

SEASONAL PERFORMANCE AND TEST OF MULTI-FUNCTION HEAT PUMP UNITS

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Abstract: The energy need for space heating is decreasing significantly in modern highly-insulated dwellings. Thus, in the recent years ventilation compact units with heat pump have been developed for low energy houses, which combine the functions space heating (SH), domestic hot water (DHW) and ventilation in one unit. To assess the overall performance, the seasonal performance factor (SPF) has to be calculated by a standardised method. In the draft standard (prEN 15316-4.2) a temperature class based calculation method (bin method) for heat pump systems with SH and DHW operation has been introduced to calculate the SPF and respectively the electricity input. This standard is to support the implementation of the Energy Performance of Buildings Directive (EPBD 2003). The method has now been extended to multi-function units with ventilation, heat recovery and optional solar collector and ground-to-air heat exchanger. The results of the calculation method have been validated with measurements of two pilot plants. Furthermore, an exergy analysis for heat pump compact units provided a method to compare system layouts by showing the influence of the different loss mechanisms on a uniform basis.

Key Words: *heat pump, standardisation, test procedure, seasonal performance*

1 INTRODUCTION

Multifunctional system configurations with heat pumps, as shown in **Figure 1**, have been introduced in the markets for low energy buildings to cover the building energy needs for space heating (SH), domestic hot water (DHW) and ventilation with one unit. A system assessment of these compact units with multifunctional system layout requires standard test procedures and calculation methods, which are presently incomplete or missing.

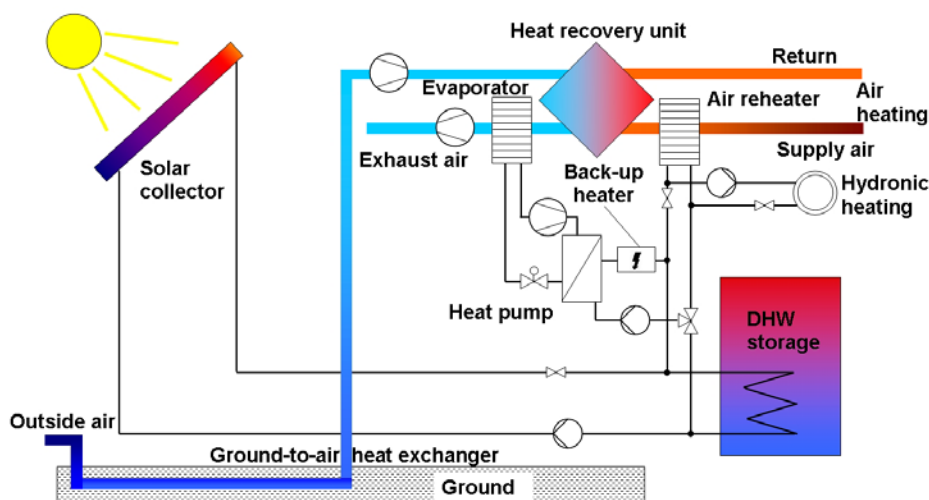


Figure 1: Principle system layout of compact units ((Afjei et al. 2007), source manufacturer)

The assessment is on the one hand needed for the holistic energy rating in the frame of the building energy certificate as currently introduced in the EU based on the Directive on the Energy Performance of Buildings (EPBD 2003). On the other hand, transparent labelling concerning energy consumption/costs and environmental impact is needed to convince consumers to apply environmentally sound technologies. Thus the main focus of the research work was to elaborate recommendations to standardisation organisations concerning

- comprehensive test procedures for combined operating heat pumps for SH and DHW with minimum testing requirements,
- subsequent transparent and easy-to-use calculation methods for the SPF of combined operating heat pumps.

Since calculation method and test procedure are meant to be introduced in standards, the testing should be uniformly applicable to the variety of marketable systems. The calculation, on the other hand, should be a “hand” calculation without extensive computational effort (e.g. simulations), taking account of the prevailing physical impacts on the seasonal performance. Thereby two types of marketable combined operating heat pump systems have to be distinguished:

- alternate combined operation, where the heat pump is switched between SH- and DHW-operation,
- simultaneous systems, where SH and DHW is produced at the same time.

Since test procedures for the SH-only, DHW-only and ventilation system already are in use, existing standards for the single operation modes have been extended to cover highly-integrated system configurations, as shown in **Figure 1**. The system boundary for the testing and calculation has been defined to include the heat pump, attached SH and DHW storages and eventually installed back-up heaters in order to cover the most common system configurations. **Figure 2** shows the system boundary in the EPBD calculation scheme.

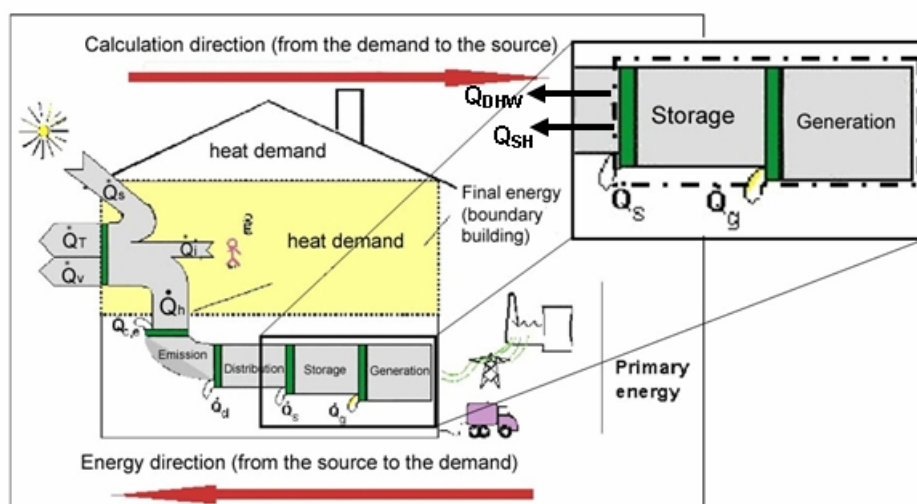


Figure 2: Calculation scheme of the EPBD (EPBD 2003)

The project was embedded in Annex 28 (Wemhöner and Afjei 2006) in the heat pump program (HPP) of the international energy agency (IEA) and comprises

- testing of the systems according to existing and extended standards,
- calculation of the system performance,
- field monitoring for the validation of the calculation results to gain experience with the functionality and for system optimisation and
- exergy analysis for heat pump compact units.

2 RESULTS

2.1 Test procedure

The testing of ventilation compact units with heat pump comprises

- Assessment of air-tightness (testing for internal and external air leakage)
- Testing ventilation efficiency
- Thermodynamic tests
- Acoustic tests
- Filter bypass leakage (optional)
- Operation/Maintenance/Safety (optional)
- Hygienic investigations (optional)

The following description refers to the thermodynamic testing as developed at HLKS of HTA Lucerne (Afjei et al. 2007), the national Swiss test centre for residential ventilation systems. No European test standard for compact units exists, yet, but only separate test standards for the ventilation part and for heat pumps covering SH and DHW. However, the heat pump has been considered as core component of the compact unit and test points have been chosen based on the European heat pump test standards EN 14511 (now amendment (EN 14511 2007)) and (EN 255-3 1997).

Since the operation conditions for the heat pump depend on the heat recovery (HR) a combined testing of the heat pump and the ventilation system is of advantage. Consequently, testing of the three basic operation modes deliver the respective characteristics of the component:

- ventilation-only operation (temperature change coefficient, fan power)
- combined ventilation and heat pump in SH operation (COP, heating capacity SH)
- combined ventilation and heat pump in DHW operation (COP, useful energy DHW)

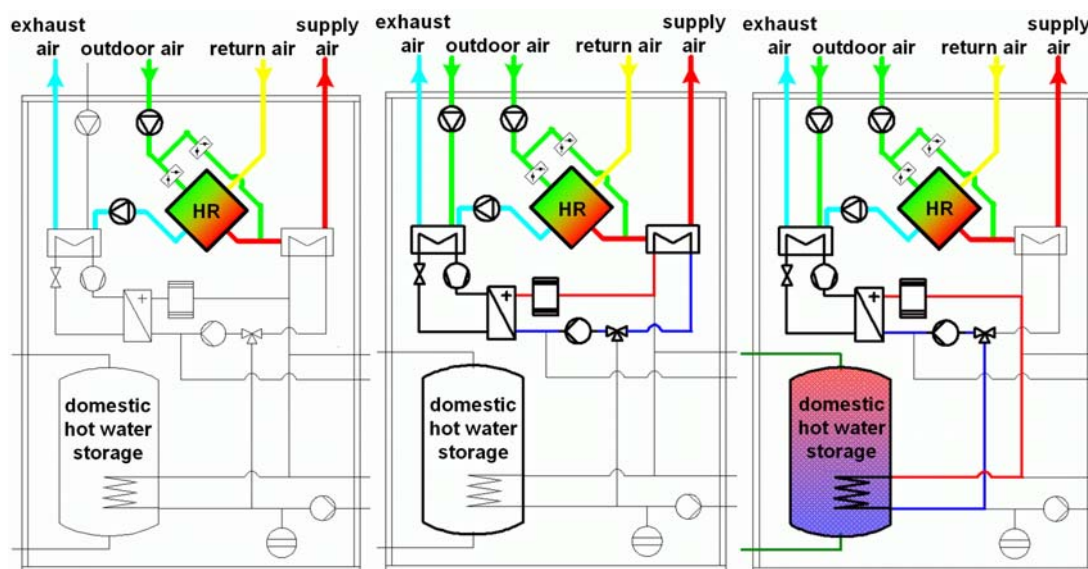


Figure 3: Operation modes for thermodynamic testing of compact units with internal circuit (left: ventilation only, middle: ventilation and SH, right: ventilation and DHW)

Figure 3 shows the different operation modes for a heat pump compact unit with an internal air ventilation circuit and heat recovery. The active part is depicted bold. The component characteristic depends on the temperatures and the volume flow rates, thus test points of EN 14511 have to be tested for different air volume flow rates. To characterise the ventilation part including heat recovery unit, the temperature change coefficient and the electrothermal amplification factor are determined.

2.2 Calculation method

The input data of the calculation method are the useful energy needs for SH and DHW (cf. **Figure 2**), the test results of the system at the tested operation points of the units, hourly outdoor air temperature data of the site, which can be produced with standard software or obtained by meteorological services, the system configuration and basic control details (e.g. characteristic temperature curve heating, balance point). Based on the existing standard calculation the extension of the calculation for combined systems is performed by a bin method. The principle of the bin method is shown in **Figure 4** for multifunctional systems. The hourly outdoor air temperature data of the site are sorted and cumulated. The resulting cumulative annual frequency of the outside temperature is divided into temperature classes (bins). In the centre of each bin, an operating point is evaluated with regard to the heat pump operation at these specific ambient conditions based on the product testing, i.e. the operating points shall be chosen at known test points of the unit. The operating point is considered to characterise the heat pump operation of the whole bin. Since the temperature difference depicted on the abscissa corresponds to a heat load, the areas correspond to the integrated heat load over the time on the ordinate and thus to the energy of each bin.

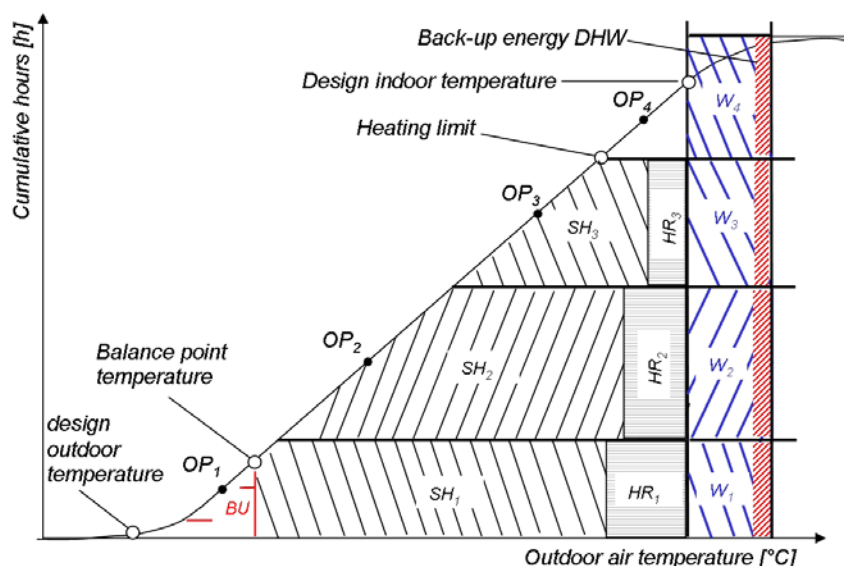


Figure 4: Principle of the bin method for multifunctional system for SH, DHW and ventilation with heat recovery unit (alternate mode) (OP - operating point, BU – back-up, SH – heat pump SH mode, W – heat pump DHW mode, HR – heat recovery unit)

Therefore, the areas can be evaluated by heating degree hours. The heating degree hours are calculated as product of the temperature difference between the fixed standard indoor temperature and the actual outdoor temperature of each temperature class, multiplied by the number of hours in the respective temperature class. A cumulation of the respective heating degree hours within the bin limits delivers the bin area, which is proportional to the heat energy fraction consumed in the respective bin. Thus, the ratio of the bin area to the total area of all bins corresponds to the energy fraction of the total energy need consumed in this respective bin. This bin energy is produced at the COP given by the operating point of the bin.

Consequently, the bin heat energy divided by the COP at the operating point delivers the electrical energy input for the heat pump. A weighting of COP-values at the operating point temperature conditions with the energy fractions of the bins and a subsequent summation of all bins yields the seasonal performance factor SPF.

Auxiliary electrical consumption of components not included in standard testing can be evaluated by the running time times the nominal electrical power input of the respective

component. This auxiliary consumption has to be added to the estimated heat pump electricity. Auxiliary consumption included in the heat pump testing is already considered in the COP. The whole energy calculation can thus be transformed to a calculation of area ratios, which can be expressed in terms of heating degree hours. The heating degree hours themselves can be calculated based on the outdoor temperature frequency as described above.

The recovered heat by a ventilation heat recovery – only if not already taken into account in the building energy calculation – is calculated by an enthalpy balance of the heat recovery and expressed in a diagram as corresponding area in the bin. The heat exchange in the heat recovery is thereby defined by the temperature change coefficient derived of the ventilation testing. The recovered ventilation loss is subtracted from the total energy need of the bin defined by the bin area, resulting in a reduced bin area for the heat pump and back-up heater (cf. **Figure 4**).

The calculation of the energy fraction of the back-up heater can be determined by a given balance point or by a more detailed power balance based on the actual heat pump capacity and the respective building heat load of each temperature class (e.g. in 1 K steps).

For DHW operation, the same bin-wise calculation can be performed, but for the COP at the operating points, the standard test results of DHW-testing, e.g. for Europe acc. to (EN 255-3 1997), must be taken. The respective tapping requirements in the bin can, for instance, be approximated by a constant daily consumption or a certain tapping profile. This calculation is sufficient to calculate the seasonal performance of alternate combined operating systems.

In simultaneous combined operating systems, a third operation mode, the combined operation, has to be considered based on the extended test procedure, since the heat pump characteristic changes at combined operation. How much combined operation takes place in the bin is estimated by a load balance which can be expressed in terms of the required running time of the heat pump needed to cover the energy requirement of each mode.

The minimum running time, which is determined by the energy need in the bin and the respective capacity of the heat pump, defines the maximum possible combined operation. The energy fraction covered in combined mode implies the energy needs to be covered in SH-only and DHW-only operation, i.e. the total energy need diminished by the energy covered in combined operation. The overall seasonal performance follows from energy weighting of respective operation modes.

2.3 Field monitoring

In order to gain experience with the field operation of ventilation compact units with exhaust air heat pump for the application in low energy houses, a detailed annual field monitoring of a unit installed in a single family low energy house in Gelterkinden, canton Baselland, has been accomplished (Afjei et al. 2007). The dwelling was certified according to the Swiss MINERGIE® standard. The basic data are given in **Table 1**.

Table 1: Design data of the single family house pilot plant

Number of inhabitants	3 (2 adult, 1 child)
Energy reference area	153 m ²
Design heat load	4.2 kW at 20 °C/-8 °C
Energy needs for space heating	44 kWh/m ² a
Flow/return temperature floor heating	30 °C/25 °C at -8 °C
Average DHW temperature	50 °C
Air exchange rate	0.33 h ⁻¹ (100 m ³ /h)
Outside & exhaust air volume flow for heat pump	600 m ³ /h

The system has balanced mechanical ventilation with a heat recovery unit and an air source heat pump using exhaust air with additional outside air as heat source. The heat pump works alternately on a floor heating emission system and a 200 l DHW storage. An electrical back-

up heater is installed to cover peak loads. The balance point for the monoenergetic operation was set to -5 °C. A sketch of the compact unit is given in **Figure 5**.

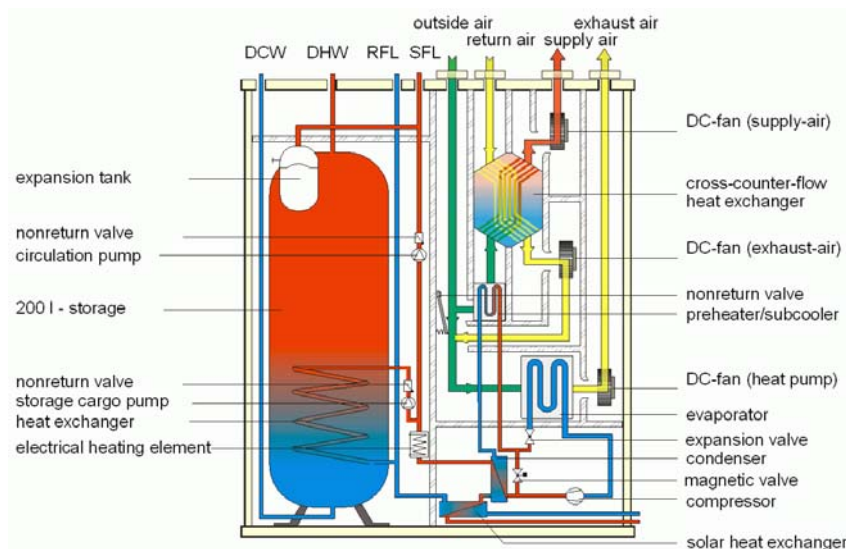


Figure 5: Sketch of the installed compact unit ((Afjei et al. 2007), source manufacturer)

The floor heating system is designed to be operated with low flow and return temperatures of 30 °C and 25 °C, respectively, at an outdoor design temperature of -8 °C in order to use the self-regulation effect, which is depicted in **Figure 6**. It refers to the reduced heat flux at increasing room temperatures and a corresponding smaller temperature difference between emission system and space.

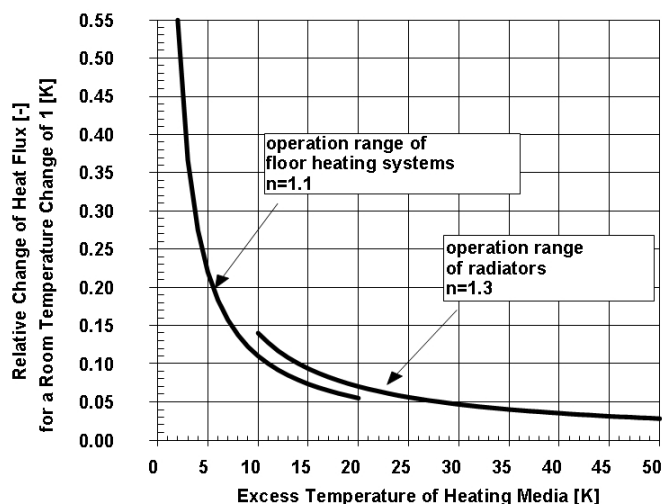


Figure 6: Self-regulation effect (n = exponent of the heating surface) (Afjei et al. 2000)

In **Figure 6** the relative change of the heat flux due to a change in the room temperature of 1 K is depicted for a floor heating system and a radiator heating system. In case of the lower excess temperature of the floor heating system the heat flux is much more affected by the 1 K change than in the case of the radiator with higher excess temperature. That means, in the case of low excess temperatures of the emission system the heat flux is strongly reduced in case of an increase of the room temperature, so that the heat flux is self-regulated by the room temperature. This results in a possible saving of installation costs for hydraulic controlling devices like thermostatic valves as well as a maximised COP of the heat pump due to the minimised temperature lift caused by the low flow temperature requirements.

The evaluation period of the field testing is April, 26th 2004 to April, 25th 2005. **Figure 7** shows the annual electrical energy consumption of the different components of the compact unit (left) and heat energy consumption (right) for the different building needs. 75% of the electrical energy consumption is used as driving energy for the heat pump compressor, while 20% is used by auxiliaries and only 5% by the electrical back-up heater. Concerning the heat energy consumption, the remarkably low DHW heat energy consumption of only 9% explains the high fraction of SH energy of 83% despite the layout as low energy house.

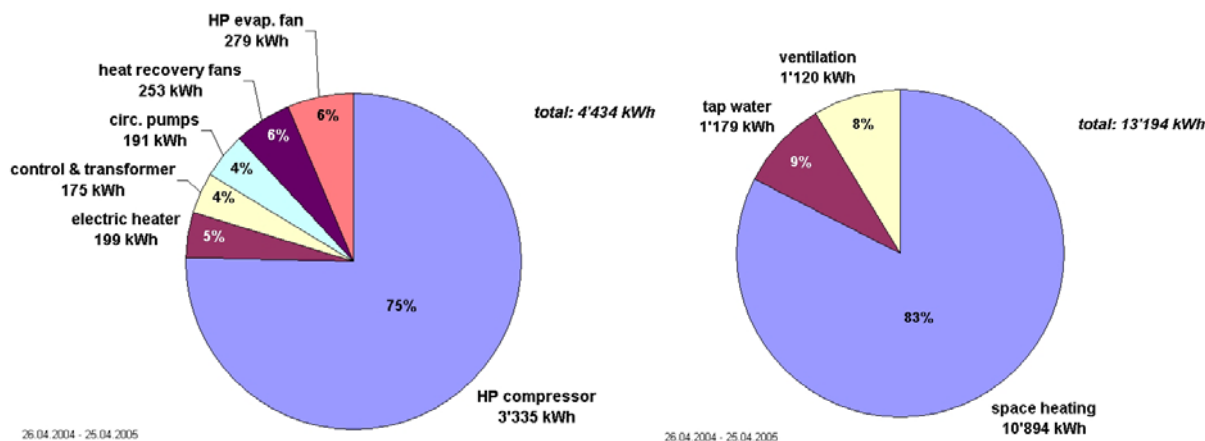


Figure 7: Annual electrical (left) and heat energy consumption (right) of the compact unit

Figure 8 presents the system boundaries used for the evaluation of performance characteristic number for the assessment of a ventilation compact unit with air-source heat pump. The same numbers have been calculated with the calculation method.

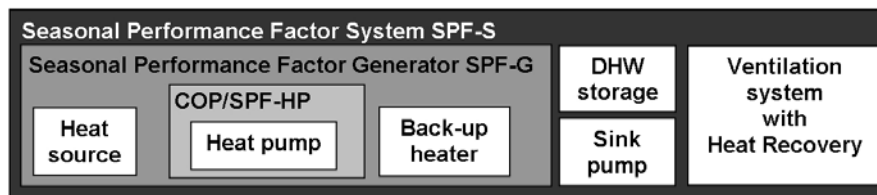


Figure 8: System boundaries for the assessment of the monitored compact unit

The SPF-HP comprises the system boundary according to the European heat pump testing. The Generator SPF (SPF-G) is the ratio of the produced energy of all generators (heat pump, back-up) to the respective electrical energy input and is well suited for the comparison to other heat generators like boilers. The System SPF (SPF-S) is related to the energy need and is calculated as ratio between the used energy (incl. the recovered ventilation energy) and the total electrical energy input to the entire system (incl. auxiliaries and electrical input to the ventilation). Results of the performance characteristic number are depicted in **Figure 9** as weekly performance factors (WPF) related to the outside air temperature.

The values for the WPF-HP presented in **Figure 9** range between 3.2 and 4.5 during winter, i.e. for outside air temperatures of 0°C to 14°C. During summer, i.e. for outside air temperatures above 15°C, the measured values for the WPF-HP range between 4.4 and 3.0, decreasing with higher outside air temperatures. The WPF-SYS ranges between 2.1 and 3.7 for outside air temperatures in the range of -4°C to 14°C (winter period) and decreases from 3.5 to 0.9 above 20°C outside air temperature. The WPF-G ranges between 2.5 and 4.2 below 14°C and between 4.0 and 2.5 in summer.

The characteristic of WPF-HP in winter can be explained with an increasing performance factor of the heat pump at higher outside air temperatures. However, above 15°C the heat pump shifts to less efficient domestic hot water production. The raising heating capacity due

to higher source temperatures combined with the limited heat flux of the DHW heat exchanger yield an increase of the temperature difference and hence an increasing heat pump supply temperature. As a result, all efficiency values in **Figure 9** deteriorate with the outside air temperature.

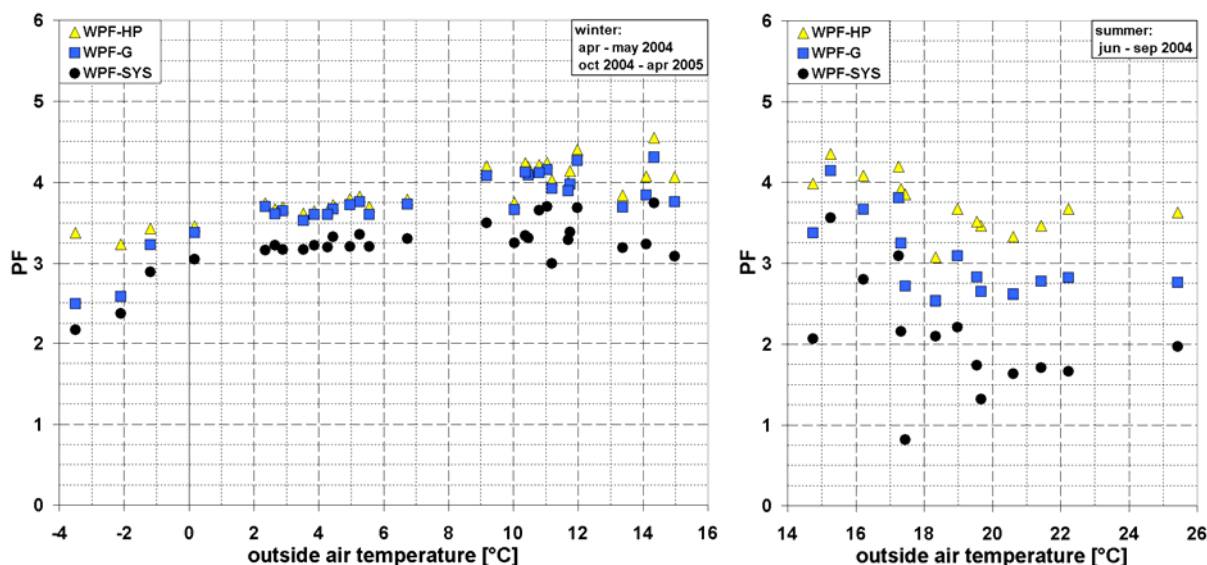


Figure 9: Weekly Energy Performance Factors WPF of the measured pilot plant

The characteristic of WPF-SYS is plausible, because system efficiency increases with lower outside air temperatures due to an increasing requirement of space heating and the ventilation heat recovery that is considered in WPF-SYS. The low system efficiency at high outside air temperatures results from relatively high standby losses in relation to the small energy requirement, but over 18 °C outside air temperature they are with 510 kWh a small part of yearly energy requirement compared with 13'200 kWh in total and thus of minor importance. The performance factors WPF-HP and WPF-G depicted in **Figure 9** include the generated heat in the subcooler for ventilation. Not considering the generated heat in the subcooler would decrease the WPF-HP and the WPF-G by about 0.2 or 5%.

2.4 Performance evaluation of the calculation method

The results of the calculation have been compared in (Wemhöner and Afjei 2006) to the field monitoring results for two direct expansion ground source heat pumps (only SH mode), one ground-source brine-to-water heat pump (SH, DHW mode) and the ventilation compact unit air source heat pump (SH, DHW and ventilation mode) described in the previous chapter. For the evaluation, the system boundaries given in **Figure 8** were used.

In the following the results for the air source heat pump compact unit are discussed. The measured performance of the field monitoring has been compared to the calculated values for the different operation modes and the different system boundaries. For the calculation the local outdoor temperature conditions monitored with the field monitoring equipment have been used as input data for the calculation in order to refer to the same boundary conditions as in field testing, which is necessary for validation purposes.

Figure 10 gives an overview of the results. Since controller settings are usually not known in detail and therefore cannot be evaluated, two limits have been considered for the calculation of the sink pump electricity demand: a running time linked to the operation of the heat pump and an operation throughout the whole heating period, which is given by the values in brackets. However, the impact on SPF values is marginal.

Regarding the comparison of the field monitoring performance to the calculation results, the space heating part is generally reproduced better than the DHW part. For the DHW

calculation a constant daily consumption has been assumed, while in reality the tapping volumes are not so evenly spread over the year. Further differences occur due to control effects. For instance, the back-up heating supports the DHW heating by the heat pump after large draw-offs to accelerate the hot water availability. These controller settings are too case-specific to be reproduced by a hand calculation.

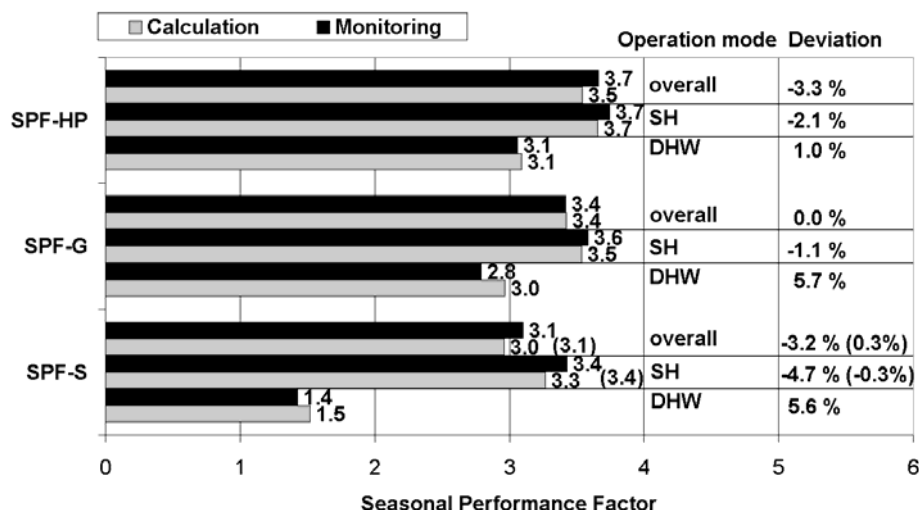


Figure 10: Seasonal performance factors of the field monitoring compared to the calculated values

Calculated overall seasonal performance factor values deviate in the range of $\pm 4\%$ from measured values, which is a satisfactory result for a hand calculation, where certain simplifications are inevitable. Some of the simplifications are given in chapter 3.1. The single operation modes show with $\pm 6\%$ a slightly higher deviation from the measured results in case of this ventilation heat pump compact unit.

2.5 Exergy analysis

Exergy is defined as the fraction of the energy that can be arbitrarily converted in each of the different energy forms. In all natural processes, exergy is converted to anergy, e.g. by irreversible heat transfer due to finite temperature differences or pressure drop due to friction, which corresponds to an exergy loss. Therefore, the exergy content is a measure of the thermodynamic quality of an energy flow or a process.

An exergetic consideration has been applied to characterise the impact of the temperature requirements of different source and sink systems used in compact units (cf. **Figure 1**) on the performance of a ventilation compact unit.

Pure air heating distribution systems have the limitation that the maximum temperature is restricted to about 50°C . Moreover, to avoid excessive exergy input to the fans, common designs limit the air volume flow to the hygienic necessary. Thus, air heating systems can only be applied in ultra low energy dwellings due to the limited heat capacity of the air and require high supply temperature up to the limit of about 50°C in colder winter periods.

Floor heating systems, on the other hand, have a large surface for the heat emission in the room, whereby supply temperatures below 30°C can be realised. However, additional exergy input for the circulation pump to transport the heat transfer medium is required.

The exergy comparison has been accomplished for an ultra-low energy house under the simplifying assumptions of adiabatic components, negligence of defrosting operation and a constant internal exergetic efficiency of the refrigerant process of $\zeta=0.5$ (exergy losses of compression, refrigerant pipe, heat exchanger friction, expansion) to focus on the exergy losses linked to the source/sink system periphery. **Table 2** shows the boundary conditions used.

Table 2: Boundary conditions and resulting source and sink temperature (HR – heat recovery)

Building	
Ambient reference temperature [°C] / Indoor design temperature [°C]	-4 / 20
Building heat load at ambient reference temperature [W]	1'200
Ventilation system	
Volume flow rate ventilation system [m ³ /h]	110
Temperature change coefficient (considered the same for supply and return air path) [-]	0.8
Pressure drop heat recovery, condenser, evaporator [Pa] / air ducts [Pa]	80 / 40
Constant fan power [W] / Pressure drop dependent fan power [W/Pa]	10/ 0.12
Heat pump	
Pinch point condenser air heating [K] / floor heating [K] / Pinch point evaporator both[K]	5 / 2 / 5
Required supply temperature air / floor heating at ambient reference temperature [°C]	48 / 29
Condensation temperature / Evaporation temperature of the air heating system [°C]	48 / -21
Condensation temperature / Evaporation temperature of the floor heating system [°C]	30 / -23
Pump power of the circulation pump [W]	40

The COP of the heat pump can be expressed by the equation

$$\text{COP} = \zeta \cdot \text{COP}_C = \zeta \cdot \frac{T_{\text{cond}}}{(T_{\text{cond}} - T_{\text{evap}})}$$

eq. 1

where COP_C denotes the Carnot COP, ζ the constant internal exergetic efficiency, T_{cond} the condensation and T_{evap} the evaporation temperature [K].

Exergy losses of air heating systems are 14% higher than losses of the floor heating system, if conditions in **Table 2** are taken, despite the exergy input to the circulation pump and the evaporation temperature decrease due to the higher anergy extraction in the evaporator in case of the floor heating system.

Figure 11 depicts the range of temperature levels in the common components of a compact unit with common source and sink systems. The bold red arrows follow the way of the heat transfer media through the compact unit. Starting at ambient conditions, the heat recovery unit raises the exhaust air temperature above ambient level, while in the evaporator it is cooled down by the refrigerant. The evaporation temperature level can be increased by a mix with an additional outdoor air flow or by a preheating of the air in a ground-to-air heat exchanger, which is often installed in order to avoid frosting of the heat recovery heat exchanger. The heat pump makes the main temperature lift from the source temperature level to the required distribution temperature level, and in the emission system the temperature level is reduced first to the room temperature, and then by the heat losses of the building to the ambient level.

The temperature levels and corresponding exergy losses depend on the design of the heat exchangers, since larger heat exchanger surfaces reduce the heat transfer resistance and lead to lower temperature differences for the heat transfer. On the other hand, the temperature levels are also defined by the capacity of the source and the sink heat transfer medium. By the choice of hygienic necessary ventilation air as heating medium, for instance, high temperature spreads are required based to the limited heat capacity due to restricted volume flow rate and the heat capacity of the air.

The theoretical minimum required temperature lift for a reversible heat transfer is defined by ambient and indoor temperature. All limitations of the heat transfer system periphery thus have to be compensated by additional exergy input to the heat pump compressor due to the elevated temperature lift. On the sink side the maximum spread is seen between an air heating and TABS emission system, on the source side between a system with only ventilation exhaust air and a preheating of a higher outside air flow.

The general advantage of exergy analysis is, that the different loss mechanisms, e.g. due to heat transfer or pressure drop, can be quantified on the same basis. However, an exergy analysis gives only the technical viewpoint, and economical consideration may result in a

different assessment. Last but not least environmental and comfort issues have to be considered, as well.

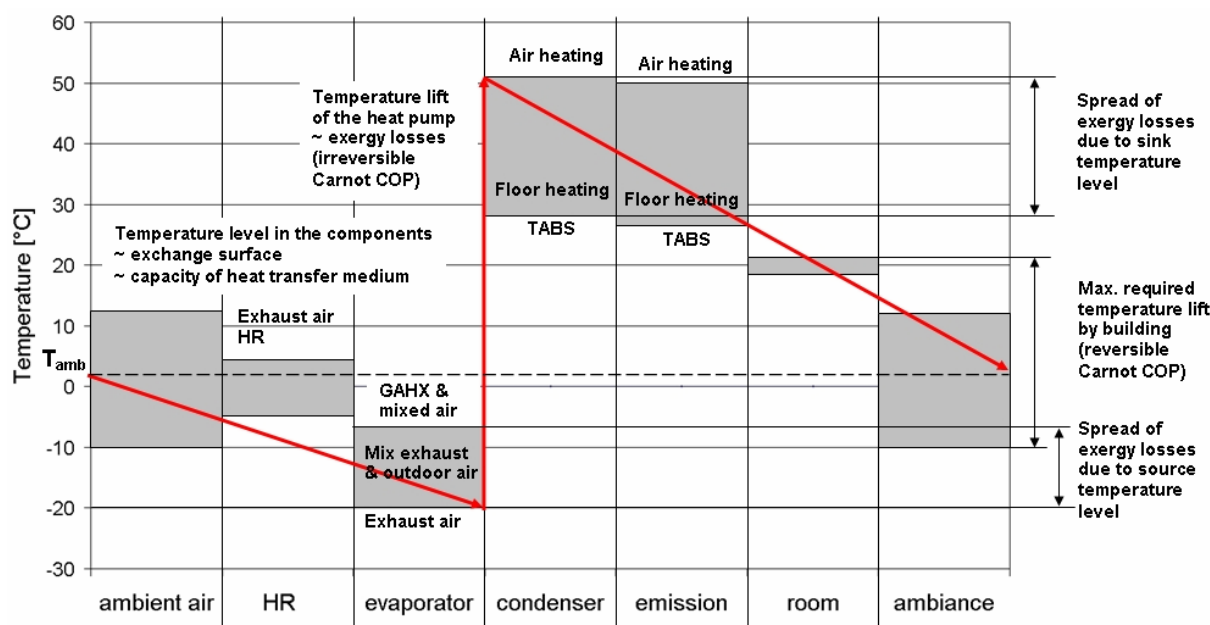


Figure 11: Temperature levels in the single components for compact unit
(GAHX – ground-to-air heat exchanger, HR – Heat recovery unit,
TABS – Thermally activated building structures, COP – Coefficient of performance)

3 CONCLUSIONS AND ACKNOWLEDGEMENT

3.1 Simplifications in the calculation method

The calculation method implies some simplifications. One assumption is the redistribution of the energy according to heating degree hours which are only dependent on the outdoor air temperature. In low and ultra-low energy houses external (solar and internal gains may have an impact on the energy redistribution, too. This can be approximated by an adjustment of the upper temperature limit for heating based on the quantity of solar and internal gains as depicted in **Figure 4**. Furthermore, it is postulated, that the heat recovery reduces the space heating energy need, the heat pump covers the space heating need up to the capacity limit and the remaining space heating energy need is supplied by the electrical back-up heating. This corresponds to the normal operation of the unit, but is, however, an idealisation, since control may have an impact on the respective fractions.

Actually, the controller impact cannot be considered in detail, since settings are often case-specific and not known. Therefore, controller impact is simplified to a standard setting dependent on the system layout, e.g. for the running time of auxiliaries, as well.

3.2 Implementation

The results are currently implemented in the respective CEN standards. The heat pump calculation method has been implemented in the European draft standard (prEN 15316-4.2 2006) used in the (EPBD 2003). As national implementation of the EPBD, Germany has introduced the method on a monthly basis in (DIN V 18599 2007).

A joint working group committed to testing of ventilation compact units with heat pump is presently in constitution with the technical committees CEN/TC 113 (heat pump testing) and CEN/TC 156 (ventilation systems). This gives the perspective of a combined testing of heat pump and ventilation part of compact units with heat pump.

3.3 Component and system optimisation

Exergy analyses are well suited for a component and process optimisation, since the impact of different loss mechanisms can be quantified on the uniform basis of exergy losses. The exergy analysis of the compact unit shows that increases of the temperature lift between source and sink system has to be supplied as additional exergy to the heat pump compressor. Therefore, system layouts guaranteeing high source temperature and low sink temperature levels, so called Low-Ex systems, lead to a significant reduction of the exergy losses and thus reduce the electrical energy input.

3.4 Acknowledgement

The IEA HPP Annex 28 was a team work and the results are based on the contribution of each member. The project management and the Swiss contribution to IEA HPP Annex 28 have been funded by the Swiss Federal Office of Energy (SFOE).

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