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Residential CO2 Heat Pumps for Combined Space Heating and Hot Water Heating - System Design, Test Procedures and Calculation of SPF

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Residential CO_2 Heat Pumps for Combined Space Heating and Hot Water Heating – System Design, Test Procedures and Calculation of SPF

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RESULT (summary)

Norway is a member of Annex 28, "Test Procedure and Seasonal Performance Calculation for Residential Heat Pumps with Combined Space and Domestic Hot Water Heating" (2003-2005), organized under the IEA Heat Pump Programme. The participating countries are AU, CA, CH, FR GE, JP, NO, SE, and the US. The main tasks of the Annex are to provide a state-of-the-art description of residential heat pump systems for combined space heating and hot water heating (*integrated heat pump systems*), to develop adequate test procedures for these systems, and to work out simple methods for calculation of the Seasonal Performance Factor (SPF).

Design: The report provides an overview of possible designs for integrated heat pumps using **carbon dioxide** (CO_2) as the working fluid. The following classification has been used: 1) Heat pumps with one, two or three external gas cooler units that are connected to the DHW storage tank by means of a closed water loop, 2) Heat pumps where the gas cooler is an integral part of the DHW tank, and the heat exchanger for the space heating system is mounted inside or outside the tank. The different system designs have distinct characteristics regarding e.g. temperature flexibility, technical complexity and attainable COP at various operating conditions.

Testing: The report presents recommended test conditions and test procedures for integrated CO_2 heat pumps during operation in *the space heating mode*, *the DHW heating mode* and *the combined heating mode* (i.e. simultaneous space heating and DHW heating). Since the thermodynamic losses in the DHW tank may have a significant impact on the COP, integrated CO_2 heat pumps should always be tested together with the DHW tank. It is also important to have specific requirements regarding e.g. the maximum supply temperature in the space heating mode (40°C for systems with tripartite gas cooler) as well as the inlet tap water temperature (10°C), the minimum outlet water temperature (55°C), the hot water flow rate (0.1 dm^3/s) and the draw-off pattern ($4 \times 0.25 \cdot V_n$) in the DHW and combined heating modes.

SPF calculation: The last part of the report briefly discusses the method for calculating the Seasonal Performance Factor (SPF) of integrated heat pump systems.

KEYWORDS

SELECTED BY AUTHOR(S)		
	Residential heat pumps	Test conditions and test procedures
	CO_2 as working fluid	Calculation of SPF

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1 INTRODUCTION

Norway is a member of Annex 28, "Test Procedure and Seasonal Performance Calculation for Residential Heat Pumps with Combined Space and Domestic Hot Water Heating" (2003-2005), organized under the umbrella of the International Energy Agency (IEA) and the IEA Heat Pump Programme (HPP). The 9 participating countries are Austria, Canada, France, Germany, Japan, Norway, Sweden, Switzerland (Operating Agent), and the USA.

The main tasks of IEA HPP Annex 28 are:

- **Task 1** – Provide a state-of-the-art description of residential heat pump systems for combined space heating and hot water heating – so-called *integrated* heat pumps.
- **Task 2** – To develop adequate test procedures for residential integrated heat pump systems, which deliver the necessary data with a minimal requirement of testing equipment and testing time.
- **Task 3 –** To work out simple methods for calculation of the SPF for integrated heat pump systems

Task 1 – The report provides an overview of possible designs for integrated heat pumps using **carbon dioxide (CO_2)** as the working fluid. The following classification has been used: 1) Heat pumps with one, two or three external gas cooler units that are connected to the DHW storage tank by means of a closed water loop, 2) Heat pumps where the gas cooler is an integral part of the DHW tank, and the heat exchanger for the space heating system is mounted inside or outside the tank. The different system designs have distinct characteristics regarding e.g. temperature flexibility, technical complexity and attainable COP at various operating conditions.

Task 2 – The report presents recommended test conditions and test procedures for integrated CO_2 heat pumps during operation in *the space heating mode*, *the DHW heating mode* and *the combined heating mode* (simultaneous space heating and DHW heating). Since the thermodynamic losses in the DHW tank may have a significant impact on the COP, integrated CO_2 heat pumps should always be tested together with the DHW tank. It is also important to have specific requirements regarding e.g. the maximum supply temperature in the space heating mode (40ºC for systems with tripartite gas cooler) as well as the inlet tap water temperature (10ºC), the minimum outlet water temperature (55ºC), the hot water flow rate (0.1 dm^3/s) and the draw-off pattern ($4 \times 0.25 \cdot V_n$) in the DHW and combined heating modes.

Task 3 – The last part of the report briefly discusses the method for calculating the Seasonal Performance Factor (SPF) of integrated heat pump systems.

2 DESIGN OF RESIDENTIAL CO_2 HEAT PUMP SYSTEMS FOR COMBINED SPACE HEATING AND HOT WATER HEATING

Task 1 – Includes a state-of-the-art description of residential heat pump systems for combined space and hot water heating – so-called integrated heat pump systems.

2.1 Heating Demands in Buildings

The heating demands in a house are caused by transmission and infiltration losses through the building envelope, ventilation losses when fresh air is supplied to the house by means of a ventilation system, and heating of domestic hot water (DHW). Owing to the implementation of more stringent building codes, the transmission and infiltration losses in new houses have been considerably reduced in recent years. Various standards for *low-energy houses* have also been established in Europe, the USA and Canada. The annual transmission and ventilation losses in these houses are typically 40 to 50% lower than that of new houses which are designed in accordance with prevailing building regulations (Breembroek and Dieleman, 2001).

Owing to the decreasing space heating demand and the fact that more than 70% of the ventilation losses in balanced ventilation systems can be recovered by heat exchange, the annual heating demand for DHW constitutes an increasing share of the total heating demand in new houses. Figure 2.1 shows, as an example, the development of the different heating demands in German single-family houses (Breembroek and Dieleman, 2001). According to Afjei (1997) and Breembroek and Dieleman (2001), the DHW ratio typically ranges from 10 to 15% in existing houses and from 20 to 45% in new houses and low-energy houses. The DHW ratio is defined as the ratio of the annual DHW heating demand [kWh/y] to the total annual heating demand of the house [kWh/y] when heating of ventilation air is excluded.

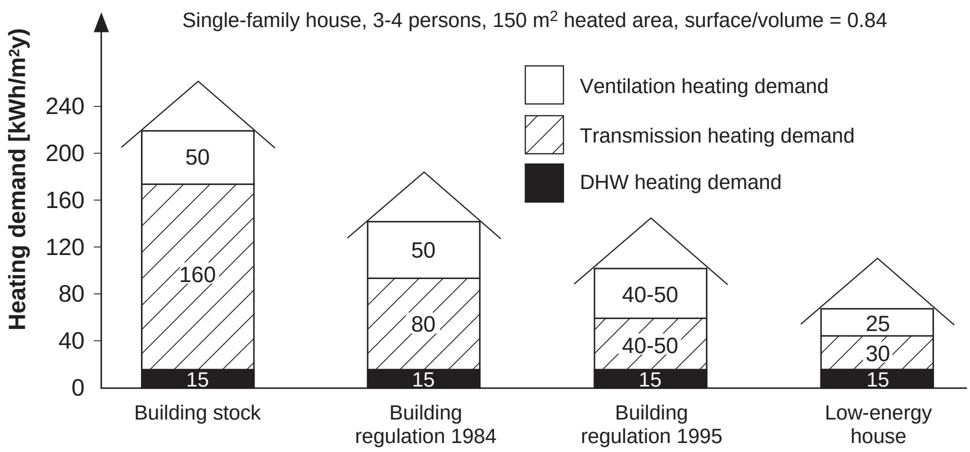

Figure 2.1 Development of the annual heating demand [kWh/(m^2year)] for German single-family houses with 150 m^2 heated area and 3-4 residents (Breembroek and Dieleman, 2001).

2.2 Heating Systems

2.2.1 Heat Distribution Systems

Heat distribution systems for residential heat pumps can be classified as *ductless air systems* (air cooled condenser – space conditioning), *central air systems* (ducted system – space conditioning and ventilation) and *hydronic systems* (space heating and heating of ventilation air).

Hydronic heat distribution system are commonly used in Europe and Canada, whereas central air heating system are dominating in the US (Breembroek and Dieleman, 2001). Hydronic heat distribution systems comprise a closed-loop piping system, circulation pumps, an expansion system as well as terminal units for rejection of heat. The latter include radiators, convectors, fan-coil units as well as floor, wall and ceiling heating systems. Typical distribution temperatures are presented in Table 2.1. A separate ventilation system is required in order to provide adequate indoor air quality in modern air-tight houses.

Table 2.1 Common temperature requirements for different types of terminal units in hydronic heat distribution systems (Breembroek and Dieleman, 2001).

System	**Radiators**	**Convectors**	**Floor Heating**	**Fan-Coils**
Temperature	60 – 80ºC	45 – 55ºC	30 – 45ºC	40 – 50ºC

2.2.2 Hot Water Systems

Domestic hot water (DHW) systems are usually designed as closed unvented systems (4 to 6 bars), where the storage tank is connected to the city water supply (cold mains). In open vented systems, which are used for example in Japan, the storage tank has an open vent to the atmosphere. The DHW is then gravity fed to washbasins, bathtubs etc., and is distributed to showers and whirlpools by means of booster pumps. The DHW storage temperature typically ranges from *55 to 80ºC*.

In conventional residential heat pump systems for combined space heating and hot water heating, the DHW storage tank is either a single-shell or double-shell tank. Reheating and back-up heating are normally provided by one or two electric immersion heaters. Figure 2.2 shows some design examples for unvented single-shell and double-shell DHW storage tanks.

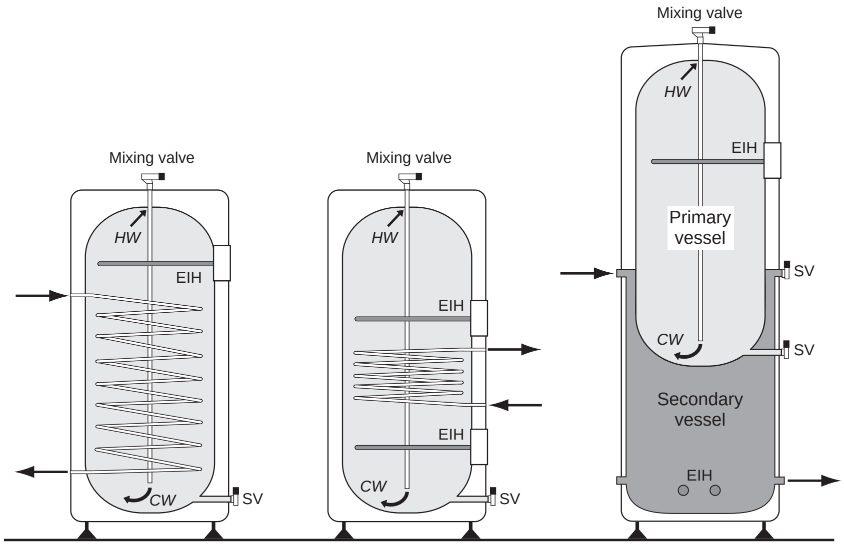

Figure 2.2 Design examples for unvented single-shell and double-shell DHW storage tanks. SV=safety valve, EIH=electric immersion heater, HW=hot water, CW=cold water.

2.2.3 Heat Pumps Using Conventional Working Fluids

Conventional residential heat pumps for combined space heating and hot water heating are charged with R404A, R407C, R410A, R134a or propane. A number of systems have been developed for hot water heating including the utilization of condenser heat, hot discharge gas (desuperheater) or both condenser heat (preheating) and hot discharge gas (reheating). Figure 2.3 shows an example of two design concepts for brine-to-water heat pumps for combined space heating and hot water heating (Stene, 2004).

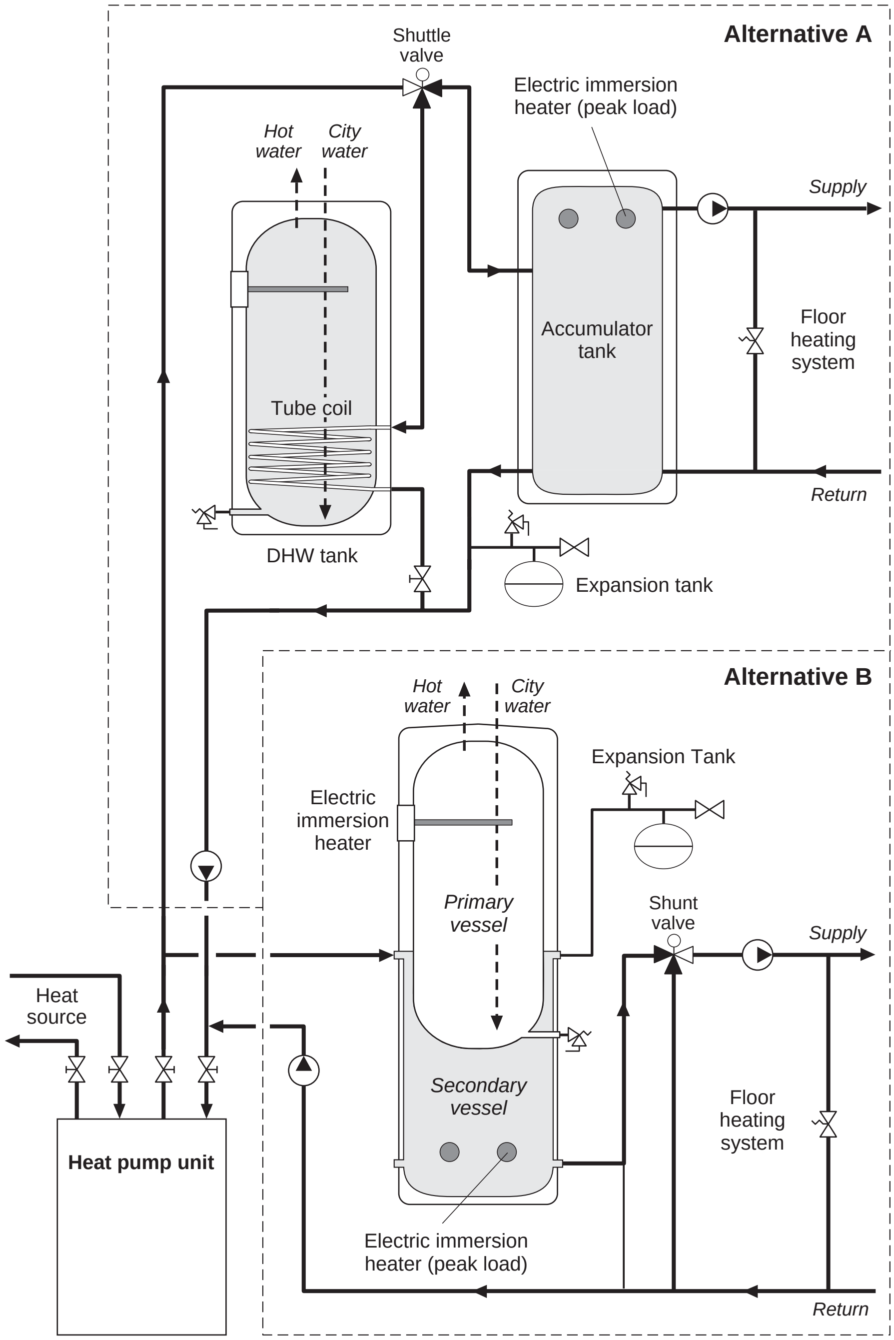

Figure 2.3 *Principle of a residential heat pump system for space heating and hot water heating. A) System with shuttle valve, single-shell DHW tank with tube-coil and separate buffer tank for the space heating system. B) System with double-shell DHW tank.*

In *alternative A*, the shuttle-valve directs the water flow from the heat pump condenser to the DHW tank or the accumulator tank for the space heating system, and heating of DHW is prioritized (alternate operation). Depending on the type of working fluid, the heat pump unit is able to heat the DHW to 45 to 60ºC. In *alternative B*, the DHW is preheated by the water in the heat distribution system. In this system there will be a trade-off between the supply water temperature from the condenser, which determines the COP of the heat pump and the degree of preheating, and the need for supplementary heating in the DHW system.

Figure 2.4 shows an example of a newly developed air-to-water R410A heat pump for combined space heating and hot water heating in residences. By means of a shuttle-valve, heat is either supplied to the space heating system or the DHW system. In the unique high-temperature heat exchanger, the cold water is preheated by condenser heat and reheated by the hot discharge gas. This enables production of high temperature DHW at a relatively low condensation temperature. The heat pump system is equipped with an inverter controlled compressor, and the total heating capacity at +7ºC ambient air temperature is 7 kW.

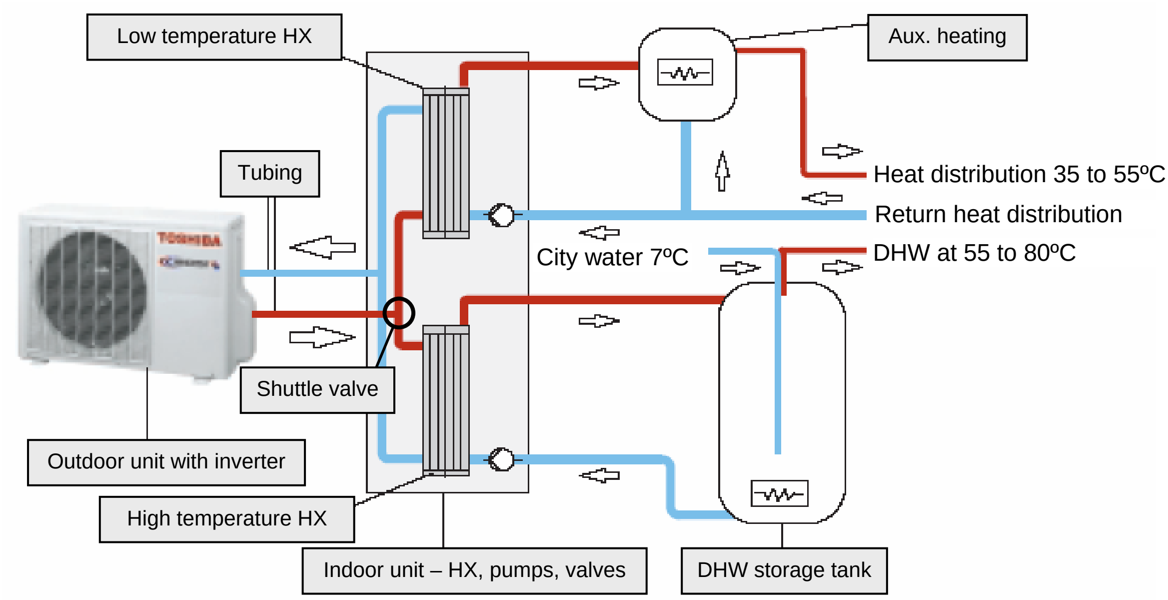

Figure 2.4 Example of an air-to-water R410A heat pump for combined space heating and hot water heating (ABK Klimaprodukter, Norway).

Reference is made to Stene (2004) regarding further information on conventional residential brine/water-water heat pumps for combined space heating and hot water heating.

2.3 Task 1 – Design of Integrated CO_2 Heat Pump Systems

2.3.1 Introduction

There is a great diversity in the design of residential CO_2 heat pump systems for combined space heating and hot water heating – so-called *integrated CO_2 heat pumps*. The main differences between the systems are related to the design and operation of the evaporator (air, water or brine), the CO_2 gas cooler, the hot water system and the space heating system (air cooled or hydronic).

Heat sources of current interest for integrated CO_2 heat pumps include ambient air, ventilation air and ground (rock, soil or lake water – indirect brine system). Residential heat distribution systems are briefly described in Chapter 2.2.1.

2.3.2 Gas Cooler Design – DHW Tank Design

Application of double-shell DHW storage tanks (ref. Chapter 2.2) in CO_2 heat pump systems will lead to a small temperature glide during heat rejection, and with that, a low COP for the heat pump (Stene, 2004). Consequently, only CO_2 heat pump systems using a *single-shell DHW storage tank* will be described in more detail.

The design of CO_2 heat pump systems can be classified as follows:

- **Units with external CO_2 gas cooler** – The gas cooler is connected to the DHW storage tank by means of a closed water loop. A small inverter controlled pump circulates the water from the bottom of the tank through the gas cooler and to top of the tank. The CO_2 heat pump is equipped with:
    - A single gas cooler unit
    - A bipartite gas cooler (two gas cooler units – serial or parallel connection on the CO_2 side)
    - A tripartite gas cooler (three gas cooler units – serial connection on the CO_2 side)

- **Units with integrated CO_2 gas cooler and DHW storage tank** – The gas cooler can be designed as a tube coil, multiport extruded (MPE) tube etc., and mounted inside the tank, at the tank surface or in a thermosyphon unit. The heat exchanger for the space heating system is located:
    - Inside the DHW tank
    - Outside the DHW tank

Figure 2.5 shows examples of E) External CO_2 gas cooler configurations, T) Single-shell DHW storage tanks for external CO_2 gas cooler connection and I) Integral design for CO_2 gas cooler units and single-shell DHW tanks. Gas cooler config. E1/E2/E3/E4 can be connected to DHW storage tank T1, while the external single gas cooler unit in E5 can be connected to DHW and space heating systems T2/T3/T4.

For system E1/E2/E3/E4, the space heating system is separated from the DHW storage tank, which means that a separate accumulator tank may be required for the space heating system. For system E5 and I1/I2/I3/I4/I5, the space heating heat exchanger is an integral part of the DHW storage tank, and the tank works both as a DHW storage tank and as an accumulator tank (buffer) for the space heating system.

The different system designs in Figure 2.5 have distinct characteristics with regard to:

- The possibility of providing high-temperature DHW (60 to 90ºC)
- The possibility of providing high-temperature space heating (>60ºC)
- Temperature flexibility
- The COP in different operating modes and at different operating conditions
- Technical complexity regarding component and system design, control algorithms etc.
- Change in heating capacity and COP when altering the boundary conditions

Table 2.2 provides an overview of important characteristics for the gas cooler configurations and DHW storage tank designs presented in Figure 2.5

Gas cooler configuration A4 (*tripartite gas cooler*) will lead to the highest possible COP for the CO_2 heat pump. This is the result of the counter-flow heat rejection in three separate gas cooler units (good temperature fit – small exergy loss), as well as moderate optimum high-side pressure during simultaneous space heating and DHW heating. The gas cooler is also capable of providing high-temperature hot water (60-90ºC). Due to the serial connection of the gas cooler units, the system is only applicable together with *low-temperature space heating systems* ($t_{w,out}$<35ºC), and both the COP and the heating capacity ratio[1)] are heavily affected by the temperature levels in the space heating system and DHW system (Stene, 2004).

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[1] *The ratio of the DHW heating capacity and the total heating capacity of the tripartite gas cooler.*

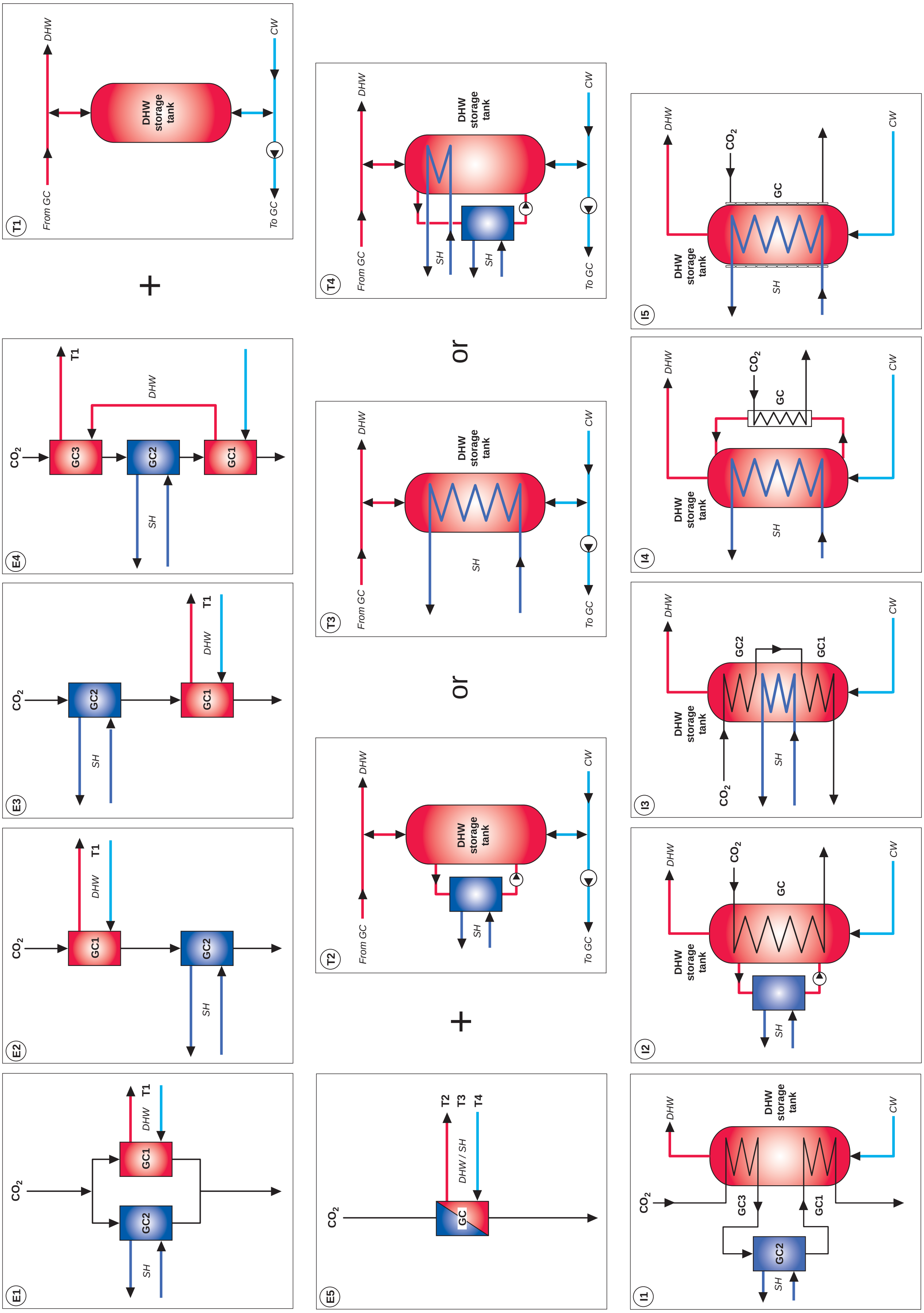

Figure 2.5 *E) Examples of external CO_2 gas cooler configurations, T) Examples of single-shell DHW storage tanks for external CO_2 gas cooler connection, I) Examples of integral design for CO_2 gas cooler units and single-shell DHW storage tanks. SH = space heating, DHW = domestic hot water, CW = city water, GC = gas cooler.*

Table 2.2 Important characteristics for gas cooler configurations and DHW storage tank designs presented in Figure 2.5.

System	E1 T1	E2 T1	E3 T1	E4 T1	E5 T2/T3/T4	I1	I2	I3	I4	I5
High-temperature space heating possible (>60ºC)	Yes	No	Yes	**No**	Yes	Yes	Yes	Yes	Yes	Yes
High-temperature DHW possible (60-90ºC)	Yes	Yes	No	**Yes**	Yes	Yes	Yes	Yes	Yes	Yes
Technical complexity[1)]	B	B	B	**C**	B	C	B	C	B	B
Temperature flexibility[1)]	B	B	B	**A**	C	A	A	A	A	A
Theoretical minimum for the CO_2 outlet temperature[2)]	θ_m	$t_{w,in}$	θ_{wc}	$\boldsymbol{\theta_{wc}}$	θ_{wc}	θ_{wc}	θ_{wc}	θ_{wc}	θ_{wc}	θ_{wc}
COP sensitivity[3)]	B	A	C	**C**	C	C	C	C	C	C

1) A = low, B = moderate, C = high 2) θ_m = CO_2 mixing temperature for GC1 and GC2, $t_{w,in}$ = return temperature for the space heating system, θ_{wc} = city water temperature 3) Relative COP reduction due to low DHW heating demand and/or thermodynamic losses in the DHW tank, A = low, B = moderate, C = high.

2.3.3 Interaction between the DHW Tank and the Gas Cooler Performance

For the transcritical CO_2 cycle, heat is given off at a supercritial pressure and gliding temperature, and the Coefficient of Performance (COP) is defined as the ratio of the specific enthalpy difference in the gas cooler (Δh_{gc}) to the specific compressor work (w).

$$\mathrm{COP} = \frac{\Delta h_{gc}}{w} \tag{2.1}$$

In order to achieve a high COP, *useful heat* has to be rejected over a wide temperature range, and the resulting CO_2 outlet temperature from the gas cooler as well as the high-side pressure must be relatively low (8-9 MPa). Hence, *the COP of an integrated CO_2 heat pump is heavily affected by the return temperature in the space heating system, the city water temperature and the thermodynamic losses in the DHW tank*. The latter, which is caused by mixing of hot and cold water and internal heat conduction in the DHW tank, may lead to a significant increase of the CO_2 outlet temperature from the gas cooler (Stene, 2004).

Figure 2.6 shows the calculated COP of a single-stage transcritical CO_2 heat pump as a function of the CO_2 outlet temperature from the gas cooler and the high-side pressure (Stene, 2004). In the calculations it has been assumed -5ºC evaporation temperature, 5 K suction gas superheat, 60% isentropic compressor efficiency and 10% heat loss from the compressor. These are operating parameters that are typical for residential brine-to-water heat pump units. The temperatures in the brackets are the CO_2 inlet temperatures for the gas cooler.

At CO_2 outlet temperatures below 30ºC, the COP curves are virtually linear, and the COP increases on average by roughly 1% per degree Kelvin drop in the CO_2 outlet temperature and by roughly 1.5 to 3.5% per 0.1 MPa drop in the gas cooler (high-side) pressure.

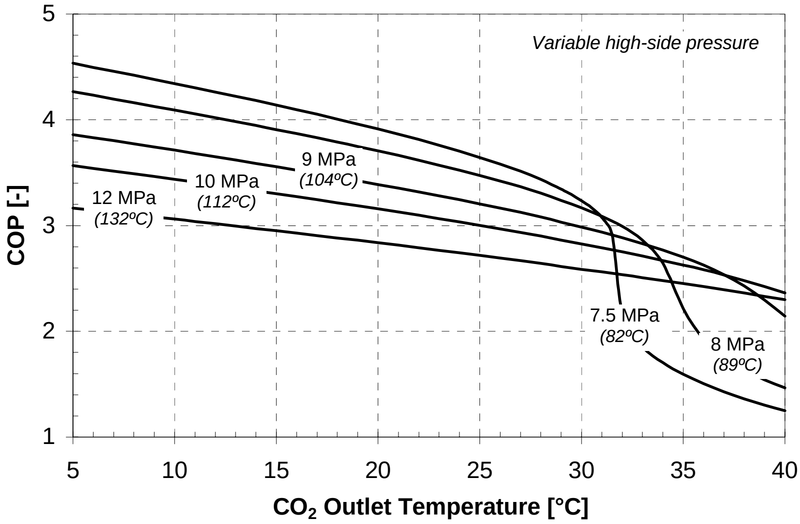

Figure 2.6 *The calculated COP for a single-stage transcritical CO_2 heat pump cycle as a function of the CO_2 outlet temperature from the gas cooler and the high-side pressure. The evaporation temperature is -5ºC (Stene, 2004).*

Figure 2.7 shows, as an example, the simulated relative COP for a residential CO_2 heat pump system using an external tripartite gas cooler (Figure 2.5, configuration E4 + T1), when operating in the *combined heating mode*[2)] at varying inlet water temperature to the DHW preheating gas cooler unit (GC1). The operating conditions were the same as in Figure 2.6, and the COP at 5ºC inlet water temperature was used as reference (i.e. COP=1.0 at 5ºC). The supply/return temperature for the space heating system (floor heating) was 35/30 or 40/35ºC, and the DHW temperature was 60 or 80ºC (Stene, 2004).

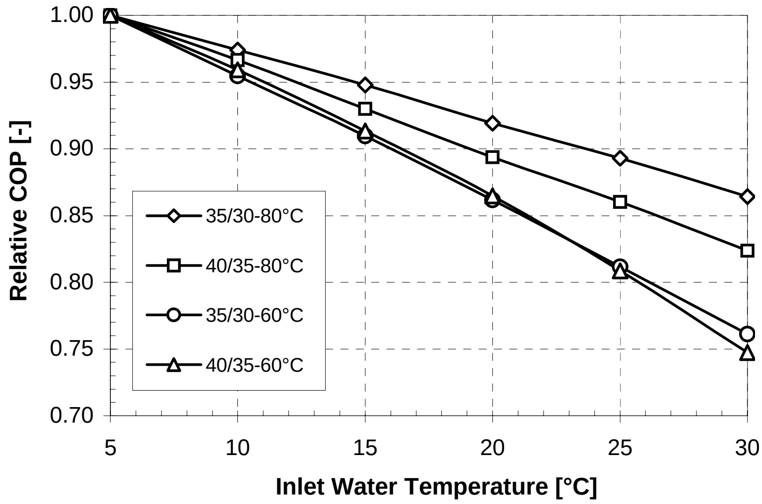

Figure 2.7 *The simulated relative COP in the combined heating mode as a function of the inlet water temperature and varying set-point temperatures for the SH and DHW systems (Stene, 2004).*

Figure 2.7 clearly demonstrates that *the COP is very sensitive to variations in the inlet water temperature*. As an example, by increasing the inlet water temperature from 5 to 10ºC, the COP drops by about 5% at 35/30ºC supply/return temperature and 60ºC hot water temperature, whereas the COP is reduced by 14 and 24% at 20 and 30ºC inlet water temperatures, respectively.

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[2] *Simultaneous space heating and hot water heating.*

Figure 2.8 shows the simulated *relative COP* for the CO_2 heat pump in the DHW mode[3)] at varying inlet water temperature to the DHW preheating gas cooler unit (GC1), and 60 or 80ºC hot water temperature. The COP of the CO_2 heat pump at 5ºC inlet water temperature was used as reference (Stene, 2004).

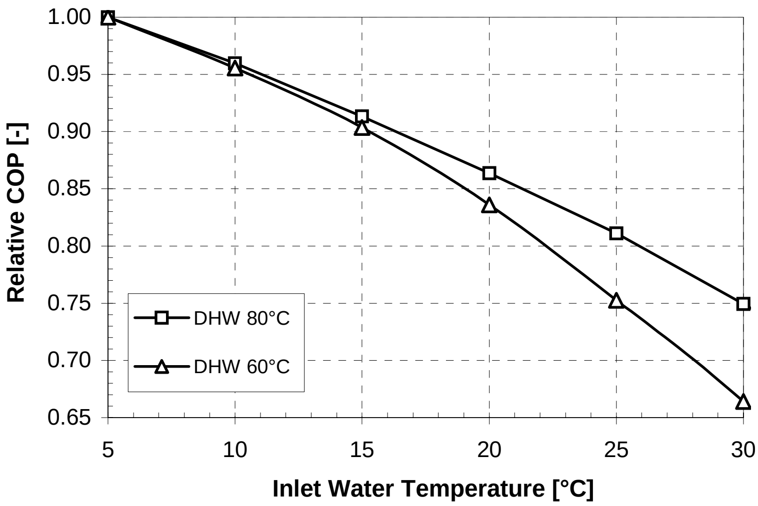

Figure 2.8 The simulated relative COP in the DHW mode as a function of the inlet water temperature and varying set-point temperatures for the DHW system (Stene, 2004).

The COP in the DHW mode was even more sensitive to variations in the inlet water temperature than that of the *combined heating mode*, since the entire heating capacity of the CO_2 heat pump unit was used for hot water production. As an example, by increasing the inlet water temperature from 5ºC to 20 and 30°C, the relative COP was reduced by 17 and 34%, respectively. This was approximately 3 and 10% percentage points higher than that in the *combined heating mode*.

From the discussion in this chapter it can be concluded that:

When measuring the performance of integrated CO_2 heat pumps using external or integral gas cooler designs (ref. Figure 2.5), it is of great importance that the thermodynamic losses in the DHW storage tank are taken into account. In order words – in order to achieve test results that reveal the real performance of integrated CO_2 heat pump systems, the CO_2 heat pumps unit should always be tested together with a DHW storage tank (integral tank or separate tank). The city water temperature (cold mains) also represents an important input parameter in the test matrix.

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[3] *Hot water heating only.*

2.4 Examples of Integrated CO_2 Heat Pump Systems

2.4.1 CO_2 Heat Pump System using an External Single Gas Cooler – EcoCute (Japan)

In April 2002, Denso Corporation Ltd. in Japan was the first company to launch a residential CO_2 heat pump water heater[4)]. The air-to-water heat pump unit had a nominal heating capacity of 4.5 kW, and two hot water tanks connected in series was charged with 85ºC hot water. In August 2002, a 6 kW multifunctional CO_2 heat pump (EcoCute) was launched. The unit provides hot water heating, space heating (floor heating), bathroom heating/drying (fan-coil) and reheating of bath tub water. The temperature level for the hot water in the tank is either 90ºC (winter) or 65ºC (summer). Figure 2.9 shows some photos of the EcoCute unit. The entire unit including the CO_2 heat pump, the DHW storage tank and the heat exchangers are mounted at the outside of the house, and hot sanitary water and hot water for the auxiliary systems are distributed by means of water loops.

Figure 2.9 The Denso CO_2 heat pump system "EcoCute" (Hihara, 2004).

The EcoCute residential CO_2 heat pump system is equipped with a single external gas cooler (Figure 2.5, configuration E5) and a 460 litre DHW storage tank with both internal and external heat exchangers for space heating and reheating of bath tub water (Figure 2.5, T4). Figure 2.10 shows the principle of the EcoCute heat pump system.

When the heat pump is running, cold water from the bottom of the DHW storage tank is circulated by means of a small inverter controlled pump through the single gas cooler unit (A), heated to the desired temperature and returned at the top of the storage tank. Ambient air is used as the heat source (B).

The hot water storage tank is utilized as follows:

[4] *CO_2 technology licenced from Shecco Technology (www.shecco.com), SHECCO™*

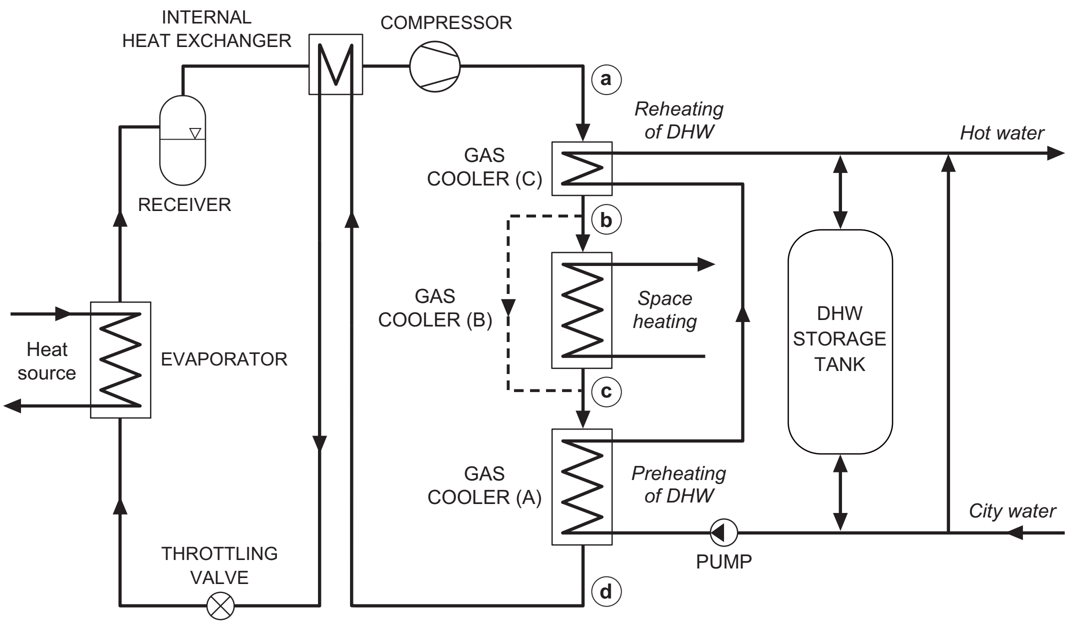

Figure 2.11 The principle of a residential CO_2 heat pump system equipped with an external tripartite gas cooler for preheating of DHW (A), low-temperature space heating (B) and reheating of DHW (C) (Stene, 2004).

A 6.5 kW prototype brine-to-water CO_2 heat pump for combined space heating and hot water heating was extensively tested and analysed (Stene, 2004). Figure 2.12 shows some photos of the installation.

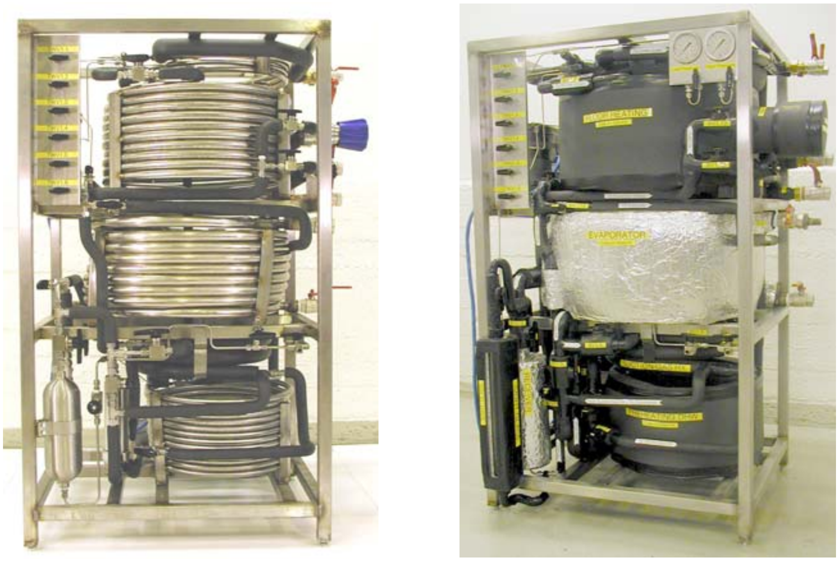

Figure 2.12 The 6.5 kW prototype brine-to-water CO_2 heat pump unit for combined space heating and hot water heating (Stene, 2004).

The CO_2 heat pump was tested for simultaneous space heating and DHW heating (Combined mode), DHW heating only (DHW mode) and space heating only (SH mode). 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 average city water temperature was about 6.5ºC, and most tests were carried out at an evaporation temperature of -5ºC. The measured maximum COP in the Combined mode, DHW mode and SH mode (SH 33/28ºC – DHW 60ºC) was approximately 4.0, 3.9 and 3.2, respectively.

Stene (2004) estimated the SPF for the prototype CO_2 heat pump and a high-efficiency residential brine-to-water R410A heat pump, 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). 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. 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. Figure 2.13 shows the estimated SPF for the three heat pump systems during monovalent operation as a function of the seasonal DHW heating capacity ratio.

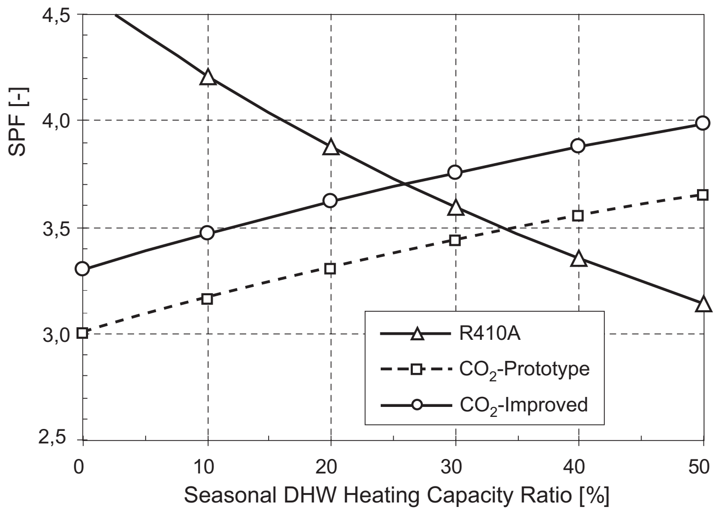

Figure 2.13 The estimated SPF during monovalent operation for the R410A heat pump, the prototype CO_2 heat pump and the improved CO_2 heat pump (Stene, 2004).

An integrated residential brine-to-water CO_2 heat pump system using an external tripartite gas cooler 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 minimized.

Stene (2004) has demonstrated that there can be considerable thermodynamic losses in single-shell DHW tanks due to mixing of hot and cold water and internal heat conduction. The losses will increase the average inlet water temperature to the CO_2 gas cooler during charging of the tank, which in turn will reduce the COP of the CO_2 heat pump. The smaller the charging volume and the lower the heating capacity of the gas cooler, the larger the relative effect of the thermodynamic losses.

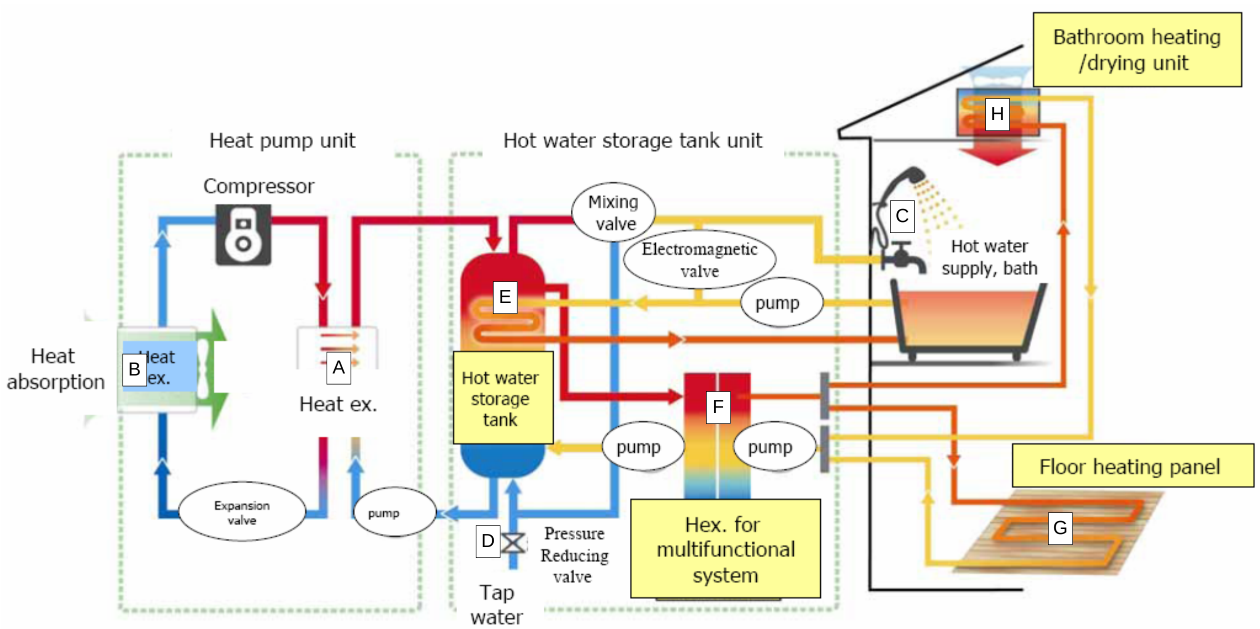

Figure 2.10 Principle layout of the EcoCute air-to-water multifunctional CO_2 heat pump for hot water heating, space heating and bathroom heating/drying (Hihara, 2004).

- Tapping – Hot water at 65 or 90ºC from the top of the storage tank is premixed with cold city water and circulated to bath tubs, showers etc. (C). During the tapping periods, cold city water enters the bottom of the storage tank (D)
- Reheating bath tub – The water in the bath tub is reheated by means of a heat exchanger located in the upper part of the hot water storage tank (E)
- Space heating and drying – Water from the top of the storage tank is circulated to a multifunctional heat exchanger (F) for space heating (G) and bathroom heating/drying (H)

The annual performance of the EcoCute CO_2 heat pump has been simulated and tested for Tokyo climate. The measured/estimated Seasonal Performance Factor (SPF) for the EcoCute was *about 3.0* at a DHW ratio[5)] of approx. 55% and 70ºC hot water storage temperature. Some details regarding the simulation method, the boundary conditions and the simulation results are presented in Appendix A (Hihara, 2004), whereas the test facilities, test methods, and test results are presented in Hihara and Horofumi (2004).

2.4.2 CO_2 Heat Pump System Using an External Tripartite Gas Cooler (Norway)

Stene (2004) carried out an extensive theoretical and experimental analysis of an integrated residential CO_2 heat pump system using an external tripartite gas cooler (Figure 2.5, configuration A4 + B1). Figure 2.11 shows the principle of the CO_2 system. Gas cooler units A (GC1) and C (GC3) 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.

[5] *Ratio of the annual hot water demand [kWh/y] and the total annual heating demand of the residence [kWh/y].*

3 TEST PROCEDURES

Task 2 – Includes the development of adequate test procedures for residential integrated heat pump systems, which deliver the necessary data to calculate the Seasonal Performance Factor (SPF) with a minimal requirement of testing equipment and testing time.

3.1 Introduction

Integrated CO_2 heat pumps should be tested in three different operating modes:

- Space heating only (*space heating mode*) – Occurs when the domestic hot water (DHW) storage tank is fully charged, and there is no current DHW demand in the house (no draw-off).
- Hot water heating only (*DHW heating mode*) – Occurs when there is a need for charging the DHW storage tank, but there is no space heating demand in the house (e.g. during the summer).
- Space heating and hot water heating (*combined heating mode*) – Occurs when there is a simultaneous demand for space heating and charging of the DHW tank.

In order to ease the standardisation work for testing of integrated CO_2 heat pumps, existing CEN standards should be used as far as possible. As a consequence, steady-state testing in the *space heating mode* should mainly be performed according to EN 14511:2004, and testing in the *DHW heating mode* should mainly be performed according to EN 255-3:1997 – with only minor modifications of the test methods/conditions.

As discussed in Chapter 2.3.3, *integrated CO_2 heat pumps using an external gas cooler should always be tested together with the DHW storage tank*. Consequently, the boundary conditions for systems with external or integral gas cooler designs will be as illustrated in Figure 3.1 (red circles). In the figure, the nomenclature for brine/water-to-water heat pumps has been used as an example.

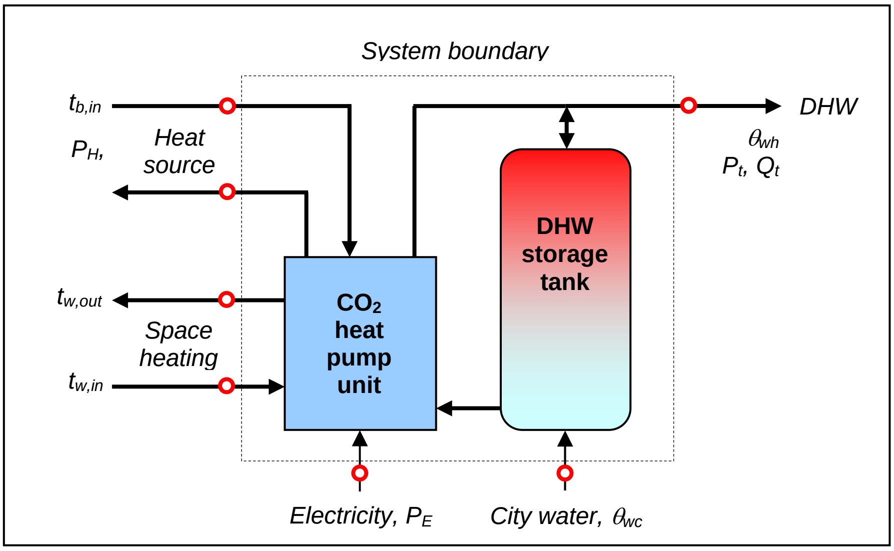

Figure 3.1 Illustration of the boundary conditions and variables during testing of residential brine/-water-to-water CO_2 heat pumps for combined space heating and hot water heating.

The following discussion of test conditions and methods is mainly focusing on the heat rejection process for space heating and DHW heating.

3.2 Proposed Test Conditions and Test Methods

The main purpose of the test results for the different heating modes is to enable a fair comparison of different integrated heat pump systems using HFCs (R404A, R407C, R410A, R134a), CO_2 and hydrocarbons as working fluids. The test results should also be applicable for calculation of the seasonal performance factor (SPF) for different integrated heat pump systems (Task 3, ref. Chapter 4) as well as when comparing the primary energy use for heat pump systems and conventional heating system.

3.2.1 Space Heating Mode (ref. EN 14511-2:2004 and EN 14511-3:2004)

On 30th April 2004, the former standard EN 255-2:1997 for the testing of heat pumps for single *space heating mode* was replaced by EN 14511:2004 (1-4).

- **Outdoor heat exchanger**
    - Temperatures –The brine/water/air temperatures for the outdoor heat exchanger at standard rating conditions should comply with EN 14511-2.
    - Flow rates – As proposed by the Swedish National Team (Haglund Stignor et al., 2004), the same brine and water flow rates should be used during standard rating and application rating conditions for the outdoor heat exchanger.
- **Indoor heat exchanger**
    - Temperatures – The water temperatures for the indoor heat exchanger should comply with EN 14511-2 with the following exception: CO_2 heat pumps using a *tripartite gas cooler* (Figure 2.5, E4+T1) are unable to supply heat to high-temperature heat distribution systems (Stene, 2004). Consequently, when testing CO_2 heat pumps with a tripartite gas cooler, the inlet and outlet temperature set-points for the space heating system should be changed from 35/30, 45/40 and 55/50ºC to *35/30 and 40/35ºC*, respectively.
    - Flow rates – As proposed by the Swedish National Team (Haglund Stignor et al., 2004), the same water flow rate should be used during standard rating conditions and application rating conditions for the indoor heat exchanger. In other words, the same flow rate should be used for floor heating and radiator heating applications.
- **Extra pumps**
    - Power input – For CO_2 heat pumps using pumps for internal circulation of water (Figure 2.5, E5+T2/T3/T4), the total power input to the pumps should be included in the effective power input to the heat pump system.
- **Integral DHW tanks**
    - Length of test period – For CO_2 heat pump systems using an integral DHW storage tank (Figure 2.5, E5+T2/T3/T4 and I1/I2/I3/I4/I5), the duration of the test period for steady-state tests should be 24 hours, consisting of a 12 hour stabilization period followed by a 12 hour measurement period. This test cycle has been proposed by the Swedish National Team for exhaust air heat pumps with a built-in storage (Haglund Stignor et al., 2004).
- **Heating Capacity – Coefficient of performance (COP)**
    - Calculation of COP – According to EN 14511-3, the COP for the CO_2 heat pump system during operation at the different test conditions in the *space heating mode* is calculated as:

$$COP_H = \frac{P_H}{P_{in}} = \left(\frac{q \cdot \rho \cdot c_p \cdot \Delta t}{P_{in}} \right) \tag{3.1}$$

where:

- ***q*** Water flow rate for the indoor heat exchanger [m^3/s]
- ***ρ*** Density of water at specified temperature [kg/m^3]
- ***c_p*** Specific heat capacity of water at specified temperature [J/(kgK)]
- ***Δt*** Difference between the outlet and inlet temperature [K]
- ***P_{in}*** Effective power input to the heat pump system including correction due to energy consumption of liquid pumps and fans [W]

  - Calculation of SPF – P_H [W], which is the heating capacity of the heat pump, COP and the corresponding temperatures for the outdoor and indoor heat exchangers, are crucial input parameters when calculating the seasonal performance factor (SPF) for the heat pump during one year of operation (ref. Task 3, ref. Chapter 4).

3.2.2 DHW Heating Mode (ref. EN 255-3:1997)

The European standard for testing of heat pumps in the DHW mode, EN 255-3, will soon be revised in connection with Mandate M/324 (M/324).

- **Outdoor heat exchanger**
  - Flow rates – As proposed by the Swedish National Team (Haglund Stignor et al., 2004), the brine and water flow rates for the outdoor heat exchanger should be the same as during testing in the *space heating mode* (ref. EN 14511).
  - Temperatures – Testing should be carried out at one heat source temperature as stated in EN 255-3. The COP and heating capacity at other heat source temperatures can be determined by using the measured heat pump characteristics from the *space heating mode*.
- **DHW storage tank**
  - Hot water temperature – For systems with external gas cooler (Figure 2.5, E1/E2/E3/-E4/E5), the outlet hot water temperature from the CO_2 gas cooler during charging of the DHW storage tank should be *minimum 55ºC*. Since the mixing and heat conduction losses in the DHW storage tank drop when the temperature is lowered, this requirement will ensure that the temperature level is not kept at an artificially low level just to increase the COP of the CO_2 heat pump. *55ºC is also the minimum temperature to prevent legionella*.
  - Inlet city water temperature – The inlet city water temperature during testing should be reduced from 15ºC (EN 255-3) to *10ºC* as defined in Mandate M/324. The water temperature should be constant during the entire test period.
  - Hot water flow rate – As proposed by the Swedish National Team (Haglund Stignor et al, 2004), the hot water flow rate in systems with V_n<600 litres should be *reduced from 0.2 to 0.1 dm^3/s* (testing without mixing valve – ref. next bullet Mixing valve). Typical water flow rates from the DHW tank (60ºC) is ~0.05-0.1 dm^3/s for a shower, 0.05 dm^3/s for a washbasin, 0.15 dm^3/s for a bath tub and 0.1 dm^3/s for washing-up (Novakovic et al., 1996). Hence, a reduced draw-off flow rate of 0.1 dm^3/s reflects the real user situation better than 0.2 dm^3/s.

- Mixing valve – The heat pumps should be *tested without a mixing valve* for the DHW storage tank. EN 255-3 states that “if the water heater is equipped with a mixing valve for the hot water, this valve shall be set at the manufacturers recommended setting throughout the test”. For systems that are not equipped with a mixing valve, the draw-off from the DHW storage tank will be considerably larger than that of systems where hot water from the tank is mixed with cold city water. As an example – at 60ºC hot water temperature in the tank, 10ºC city water temperature and 40ºC water temperature at the tapping site, the ratio of the drawn-off volume from the tank and the tapped volume is 0.6. Since the volume of the hot water at the tapping site should be equivalent to $0.25 \cdot V_n$, the volume of tapped hot water from the tank will be different for systems with and without a mixing valve. This will in turn have a considerable impact on the measured COP of the CO_2 heat pump systems (Stene, 2004). Consequently, in order to obtain equal test conditions for all kinds of CO_2 heat pump systems, testing should be carried out without a mixing valve.
- Supplementary heating – All supplementary heating systems for reheating of the DHW (e.g. electric immersion heaters) should be switched off during testing, not only manually operated systems as stated in EN 255-3.
- Extra pumps – For CO_2 heat pumps using an external gas cooler (Figure 2.5, E1/E2/E3/E4/E5), the power input to the circulation pump for the hot water circuit should be included in the effective power input to the system. According to EN 255-3, the power input should be fully included in the effective power absorbed by the heat pump.

- **Accomplishment of the testing**
  - Heating up period – Should comply with EN 255-3.
  - No. of draw-offs – As proposed by the Swedish National Team (Haglund Stignor et al., 2004), the test cycle should include *4 subsequent draw-offs* where each draw-off constitutes *25%* of the nominal volume of the DHW storage tank (i.e. $0.25 \cdot V_n$). The draw-off cycle will therefore comprise an initial heating up period followed by 4 subsequent draw-off/heating up periods. The total draw-off volume during testing will be the same as in EN 255-3 ($V_{tot}=V_n$). The COP of CO_2 heat pump systems is much more sensitive to the draw-off profiles than heat pumps with conventional working fluids. This is because the inlet water temperature to the gas cooler is determined by the draw-off volume and the degree of mixing and conductive heat transfer between hot and cold water in the DHW tank (Stene, 2004). The draw-off-profiles defined in Mandate M/324 include a large number of small draw-offs, 11 (36 litres), 23 (100 litres) and 24 (200 litres), and the Swedish proposal represents a good compromise between the draw-off patterns of EN 255-3 and M/324.
  - Coefficient of performance – As proposed by the Swedish National Team (Haglund Stignor et al., 2004), only the last (fourth) draw-off and the following heating up period should be used for determination of the COP, provided that the energy content of this draw-off does not differ more than *5%* (10% in EN255:3) from the preceding draw-off. Otherwise, further draw-off cycles should be carried out until this criterion is fulfilled. According to EN 255-3, the COP for the CO_2 heat pump system for tapping hot water (*DHW heating mode*) is calculated as:

$$COP_t = \frac{\int_0^{t_t} q_{wh} \cdot \rho_{wh} \cdot c_{pw} \cdot (\theta_{wh} - \theta_{wc})\, dt}{W_{et} - P_{es} \cdot t_s} \tag{3.2}$$

where:

q_{wh} DHW flow rate [m^3/s]

ρ_{wh} Density of water at specified temperature [kg/m^3]

c_{pw} Specific heat capacity of water at specified temperature [J/(kgK)]

θ_{wh} Temperature of the outgoing hot water [ºC]

θ_{wc} Temperature of the incoming cold water [ºC]

W_{et} Total energy input to the heat pump during the test period (including correction due to energy consumption of liquid pumps and fans, ref. EN 255-3) [Wh]

P_{es} Standby power input [W]

t_t Duration of the heating up period [s]

t_s Duration of the standby period [s]

- Heating capacity – The average heating capacity [W] of the heat pump during the last draw-off and subsequent heating period is calculated as:

$$\overline{P}_t = \frac{\int_0^{t_t} q_{wh} \cdot \rho_{wh} \cdot c_{pw} \cdot \left(\theta_{wh} - \theta_{wc}\right) dt}{t_t} \tag{3.3}$$

where:

t_t Duration for the final heating up period [s]

- Other parameters – Determination of the other parameters in the test, i.e. the reference hot water temperature (*θ_{wr}*), the standby power input (*P_{es}*) and the maximum quantity of usable hot water (*Q_{max}*), shall comply with EN 255-3.

3.2.3 Combined Heating Mode

During the year, there will be periods of varying duration where there will be a simultaneous demand for space heating and charging of the DHW storage tank. All the integrated CO_2 heat pump systems/configurations presented in Figure 2.5 are able to provide simultaneous space heating and hot water heating (*combined heating mode*).

As pointed out in Chapter 2.3.2, the CO_2 heat pump systems can be divided into two main groups:

- The space heating system is separated from the DHW storage tank. A separate accumulator tank may be required for the space heating system – System E1/E2/E3/E4 (ref. Figure 2.5)
- The space heating heat exchanger is an integral part of the DHW storage tank, and the tank works both as a DHW storage tank and as a accumulator tank (buffer) for the space heating system – System E5 and I1/I2/I3/I4/I5 (ref. Figure 2.5).

If possible, the test conditions and test methods should be the same for all system configurations.

During operation in the *combined heating mode*, it is not possible to split up the power consumption for space heating and DHW heating, which means that the COP cannot be calculated according to EN 14511 and EN 255-3. Consequently, an *average COP* should be calculated on the basis of the total quantity of heat delivered from the heat pump and the effective energy input during the test period.

The following proposal for test conditions and test procedure for CO_2 heat pump systems in the *combined heating mode* is partly based on a proposal from the Swedish National Team regarding testing of exhaust air heat pumps with a built-in storage (Haglund Stignor et al, 2004):

- **PROPOSED TEST/BOUNDARY CONDITIONS**
  - Outdoor heat exchanger – The test conditions for the outdoor heat (heat source) exchanger should be as in EN 255-3 (*i.e. 0ºC for brine, 10ºC for water, +7ºC for ambient air, 20ºC for exhaust air*). The brine/water flow rate for the outdoor heat exchanger should be the same as during testing in the *space heating mode* (EN 14511).
  - Indoor heat exchanger – The combined test should be performed at two different inlet and outlet water temperatures for the indoor (space heating) heat exchanger, *35/30ºC* and *40/45ºC* The highest temperature level is not feasible for systems with tripartite gas cooler (ref. Table 2.2). The inlet/outlet temperatures should be constant during the entire test period. Since the heating capacity of the space heating heat exchanger may vary considerably during the test period, this requirement can be met by using an inverter controlled pump in the water circuit.
  - Inlet city water temperature – The inlet water temperature should be *10ºC* during the entire test period as defined in Mandate M/324.
  - Hot water temperature – For systems with external gas cooler, the outlet hot water temperature from the CO_2 gas cooler during charging of the DHW storage tank should be *min. 55ºC.*
  - Hot water flow rate – The hot water flow rate during all draw-off periods should be *0.1* dm^3/s.
  - Supplementary heating – All supplementary heating for the DHW and space heating system should be *switched off* during testing.
  - Mixing valve – The heat pumps should be tested *without a mixing valve* for the DHW system.
  - Extra pumps – For CO_2 heat pumps using pumps for internal circulation of water, the *total power input* to the pumps should be included in the effective power input to the system.

- **PROPOSED TEST PROCEDURE**
  - DHW tank, heating up period – The procedure for the heating up period for the water in DHW storage tank should comply with EN 255-3. The heat pump should not reject heat to the space heating system during the heating up period.
  - DHW tank, stabilisation period – When the DHW storage tank is fully charged and the thermostat shuts off the heat pump, a volume of hot water is tapped equivalent to $0.25 \cdot V_n$. The heat pump is switched on, and heat is supplied to both the DHW system and the space heating system at the specified temperature conditions for the outdoor heat exchanger, indoor (space heating) heat exchanger and DHW tank (see Proposed Test/Boundary Conditions). The heat pump will be running in Combined mode until the thermostat in the DHW storage tank reaches the set-point temperature. The procedure with draw-offs and DHW/space heating by the CO_2 heat pump is repeated until the difference in energy content between two successive draw-offs from the DHW storage tank is equal to or less than 5%.
  - Test period – A volume of hot water is tapped equivalent to $0.25 \cdot V_n$ (draw-off no. 1). The CO_2 heat pump is switched on, and heat is supplied to both the DHW system and the space heating system. When the thermostat in the DHW storage tank reaches the set-point temperature,

draw-off no. 2 is carried out, followed by a new heating period. This procedure continues until 4 draw-offs and 4 subsequent heating periods have been carried out. The heat pump should supply heat to both the space heating and the DHW systems during the entire test period.

- **CALCULATION OF COP**
  - The coefficient of performance (COP) for the CO_2 heat pump system during operation in the *combined heating mode* is calculated as:

$$COP_{COMB} = \sum_{i=1}^{i=4}\left(\frac{Q_H + Q_t}{W_{tot}}\right) = \sum_{i=1}^{i=4}\left(\frac{\int_0^{t_h}\left[q_w \cdot \rho_w \cdot c_{pw}\left(t_{w,out} - t_{w,in}\right) dt\right]_H + \int_0^{t_h}\left[q_w \cdot \rho_w \cdot c_{pw} \cdot \left(\theta_{wh} - \theta_{wc}\right) dt\right]_t}{W_{tot}}\right) \quad (3.4)$$

where:

Q_H	Quantify of heat delivered to the space heating system during the test period [Wh]
Q_t	Quantity of heat delivered to the hot water system during the test period [Wh]
W_{tot}	Total energy input to the heat pump during the test period (including correction due to energy consumption of liquid pumps and fans, ref. EN 255-3) [Wh]
q_w	Water flow rate for the space heating (H) and DHW (t) systems [m^3/s]
ρ_w	Density of water at specified temperature [kg/m^3]
c_{pw}	Specific heat capacity of water at specified temperature [J/(kgK)]
$t_{w,out}$	Outlet water temperature for the space heating heat exchanger [ºC]
$t_{w,in}$	Inlet water temperature for the space heating heat exchanger [ºC]
θ_{wh}	Temperature of the outgoing hot water [ºC]
θ_{wc}	Temperature of the incoming cold water [ºC]
t_h	Duration of the test period
i	Refers to 4 draw-off periods and 4 reheating periods for the DHW system (i=1-4) [s]

- **HEATING CAPACITY**
  - The average heating capacity [W] of the integrated heat pump for space heating (*H*) and hot water heating (*t*) during the test period is calculated as:

$$\overline{P}_H = \frac{\int_0^{t_h} q_w \cdot \rho_w \cdot c_p \cdot \left(t_{w-out} - t_{w-in}\right) dt}{t_h} \quad (3.5)$$

$$\overline{P}_t = \frac{\int_0^{t_h} q_w \cdot \rho_w \cdot c_p \cdot \left(\theta_{wh} - \theta_{wc}\right) dt}{t_h} \quad (3.6)$$

where:

t_h	Duration of the test period, i.e. 4 draw-off periods and 4 reheating periods [s]

4 CALCULATION OF THE SEASONAL PERFORMANCE FACTOR

Task 3 of IEA HPP Annex 28 includes the development of simple methods for calculation of the Seasonal Performance Factor (SPF) for integrated heat pump systems.

4.1 Introduction

The seasonal performance factor (SPF) is a key parameter when evaluating the energy efficiency of residential heat pump systems for combined space heating and hot water heating. The SPF is defined as the ratio of the total annual heat supply from the heat pump system, Q_{tot} [kWh/y] to the total annual energy supplied to operate the system, W_{tot} [kWh/y].

$$SPF = \frac{Q_{tot}}{W_{tot}} = \left(\frac{Q_{H-tot} + Q_{t-tot}}{W_{tot}} \right) \quad (4.1)$$

The SPF is influenced by a large number of factors including:

- **Heating demands** – climatic conditions, standard of the building envelope, user behaviour etc.
  - Space heating – maximum heat load, variations over the day/year
  - DHW heating – tapping profiles
- **System design**
  - Monovalent or bivalent heating system
  - Heat source – type (water, ambient air, exhaust air, ground), direct or indirect system
  - Heat distribution system(s) – water (hydronic) and/or air (ductless, ducted)
  - Overall system design
- **Heat pump unit – test results (Task 2)**
  - Heating capacity and COP at different heating modes and temperature conditions
    - Space heating mode
    - DHW heating mode – incl. stand-by losses
    - Combined heating mode (alternate or simultaneous operation)
  - Maximum supply temperature for DHW heating (need for reheating)
- **Secondary systems**
  - Heat source system – type (water, brine, air), temperature variations
  - Heat distribution system – type (water, air), temperature variations
  - Ventilation system – type (exhaust, balanced), temperature variations
  - DHW system – set-point temperature, heat loss rate
- **Auxiliary systems**
  - Peak load system, space heating – type, design and efficiency
  - Peak load system, DHW heating – type, design and efficiency
  - Pumps, fans – energy consumption
- **Control system**
  - Control algorithms (incl. stand-by losses and cycling losses)

4.2 Calculation Method

The objectives of the calculation method to be developed within the framework of IEA HPP Annex 28 is to be transparent, as far as possible easy-to-use, applicable without extensive computation and capable of delivering a seasonal performance factor (SPF) for integrated heat pump systems for combined space heating and domestic hot water (DHW) heating as a basis for comparison of heat pump systems.

The participants of the IEA HPP Annex 28 have agreed on to use the *bin method* for calculating the SPF (Wemhöner and Afjei, 2005). The bin methodology is based on the cumulative annual frequency of the ambient temperature, which has a major impact on the space heating demand of a residence – i.e. the space heating demand is more or less proportional to the difference between the indoor and ambient temperatures. Figure 4.1 demonstrates the principle of the bin methodology (Wemhöner and Afjei, 2005).

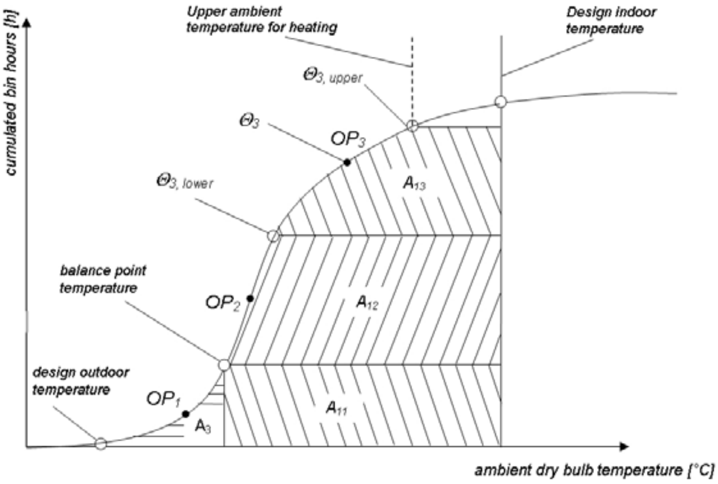

Figure 4.1 Principle of the bin methodology on the basis of cumulative annual frequency of the ambient dry bulb temperature at the site (Wemhöner and Afjei, 2005).

The cumulative annual frequency of the ambient dry bulb temperature is divided into temperature intervals called bins. The method assumes that the operating conditions including the heat source and heat sink temperatures are valid for the entire bin, i.e. the operating conditions characterises the mean operating conditions of the bin and therefore are set in the centre of the bin. The power/energy demand for heating of domestic hot water (DHW) is assumed to be constant during the year.

With reference to Chapter 3.2, *Proposed Test Conditions and Test Methods*, the heating capacity and COP of the integrated heat pump at different operating conditions in the:

- Space heating mode – COP_H, P_H
- DHW heating mode – COP_t, P_t
- Combined heating mode (simultaneous operation) – COP_{COMB}, P_H, P_t

are combined with the information of the space heating and DHW heating demand of each bin in order to calculate the seasonal performance factor (SPF) of the heat pump. The number of hours for the Combined mode is determined by the ratio $X=Q_H/Q_t$ for the bin. For bivalent heat pump systems, a weighting with energy fractions for the heat pump unit and the peak load system is performed in order to calculate the seasonal performance factor of the total heating system (SPF_{sys}).

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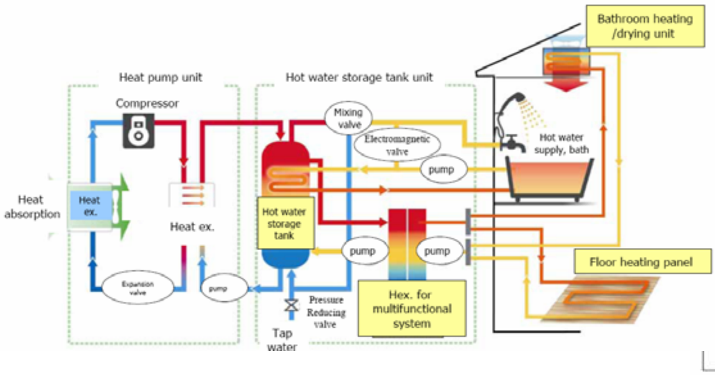
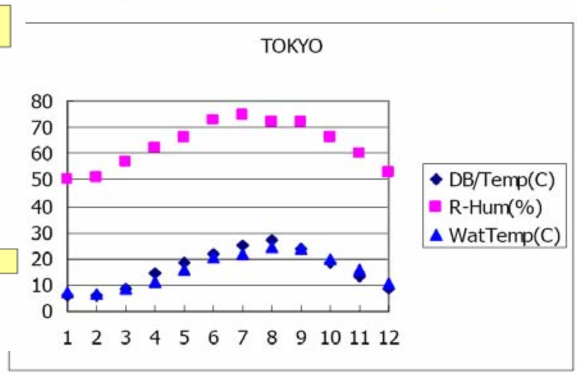
6 APPENDIX A – Simulation and Annual Performance of CO_2 Heat Pump with Floor Heating and Hot Water Supply (Hihara, 2004)

Air-to-water heat pump unit with combined floor heating and hot water heating

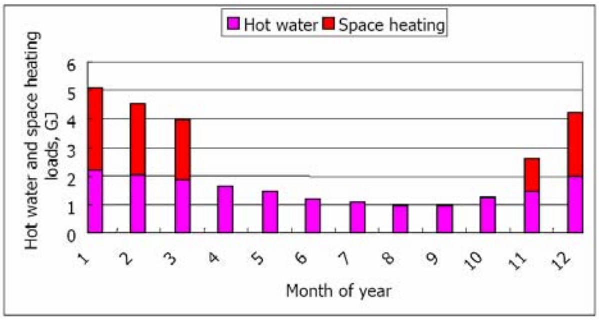


Monthly-averaged ambient parameters in Tokyo



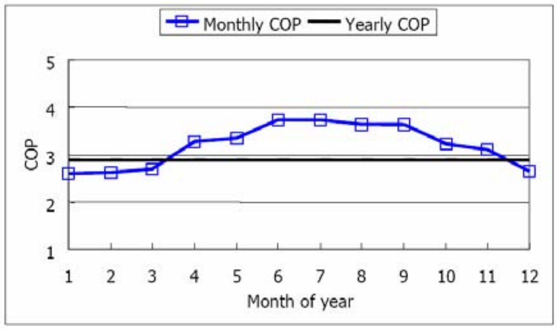
Simulation method

- Ambient parameters: Tokyo city (JRA 4050:2001)
- Simulated house: Typical (averaged) Japanese dwelling with 4 residents
- Performance of carbon dioxide heat pump unit
  (1) COP versus inlet water temperature at given outlet temperatures (65/90C) and certain ambient DB/WB temperatures (Correlated from measured data from a typical Japanese product)
  (2) COP versus ambient temperatures (Theoretical simulation with product sizes and operation conditions)
- Temperature of hot water in the tank: 65/C intermediate and summer seasons
  90/C winter and intermediate seasons
- Heat loss coefficient through tank surface by convection and radiation: 1.5W/m²/K (generally, heat loss is less than 10% within 13 hours)
- The temperatures of supplied hot water from tank for domestic hot water, space heating and bathroom drying are almost constant except the temperature decrease due to the surface heat loss.
- The proposed tank volume for simulation is 460 liters.
- The produced and stored hot water in the tank during night time (0:00-6:00) is assumed to match the loads during the day time (6:00am-23:00pm)
- The proposed house with heat conductance 3.4W/m²/K, floor heating area 13.2m², room temp. 18/C.

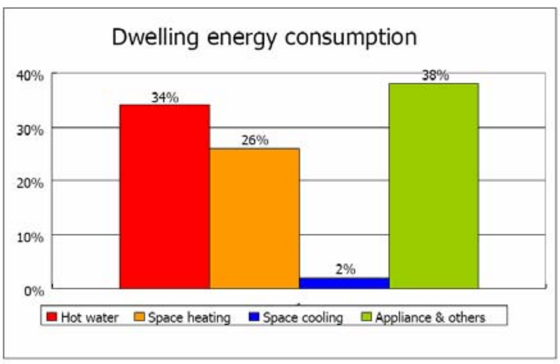


Monthly COP and annual COP

$$Annual \rightarrow COP = \sum_{i=1}^{i=12} Q_i / \sum_{i=1}^{i=12} P_i$$



Dwelling energy consumption in Japan



Conclusion

- Our simulation shows that the seasonal COP of carbon dioxide heat pump with combined water heating and floor heating is about 3.0.
- Further performance measurements are needed to obtain the assessment method of annual performance.
- We will obtain some performance data from a test-rig machine by September, and will propose testing and calculation methods by November.

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