating systems (electric immersion heater, heat pump, solar heater).

3   Conclusion

Due to the high energy efficiency and the favourable environmental properties of carbon dioxide (CO_2) as a working fluid, CO_2 heat pump water heater (HPWH) systems are regarded a promising technology for centralized DHW heating in e.g. hotels, hospitals, nursing homes, sport centres, block of flats and apartments buildings. In terms of energy efficiency and utilization of renewable heat the CO_2 system even outperforms state-of-the-art solar heating systems. Air-to-water and water-to-water CO_2 HPWH systems in the capacity range from about 5 to 60 kW are now available in Japan and Europe from a number of Japanese manufacturers (ref. *www.R744.com*).

4   References

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