Research program
heat pumping technologies, cogeneration, refrigeration

Calculation method for the seasonal performance of heat pump compact units and validation

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Summary

For the heating of low energy houses according to the MINERGIE® or MINERGIE-P® Standard with significantly reduced heat energy needs heat pump compact units have been developed in the recent years, which combined the functions space heating, domestic hot water and ventilation in one unit. To assess the overall performance, the seasonal performance factor (SPF) has to be calculated by a standardised method. The SPF is one basic number for the buildings energy certificate, as it is introduced in EU member countries based on the EU Building Energy Performance Directive (EPBD).

In this project charged by the Swiss Federal Office of Energy (SFOE) a test procedure has been elaborated at the University of Technics and Architecture HTA Luzern in order to deliver the required component characteristic for the calculation of the SPF by test rig measurements. The test procedure is based on European standards and test methods of the German Passive House Institute and the German Institute of Building Technologies DIBt. The Institut of Energy in Building of the University of Applied Sciences Northwestern Switzerland (up to 31.12.2005: FHBB, Institute of Energy) developed a calculation method based on temperature classes („bin method“) for the calculation of the SPF of combined heat pump systems for space heating and domestic hot water (DHW). In the frame of the EPBD the method has been implemented in the European draft standard prEN 15316-4.2, which will soon be sent out to formal vote. In this project the method has been extended to compact units.

In order to gather experience with the real behaviour of pilot plants detailed field monitoring of two compact units has been performed. One pilot plant is installed in a single family house according to the MINERGIE® standard in Gelterkinden (canton Baselland, CH), using a compact unit designed for heat loads of MINERGIE® houses with hydronic distribution by a floor heating system.

The other pilot plant is a single family house in Zeiningen (canton Aargau, CH) according to MINERGIE-P® with a compact unit designed for ultra-low energy houses. The compact unit is originally designed for air heating distribution, but has been modified to a prototype application with hydronic heat emission by thermally-activated building structures for this project.

The pilot plant in Gelterkinden delivered the results that in wintertime, 78% of the electrical energy is used by the heat pump, 5% by the electrical back-up heating and 17% for auxiliaries including the ventilation fans and circulation pump of the heating system. The resulting overall SPF-HP of the heat pump is 3.8, while the SPF-S of the system related to the energy need of the space heating and DHW distribution varied between 2.4 in summertime and 3.1 in wintertime at a measurement uncertainty of 4 - 7%.

Calculated values correspond to the field monitoring in the range of $\pm$6% and thereby are in the range of the exactness of the field monitoring on the one hand and the component characteristics on the other hand. Most important impact on the SPF is the source and sink temperature level, since the heat pump is the core component of the system.

The field monitoring of the pilot plant in Zeiningen yielded, that in wintertime 36% of the electrical energy was used by the heat pump, 47% by the electrical back-up heater and 17% by auxiliary components. The SPF-HP of the heat pump was 2.8 while the SPF-S of the overall system is 1.8 due to the higher fraction of electrical back-up energy. Reasons for the higher back-up energy use was the low heating capacity of the heat pump of 1.5 kW related to a building design heat load of 2.5 kW.

Zusammenfassung

Zur Beheizung moderner Niedrigenergiegebäude nach dem MINERGIE® oder MINERGIE-P® Standard mit einem wesentlich reduzierten Heizwärmebedarf sind in den vergangenen Jahren Wärmepumpenkompaktgeräte entwickelt worden, die neben der Heizung auch die Funktionen Warmwasserbereitung und Wohnungslüftung mit Wärmerückgewinnung in einem Gerät vereinigen. Um eine Aussage über die Gesamteffizienz machen zu können, muss der Jahresnutzungsgrad nach einem standardisierten Verfahren berechnet werden. Diese Grösse bildet die Grundlage für den Gebäude-Energieausweis, der im Rahmen der EU-Gebäudeenergierichtlinie (EPBD) in den Ländern der EU eingeführt wird.

In dem vom Bundesamt für Energie (BFE) in Auftrag gegebenen Projekt wurde von der HTA Luzern ein Testprozedere ausgearbeitet, mit dem die zur Berechnung des Jahresnutzungsgrades notwendigen Leistungsdaten von Wärmepumpenkompaktgeräten auf dem Prüfstand ermittelt werden. Das Testprozedere stützt sich auf europäische Normen und Prüfverfahren des Passivhausinstituts und des Deutschen Instituts für Bautechnik DIBt. Vom Institut Energie am Bau der Fachhochschule Nordwestschweiz (bis 31.12.2005: FHBB, Institut für Energie) wurde ein auf der Temperaturklassenmethode („bin method“) basierendes Rechenverfahren zur Berechnung des Jahresnutzungsgrads für kombinierte Wärmepumpensysteme zur Raumheizung und Wassererwärmung entwickelt. Im Rahmen der CEN Normung zur EPBD wurde das Verfahren in den europäischer Normentwurf prEN 15316-4.2 integriert, der in Kürze in die Schlussabstimmung geht. In diesem Projekt wurde eine Erweiterung des Verfahrens auf Kompaktgeräte vorgenommen.

Für Aussagen zum Betriebsverhalten der Heizgeräte im Feldeinsatz und zur Validierung des Verfahrens wurden zwei Pilotanlagen detailliert gemessen. Ein nach MINERGIE® zertifiziertes Objekt in Gelterkinden (BL) ist mit einem Wärmepumpenkompaktgerät ausgerüstet, das an den Energiebedarf von MINERGIE®–Häusern angepasst ist und eine hydraulische Heizverteilung erlaubt.

Das andere nach Minergie-P® zertifizierte Objekt in Zeiningen (AG) ist mit einem Wärmepumpenkompaktgerät ausgestattet, das für MINERGIE-P®- bzw. Passivhäuser mit Luftheizung konzipiert ist. In der Pilotanlage wurde es als Prototyp-Anwendung auf eine hydraulischen Wärmeverteilung modifiziert, die gekoppelt mit einer Bauteilaktivierung arbeitet.

Die Feldmessungen in Gelterkinden zeigten, dass im Winter 78% des elektrischen Energieverbrauchs für die Wärmepumpe, 5% für die elektrische Zusatzheizung (für Heizung bis -5°C und WW-Betrieb) und 17% für Hilfsaggregate aufgewendet werden müssen. Die Wärmepumpe erreichte eine Jahresarbeitszahl von 3.8, der bedarfsseitige Systemnutzungsgrad variierte zwischen 2.4 (Sommer) und 3.1 (Winter) bei einer Messunsicherheit von 4 - 7%.

Das Rechenverfahren stimmt mit einer Unsicherheit von ±6% mit den Messungen überein und liegt damit im Bereich der Messgenauigkeit der Eingangsdaten einerseits, insbesondere des Wärmepumpenkennfeldes, und der Feldmessungen andererseits.

Die Feldmessungen in Zeiningen ergaben, dass im Winter 36% des elektrischen Energieverbrauchs von der Wärmepumpe, 47% von der elektrischen Zusatzheizung und 17% von den Hilfsaggregaten verursacht werden. Die Wärmepumpe erreichte eine Jahresarbeitszahl von 2.8, der bedarfsseitige Systemnutzungsgrad erreichte wegen des hohen Elektroanteils nur 1.8. Grund war die im Vergleich zur Norm-Heizlast (2.5 kW) geringe Wärmepumpennennleistung von 1.5 kW.

Das Projekt war ein nationaler Beitrag der Schweiz zum Annex 28 im Wärmepumpenprogramm (HPP) der Internationalen Energieagentur IEA. Am Annex 28 nahmen acht weitere Länder teil: AT, CA, DE, FR, JP, NO, US, SE.

Contents

1 OBJECTIVES

The objective of the project is to develop a calculation method for the seasonal performance factor of heat pump compact units with heat recovery unit, ground-to-air heat exchanger and optional use of solar energy. The method is based on the FHNW method for the calculation of combined operating heat pumps for space heating and domestic hot water production developed in an SFOE project in the frame of IEA HPP Annex 28 [25] at the Institute of Energy in Building of the University of Applied Sciences Northwestern Switzerland. The method shall be composed in that way that the most important impacts on the seasonal performance can be estimated, but input data shall be limited to publicly available data from manufacturer information or component testing. Since the calculation is mainly based on the product characteristic of the heat pump compact unit, uniform results of an adequate test procedure have to be available. Thus, the calculation method is based upon a test procedure that is developed at the HLKS test centre at the University of Applied Sciences HTA Lucerne.

The calculation method is to be validated by comparison with field measurements. Existing data of a brine-to-water ground source heat pump are evaluated based on field monitoring accomplished in the project "Low cost low temperature heating systems with heat pump" [16] of the Swiss Federal Office of Energy SFOE.

Since compact units are introduced quite recently in the market, little experience and field data exist. Therefore, two systems have been field monitored in the project. One system, the LWZ 303 SOL of the German manufacturer Stiebel Eltron, is installed in a house according to the Swiss MINERGIE® standard (www.minergie.ch). The other system, the Vitotres 343 of the manufacturer Viessmann, is installed in a house according to the MINERGIE-P® standard, which is similar to the German standard "Passivhaus". Besides the validation of the calculation method another objective of the field monitoring is to gather more experience with the real behaviour of compact units in the field application to prove the functionality and give a feedback to the manufacturer on optimisation potentials.

The seasonal performance calculation of the units is needed on the one hand to compare the performance of compact units among each other or to other heating systems regarding the market competitiveness. On the other hand, the seasonal performance calculation is needed for labelling, which is required e.g. for building standards like the Swiss MINERGIE® standard or for a building certificate according to the European Directive on the Energy Performance of Buildings (EPBD) [13].

The calculation method and the test procedure shall serve as a recommendation for a European standard in CEN Standardisation Committees.

The test method developed at the University of Applied Sciences Lucerne (HTA) consists of a set of testing standards for the various types of units, conforming to international requirements. These test standards define test methods and procedures. The duration of tests is to be kept short whilst delivering the data required for the SPF calculation.

In parallel to this project a test facility University of Applied Sciences Lucerne (HTA) has been designed, installed and commissioned, which is capable of performing a complete technical test of compact ventilation units equipped with a heat recovery system and heat pump at a reasonable cost. The tests are to provide not only thermal and flow measurements, but also complementary acoustic measurements. These shall comprise the power level of the sound emitted by the unit into the room in which it is installed on the one hand and the power level of the sound emitted into the duct system over the air connections on the other hand. Furthermore, tests should identify leakages including filter bypass leakage. Finally, a comprehensive appraisal is to cover operation and maintenance as well as the materials utilised and any problems detected, such as thermal bridges.

2 STATE OF THE ART

2.1 Typology of compact units

Heat pump ventilation compact units, colloquially “compact units”, have been introduced in the market in the end of the Nineties in connection with the upcoming passive house standard (http://www.passiv.de) in Germany. Currently, many manufacturers in heating industry offer a compact unit in the one or other form. Even though these available compact units differ with regard to the internal system layout, the core components contained in every configuration are a heat pump connected to the exhaust path of a ventilation heat recovery and a domestic hot water (DHW) storage tank. The compact units can be classified according to the following criteria:

- **Heat source for the heat pump**
    - Pure exhaust air of the ventilation heat recovery
    - mixed exhaust-/outdoor air
    - ground source
    - inlet air preheating (e.g. ground heat exchanger preheating the outdoor air)
- **Distribution system**
    - Air heating
    - Hydronic heating
    - Both possibilities
- **Functionality**
    - Space heating/Domestic hot water/Ventilation
    - optional cooling function by reversible heat pump operation
    - humidification function, enthalpy recovery
    - solar DHW production
- **Field of application/range of heating capacity**
    - ultra low energy houses (heating capacity of the heat pump ~ 1.5 – 2 kW)
    - low energy houses (heating capacity of the heat pump ~ 4 – 6 kW)

Typical advantages of compact units are:

- combined covering of different buildings needs (heat recovery)
- compact design enables to place the unit inside the heated space and thus losses of the unit can be used for space heating in wintertime (adequate noise protection required!)
- planning and installation expenses (installation, commissioning) can be kept at low level
- monoenergetic operation with electricity, no fees for connection to natural gas network, no space needed for fuel

2.2 Test standards for compact units

Neither national nor international test standards are available for compact ventilation units equipped with heat pumps. Current standard testing of heat pumps as core component and different national methods for the testing of compact units from Germany are shortly presented in the following.

2.2.1 Testing of heat pumps

2.2.1.1 Space heating operation

For the space heating (SH) only mode heat pumps are tested in steady-state operation according to the European standard EN 14511 [8] which was introduced in 2004 and replaced the former standard EN 255-2 [20] . In both standards a transient test for frosting conditions of air-source heat pumps is described, as well. Test points of both standards are shown in Fig. 1.

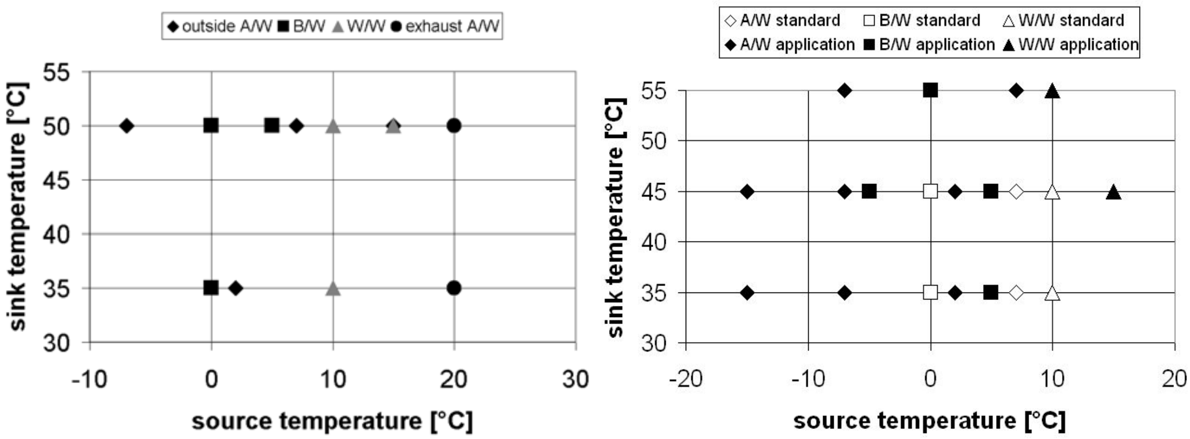


Fig. 1: Test points according to previous EN 255-2 (left) and present EN 14511-2 (right)

The basic results are the COP and the heating capacity at the respective test points. Results of the extended heat pump testing of the Swiss national heat pump test centre WPZ are published in the WPZ-Bulletin at http://www.wpz.ch.

2.2.1.2 DHW operation

To take into account the interaction between the heat pump and domestic hot water (DHW) storage, heat pumps for domestic hot water operation are tested in a system test. In contrary to the steady-state testing, the test standard for DHW operation EN 255-3 [7] uses a tapping cycle and thereby instationary effects are contained in the test results. The testing cycle and the test points - each one for different types of heat pumps - are presented in Fig. 2. The testing cycle consists of 5 phases as depicted in Fig. 2, right: The hot water storage is heated-up to the reference temperature (phase 1), a cyclic extraction of half the storage volume and reheating is repeated until the energy content of the drawn-off hot water stays within 10% (phase 2). Afterwards an extraction down to a water temperature of 40°C is accomplished (phase 3) and the storage is reheated. Then, the system is left in stand-by operation for at least 24 h, until the storage stand-by losses are reheated (phase 4). In the end, a second extraction until a temperature of 40°C is performed (phase 5). More details are given in the test guidelines for the testing of compact units in Appendix A.

By the stand-by testing in phase 4 the electrical power input to cover the DHW storage losses is determined as an output of the testing. However, the electrical power input cannot be directly recalculated to thermal losses, since other auxiliary energies are included in the values and COP value for the reheating is usually not known, as temperature conditions differ from the COP_t for the extraction of DHW evaluated in phase 2.

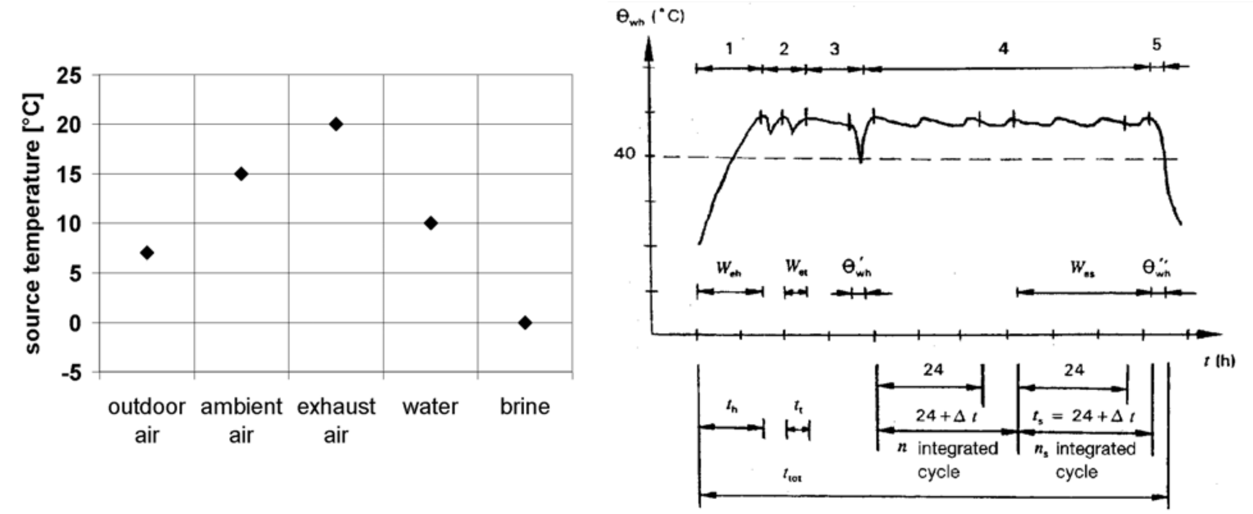


Fig. 2: Test points dependent on heat pump source (left) and test cycle (right) of EN 255-3

The delivered output values of the EN 255-3 are listed in Tab. 1.

Tab. 1: Output values of EN 255-3 testing for heat pumps with DHW operation

Value	Phase	Unit
Heating-up time (t_h)	1	s
Heating-up energy input (W_{eh})	1	kWh
Coefficient of performance for tapping of hot water (COP_t)	2 (corrected by 4)	-
Maximum volume of water in a single tapping (V_{max})	3 and 5	dm^3
Hot water reference temperature (θ_{wr})	3 and 5	°C
Stand-by power input (P_{es})	4	W

2.2.2 Testing of compact units acc. to DIBt

In Germany every generator for space heating appliance sold on the market has to be certified by the German Institute of Building Technology DIBt (= Deutsches Institut für Bautechnik, www.dibt.de). The following tests are applied:

- Safety testing (visual inspection - mechanical; electrical)
- Leakage testing
- Ventilation testing
- Thermal testing (heat recovery, heat pump)

Test conditions for the thermal testing are given in 3.3.2.4.

Results of the testing are

- Internal and external leakage
- Pressure–volume flow rate characteristic (pressure increase, specific electrical input)
- <u>heat recovery unit</u>
  - heat recovery efficiency η'_{WRG} (includes defrosting, heat losses, volume flow balance) according to the equation

$$\eta'_{WRG} = \frac{\dot{H}_{sa} - \dot{H}_{oa}}{\dot{H}^{*}_{sa} - \dot{H}_{oa}} \qquad [-]$$

eq. 1

where

η'_{WRG}	heat recovery efficiency	[-]
$\dot{H}_{sa}$	enthalpy flow of the supply air	[J]
$\dot{H}^{*}_{sa}$	enthalpy flow at indoor air temperature and moisture content of outdoor air x_{oa}	[J]
$\dot{H}_{oa}$	enthalpy flow of the outdoor air	[J]

For the heat recovery efficiency measured values are corrected based on the applied system, e.g. in case of a prescribed ground-to-air heat exchanger the correction factor for defrost operation is 0.

- heat pump
  - COP heat pump (based on electrical energy input to the compressor)
  - COP unit (based on compressor and fans, not on system boundary of EN 14511)
  - COP_t for extraction of DHW based on EN 255-3

  at different volume flow rates, see chap. 3.3.2.2.

The same test procedure is also applied at TÜV Süd Group (http://www.tuev-sued.de) and IKE Stuttgart, Chair for space heating- and ventilation technique (http://www.lhr.ike.uni-stuttgart.de).

2.2.3 Testing of Fraunhofer Institute of Solar Energy Systems in Freiburg (DE)

The testing of compact units according to Fraunhofer Institute of Solar Energy Systems (FhG-ISE) in Freiburg, Germany, has been deduced from test reports and incorporates the following testing

- Leakage testing (accomplished by Tracergas measurements)
- Ventilation system testing (temperature change coefficient supply air acc to eq. 5, fan efficiency)
- Thermal testing (heat pump and heat recovery, basically acc. to EN 255-3)

Not all tests are necessarily applied. Measurements take place inside the unit, too.

The leakage testing is accomplished with the inert tracergas SF_6 to quantify the leakage of the return air flow into the supply air flow and the outdoor air flow into the exhaust air flow [15].

The ventilation system testing basically delivers the temperature change coefficient, which is differentiated for the heat exchanger and the unit. Measurements of the heat exchanger, however, have high uncertainties, since air temperature varies over the cross section of the air duct directly behind the heat exchanger. Furthermore, testing with a balanced and unbalanced ventilation may be accomplished.

The thermal testing is mostly based on EN 255-3, i.e. the testing of heat pumps in DHW operation described in chap. 2.2.1.2. One complete test cycle of EN 255-3 is to deliver the output values of the EN 255-3 acc. to Tab. 1, in particular the COP_t and the electrical power input to cover the storage losses. For a second test point a reduced cycle is performed to deliver the COP_t. Results of the thermal testing are basically a so-called winter-COP (outdoor air inlet in the range of 7°C – 10°C) to characterise a combined space heating/DHW operation (winter) and a summer-COP (outdoor air inlet at 20°C) for DHW-only (summer) operation. Both COP values are evaluated at a fixed volume flow rate, which is not necessarily the same for winter- and summer-COP. Depending on the internal system configuration, additional testing of different operation modes may be performed, i.e. space heating-only operation mode or DHW tests with activated and deactivated back-up heater.

In addition, the COP in the heating-up phase (phase 1 of EN 255-3) and the same characteristic numbers of the heat recovery system as in the DIBt testing are evaluated. By an overall energy balance the heat rejection of the unit into the room (or the heat gain of the unit from the room) are calculated, as well.

Since the calculation method of the Passive House Institute (www.passiv.de) is strongly related to the two COP values, some manufacturers give these values in their technical data sheet.

2.2.4 Assessment of existing test procedures

The existing test methods apply a combined testing of the heat recovery and the heat pump, which is advantageous, since the impact of a possible preheating and the heat recovery on the inlet temperature of the heat pump are included in the test results.

A general problem, however, is, that the test results often cannot be applied to different operating conditions of the system, since on the one hand temperature conditions are not specified or on the other hand, only one sink temperature condition or only one volume flow rate is tested. Therefore, if operation conditions deviate from that in the testing, the values do not describe the real performance.

2.3 Existing performance calculation methods

A survey of existing calculation methods revealed that two Excel-sheets both integrated in tools to prove compliance with building standards can be used for heat pump compact units.

The Swiss "MINERGIE® association" (www.minergie.ch) has published an Excel-Tool for the calculation of heat pump systems for heating and alternate domestic hot water production including the electrical back-up systems, while the German "Passivhaus-Institut" has included a sheet for compact units in the “Passivhaus-Projektierungs-Paket” (PHPP) since the update of 2003. The actual version is the PHPP 2004 [1].

In a research project of the Swiss Federal Office of Energy (SFOE) in the frame of Annex 28 [5] in the Heat Pump Program (HPP) of the International Energy Agency (IEA) (http://www.heatpumpcentre.org), a calculation method to cover combined operating heat pump systems with alternate or simultaneous domestic hot water production has been developed at the FHNW. This method has been integrated in the heat pump draft of the European standardisation organisation CEN in the framework of the revision of building and building technologies standards of the Directive on the Energy Performance of Buildings (abbreviated Energy Performance Building Directive EPBD [13]).

The three methods are shortly described in the following based on the basic principle of the calculation outlined in the next chapter.

2.3.1 Background of the calculation

Since all three methods are based on an outdoor temperature bin approach, the principles of the bin method are shortly described in the next paragraph. Moreover, the bin method is introduced in national calculation methods (e.g. as described in [16], guidelines as VDI 2067 [11] and standards as ASHRAE 116-1995 [10]).

2.3.1.1 Bin method

The heating capacity and the COP of a heat pump depend strongly on the source and sink temperature conditions. These values may change over the entire operation range of the heat pump. However, as described in the last paragraph, performance values of the heat pump are only known at fixed standard test points. Thus, a weighting using these available information at fixed operating conditions with the respective energy production at these operating conditions is accomplished by the so-called bin method.

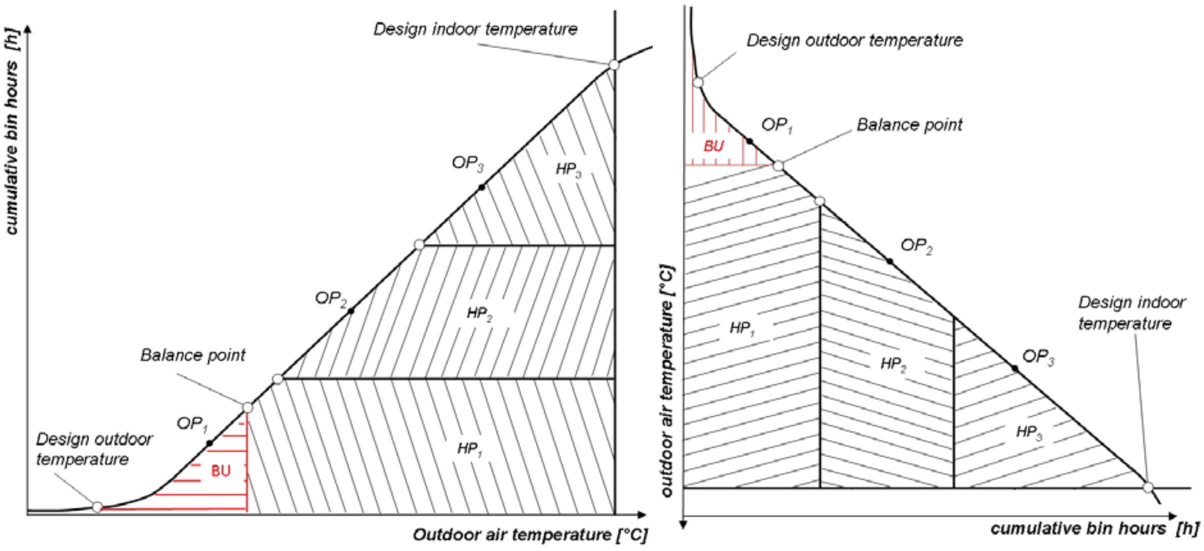


Fig. 3: Principle of the bin method (left: cumulative frequency with heat pump and parallel back-up operation, right: duration curve with heat pump and back-up operation)

The principle of a bin method is depicted in Fig. 3. The cumulative annual frequency of the outdoor temperature depicted in Fig. 3 left is divided into temperature classes (temperature bins). A 90°-clockwise rotation of the cumulative annual frequency is known as annual duration curve depicted in Fig. 3 right. In the centre of each bin, an operating point is evaluated with regard to the heat pump performance, i.e. heating capacity and COP at these specific operating conditions, which is determined by the standard component testing described in the preceding chapter. The operating point is considered to characterise the heat pump operation of the whole bin.

The temperature difference between the indoor design and the actual outdoor air temperature, i.e. a distance in the diagram, corresponds to a heat load. Thus, the area under the cumulative frequency, i.e. the cumulative heating degree hours, correspond to an energy, i.e. the space heating energy needs, since the heat load (temperature difference) is integrated over the time. Therefore, a weighting of the operating conditions with this energy fraction evaluated by the areas or heating degree hours of the bin and a subsequent summation of all bins delivers the seasonal performance.

The contribution of an electrical back-up heater can be estimated by an evaluation of the respective area in the cumulative frequency diagram, in Fig. 3 the area BU with the balance point as upper boundary and eventually a low-temperature cut-out as low boundary depending on the operation mode of the back-up heater. In Fig. 3 an alternative parallel operation of the heat pump and the back-up is depicted.

For the domestic hot water operation, a similar calculation can be performed based on the weighting with the domestic hot water requirement and testing results of the hot water testing according to EN 255-3.

The overall seasonal performance can be calculated by an energy related weighting of the seasonal performance factors for space heating and domestic hot water operation, respectively. To derive the seasonal performance factor for the generator or system, additional losses (e.g. due to storage components) and additional auxiliary energy expenses, which are not or not entirely considered in the standard testing (e.g. the source pump), have to be taken into account.

2.3.2 PHPP 2004 Tool

2.3.2.1 Background

The "Passiv-Haus-ProjektierungsPaket" (PHPP) is a stationary monthly energy balance based on the former European standard EN 832. The objective is to deliver the primary energy demand of a building. Therefore, the installed HVAC system is considered, as well. The calculation is implemented in MS Excel and consists of a collection of Excel spreadsheets for the different building and system components. The main intention of the calculation is to prove the compliance with the German building standard "Passivhaus". However, the PHPP is also a powerful tool for system design purposes in general. One of the Excel sheets in the file of PHPP 2004 is dedicated to compact units.

2.3.2.2 Summary

The approach has the objective to evaluate the electricity input to the compact unit necessary to cover the space heating and DHW requirement. The calculation is based on the design heat load evaluated by two typical days, one clear cold day at outdoor design temperature (e.g. -6°C) with high solar irradiation and one cloudy day at moderate temperature (e.g. -1°C) with lower solar irradiation are calculated, i.e. solar gains are considered in this calculation of the design heat load. The maximum of the two calculated heat loads is taken as design heat load and the load profile is considered linear between the design heat load and the end of the heating period. This leads to a duration curve as presented as grey triangle in Fig. 4 (90°clockwise turned transformation of the curve depicted in Fig. 3). The domestic hot water load is considered constant over the whole year and depicted as dark grey rectangular in Fig. 4. The electrical back-up energy use is calculated for an alternative parallel operation of heat pump and back-up heater as depicted in Fig. 4. The heat pump is characterised by the heating capacity at the balance point $\phi_{HW,hp,bal}$. If $\phi_{HW,hp,bal}$ is higher than the sum of the SH and DHW load $\phi_{HW,nd,max}$ no electrical back-up contribution results, since the balance point is lower than the design outdoor air temperature.

If $\phi_{HW,hp,max}$ is lower than the sum of domestic hot water load and space heating load $\phi_{HW,nd,max}$ the direct electrical fraction is calculated as the area of the upper red triangle in Fig. 4 by the equation [1]

$$Q_{HW,bu} = Q_{H,dis,in} \cdot \frac{(\phi_{HW,nd,max} - \phi_{HW,hp,bal})^2}{(\phi_{HW,nd,max} - \phi_{W,nd})^2} \qquad [J]$$

eq. 2

where

$Q_{HW,bu}$	produced heat energy by an electrical back-up heater for SH and DHW	[J]
$Q_{HW,dis,in}$	Heat energy need of the space heating distribution system	[J]
$\phi_{HW,nd,max}$	max. heat load for space heating and DHW operation	[W]
$\phi_{HW,hp,bal}$	heating capacity of the heat pump at the balance point	[W]
$\phi_{W,nd}$	heat load for DHW operation	[W]

The rest of the energy requirements for space heating and DHW are covered by the heat pump. To determine the electrical energy consumption for the heat pump operation, two COP values are used, one for the wintertime (same COP for space heating and DHW operation) and one for the summertime.

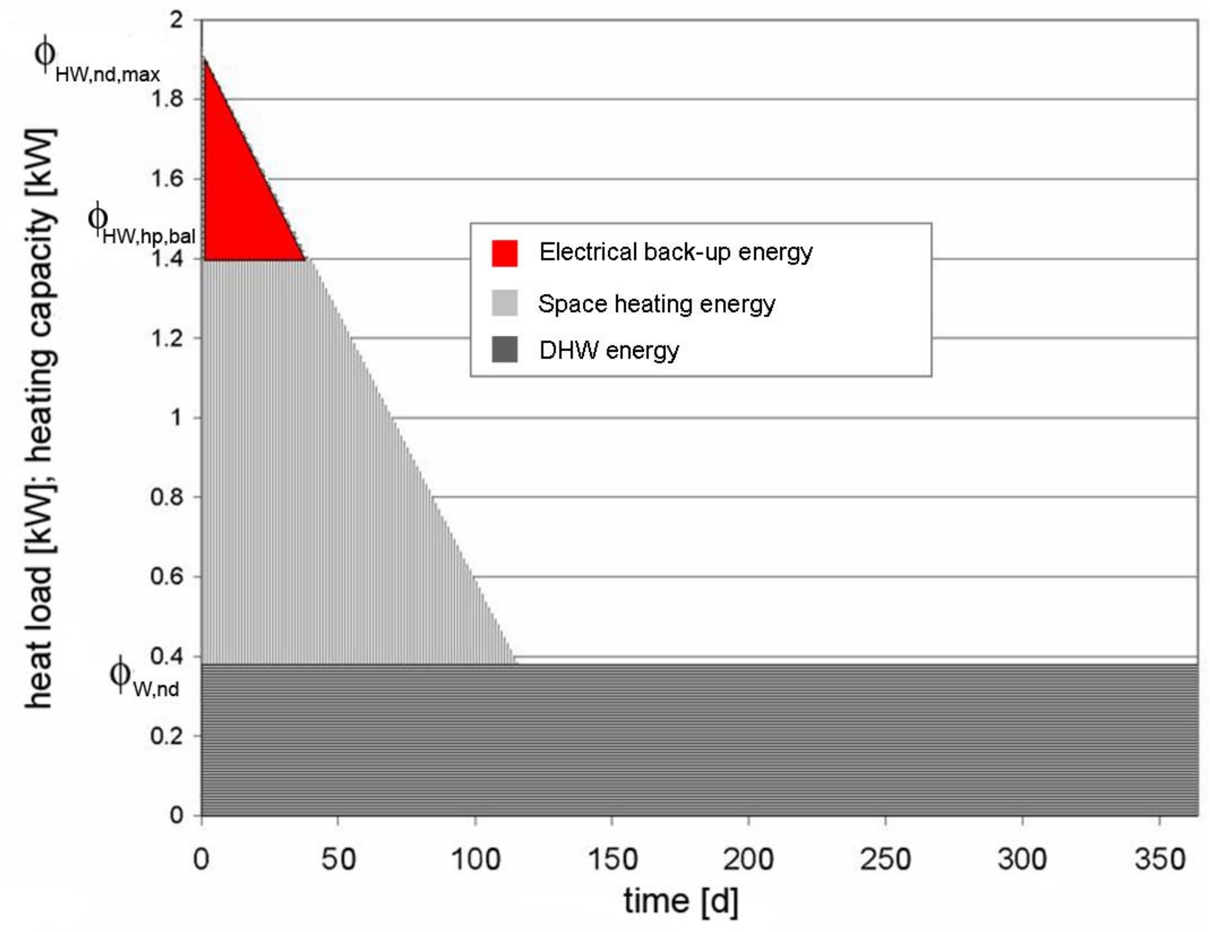


Fig. 4: Calculation approach integrated in the PHPP (based on [1])

The splitting in winter and summer period is comparable to a two-bin method, where the $COP_{HW,winter}$ and $COP_{HW,summer}$ are considered as performance factors for the respective periods. The length of the winter period is defined by the heating period, which is calculated by the space heating energy requirement divided by the design heat load evaluated as described above. If more than one heat generator is installed, e.g. the compact unit and an additional boiler, the fraction of each generator is an input.

2.3.2.3 Approximations

- Both heating capacity and COP values are set to the same value for SH and DHW mode in winter operation without differentiation of the required temperature levels. The COP values are taken as performance factors for each the winter and summer operation.
- The default value for a summer COP is considered lower than the winter COP according to the PHPP manual [1], since compressor heat losses of the heat pump may be used for space heating in wintertime.
- Operation limits of the heat pump are not considered, i.e. the hot water temperature is postulated to be below the operating limit of the heat pump or back-up operation is included in the testing.
- The electrical back-up heater is operated in alternative parallel operation, i.e. the heat pump delivers the maximum heat possible and back-up delivers the missing energy.^

2.3.2.4 Assessment

The implemented method is easy-to-use, since only two COP values and the heating capacity at the balance point, which has to be known, is required. Based on the approximations the approach seems feasible, if both source temperature and sink temperature of the space heating and the DHW are nearly the same and mostly constant over in the winter operation range and testing is representative for the average winter or summer conditions. Therefore, the approximations are best fulfilled for specific types of compact units with a heat pump with installed ground-to-air heat exchanger (quite constant source conditions at inlet heat recovery), which uses only the exhaust air of the heat recovery (constant source conditions for the heat pump after the heat recovery), and air heating distribution system (constant sink temperature in the range of the DHW temperature). However, due to the heat recovery unit the inlet temperature range of the heat pump is limited without or with moderate outdoor air fraction, and increasing DHW energy shares decrease the impact of the supply temperature level on the overall performance. Testing has to deliver COP-values, which reflect the average conditions of the combined space heating/DHW operation and DHW operation in summer conditions, since these COP values are taken as performance factors for the respective operating periods. Actually, the calculation itself does not evaluate the seasonal performance of the heat pump or compact unit, but basically the required electrical back-up energy input due to a lack of capacity for an alternative parallel back-up operation. Back-up energy due to a temperature operation limit of the heat pump is not considered or has to be included in the test results. If the compact unit is of a different type, e.g. with a hydronic floor heating emission system at low temperature conditions or large outdoor air fraction, it is difficult to perform the calculation, since temperature conditions are not explicitly taken into account.

2.3.2.5 Update of the calculation method in PHPP 2007

After the final project meeting, updates depicted in Fig. 5 of the PHPP calculation method to be implemented in the PHPP Tool 2007 have been presented.

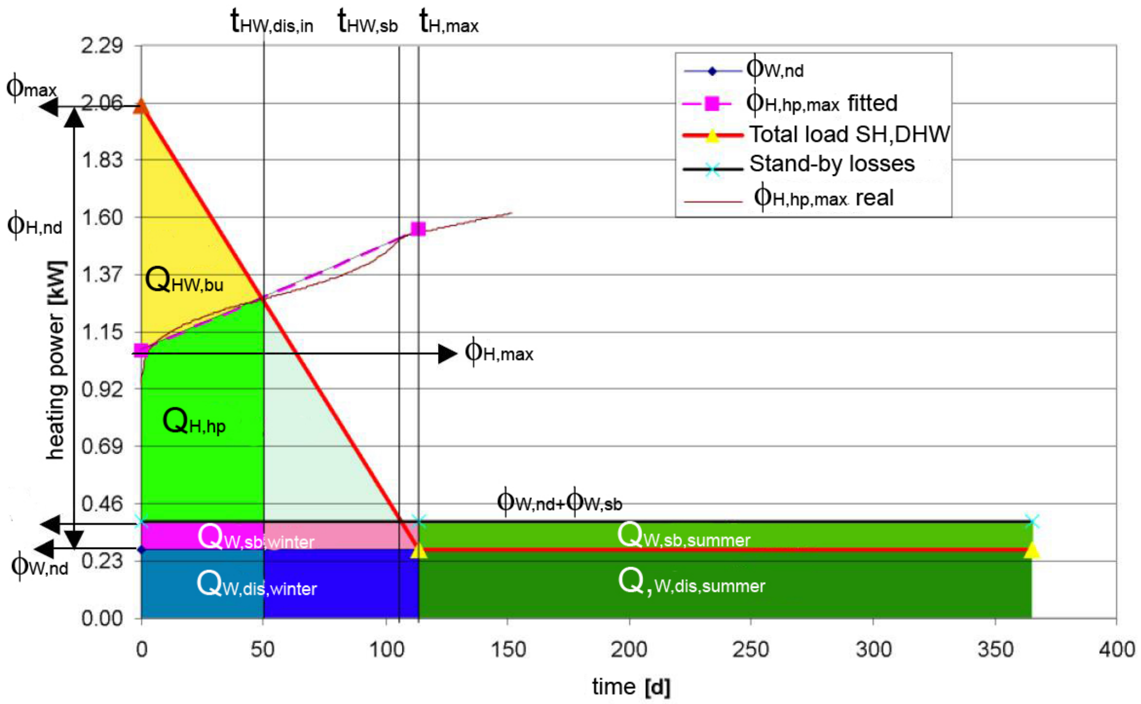


Fig. 5: Calculation approach to be implemented in the PHPP 2007 (based on [52])

Changes to the previous version of the PHPP 2004 in the previous chap. 2.3.2.1 – chap. 2.3.2.4 comprise the following extensions:

- Separation of DHW energy need and DHW stand-by energy

  DHW stand-by energy is considered in the frame of the calculation of the compact unit as shown by the different areas in Fig. 5. Stand-by losses of the DHW operation reduce the energy need for the space heating operation.
- Heat pump heating capacity

  Instead of the constant heating capacity used in the PHPP 2004, a linearised heating capacity of the heat pump is used.
- Consideration of ground-to-air heat exchanger, additional outdoor air volume

  The impact of the temperature rise of the ground-to-air heat exchanger on the heating capacity and COP of the heat pump is considered evaluating the outlet temperature of the ground-to-air heat exchanger, which is calculated by an effective heat recovery efficiency. In case of an additional outdoor air volume flow rate, which does not pass the ventilation heat recovery unit, the temperature at the entry of the heat pump is evaluated by an enthaly balance based on the input of the volume flow rates.
- Priority for space heating and DHW

  The thermal power is calculated based on the running time for space heating, DHW and stand-by operation. For the calculation the priority for the space heating or DHW operation is taken into account. For instance, if the priority is on DHW operation, the heat pump can only produce space heating energy in the remaining time (if any time is remaining, at all).
- Calculation of SPF-values based on the temperature depend COP characteristic

  The temperature impact on the COP characteristic is taken into account by evaluating the COP from the standard testing for the single areas shown in Fig. 5. The SPF or the total electrical energy input, respectively, is calculated by summing-up the electrical energies for the single areas.

2.3.3 Excel-Tool WPEsti

2.3.3.1 Background

In charge of the Swiss MINERGIE® association (www.minergie.ch), the Swiss Heat Pump Organisation "Fördergemeinschaft Wärmepumpen Schweiz FWS" (www.fws.ch) and the cantonal office for waste, water and energy AWEL in Zurich (www.awel.zh.ch), the engineering company Huber Energietechnik AG (www.igjzh.com) has developed the Excel-Tool WPEsti [3]. The tool can be downloaded from the respective websites as Excel spreadsheet. It can be used separately, but is also integrated in the calculation procedure to prove compliance with the Swiss building standard MINERGIE®.

The calculation is based on the seasonal performance calculation method developed in the NTH project of SFOE [16]. The description refers to the version V. 2.0 which has been published in April 2006. The calculation covers heat pump systems for space heating-only operation, domestic hot water-only operation or alternate combined operation. The ventilation system is not treated but covered in a separate Excel-Sheet of the MINERGIE® calculation procedure, however, without a direct link between the calculations. Solar energy contributions or ground-to-air heat exchanger are not considered.

2.3.3.2 Summary

In the following an overview of the calculation is given, more details to the approaches are described in [4]. The calculation is based on the input data of the Swiss building regulation SIA 380/1 [26], the old heat pump standard testing acc. to EN 255-2 [20] described in chap. 2.2.1.1 and further product data, e.g. source pump power for brine-to-water heat pumps or storage size.

The calculation can be split in two parts: On the one hand the calculation of the electrical back-up energy is performed by a detailed power balance and on the other hand, the SPF values for SH, DHW and overall are calculated based on the COP values from standard testing.

Back-up calculation

The back-up energy is calculated by a power balance of the space heating and DHW load on the one hand and the heating capacity of the heat pump on the other hand using the following bin scheme for air-to-water heat pumps: One cumulative bin below -7°C, 1 K temperature bins between -7°C and 7°C and each one cumulative bin [8..20°C] and above 20°C. The heating capacity for air-to-water heat pumps is interpolated for the source temperature based on the standard test points A-7/W35, A2/W35 and A7/W35 according to the bin scheme. For brine-to-water and water-to-water heat pumps, a constant heating capacity at B0/W35 and W10/W35 is used, respectively. The heating capacity is corrected with cumulated relative losses (e.g. cycling, carter heating, heating storage) listed in Tab. 2 for the respective operation mode.

The design heat load is recalculated by the transmission and ventilation losses and the space heating energy need according to SIA 380/1, the heating degree days and outdoor design temperature according to the equation [4]

$$\phi_{H,nd}(\theta_{oa} = \theta_{oa,des}) = \frac{(Q_{tr} + Q_{ve}) \cdot A_E \cdot 1000\,[W/kW]}{3.6\,[MJ/kWh] \cdot DD_{H,20/12} \cdot 1.065 \cdot 24\,[h]} \cdot (20°C - \theta_{oa,des}) \quad [W]$$

eq. 3

where

$\phi_{H,nd}$	design heat load (estimated, not according to SIA 384.201)	[W]
θ_{oa}	outdoor air temperature	[°C]
$\theta_{oa,des}$	outdoor design temperature	[°C]
Q_{tr}	space heating energy need due to transmission losses	[MJ/m^2]
Q_{ve}	space heating energy need due to ventilation losses	[MJ/m^2]
A_E	energy reference area	[m^2]
$DD_{H,20/12}$	heating degree days at indoor temperature 20°C up to the heating limit 12°C	[Kd]

This value is given as proposal. If the input of the heat load is less than a minimum value of 90% of the proposal, the minimum value is used for the calculation. An average DHW load derived by the annual DHW energy need according to SIA 380/1 is added to the space heating load.

If the heat pump heating capacity, which is multiplied by weighted efficiencies for space heating and DHW operation according to eq. 5 is below the sum of SH and DHW load, the resulting lack of capacity is multiplied with the cumulative hours of the bin. Cumulative frequency is evaluated based on data of Zurich SMA Design Reference Year (DRY), Davos and Lugano. Other sites are corrected based on the design outdoor temperature. The cumulative frequency is diminished by the fraction of solar gains, which are calculated based on the SIA 380/1 ratio of the space heating energy need and the sum of transmission and ventilation losses.

The resulting back-up energy is entirely accounted to the space heating operation, unless the heating capacity of the heat pump is smaller than the DHW load. Further back-up energy for the DHW operation is calculated based on the heat pump operation limit temperature. The back-up energy is corrected with electrical back-up losses based on the configuration given in Tab. 2.

SPF calculation:

For air-to-water and space heating mode a time weighting of the COP values from standard testing for three bins with operating points corresponding to standard test points is done. The

COP values are corrected to the design supply temperature of the heating system by the method of constant exergetic efficiency given in Appendix C and regarding differing mass flow conditions in testing and operation (see chap. 4.2.2). For brine-to-water and water-to-water heat pumps the COP on the test point is set to the SPF values without weighting, but with corrected source and sink temperature.
In DHW operation, the COP at the respective test point A7/W50, B0/W50 and W10/W50 of the space heating characteristic is corrected to the DHW temperature, that can be reached by the heat pump and taken as SPF. For ground source heat pump, the source temperature is corrected, as well.
For both operation modes the SPF values are multiplied by the factor acc. to the equation [4]

$$\eta_H = 1 - \sum k_{H,ls} \qquad \eta_W = 1 - \sum k_{W,ls} \qquad [-]$$

eq. 4

where

η	efficiency for the space heating mode or DHW mode respectively	[-]
k_{ls}	percental losses for the respective operation mode acc. to Tab. 2	[%]

where the respective percental system losses k_{ls} are taken from Tab. 2. Therefore, the resulting seasonal performance factor is related to the energy need and not to the produced heat energy (corresponding to the system boundary JAZ in Fig. 38). Finally, the seasonal performance factors are weighted by the respective energy needs for space heating and DHW to an overall seasonal performance factor.

Tab. 2: Default-values for system losses in WPEsti (translated from [4])

Type of loss	$\mathbf{k_{ls}}$
Losses heating buffer storage	**3 %**
Losses DHW storage	**4 %**
Additional losses storage loading with instantaneous electric heater	**2 %**
Additional losses storage loading with electric back-up heater installed in the storage	**4 %**
Additional losses storage loading with electrical back-up heater installed in the storage and inactive during loading of the heat pump	**1 %**
DHW additional electricity demand with back-up heater in parallel operation	**5 %**
DHW additional electricity demand with back-up heater blocked at heat pump operation	**0 %**
Start-up losses of the heat pump, carter heating in heating mode	**4 %**
Start-up losses of the heat pump, carter heating in DHW mode	**4 %**

2.3.3.3 Approximations

- Weighting factors for the calculation of the SPF are time-based values of the cumulative frequency and not energy-based.
- The heating capacity for SH and DHW is approximated by the SH capacity at a temperature level of the testing at 35°C.
- Back-up energy is calculated with the design heat load, i.e. no solar and internal gains are taken into account to determine, if back-up is required or not.
- Redistribution of solar and internal gains are proportional to the outdoor air temperature frequency
- Supply temperature for the space heating is set constant to the design values, i.e. no heating curve is taken into account
- Cumulative frequencies for the site are derived based on two climatic regions (correction by outdoor design temperature)
- Conservative estimation of back-up energy use: If the set point for hot water temperature

is equal to the maximum DHW temperature reachable with the heat pump, still 5% back-up energy use for DHW operation is calculated.

- Brine-to-water operation is calculated at constant source and sink temperature
- Losses of the system components are taken into account as relative values (percentage)
- Running time restrictions of the heat pump, i.e. a cut-out time of the electricity supply by the utility is not considered.

2.3.3.4 Assessment

A detailed calculation is accomplished by a highly resoluted heat load balance in 1 K steps to determine the contribution of heat pump and back-up system. The calculation takes into account the main impacts on the heat pump performance and is presented in an ergonomic and easy-to-use Excel spread sheet. In order to reduce required input values, default values are used for impacts of minor importance on the system performance.

However, the following assumptions rather restrict the tool to the calculation of low-energy houses as e.g. MINERGIE® houses with floor heating system regarding the objective to prove compliance with the standard, where a conservative estimation is desired.

- In general a worst case estimation is calculated by the use of a constant supply temperature fixed at the design value.
- No consideration of the ventilation system. This is due to the fact that in the MINERGIE® calculation, the ventilation system is considered separately without interaction to the heat pump. As long as the impacts on the inlet temperature by the ventilation or other installed preheating systems are considered in the COP and heating capacity data, this assumption is feasible (see chap. 5.2.3). However, COP and heating capacity characteristic of the heat pump may depend on the volume flow rate, as well.
- Fixed default losses do not allow to calculate variants nor to differentiate system layouts by losses.
- Fixed heating capacity at 35°C. For higher supply temperature, correction of COP values of 35°C sink temperature by the constant exergetic efficiency method given in Appendix C may become rough, i.e. the approximations seem less suited for air heating systems used in ultra-low energy houses.
- Time-based weighting of the COP characteristic: By using the frequency and not the energy for weighting the COP values, the impact of change in heat load is neglected.
- The heating capacity for the space heating and DHW system is set to the same value of the space heating temperature conditions regardless of different temperature requirements for space heating and DHW. This approximation gets worse with higher ratios of DHW to SH requirement, e.g. in MINERGIE-P® houses, depending on the change in heating characteristic.

Summarising, the calculation seems well suited for the purpose of proving the MINERGIE® compliance, since distribution temperatures are commonly low (i.e. the impact of temperature change of supply temperature is not so dominant). However, for more general calculation, in particular the approximation of a constant sink temperature may be critical.

2.3.4 FHNW method for heat pumps with combined SH and DHW

2.3.4.1 Background

The FHNW-method has been developed in an SFOE research project [6] as Swiss national contribution for the IEA HPP Annex 28 [5]. The calculation is also based on the method developed in the NTH project of SFOE [16].
Principles for the method were

- Hand calculation, e.g. no computer simulation required
- Based on publicly available data and existing standards
- As far as possible transparent calculation without use of default values
- Applicable to the most common system configurations on the market

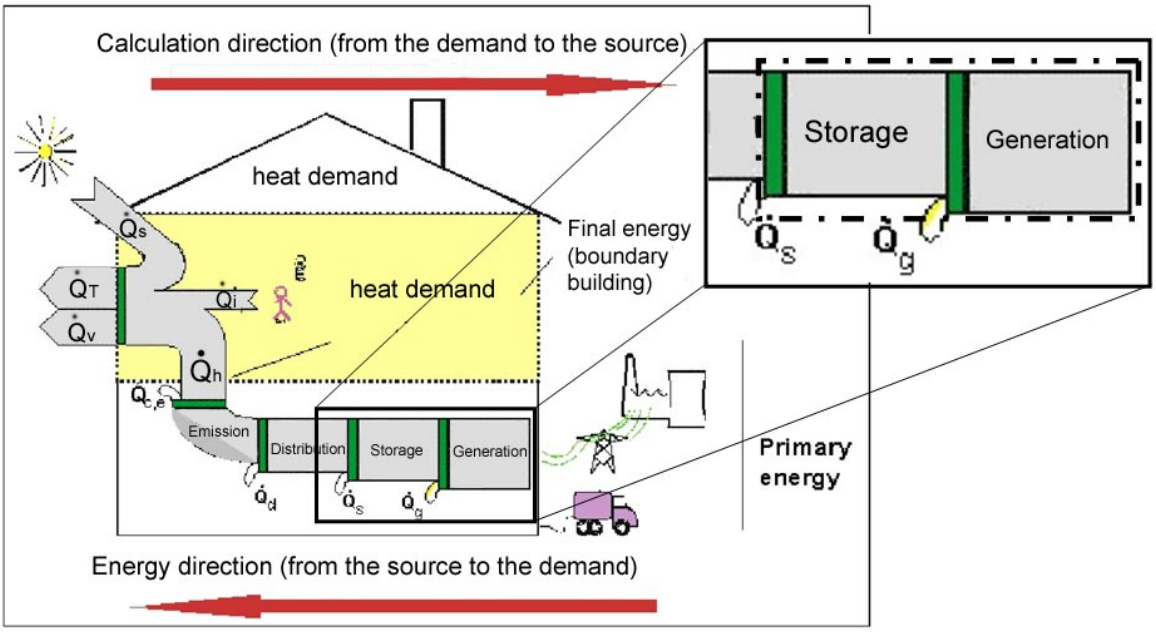


Fig. 6: System boundary for the calculation method in the calculation scheme of the prEN 15316-1 [12]

The objective and focus was the extension to simultaneous combined operating heat pumps, where the heat pump produces space heating and DHW energy at the same time, e.g. using refrigerant desuperheating or condensate subcooling and cascade cycle layouts. The results were intended as basis for an uniform international standard, so the system boundary has been chosen to include the heat generator, additionally installed storage systems and back-up heaters in compliance with the prEN 15316-1 [12] (formerly prEN 14335-1).

In Fig. 6 the system boundary is depicted in the calculation scheme of the prEN 15316-1, i.e the calculation comprises the heat pump, attached space heating and DHW storages and eventually installed back-up heaters. The input is the space heating energy need of the distribution system.

The prEN 15316-1 describes the general calculation scheme for the EU Directive on the Energy Performance of Buildings (EPBD) [13], which incorporates the building and building technology calculation to deliver an overall energy performance in terms of primary energy consumption or carbon emission. The calculation method has been implemented in the heat pump draft (prEN 15316-4.2 [51]) of the EPBD standard series. By the end of March 2006 received comments on the draft from a six-month public enquiry in the 25 member states of CEN have been implemented. The final version is expected to be published by mid of 2007.

2.3.4.2 Approximations

The method implements the bin method as described in chap. 2.3.1.1. A more detailed description of the method is included in the sample calculation for compact units given in chap. 4.2.

Basically the following assumptions are made:

- Redistribution of the space heating energy need of the distribution system according to the system boundary in Fig. 6 is made by heating degree hours and an upper temperature limit for heating (see details in chap. 5.4)

- Part load operation, i.e. cyclic losses, are only considered, if test data are available to quantify eventual losses, otherwise, they are neglected.
- Heat losses of the heat pump over the envelope are neglected unless values for the heat losses exist.
- Controller impact on the performance is evaluated by default situations, e.g. the sink pump is either coupled to the running time of the heat pump or is running through the entire heating period.

2.3.4.3 Assessment

The calculation method implies the above simplifications in order to keep the calculation and the input data requirement simple.

Control effects are hard to assess, since normally, the information is not available, and therefore, default settings are applied.

One shortcoming with regard to low and ultra-low energy houses may be the redistribution of the energy needs to the bins that is done dependent on the heating degree hours and thereby mainly dependent on the outdoor temperature. However, a correction is applied by defining an upper temperature limit for heating. The assessment of this approach is described in chap. 5.4.

A detailed comparison to results of the field monitoring is given in chap. 5.

Tab. 3 gives a summary of the calculation methods.

Tab. 3: Comparison of existing calculation methods for compact units

	PHPP	MINERGIE	FHNW-method
Energy requirement	Effective requirement of the space heating and DHW system (derived within the PHPP on other sheets)	SIA 380/1 incl. the ventilation heat recovery	Depending on input data (SIA 380/1 compliance or optimisation) recovered ventilation losses in- or excluded
System losses	Storage/ distribution	Default values acc to Tab. 2	Electrical input acc. to EN 255-3, (U·A)-value or Swiss energy regulation
Change of supply temperature	No	Constant supply temperature or storage requirements	Heating curve or storage requirements
Change of source temperature	No	Only heating operation	Heating and DHW operation, if required
Ventilation	Subtracted before calculation constant heat recovery efficiency	Subtracted before calculation constant heat recovery efficiency	Depending on input data subtracted with constant heat recovery efficiency
Impact of ventilation on inlet temperature heat pump	No	No	Depending on measured characteristic of compact unit
Solar energy by collector	Subtracted from the requirement within the calculation	Not considered	Subtracted from the bin requirement within the calculation
Back-up configuration	No (default efficiency 1)	Default values for different configurations	Input of efficiency, default 0.95
Redistribution of used solar gains	No redistribution (only 1 bin for SH) but heat load calculation integrated	Constant ratio in every bin (proportional)	According to heating degree hours with adapted/default heating limit (annual) or information of SIA 380/1 (monthly), see 5.4
Resolution of bins	1 bin SH/DHW; 1 bin DHW-only	3 bins space heating / 1 bin DHW for A/W	3 bins space heating / 4 bins DHW for A/W
Back-up energy	Evaluation of balance point within the calculation (power balance)	Balance of building demand, hot water demand and heating capacity heat pump	Evaluation of running time based on SH/DHW energy needs or balance point
Source pump	Nominal power and running time on other point of the calculation	Explicitly for B-W, W-W, otherwise included in COP	Explicitly for brine-to-water, water-to-water, otherwise included in COP
Sink pump	Nominal power and running time on other point of calculation	Not included (System boundary "generator"), see Fig. 17	Explicitly by nominal power (System boundary “system”), see Fig. 17
Alternate combined operation	Yes	Yes	Yes
Simultaneous operation	Limited (no operation modes)	Limited (no operation modes)	Yes (additional testing required)
Calculation period	Year	Year	Year or month (see 5.4)

3 HTA TEST METHOD FOR HEAT PUMP COMPACT UNITS

The University of Applied Sciences of Lucerne (HTA) has developed a set of testing standards defining testing methods and procedure, aspiring to keep the duration of tests short while delivering all the data required for the calculations.

Elaboration of the details of the testing standards proved more difficult than had been anticipated. There is a maze of standards in Switzerland, and the situation in Germany is unclear as well. Although there is a European Convention (EN 13141-7 [37]), the regulations of the DIBt (Deutsches Institut für Bautechnik, Berlin) are still in effect. We have heard, furthermore, of plans to introduce a national German convention in Germany beside CEN efforts. This will, however, take some time. In addition, the standards for heat pumps have been revised. It is not yet clear whether these conventions are to apply also for compact ventilation units equipped with heat pumps using the return air. Should these standards apply also for compact ventilation units equipped with heat pumps using the return or outside air, tests will need to be performable at outside temperatures of down to -15°C. In developing the test standards care was taken to integrate current international standardisation efforts in the areas of compact ventilation units, domestic hot water tapping models, heat pumps and acoustics. Air conditions were specified based on experience values if they had not been defined. The resulting proposal is to be introduced into international standardisation commissions in order to develop a European convention.

3.1 General approach

In future, testing of compact ventilation units will be influenced by work on international conventions in the realms of residential ventilation, heat-pump and compact air-conditioning units. IEA HPP Annex 28 "Test Procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating" [5] defines a testing procedure for heat pump compact units which takes the content of current normative discussions into account, minimises the duration of tests and thus yields lowest possible testing costs. Although various work has not yet been finalized, a detailed set of testing standards has been developed which covers all the important points required for permits (for the time being predominantly in Germany). Additionally, acoustical measurements and research in the areas of operation, maintenance and filter bypass leakage are being performed.

3.2 Test rig

The work relating to conventions complicated the design and realisation of the test rig. The test rig was designed and built to satisfy present and future requirements as completely as possible, especially key data regarding air conditions. With this test rig all important operating conditions for comfort ventilation systems can be replicated. The following types of appliances (and combinations thereof) can be tested within a volume flow range of 50 m^3/h to 1200 m^3/h:

- units with supply and return air, heat recovery and exhaust-air heat pump to heat the supply air
- units with return air, exhaust-air heat pump to heat domestic hot water
- units with return air, exhaust-air heat pump which heats domestic hot water and/or transfers heat to another water-bearing system
- units with supply and return air and heat recovery (recuperative and regenerative)

Fig. 7: Acoustic chamber for tests (left) and test equipment (right) of the HTA test rig

As the test facility is especially designed to accommodate acoustic measurements, the acoustic characteristics of compact units can be tested during thermal performance. Thanks to the 2-part, double-walled testing chamber, the base sound level can be minimised, allowing measurement of the sound power level radiated by the casing in the testing chamber (i.e. in any room where the unit is installed) – as well as the sound power level emitted through the air connections into the ducts.

The air can be conditioned to the specifications given in Tab. 4 for testing purposes

Tab. 4: Test points for compact ventilation unit with ground-air heat exchanger

Air conditions	**outside air**	**return air**	**ambient air in room**
Temperature [°C]	-12 to +21	21 to 26	10 to 26
Humidity [% r.h.]	70 to 80	30 to 60	-
Air volume flow [m^3/h]	50 to 1200	50 to 1200	-

3.3 HTA Test method for heat pump compact units

3.3.1 Black Box Testing

The general approach for the test procedure is a black-box testing, i.e. only results that can be measured from outside the system are taken into account. This concept has the advantage that different internal system configurations can be treated in the same way. Otherwise, testing would have to refer to the internal system configuration and would have to be adapted for each system. Due to the variety of different system configurations on the market, this should be avoided as far as possible.

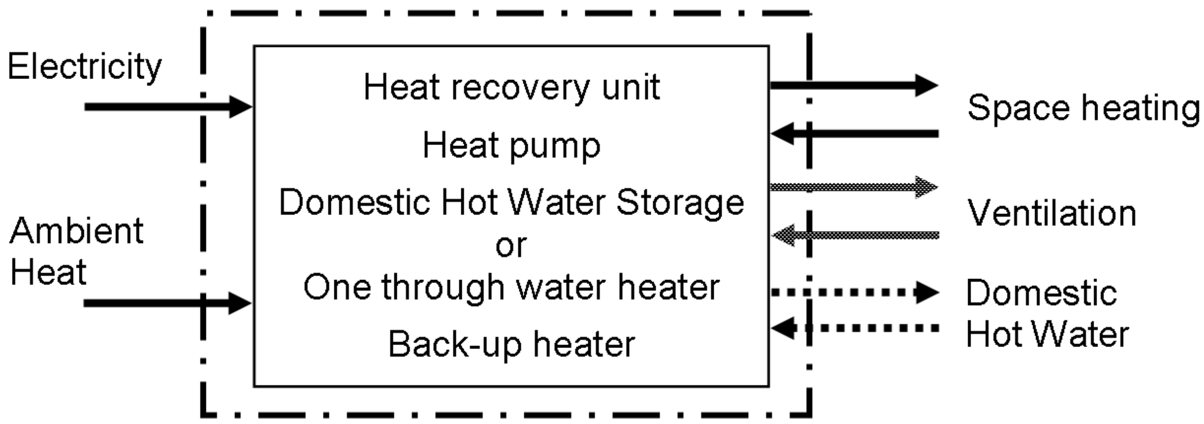


Fig. 8: System boundary for the black-box testing

However, that does not imply that the internal system configuration cannot be considered in the evaluation of the test results as far as possible, if known. Fig. 8 depicts the system boundary of the approach for the test procedure.

3.3.2 Single Tests for the different functionalities

A comprehensive appraisal of a compact ventilation unit equipped with heat recovery and a heat pump comprises the following tests:

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

A number of the items of the testing standards are listed below. Details can be found in the set of testing standards in Appendix A. Calculations of the individual parameters are also to be found in the body of the testing standards and/or in the individual conventions.

3.3.2.1 Assessment of air-tightness (testing for internal and external air leakage)

Units are tested for internal and external air leakage according to EN 308 [38] and EN 13141 [37]. Alternatively, internal leakage can be ascertained using tracer gas. This procedure determines the leakage mass flow from the return air to the supply air during operation.

3.3.2.2 Determination of the number of measuring points and volume flows

The correct operating range is determined according to specifications established by the Passive House Institute, Darmstadt[1] and is calculated as follows:

The maximum volume flow rate is determined at which the unit can deliver a pressure increase of 100 Pa·$1{,}3^2$=169 Pa in its highest operating level or maximum fan speed.

The minimum volume flow rate is determined at which the unit can deliver a pressure increase of 49 Pa in its lowest operating level at minimum fan speed.

If the ratio of the maximum to the minimum volume flow is greater than 1.6:1, the range bet-

[1] **Passivhaus-Institut**, Rheinstr. 44/46, D-64283 Darmstadt

ween the minimum and maximum flow must be subdivided into subranges and within each subrange a nominal volume flow is determined. For the determination of the subranges and their respective nominal flows the client can choose between the DIBt[2] and the Passive House Institute guidelines.

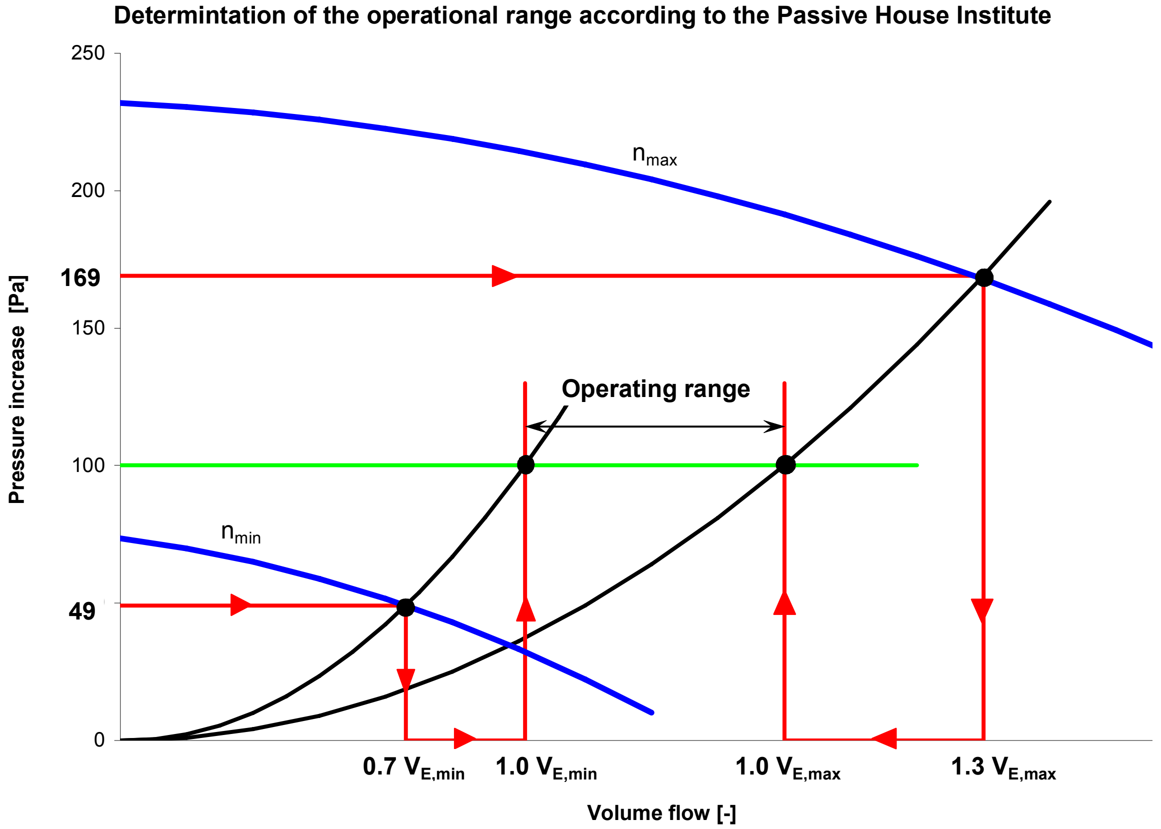


Fig. 9: Determination of operating range of test item

3.3.2.2.1 Determination according to the DIBt

The required number of subranges *n* is determined by:

$$n \geq 2.13 \cdot \ln\left(\frac{\dot{V}_{max}}{\dot{V}_{min}}\right)$$

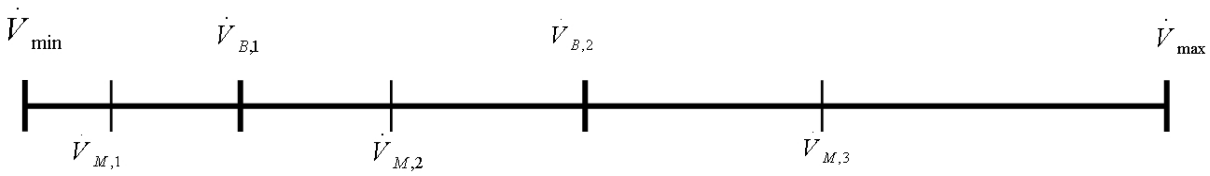
The upper interval boundary $\dot{V}_{hlim,1}$ is determined by:

$$\dot{V}_{hlim,m} \geq \dot{V}_{min} \cdot \sqrt[n]{\left(\frac{\dot{V}_{max}}{\dot{V}_{min}}\right)^{m}}$$

The nominal interval volume flow $\dot{V}_{M,m}$ is determined by:

[2] DIBt: (German Institute for Construction) **Deutsches Institut für Bautechnik**, Kolonnenstraße 30 L, D10829 Berlin

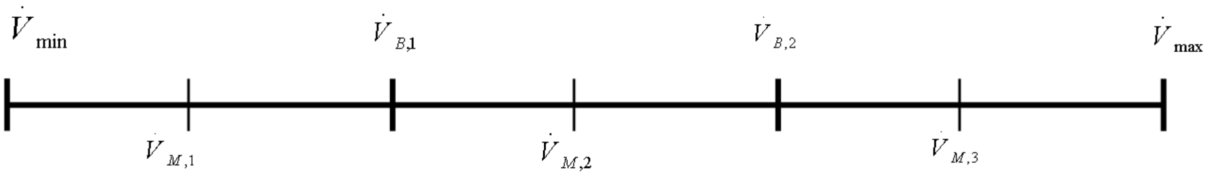
$$\dot{V}_{M,m} \geq \dot{V}_{\min} \cdot \sqrt[2n]{\left(\frac{\dot{V}_{\max}}{\dot{V}_{\min}}\right)^{(2m-1)}}$$



If the ratio between the higher and lower values is larger than 1,6:1, the volume flow range must be subdivided into equal subintervals, such that the corresponding ratios remain ≤ 1,6:1. Within these subranges measurements will be carried out at the nominal interval volume flow value.

3.3.2.2.2 Determination according to the Passive House Institute

If the ratio between the higher and lower values is larger than 1.6:1, the volume flow range must be subdivided into equal subintervals, such that the ratio ≤ 1.6:1 is maintained. Within these subranges measurements will be carried out at the mean volume flow value.



3.3.2.3 Testing the ventilation

Characteristic curves for air volume flow rate vs. pressure must be determined for both the supply and return air flows according to DIN 24163 [39].

For units with **fixed-speed fans** characteristic curves for air volume flow rate vs. pressure must be recorded for each operating level. Data points for each curve must be obtained within the stable operating range for pressure increments not over 30 Pa.

For units with **continuously-variable-speed fans** the maximum and minimum volume flow rates must be established according to the above procedure. For each of these two flow rates characteristic curves for air volume flow rate vs. pressure must be recorded. Data points for each curve must be obtained within the stable operating range for pressure increments not over 30 Pa. No measurements are required for curves between the maximum and minimum flow rates.

For units with **constant volume flow-rate control fans** the maximum and minimum volume flow rates must be established as described above. For each of these two flow rates characteristic curves for air volume flow rate vs. pressure at constant fan speed (fixed setting) must be recorded. Data points for each curve must be obtained within the stable operating range for pressure increments not over 30 Pa. No measurements are required for curves between the maximum and minimum flow rates.

3.3.2.4 Thermodynamic Tests

Thermodynamic tests are performed according to EN 14511 [8]. At present return air and outside air conditions are transferred from EN14511 [8]. If normative discussions in Germany and Austria result in different test points, these will have to be adapted. Standard test procedure requires 3 temperatures per determined volume flow for thermal tests. (cf. Tab. 5).

Tab. 5: Test points for a compact ventilation unit without ground-air heat exchanger

	Outdoor air temperature [°C]		Exhaust air temperature [°C]		Heating		DHW	Comment
	Inlet dry bulb tem perature $\theta_{oa,in}$ [°C]	Inlet wet bulb tem perature $\theta_{oa,wb}$[°C])	Inlet dry bulb tem perature $\theta_{ea,in}$[°C]	Inlet wet bulb tem perature $\theta_{ea,wb}$[°C]	Inlet temperature $\theta_{H,w,in}$[°C]	Outlet temperature $\theta_{H,w,out}$[°C]	Outlet tempe rature $\theta_{W,w,out}$[°C]	
1	**7**	6	20	12	40	**45**	60	heating
2	**-15**	-	20	7*	a	**45**	60	heating, optional
3	**-7**	-8	20	7	a	**45**	60	heating
4	**2**	1	20	10	a	**45**	60	heating
5	**-7**	-8	20	7	a	**35**	60	heating
6	**7**	6	20	12	a	**35**	60	heating
7	**15**	10	20	14	a	**35**	60	heating optional
8	**7**	6	20	12	-	**-**	60	DHW

A The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)
* The wet bulb temperature is based on a humidity gain of 2.5 g/kgdry air per outside temperature step based on the given rating point 20(12) in EN 14511

Tab. 6: Test points for compact ventilation unit with ground-air heat exchanger

	Outdoor air temperature [°C]			Exhaust air temperature [°C]		Heating		DHW	Comment
	Dry bulb outdoor tempe-rature θ_{oa} [°C]	wet bulb outdoor tempe-rature $\theta_{oa,wb}$ [°C]	Inlet temperature after ground-air-HX $\theta_{oa,ihru,n}$ [°C]	Inlet dry bulb tem-perature $\theta_{ea,in}$ [°C]	Inlet wet bulb tem-perature $\theta_{ea,wb,in}$ [°C]	Inlet tempe rature $\theta_{H,w,in}$ [°C]	Outlet tempe-rature $\theta_{H,w,out}$ [°C]	Outlet tempe-rature $\theta_{W,w,out}$ [°C]	
1	7	6	**7**	20	12	40	**45**	60	Heating
2	-15	-	**-7**	20	7*	a	**45**	60	Heating, optional
3	-7	-8	**2**	20	7	a	**45**	60	Heating
4	2	1	**5**	20	10	a	**45**	60	Heating
5	-7	-8	**2**	20	7	a	**35**	60	Heating
6	7	6	**7**	20	12	a	**35**	60	Heating
7	15	10	**…**	20	14	a	**35**	60	Heating optional
8	7	6	**7**	20	12	-	**-**	60	DHW

a The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)
* The wet bulb temperature is based on a humidity gain of 2.5 g/kgdry air per outside temperature step based on the given rating point 20(12) in EN 14511

If the tests are being performed for a DIBt permit, measurements are taken for air conditions according to DIBt guidelines:

Tab. 7: Test conditions according to the DIBt

	Measuring Point 1	Measuring Point 2	Measuring Point 3
Return air temperature	21°C	21°C	21°C
Return air humidity	36% r.h.	46% r.h.	56% r.h.
Outside air temperature	-3°C	4°C	10°C
Outside air humidity	80% r.h.	80% r.h.	80% r.h.

As various units run with ground-air heat exchangers, the outside air temperature posterior to the heat exchanger was additionally established. This data (cf. Tab. 6) is valid for the mid-land of Switzerland. For other regions the outside air temperature at the outlet of the ground-air heat exchanger must be verified and/or adjusted.

3.3.2.4.1 Testing of different components and operation modes of compact units

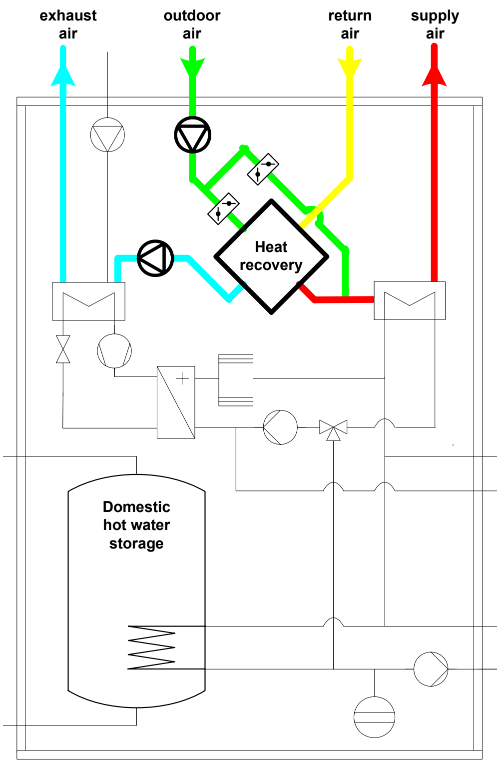
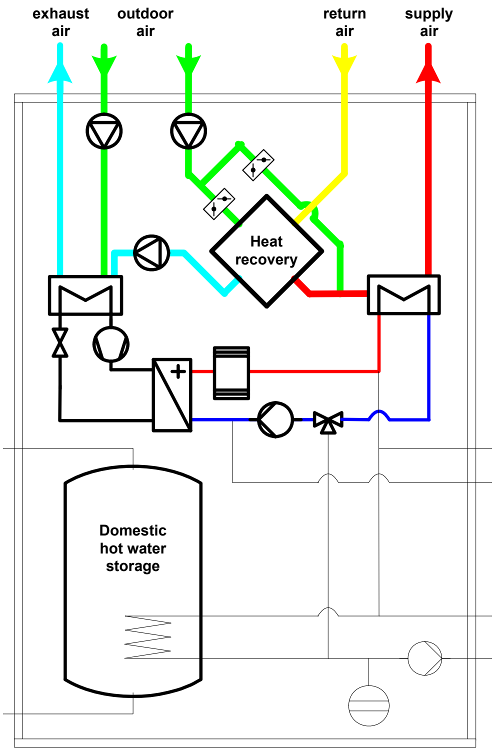


Fig. 10: Ventilation operation

1. Determination of the temperature chance coefficient and the electrothermal amplification factor

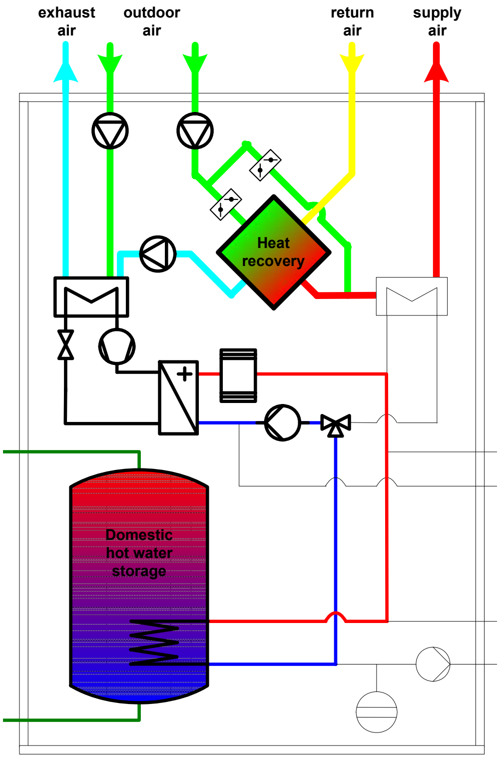
Only the fans and the unit's controls are in operation while the temperature chance coefficient and the electrothermal amplification factor are being determined. The heat pump and electric heating elements are turned off. The hot water storage has ambient temperature (+/- 5 K). The thermal measurements are performed for all of the pre-scribed temperatures and volume flows – at constant air temperatures and humidities.



2. Determination of COP for air heating

The fans and the heat pump are in operation while the COP for air heating is being determined. The heat pump dispenses heat only to the supply air. Heat must be stored in the domestig hot water storage so as to ensure an operational temperature of 60°C. No domestic hot water may be drawn from nor any thermal energy delivered to an external water-based system. Any electrical heating elements in the system must be switched off, but the ventilator is in operation. The heat pump must be operated solely to heat the supply air. Thermal measurements are performed for all of the prescribed temperatures and volume flows. The supply-air temperature may not be limited. The remaining temperatures and humidities are held constant throughout the measurements.

Fig. 11: Operation of ventilation and heat pump for air heating



3. Determination of COP for providing domestic hot water

The fans and the heat pump are running while the COP for providing domestic hot water is being determined. The heat pump dispenses heat only to the domestic water. Following EN 255-3 [7], measurements are taken at an outside temperature of +12°C. At the outset, the hot water storage must have an operational temperature of +15°C. Warm water is withdrawn according to the procedural guidelines of EN 255-3 [7]. At the manufacturer's special request the procedural guidelines of M/324 [42] can be followed instead of the guidelines of EN 255-3 [7]. Any existing electrical heating elements are turned off. The ventilator is in operation. Thermal measurements are performed on only one air volume flow. If the operating range of the unit yields two thermodynamic measuring points, measurements are carried out on the greater volume flow of the two; in case of three thermodynamic measuring points, the intermediate volume flow is measured. Measurements are performed at constant air temperatures and humidities.

Fig. 12: Operation of ventilation and heat pump for providing domestic hot water

In order to reduce testing time, Phase 4 (determination of power consumption during the operational readiness phase) is reduced from five to two heating phases. The duration of the test can thus be shortened by ca. three days. The remaining phases are carried out as described in EN 255-3 [7].

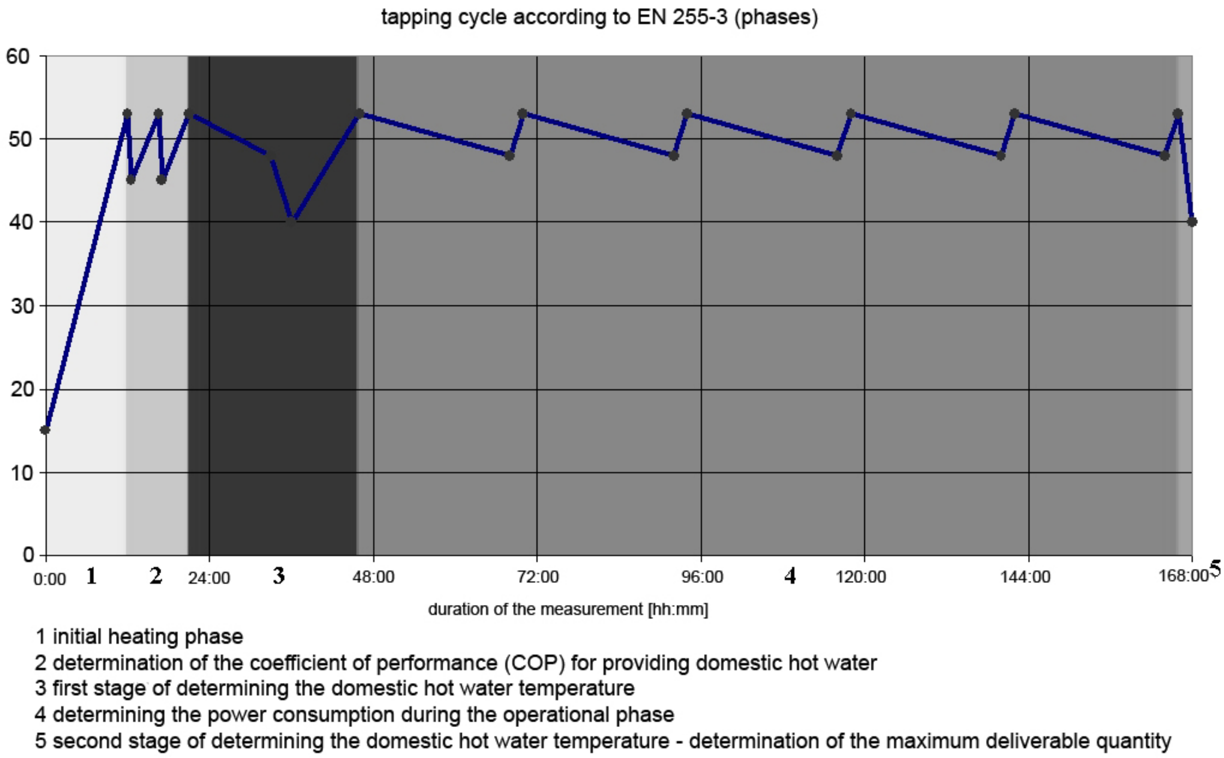


Fig. 13: Diagram of warm water withdrawal model according to EN 255-3

If the compact unit delivers heat to a water-bearing system, tests to determine the COP are performed as follows:

Ventilation unit and heat pump are in operation. The heat pump must dispense heat only to a water-bearing system. Thermal measurements are performed at all prescribed air temperatures and air volume flows. The appropriate value for the water volume flow is specified by the manufacturer so as to provide a supply temperature of 35°C (floor heating) or 45°C (radiator). Heat must be stored in the hot water storage tank so as to ensure an operational temperature of 60°C for the duration of the tests. No domestic hot water must be withdrawn nor heat dispensed to the supply air. All measurements are performed at constant air temperatures and humidities.

3.3.2.5 Stand-by Mode

Apart from the thermal measurements, the total electric power consumption of the unit (incl. controls) is recorded while it is in standby mode.

3.3.2.6 Testing the frost-protection system / Cyclic test (option, following DIBt)

Starting with the thermodynamic measurement point the outside-air temperature must be lowered from -3°C (return air 21°C and 36% r.h.) to max. -12 C in order to trigger the frost-protection system. The rate of lowering the outside-air temperature must not exceed 1K every 5 minutes. A cyclic test is performed to ascertain whether the manufacturer’s frost-protection strategy is adequate.

At an outside-air temperature 4K below the threshold required to activate the frost-protection device, but no lower than -12°C, the thermodynamic data (temperature, humidity) of the entering and exiting air flows are to be recorded for a duration of at least 30 minutes, so as to include at least 3 switching cycles. If the data curves recorded during 3 switching cycles or 30 minutes of testing show insufficient evidence of repeated cyclic behaviour (e.g. constancy of exhaust-air temperature or of pressure drop), the test is to be continued for three more switching cycles. The cyclic test is to be performed at the volume flow corresponding to one of the measurement points in the thermodynamic test and at a return-air relative humidity of 36%. If the operational range of the unit results in two prescribed measuring points, the lower of the two volume flows is to be used for the cyclic test. In case of three measuring points the intermediate volume flow should be used.

3.3.2.7 Acoustic Tests

Acoustic tests are carried out to assess sound emission through the housing of the unit under test and sound emission into the connected ventilation ducts. The measurements are performed in accordance with the sound intensity conventions of ISO 9614 [43]. The acoustical measurements are carried out in real operation, i.e. with conditioned air.

Measurements must be made as given in Tab. 8:

Tab. 8: Test prescriptions for sound testing

Sound source	**Volume flow**	**Pressure difference**
Emission from unit	max. volume flow*	100 Pa
Exhaust- and outside-air ducts	max. volume flow*	100 Pa
Supply- and return-air ducts	min. and max. volume flow*	70/100/130 Pa

**Minimum and maximum volume flow is determined according to specifications established by the Passive House Institute.*

3.3.2.8 Filter Bypass Leakage (optional)

Filter bypass leakage, an indicator for hygiene, is determined following EN 1886 [44].

3.3.2.9 Hygienic tests (optional)

Hygienic tests are performed according to SWKI 2003-5 [45].

3.3.2.10 Operation/Maintenance/Safety (optional)

Criteria for the assessment of operation, maintenance and safety of compact ventilation units are yet to be established.

3.3.3 Output values of the proposed testing for the performance calculation

3.3.3.1 Heat recovery

- Temperature change coefficient (see eq. 5 and description in chap. 4.1.2) or ventilation heat recovery efficiency respectively of the compact unit for ventilation-only operation.
- Temperature change coefficient of combined ventilation and space heating operation (only in case of water-based heating system)
- Temperature change coefficient of the compact unit for combined ventilation and DHW operation

  => directly recovered losses by the ventilation system during the heat pump operation in space heating in the case of a water based heating system and DHW operation are included in the heat recovery characteristic. The calculation is carried out with the temperature change coefficient of ventilation-only operation, or, if significant recovered losses occur, a weighted temperature change coefficient of periods with and without heat pump operation.

- Electrical power input to the ventilation system

If no preheating of the outdoor air is provided, the temperature change coefficient shall be reduced by a factor of 0.03 for the intermittent operation due to defrosting, as it is done by DIBt.

3.3.3.2 Heat pump

- COP and heating capacity of the heat pump for water heating systems (evaluated only with the heat input to the water heating system)
- COP and heating capacity of the heat pump for air heating systems (only additional input to the supply air by the heat pump as difference between heat pump and heat recovery)

  => directly recovered losses of the heat pump by the ventilation system are included in the COP value of the heat pump

  => effects of preheating are included in the COP of the heat pump
- COP and output capacity for DHW operation
- Storage losses or electrical power input to cover storage stand-by losses

3.3.3.3 Compact unit

- Stand-by power input
- Surface temperature of the compact unit
- Ambient room temperature of the compact unit

4 DESCRIPTION OF FHNW CALCULATION METHOD

The objective of this project was the extension of the FHNW-method to the calculation of heat pump compact units including a ventilation heat recovery and a validation of the calculation method in order to assess the exactness of the results. This chapter gives the approaches to cover compact units, a validation is described in chap. 5 based on the results of field monitoring of pilot plants.

4.1 General approach

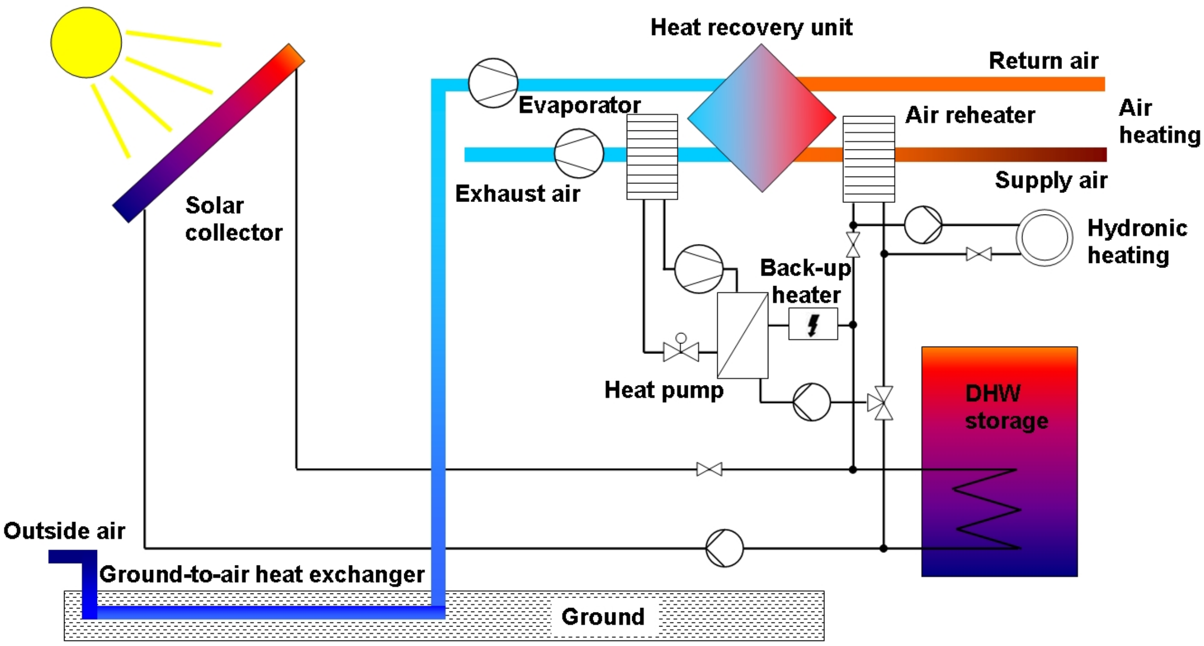


Fig. 14: Principle layout of a compact unit (modified from [23])

The FHNW-method covers the space heating and the domestic hot water system. In more integrated systems, in particular in compact units, the ventilation system is also integrated in the same casing. Moreover, a solar collector and preheating of the inlet air, e.g. by a ground-to-air heat exchanger (GAHX), can be found. Fig. 14 shows the principle layout of a compact unit.

The ventilation heat recovery reduces the ventilation losses. Thus, the coupling of the heat recovery unit affects the system operation with regard to

- Reduction of the space heating energy requirement to be covered by the heat pump (and in case the back-up heater)
- Change of inlet temperature to the heat pump

Furthermore, a ground-to-air heat exchanger (GAHX) is often connected to the ventilation system. The GAHX has the following advantages

- Preheating of the outdoor air
  Efficient heat input to reduce the space heating requirement. However, the heat recovered by the ventilation heat recovery unit is reduced due to higher inlet temperatures.

- Avoidance of defrost operation of the heat recovery
  Higher reduction of the ventilation losses by longer running times of the heat recovery without defrosting interruption.
- Higher inlet temperature to the heat pump
  Higher heating capacity and COP of the heat pump. This is in particular interesting for pure exhaust air units which have a limited capacity for the reheating of the supply air after the heat recovery unit, as depicted in Fig. 15.

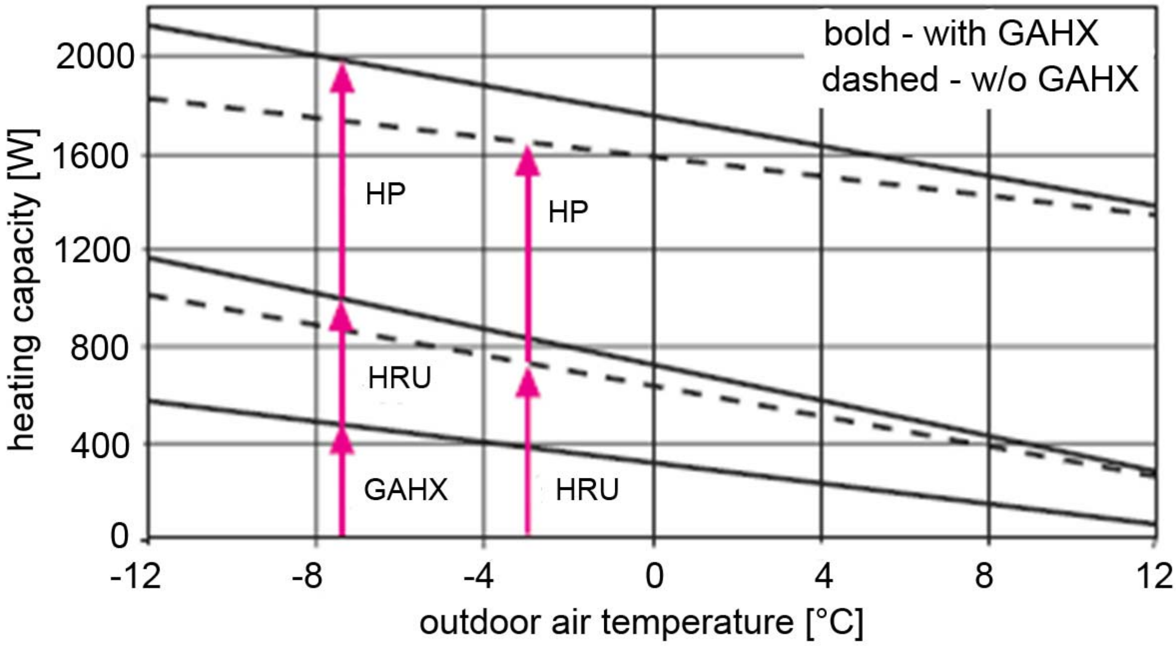


Fig. 15: Illustration of the different contributions to the heating capacity (taken from [9]) (HP–heat pump, HRU–heat recovery unit,GAHX–ground-to-air heat exchanger)

- Optional cooling operation in summertime
  Bypass of the heat recovery in summer operation and ventilation with precooled air by the GAHX

An installed solar thermal system can further contribute to cover the heat requirement. Depending on the system configuration parts of the DHW production can be covered by solar energy or even a share of the space heating requirement can be supplied by the solar system.

Concerning the calculation of the compact unit, the solar collector has the following impact

- Reduction of DHW and eventually the space heating requirement
- Elevated electrical energy input by additional collector circuit pump

4.1.1 Assumptions for the calculation of compact units

As for the calculation of the heat pump certain assumptions and simplifications are made in order to keep the calculation simple.

- The ventilation system is assumed to operate the entire heating period.
  This assumption seems realistic, since in wintertime, normally no window-airing is applied to minimize ventilation energy losses. This assumption could be replaced by a requirement for the ventilation, which would allow calculating the operation time of the ventilation system, or a temperature limit for the operation.

- The control system is idealized in that way, that the heat recovery is operated and reduces the space heating energy requirement. For the rest of the heat requirement the heat pump is applied up to the limit of the heating capacity and only the rest is covered by the electrical back-up heating.

These assumptions reflect the usual operation of compact units.

In case of an installed solar collector, the following assumptions are made

- The solar energy is supplied to the DHW system. For an annual calculation, the solar energy of the month Dec., Jan and Feb. are allocated to the lower bin, of the month March and Nov. to the second bin, of the month April and October to the third bin and of May to Sept. to the upper bin.

If a monthly calculation is applied, the allocation could be considered month to month. The presented method in chap. 4.1.4 can also take into account solar energy supplied to the space heating system. However, common collector sizes installed with compact units rather refer to a DHW operation, even though some of the compact units on the market would have the possibility to supply solar energy for space heating due to the internal configuration, while others restrict the possibilities to pure DHW use.

Fig. 16 shows the repartition of energies in the cumulative frequency diagram for a system with heat recovery and solar collector.

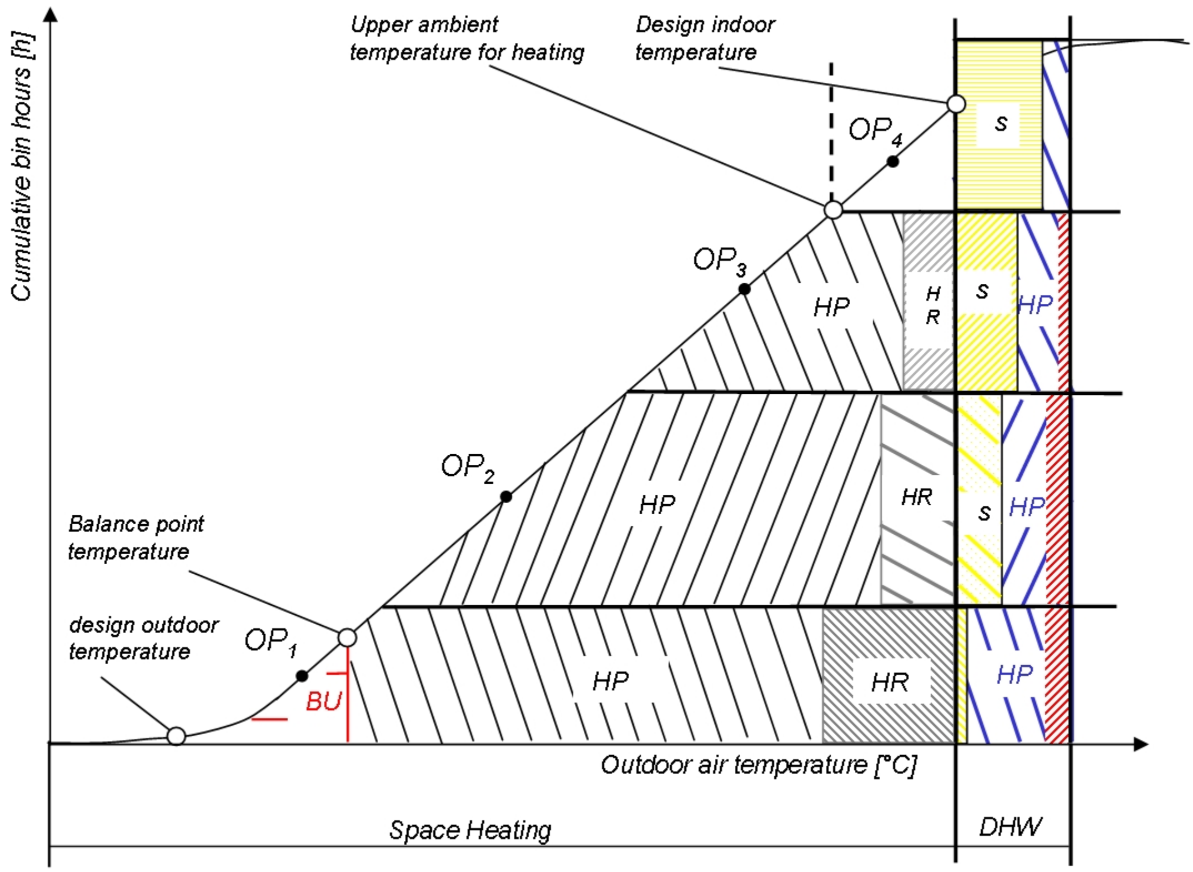


Fig. 16: Extended bin-method for heat pump compact units with coverage by heat recovery (HR), solar collector for DHW (S), heat pump (HP) and back-up heater (BU)

4.1.2 Consideration of the heat recovery

The heat recovery is characterised by the temperature change coefficient that can be defined for the supply air side and the return air side of the heat recovery by the equation

$$\Phi_{oa} = \frac{\theta_{sa} - \theta_{oa}}{\theta_{ra} - \theta_{oa}} \qquad \Phi_{ra} = \frac{\theta_{ra} - \theta_{ea}}{\theta_{ra} - \theta_{oa}} \qquad [-]$$

eq. 5

where

Φ_{oa} = temperature change coefficient on outdoor air (air flow from outside to building) [-]
Φ_{ra} = temperature change coefficient on exhaust air (air flow from building to outside) [-]
θ_{ea} = air temperature of exhaust air exiting the heat recovery [°C]
θ_{sa} = air temperature of supply air exiting the heat recovery [°C]
θ_{oa} = outdoor air temperature [°C]
θ_{ra} = air temperature of return air (exiting the building and entering the heat recovery) [°C]

Due to the black box testing approach described in chap. 3.3.1, the temperature change coefficient does not refer to the system boundary of the heat exchanger, but to the system boundary of the compact unit. Therefore, additional heat input, for instance by fans, is considered in this value, but the testing refers only to the ventilation part, so no pre- or reheating coupled to the heat pump or electrical back-up heater operation are included. In case of a balanced ventilation system as it is the standard for low energy house application, the temperature change coefficient corresponds to the heat recovery efficiency according to DIBt given in eq. 1. In case of a disbalance the heat recovery efficiency is to be applied.
In case of a combined testing of the heat recovery and heat pump as proposed in chap. 3.3 the change of inlet temperature is included in the heat pump characteristic. However, volume flow conditions have to be considered, so testing has to cover different volume flow rates. In case of separate test results of the heat recovery and the heat pump, the return air side temperature change coefficient can be used to determine the inlet temperature of the heat pump, eventually taking into account the mix with an additional outdoor air flow. Based on the temperature change coefficient of the heat recovery, the reduction of the ventilation heating need can be calculated unless already included in the space heating need calculation.
Defrosting of the heat recovery is normally taken into account by a reduced temperature change coefficient in the testing, as described for the DIBt in chap. 2.2.2. In case of an air preheating to avoid defrosting, no correction to the temperature change coefficient is applied. Additional electrical energy input is considered by the fan power and the operation time of the heat recovery unit.

4.1.3 Consideration of a ground-to-air heat exchanger

In case of an installed GAHX, the outlet temperature after the GAHX is calculated based on the input data

- number of pipes, length, diameter and material of the piping in the ground
- ground parameters as heat conductivity, specific heat and density defined by the ground type
- depth of the GAHX below ground surface

For the calculation of the outlet temperature a static model described in [34] is used. The model takes into account the heat transfer from the ground to the pipes, the heat transfer through the pipe walls to the air flow. The ground temperature in the depth of the GAHX is calculated based on the phase-shifted and amplitude-damped outdoor air temperature.
In order to calculate the additional electrical energy input due to the pressure drop in the GAHX, the pressure drop is calculated based on the pipe friction and friction in single fittings

(suction valve at inlet, bending of the pipe, etc. given in [35]). Details of the calculation of the outlet temperature are given in Appendix G.

4.1.4 Consideration of solar energy

The contribution of solar energy is considered as the draft standard prEN 15316-4.3 for component test data according to EN 12975-2 [30] as they are provided e.g. by SPF in Rapperswil (http://www.spf.ch). To evaluate the yearly amount of solar energy input the total amount is estimated by a monthly evaluation based on the input data:

- Aperture surface of the collector (acc. to EN 12975-2 [30]), inclination and azimuth angle
- Efficiency of the collector according to EN 12975-2
- Length of piping from the collector to the storage, length specific heat loss coefficient
- Nominal electrical power consumption of the collector circuit pump
- Monthly sum of the global irradiation on the horizontal given in SIA 382/1 [27]

The global irradiation on tilted surface can be calculated based on the three components direct and diffuse irradiation from the sky and reflected diffuse irradiation according to eq. 6

$$G_t = G_{bh} \cdot \frac{\cos\theta_i}{\cos\theta_z} + G_{dh} \cdot \frac{1+\cos\beta}{2} + G_h \cdot \rho_g \cdot \frac{1-\cos\beta}{2} \qquad [\mathrm{W/m^2}]$$

eq. 6

where

G_t	global solar irradiation on tilted surface in the month	[W/m^2]
G_{bh}	direct irradiation on horizontal	[W/m^2]
G_{dh}	fraction of diffuse irradiation from sky	[W/m^2]
G_h	global irradiation on horizontal	[W/m^2]
ρ_g	ground reflectance	[-]
β	inclination of collector surface	[rad]
θ_i	incidence angle for inclined surface	[rad]
θ_z	zenith angle	[rad]

In order to determine the ratio between the direct irradiation and the diffuse irradiation a clearness index is calculated using the theoretical global irradiation on a horizontal surface derived by an isotropic sky model and the measured value on horizontal based on the monthly values given in SIA 382/1 for the 15th day of each month. Details on the calculation are given in Appendix B.

The solar energy output is calculated according to the equation

$$Q_{sol,out} = (aY + bX + cY^2 + dX^2 + eY^3 + fX^3) \cdot Q_{nd,sol} \qquad [\mathrm{kWh}]$$

eq. 7

where

$Q_{sol,out}$	heat energy output of the solar system	[W/m^2]
$Q_{nd,sol}$	energy need of the distribution system to be covered by solar energy	[kWh]
a,b,c,d,e,f	correlation factors to take into account the system configuration	[W/m^2]
X	dimensionless value reflecting a loss-load ratio	[-]
Y	dimensionless value reflecting a solar-load ratio	[-]

The energy need for the solar system depends on the system type, i.e. solar contribution to DHW-only, SH-only or solar combi-system for DHW and space heating. In case of a combi-system the calculation is accomplished two times with the respective needs for SH and DHW. The collector aperture surface and the storage volume (in case of a single common storage) are divided according to the energy need ratio. However, for the storage volume, the redistribution has to be made with the monthly ratio, since otherwise too small storage volumes for the DHW operation would result. For the respective energy need the calculation is carried out independently and solar energy output is added up.

Tab. 9: Input data of the calculation (based on pilot plant Gelterkinden, see chap. 6.1)

Correlation coefficients for the calculation of solar heat energy output						
	a	b	c	d	e	f
Water storage system	1.029	-0.065	-0.245	0.0018	0.0215	0
Direct solar floor	0.863	-0.147	-0.263	0.008	0.029	0.025

The correlation factors a,b,c,d,e,f are in given in Tab. 9.

The dimensionless factor X takes into account the losses of the solar system and is calculated according to the equation

$$X = A_{col} \cdot (k_1 + UL / A_{col}) \cdot \eta_{loop} \cdot (\theta_{ref} - \theta_{oa,avg}) \cdot c_{cap} \cdot N_{month} / (Q_{nd,sol} \cdot 1000\,[Wh/kWh]) \quad [-]$$

eq. 8

where

X	dimensionless value reflecting a loss-load ratio	[-]
A_{col}	collector aperture surface according to EN 12975-2	[m^2]
UL	heat loss of the collector circuit pipes in collector field (pipework between collector and storage included in η_{loop})	[W/(m^2·K)]
k_1	linear collector loss coefficient according to EN 12975-2	[W/(m^2·K)]
η_{loop}	efficiency of the collector loop (set to 0.8)	[-]
θ_{ref}	reference temperature for the respective application	[°C]
$\theta_{oa,avg}$	average outdoor air temperature in the month	[°C]
c_{cap}	Correction factor for the storage capacity	[-]
N_{month}	number of hours of the respective month	[h]
$Q_{nd,sol}$	energy need of the distribution system to be covered by solar energy	[kWh]

The reference temperature depends on the application. For space heating, it is set to $\theta_{H,ref}$ = 100°C, for DHW it is calculated according to the equation

$$\theta_{W,ref} = 11.6 + 1.18 \cdot \theta_{hw} + 3.86 \cdot \theta_{cw} - 2.32 \cdot \theta_{oa} \quad [°C]$$

eq. 9

where

$\theta_{W,ref}$	reference temperature for a DHW system	[°C]
θ_{hw}	DHW hot water reference temperature	[°C]
θ_{cw}	DHW cold water inlet temperature (identical for all month, e.g. for Zurich 9.7°C)	[°C]
θ_{oa}	outdoor air temperature	[°C]

The dimensionless factor Y takes into account the output of the solar system and is calculated according to the equation

$$Y = A_{col} \cdot IAM \cdot \eta_0 \cdot \eta_{loop} \cdot G_{month} \cdot N_{month} / (Q_{nd,sol} \cdot 1000\,[Wh/kWh]) \quad [-]$$

eq. 10

where

Y	dimensionless value reflecting a solar-load ratio	[-]
A_{col}	collector aperture surface according to EN 12975-2	[m^2]
IAM	incidence angle modifier according to EN 12975-2	[-]
η_0	optical efficiency according to EN 12975-2	[-]
η_{loop}	efficiency of the collector loop (set to 0.8)	[-]
G_t	global irradiation on tilted surface in the month	[W/m^2]
N_{month}	number of hours of the respective month	[h]
$Q_{nd,sol}$	energy need of the distribution system to be covered by solar energy	[kWh]

Auxiliary electrical energy input to the collector circuit can be calculated by the nominal pump power and running time according to the equation

$$E_{HW,aux,sol,in} = P_{W,aux,sol} \cdot t_{op,sol} \qquad [J]$$

eq. 11

where

$E_{HW,aux,sol,in}$	auxiliary electrical energy use for the solar circuit pump	[J]
$P_{W,aux,sol}$	nominal power of the solar circuit pump	[W]
$t_{op,sol}$	operation time of solar circuit pump (set to 2000 h/a according to EN 12976)	[s]

In EN 12976 the operation time of the solar system is set to 2000 h/a. The redistribution of the annual operation to the month is done by the ratio of solar irradiation in the month to the total annual solar irradiation.

4.2 Sample of the SPF calculation

In the following a step-by-step sample of the SPF-calculation is given for the pilot plant in Gelterkinden, which is treated in chap. 6.1. The calculation is accomplished for three bins for space heating and four bins for DHW operation. The operating points are chosen according to the test conditions of the heat pump testing according to EN 14511 as described in the test procedure in chap. 3.3. However, since for the MINERGIE® calculation manufacturer data of EN 255-2 testing [2] were available, these values have been used for the calculation. The data of the outdoor air temperature of the site are given in Appendix E.4, Tab. E 4 linked to tabulated energies and characteristic number for different system boundaries.
This calculation sample reflects the calculation for the comparison with field results of the pilot plant Gelterkinden and values have been used for the validation of the calculation method with the field monitoring results which is given in chap. 5.2.

4.2.1 Input data

In Tab. 10 the input data for the calculation of the pilot plant in Gelterkinden are specified. The pilot plant is equipped with the LWZ 303 SOL [36] of the German manufacturer Stiebel Eltron.

If the recovered energy of the ventilation system is to be included in the SPF calculation or not depends on the input data. In case of the SIA 380/1 [26], two cases are to be differentiated. For the proof of compliance standard utilization of the building is set and no heat recovery is considered. In this case the recovered energy of the ventilation system has to be considered in the SPF-calculation. However, to correctly consider the used gains, the input of the monthly transmission, ventilation and solar and internal gains would be required. On the other hand, SIA 380/1 gives the possibility to calculate an optimisation taking into account the ventilation heat recovery. In this case, the recovered heat is already included in the input for the SPF-calculation and must not be considered in the SPF-calculation of the compact unit. Solar and internal gains are taken into account correctly. However, care shall be taken, that the correct heat recovery efficiency is used in SIA 380/1. Note, that only the calculation of the recovered energy would not be required, but input values of the ventilation system like volume flow rate and temperature change coefficient are still required for the calculation, since the heat pump characteristic depends on the volume flow rate.
For the sample calculation, the measured values, e.g. the space heating energy needs supplied to the water heating system and the ventilation system of the pilot plant in Gelterkinden have been used, so the calculation of the heat recovery is explicitly given in the calculation of the compact unit.
The set value for the DHW temperature was changed during the operation period. The two temperatures given in Tab. 10 refer to a daytime and a night-time temperature. Every evening at 10 p.m. the storage is loaded to the higher temperature. For reheating during daytime the lower temperature is relevant.

Tab. 10: Input data of the calculation (based on pilot plant Gelterkinden, see chap. 6.1)

Space heating mode				
Hourly values of the outdoor air temperature (given in Appendix E.4, Tab. E 4)				
Design indoor temperature $\theta_{i,des}$ [°C]	20			
Space heating energy need in heating period [kWh_{th}]	12014 (measured value)			
Design outdoor temp. $\theta_{oa,des}$/heating limit $\theta_{H,\theta hlim}$ [°C]	-8/14			
Temp. at $\theta_{oa,des}$/ $\theta_{H,\theta hlim}$ [°C] (30% flow, 70% return)	28.7/23.1			
Heat pump characteristic space heating	*A-7/W35*	*A2/W35*	*A7/W35*	*A20/W35*
Heating capacity [kW] taken from [2]	3.36	4.24	4.66	6.48
COP [-] [2]	2.91	3.27	3.54	4.54
	A-7/W50	*A2/W50*	*A7/W50*	*A20/W50*
Heating capacity [kW] [2]	2.38	3.11	3.95	5.69
COP [-][2]	2.02	2.29	2.71	3.41
Standard mass flow rate [l/h] [2]	720			
Mass flow rate in operation [l/h]	960			
Heat recovery characteristic (acc. to DIBt) [2]				
	A-3	*A4*	*A10*	
Volume flow rate 101 m^3/h	0.82	0.85	0.83	
Volume flow rate 162 m^3/h	0.80	0.82	0.82	
Average volume flow rate heat recovery [m^3/h]	47 (measured value)			
Power for both heat recovery fans [W]	28.9 (measured value)			
Control heat recovery [W]	10 (measured value)			
Auxiliary energy				
Nominal power of the circulation pump [W]	54 (measured value)			
Stand-by input heat pump [W]	10 (measured value)			
Electrical back-up heating				
Efficiency electrical back-up heating	0.95			
Domestic hot water mode				
Domestic hot water energy need [kWh_{th}]	1179 (measured value)			
Average hot water loading temperature $\theta_{w,avg}$ [°C]	46°C (measured value)			
Heat pump characteristic for DHW	Heating characteristic evaluated at 46°C			
Set domestic hot water temperature [°C]	Apr. – Nov.	Nov. – Apr.		
	42/48	50/55		
DHW storage losses [W_{th}]	122,5 (measured average annual value)			
Nominal power storage loading pump [W_{el}]	33 (measured value)			

The heat pump and the heat recovery characteristic are taken from a data sheet of the MINERGIE calculation delivered by the manufacturer [2]. The heat pump characteristic is based on a combined test of heat pump and ventilation system, so the possible impact of the heat recovery on the inlet temperature of the heat pump is considered in the COP-values.
Since no values according to EN 255-3 were available, the heating characteristic at an average loading temperature of $\theta_{w,avg}$=46°C has been used for the calculation.

A solar collector is not installed. The back-up operation in the heating mode is limited to an outdoor air temperature below -5°C, for the back-up operation of the domestic hot water a time delay of 90 minutes is applied for outdoor temperatures above -2°C. A comparison of this calculation with results of the field monitoring follows in chap. 5.2.

The upper temperature limit for heating has been defined according to the controller setting in the pilot plant to $\theta_{H,\theta hlim}$=14°C.

4.2.2 Calculation steps of the FHNW method

For each step of the calculation, the equations are given in SI-units. If common input data are given in non SI units, conversion factors are given in the equation.

Below each equation, the same equation with the values for the first bin and the results are given. The results for the further bins are given as tabulated values.

<u>Step 1: Calculation of the energy requirement in the bin</u>

Based on the input data, heating degree hours are evaluated according to ISO EN 15927-6 [29] for a base temperature of the indoor design temperature of $\theta_{i,des}$ = 20°C. The evaluation of the hourly averaged data of the outdoor temperature based on the on-site measurements in Gelterkinden is given in Appendix E4, Tab. E4.

The weighting factors and the energy need for space heating in the respective bin is evaluated by the equation

$$Q_{H,dis,in,j} = Q_{H,dis,in} \cdot w_{H,j} = Q_{H,dis,in} \cdot \left(\frac{DH_{H,\theta hlim,j} - DH_{H,\theta llim,j}}{DH_{H,\theta hlim}}\right) \qquad [J]$$

$$Q_{H,dis,in,1} = 12014\,kWh \cdot \left(\frac{7940\,Kh - 0\,Kh}{71183\,Kh}\right) = 1340\,kWh \qquad [kWh]$$

eq. 12

where

$Q_{H,dis,in,j}$	heat energy need of the space heating distribution system in bin j	[J]
$Q_{H,dis,in}$	heat energy need to the space heating distribution system	[J]
$w_{H,j}$	weighting factor for SH operation in bin j	[-]
$DH_{H,\theta hlim}$	cumulative heating degree hours up to heating limit	[Kh]
$DH_{H,\theta hlim,j}$	cumulative heating degree hours up to the temperature at upper limit of bin j	[Kh]
$DH_{H,\theta llim,j}$	cumulative heating degree hours up to the temperature at lower limit of bin j	[Kh]

The heat energy need of the domestic hot water distribution system in bin j is calculated based on a constant daily domestic hot water consumption, so DHW energy needs are considered dependent on the time of the bin

$$Q_{W,dis,in,j} = Q_{W,dis,in} \cdot w_{W,j} = Q_{W,dis,in} \cdot \frac{N_{ho,j}}{N_{ho,tot}} \qquad [J]$$

$$Q_{W,dis,in,1} = 1179\,kWh \cdot \frac{330\,h}{8760\,h} = 1179\,kWh \cdot 0.04 = 44\,kWh \qquad [kWh]$$

eq. 13

where

$Q_{W,dis,in,j}$	heat energy input to the domestic hot water distribution system in bin j	[J]
$Q_{W,dis,in}$	heat energy input to the domestic hot water distribution system	[J]
$N_{ho,j}$	cumulated hours in bin j	[h]
$N_{ho,tot}$	cumulated hours of DHW operation (e.g. 8760 for year-round operation)	[h]

Tab. 11: Evaluated outdoor air temperature for the calculation based on hourly-averaged values of the monitoring in Gelterkinden

Meteorological data/ Energy needs				
	SH/DHW alternately			**DHW-only**
	Bin1	**Bin 2**	**Bin 3**	**Bin 4**
Lower temperature limit $\theta_{\theta llim,j}$ [°C]	$\theta_{oa,min}$	>-2	>4	>14
Operating point θ_{op} [°C]	-7	2	7	20
Upper temperature limit $\theta_{\theta hlim,j}$ [°C]	-2	4	14	$\theta_{oa,max}$
Bin time t_j (equal to effective bin time $t_{eff,j}$) [h]	330	1604	3262	3564
Cumulative DH_H up to upper bin limit [Kh]	7940	37348	71183	-
Weighting factors heating $w_{H,j}$ [-]	0.11	0.41	0.48	-
Energy need of the SH distribution $Q_{H,dis,in,j}$ [kWh]	1340	4963	5711	
Weighting factors DHW $w_{W,j}$	0.04	0.18	0.37	0.41
Energy requirement DHW $Q_{W,dis,in,j}$	44	216	439	480

In case of defined cut-out periods of the electricity supply of the heat pump due to an advantageous tariff structure, the bin time has to be reduced to an effective bin time of the heat pump operation according to the equation

$$t_{eff,j} = t_j \cdot \frac{24\,[h] - t_{co}}{24\,[h]} = (N_{ho,\theta hlim,j} - N_{ho,\theta llim,j}) \cdot 3600[s/h] \cdot \frac{24\,[h] - t_{co}}{24\,[h]} \qquad [s]$$

$$t_{eff,1} = 330\,h \cdot \frac{24\,[h] - 0\,h}{24\,[h]} = (330\,h - 0\,h) \cdot \frac{24\,[h] - 0\,h}{24\,[h]} = 330\,h \qquad [h]$$

eq. 14

where

$t_{eff,j}$	effective bin time for the heat pump operation in bin j	[s]
t_j	bin time in bin j	[s]
t_{co}	cut-out time	[h]
$N_{ho,\theta hlim,j}$	cumulated hours in bin j	[h]
$N_{ho,\ \theta llim,j}$	cumulated hours of DHW operation (e.g. 8760)	[h]

The bin distribution, the bin time and the respective energy needs for space heating and DHW are given in Tab. 11. In case of the pilot plant Gelterkinden, there is no cut-out time. In case of an installed solar collector, the DHW energy need has to be reduced by the solar contribution as described in chap. 4.1.4. The heating period is defined by the bin time to the upper temperature limit for heating.

Step 2: Calculation of the heat recovery

For compact units the space heating energy need is reduced by the recovered ventilation losses of the heat recovery unit. If the heat recovery is not taken into account in the energy input values, it has to be reduced within the calculation of the compact unit. The heat recovery unit is calculated according to eq. 15. Instead of the bin-wise calculation the total recovered heat can be calculated with average values.

$$Q_{H,hru,rvd,j} = \rho_{oa,j} \cdot \dot{V}_{oa,j} \cdot c_{p,1+x} \cdot \Phi_{oa,j} \cdot (DH_{H,\theta hlim,j} - DH_{H,\theta lim,j}) \quad [J]$$

$$Q_{H,hru,rvd,1} = 1.32\frac{kg}{m^3} \cdot 47\frac{m^3}{h} \cdot 1.008\frac{kJ}{kg \cdot K} \cdot 0.82 \cdot (7940Kh - 0Kh) \cdot \frac{1}{3600kWh/kJ} = 113kWh \quad [kWh]$$

eq. 15

where

$Q_{H,hru,rvd,j}$	heat energy recovered by the heat recovery unit in bin j	[J]
$\rho_{oa,j}$	density of the outdoor air at temperature conditions in bin j	[kg/m^3]
$\dot{V}_{oa,j}$	volume flow rate of the supply air of the heat recovery in bin j	[m^3/h]
$c_{p,1+x}$	average specific heat capacity of moist air	[kJ/(kg·K)]
$\Phi_{oa,j}$	temperature change coefficient of the HR for outdoor air to supply air in bin j	[-]
$DH_{H,\theta}$	heating degree hours up to the respective temperature limit of bin j	[Kh]

Tab. 12: Calculation of the recovered energy by the heat recovery unit

Heat recovery unit [1)]			
	Bin 1	**Bin 2**	**Bin 3**
Average temperature change coefficient (acc. to [2])	0.82	0.82	0.82
Reduction SH energy requirement by heat recovery [kWh]	113	406	533
Effective energy for the heat pump and back-up [kWh]	1227	4557	5178

1) Density in the single bins: $\rho_{oa,1}$=1.32, $\rho_{oa,2}$=1.28 and $\rho_{oa,3}$=1.25, $DH_{H,\theta hlim,3} = DH_{H,\theta hlim}$ ($\theta_{i,des}$=20°C)=76867 Kh

First, the temperature change coefficient is interpolated for the volume flow rate and temperatures of the operating points. After that, the energy reduction by the ventilation heat recovery can be calculated by equation eq. 15. However, often, only an average temperature change coefficient is given by the testing, which has to be used in all bins, so the averaged value from [2] has been used for all bins. Tab. 12 gives the recovered heat for the pilot plant in Gelterkinden.

The electrical expense for the heat recovery is evaluated in step 6 for the calculation of auxiliary energy.

Generally, in case of the LWZ 303 SOL the characteristic of the heat recovery unit depends on the operation state of the heat pump due to the subcooling, i.e. a special feature of this unit. Evaluating the monitoring data, however, it was found that the outlet temperature and thus the supply air temperature did not change significantly depending on the operation state of the heat pump. This is due to the effect, that with heat pump operation, the subcooler heats the outdoor air and therefore, the temperature spread in the heat recovery is reduced, on the other hand, without heat pump operation, the outdoor air enters much colder into the heat recovery, so the heating of the supply air is done in the heat recovery. After the heat recovery the supply air temperature is in the same range, so for the calculation the impact is not important.

Moreover, the heat recovery of the compact unit has an impact on the resulting inlet temperature of the heat pump. In case of combined testing as proposed in chap. 3.3, this impact is already included in the test results. If only values of separate testing of the heat pump and the heat recovery unit are available, the effect can be taken into account depending on the type of heat pump.

In case of compact units using only exhaust air, the inlet temperature of the heat pump corresponds to the outlet temperature of the heat recovery and can be calculated with the temperature change coefficient for the return air path according to the equation

$$\theta_{hp,evap,in,j} = \theta_{ra} - \Phi_{ra,j}(\theta_{ra} - \theta_{oa,j}) \quad [°C]$$

$$\theta_{hp,evap,in,1} = 20°C - 0.8 \cdot (20°C - (-7°C)) = -1.6°C \quad [°C]$$

eq. 16

where

$\theta_{hp,evap,in,j}$	inlet temperature to the heat pump evaporator in bin j	[°C]
θ_{ra}	air temperature of return air (exiting the building and entering the heat recovery)	[°C]
$\Phi_{ra,j}$	temperature change coefficient on return path in bin j	[-]
$\theta_{oa,j}$	outdoor air temperature in bin j	[°C]

In case of an installed ground-to-air heat exchanger, the inlet temperature of the heat pump can be calculated with the same equation, but the outdoor air temperature is to be replaced with the temperature at the outlet of the ground-to-air heat exchanger, calculated as described in chap. 4.1.3.

For heat pumps using a mix of exhaust air and outdoor air, the temperature conditions at the heat pump inlet can thus be calculated for a mixture of outdoor air and the outlet air of the heat recovery unit by an enthalpy balance according to the equation

$$\theta_{hp,evap,in,j} = \frac{\dot{m}_{ea} \cdot c_{p,1+x} \cdot \theta_{ea,j} + \dot{m}_{oa} \cdot c_{p,1+x} \cdot \theta_{oa,j}}{\dot{m}_{ea} c_{p,1+x} + \dot{m}_{oa} c_{p,1+x}} \qquad [°C]$$

$$\theta_{hp,evap,in,j} = \frac{0.02\frac{kg}{s} \cdot 1.008\frac{kJ}{kgK} \cdot (-1.6°C) + 0.201\frac{kg}{s} \cdot 1.008\frac{kJ}{kgK} \cdot (-7°C)}{0.02\frac{kg}{s} \cdot 1.008\frac{kJ}{kgK} + 0.201\frac{kg}{s} \cdot 1.008\frac{kJ}{kgK}} = -6.5°C \qquad [°C]$$

eq. 17

where

$\theta_{hp,evap,in,j}$	inlet temperature of the heat pump evaporator in bin j	[°C]
$\dot{m}_{ea}$	massflow rate of the exhaust air from the heat recovery unit	[kg/s]
$\dot{m}_{oa}$	massflow rate of the outdoor air	[kg/s]
$\theta_{oa,j}$	temperature of the outdoor air	[°C]
$\theta_{ea,j}$	temperature of the exhaust air	[°C]
$c_{p,1+x}$	average specific heat capacity of moist air	[kJ/(kg·K)]

If this impact is significant or not depends basically on the volume flow ratio of the outdoor air to exhaust air, so it is especially relevant for ultra-low energy houses, where the outdoor air flow is smaller due to the lower heating needs. In case of ultra-low energy house units the ratio can reach 1. Therefore, the impact on the inlet temperature is significant, while for higher ratios of 5 to 6, it is negligible.

Here, the impact would be negligible, but is included due to the combined testing anyway. Since there may be a preheating of the outdoor air as in case of the LWZ303 SOL which is not included in the pure heat recovery characteristic, a combined testing of heat recovery and heat pump has advantages anyway.

Step 3: Storage losses

Storage losses of an eventually integrated heating buffer storage are considered by the input value of storage heat losses in terms of an (U·A)-value or a maximum heat loss in 24 h based on the size of the storage as defined in the Swiss energy directive [24] according to the equation

$$Q_{H,st,ls,sb} = \frac{\theta_{st,avg,j} - \theta_{st,amb}}{\Delta\theta_{st,ls,sb,nom}} \cdot \frac{Q_{st,ls,sb,nom} \cdot 1000\,[W/kW] \cdot t_j}{24\,[h/d]} \qquad [J]$$

eq. 18

where

$Q_{H,st,ls,sb}$	storage stand-by heat loss to the ambiance of the heating buffer storage in bin j	[J]
$\theta_{st,avg,j}$	average storage temperature in bin j	[°C]
$\theta_{st,amb}$	ambient temperature at the storage location	[°C]
$\Delta\theta_{st,ls,sb,nom}$	temperature difference in storage stand-by testing	[K]

$Q_{st,ls,sb,nom}$	stand-by heat loss in storage stand-by testing	[kWh/d]
t_j	bin time in bin j	[s]

In case of the (U·A)-value, storage losses can be estimated taking into account the supply temperature, the return temperature and the ambient temperature at the storage location. In case of the heat losses in 24h, the values have to be corrected for the relevant temperature difference between average storage temperature and the ambiance of the storage as done in eq. 18.

For the DHW operation the DHW-testing according to EN 255-3 delivers an electrical power input to cover storage losses, but no thermal losses of the storage, so if no values of the storage are known, the default values according to the Swiss energy directive [24] based on the storage volume or data of the manufacturer are to be used.

For the pilot plant, no EN 255-3 testing was available, so an average value according to the monitoring of $\phi_{st,ls,sb}$=122,5 W is used for the storage. Thus, in the respective bins, the storage losses given in Tab. 13 have to be covered.

Tab. 13: DHW storage losses

DHW storage losses				
Bin number	**Bin 1**	**Bin 2**	**Bin 3**	**Bin 4**
Upper temperature limit [°C]	-2	4	14	$\theta_{oa,max}$
Bin time domestic hot water [h]	330	1604	3262	3564
Storage losses [kWh]	40	197	400	437

In case of an installed solar energy collector the contribution for the DHW energy needs can be assessed as described in chap. 4.1.4 and thereby the DHW heat energy produced by the heat pump is reduced.

<u>Step 4: Running time</u>

Based on the heat produced by the heat pump, corresponding to the heat energy needs for space heating, domestic hot water operation and the storage losses, respectively, the running time can be evaluated based on the heat pump heating capacity in bin j according to the equation

$$t_{H,hp,op,j} = \frac{Q_{H,hp,j}}{\phi_{H,hp,j}} \qquad t_{W,hp,op,j} = \frac{Q_{W,hp,j}}{\phi_{W,hp,j}} \qquad [s]$$

$$t_{H,hp,op,1} = \frac{1228\text{kWh}}{3.13\text{kW}} = 311\text{h} \qquad t_{W,hp,op,1} = \frac{85\text{kWh}}{2.64\text{kW}} = 32\text{h} \qquad [h]$$

eq. 19

The total running time is obtained by a summation. The total running time has to fulfill the boundary conditions to be lower than the effective bin time according to the equation

$$t_{hp,op,tot,j} = \min(t_{eff,j};(t_{H,hp,op,j} + t_{W,hp,op,j})) \qquad [s]$$

$$t_{hp,op,tot,1} = \min(330\text{h};(311\text{h} + 32\text{h})) = 330\text{h} \qquad [h]$$

eq. 20

where

$t_{hp,op,j}$	running time of the heat pump for the respective operation mode in bin j	[s]
$Q_{hp,j}$	generated heat of the heat pump for the respective operation mode in bin j	[J]
$\phi_{hp,j}$	heating capacity of the heat pump for the respective operation mode in bin j	[W]
$t_{hp,op,tot,j}$	total running time of the heat pump in bin j	[s]
$t_{eff,j}$	effective bin time in bin j	[s]

The running time is used to evaluate the energy contribution of the auxiliary components. Resulting running times of the bins are given in Tab. 14.

Tab. 14: Running time of the heat pump

Running time of the heat pump				
Bin number	**Bin 1**	**Bin 2**	**Bin 3**	**Bin 4**
Running time for SH [h]	311	974	1038	-
Running time for DHW [h]	32	121	203	152
Total running time [h]	**343**	1095	1241	152
Effective bin time [h]	**330**	1604	3262	3564
Corrected running time [h]	**330**	1095	1241	152

In the first bin, the sum of the single running times for space heating and DHW is longer than the effective bin time, so the effective bin time defines the total running time. In this case the heat pump heating capacity is not sufficient to cover both needs and back-up energy is required, see next chapter.

Step 5: Back-up energy

Back-up energy can be required for two reasons, due to the operation limit temperature of the heat pump or due to a lack of capacity.

On the one hand, the heat pump has a temperature operation limit, i.e. a maximum temperature which can be delivered by the heat pump operation dependent on the process and the refrigerant used. If the temperature requirements are higher than this maximum temperature, the temperature difference has to be produced by the back-up. This is mostly the case for DHW operation, since in systems with compact units the space heating emission system is normally designed to required temperatures below the temperature operation limit of the heat pump in new buildings, or is limited to 50°C in air heating systems.

For domestic hot water operation the supplied back-up energy depends also on the hot water temperature and operation limit of the heat pump. Back-up requirements due to the operation limit of the heat pump can be evaluated by

$$Q_{W,bu,op,j} = (Q_{W,dis,in,j} + Q_{W,st,ls,sb,j}) \cdot \frac{\theta_{hw} - \theta_{hp,op}}{\theta_{hw} - \theta_{cw}} \quad [J]$$

$$Q_{W,bu,op,1} = (44\text{kWh} + 41\text{kWh}) \cdot \frac{55°C - 55°C}{55°C - 15°C} = 0\text{kWh} \quad [J]$$

eq. 21

where

$Q_{W,bu,op,j}$	heat energy generated by the back-up heater due to operation limit of the heat pump in bin j	[J]
$Q_{W,dis,in,j}$	heat energy input to the domestic hot water distribution system in bin j	[J]
$Q_{W,st,ls,sb,j}$	storage stand-by heat loss to the ambiance of the DHW storage in bin j	[J]
$\theta_{hp,op}$	max. possible temperature by heat pump operation due to the operation limit	[°C]
θ_{hw}	temperature of the delivered hot water in bin j	[°C]
θ_{cw}	temperature of cold water inlet in bin j	[°C]

For the compact unit in Gelterkinden, the operation limit was not reached, since the maximum water temperature of 55°C could be produced with the heat pump. The energy to be delivered by the heat pump follows as difference of the generated heat energy and back-up energy.

On the other hand, the design of the system can be monoenergetic, i.e. heat pump and electric resistance heater as back-up, or bivalent, i.e. any generator with any fuel as back-up, so the heat pump is not designed for the total energy needs and at lower heating capacities of the heat pump the back-up covers a fraction or the total heat requirement depending on the operation mode.

A short-cut method to calculate the required back-up energy is based on the areas under the cumulative frequency (see area BU in Fig. 3) and can be evaluated, if the balance point is given. However, this implies that all relevant influences have been taken into account for the calculation of the balance point. The formulas for a bin-wise approximation of the fraction of back-up energy in the respective bin for the different operation modes are given in the Appendix H. The advantage is that the method is easier to use. However, exactness depends on the calculation of the balance point, so for complex system, the detailed energy balance described in the following should be applied.

The required back-up energy can also be calculated by an energy balance, which can be evaluated by the required running time on the background of the boundary condition for the running time acc. to eq. 20, for higher exactness on a 1 K basis for the lower temperature range up to the temperature where no back-up energy is needed anymore, which is similar to the power balance performed in WPEsti [3]. The detailed calculation has the advantage that different heating capacities in SH and DHW mode can be taken into account and that no balance point is needed as input. Moreover, a time restriction can be considered by the effective bin time.

Tab. 15 gives an overview of the required calculation and the results for the 1 K bin balance for the following calculation steps:

Tab. 15: Detailed balance for the electrical back-up heater

Electrical back-up heater										
Outside temperature [°C]	**Generated heat SH [kWh]**	**Generated heat DHW [kWh]**	**Output capacity SH [kW] (interpolated to supply temperature)**	**Output capacity DHW [kW] (interpolated to average loading temperature)**	**Running time SH [h]**	**Running time DHW [h]**	**Required running time [h]**	**Effective bin time = Maximum running time [h]**	**Missing running time [h]**	**Back-up energy [kWh]**
-11	14.3	0.8	3.2	2.3	4.4	0.3	4.7	3	1.7	5.4
-10	32.3	1.8	3.3	2.4	9.7	0.8	10.5	7	3.5	11.6
-9	35.7	2.1	3.4	2.5	10.4	0.8	11.2	8	3.2	10.9
-8	51.7	3.1	3.5	2.6	14.6	1.2	15.8	12	3.8	13.3
-7	99.8	6.2	3.6	2.6	27.4	2.3	29.7	24	5.7	20.5
-6	132.1	8.5	3.8	2.7	35.2	3.1	38.3	33	5.3	20.1
-5	119.3	8.0	3.9	2.8	30.8	2.8	33.6	31	2.7	10.5
-4	96.1	6.7	4.0	2.9	24.1	2.3	26.4	26	0.4	1.6
-3	255.0	18.5	4.1	3.0	62.2	6.2	68.4	72	-3.6	0
Σ (w/o -3)	**581.3**	**37.2**			**156.6**	**13.6**	**170.2**	**136**	**26.3**	**93.9**

Basis is the generated heat energy for space heating and DHW, i.e. energy need plus storage losses. The required running time can be calculated by the respective heating capacities of the operation mode based of the temperature conditions. If the required running time resulting from this calculation is higher than the effective running time according to eq. 20 back-up energy is required.

The control of the system defines, if the DHW or the space heating requirement is heated by the back-up. The back-up heat can be calculated by the equation

$$Q_{bu,cap,j} = \left(\sum_{i=1}^{n_{bins}} t_{HW,hp,op,tot,j} - t_{eff,j}\right) \cdot \phi_{hp,j} \qquad [J]$$

$$Q_{bu,cap,1} = (4.7\,h - 3\,h) \cdot 3.2\,kW = 5.4\,kWh \qquad [kWh]$$

eq. 22

where

$Q_{bu,cap,j}$	back-up energy due to lack of capacity of the heat pump for the respective operation mode in bin j	[J]
$t_{HW,hp,op,tot,j}$	total running time of the heat pump in bin j	[s]
$t_{eff,j}$	effective bin time in bin j	[s]
$\phi_{hp,j}$	heating capacity of the heat pump for the respective operation mode in bin j	[W]

$\phi_{hp,j}$ is to be set to the respective heating capacity of the heat pump according to the control strategy. In Tab. 15 it is assumed that the back-up energy is used for space heating operation up to -5°C, since the balance point is set to -5°C in case of the pilot plant. For higher temperatures it is accounted to the DHW operation. If no control strategy is known, half of the running time is allocated to space heating and half to DHW operation as default. The back-up energy is already considered in Tab. 14, however, if only the running time on the basis of 3 bins would be evaluated, 13 hours and respectively about 50 kWh more back-up energy would be required in comparison to 1K bin balance performed in Tab. 15 due to the average heating capacity used.

<u>Step 6: Auxiliary energy input</u>

Energy input to the respective auxiliary component k of the compact unit can be calculated by

$$E_{cu,aux,k,in} = P_{cu,aux,k} \cdot t_{cu,op,aux,k} \qquad [J]$$

$$E_{H,aux,sk,in} = 54W \cdot 5196h = 281kWh \qquad [kWh]$$

eq. 23

where

$E_{cu,aux,k,in}$	auxiliary energy use of the compact unit not contained in the characteristics of standard testing	[J]
$P_{cu,aux,k}$	nominal power of the auxiliary component k of the compact unit	[W]
$t_{cu,op,aux,k}$	running time of the auxiliary component k of the compact unit	[s]
$E_{H,aux,sk,in}$	auxiliary energy used for the space heating circulation pump	[J]

The running time of the auxiliary component has to be evaluated by the used control strategy. If the control strategy is not known, the running time of the heat pump is assumed for the source pump and the heating period for the circulation pump. For the stand-by input the difference between heating period and running time of the heat pump is set. The heat recovery fans are assumed to run the entire heating period as in the evaluation of the field monitoring.

An overview of the calculated auxiliary energies based on the measured power of the components of the compact unit in Gelterkinden is given in Tab. 16.

Tab. 16: Auxiliary energies of the compact unit

Auxiliary energy use of the compact unit				
Auxiliary component	**Running time acc. to**	**Power [W]**	**Running time [h]**	**Energy [kWh]**
Heat recovery fans, control, transformers	Up to 20°C	40	7398	296
Source pump/fan SH	Heat pump	60	2310	Contained in COP
Heat pump stand-by SH	Heating period – heat pump	10	2873	29
Sink pump SH	Heating period or Heat pump	54	5196	281
Source pump/fan DHW	Running time DHW	60	508	Contained in COP
Storage loading pump DHW	Heat pump DHW	33	508	17
Heat pump stand-by DHW	Total DHW activation time – Heating period – DHW running time	10	3056	31
Total auxiliary energy use				654

Step 7: Electrical energy input

The heat energy needs are transformed to electrical energy input by means of the COP-characteristic according to test procedure described in chap. 3.3, which is based on EN 14511 testing of the heat pump. If no values of DHW testing according to EN 255-3 are available – as here for the LWZ 303 SOL – the characteristic of the space heating operation can be evaluated at an average loading temperature for the storage.

However, this is only possible for alternate combined operating heat pump systems, since in these system configurations, the heat pump is switched either to the space heating or to the DHW operation. For simultaneous systems, additional test results for the simultaneous operation have to be available, since the characterstic may change significantly.

The relevant supply temperature for the heat pump increases from this colder water temperature during the loading to temperatures slightly above the maximum hot water temperature due to the required temperature difference for the heat transfer. Therefore, the average temperature for the loading is lower than the maximum hot water temperature that can be reached by the heat pump operation.

In the case of Gelterkinden, the maximum temperature was changed during the yearly operation, thus for the validation the average temperature has been evaluated from the measurements to $\theta_{w,avg}$=46°C.

Evaluation of the COP dependency on the source and sink temperature is only correct, if the mass flow rate corresponds to the mass flow rate used during the standard testing, since otherwise, there are different temperature conditions at the heat pump condenser.
Therefore, the temperature spread of the heat pump based on the mass flow rate defined by the design of the emission system has to be taken into account.

In case of the proposed testing in chap. 3.3 based on EN 14511 the temperature spread at the standard rating point (A7/W35 for air-to-water heat pumps and floor heating application) is fixed to 5 K. With the temperature spread, the mass flow rate for the testing is determined and applied to all test points. Thus, the temperature spread during testing for the different operating points can be determined. The mass flow in operation can be determined by the design conditions of the heating system, i.e. heating capacity of the heat pump and design temperature spread at outdoor design temperature. For the LWZ 303 SOL the water volume flow in the testing is 720 l/h acc. to [2], the volume flow in operation in the pilot plant Gelterkinden is 960 l/h. Thus, the temperature spread can be calculated according to the equation

$$\Delta\theta_{H,nom,j} = \frac{\phi_{H,hp,nom,j}}{\dot{m}_{H,nom} \cdot c_w} \qquad [K]$$

$$\Delta\theta_{H,nom,1} = \frac{3360\,W}{0.2\,kg/s \cdot 4182\,J/kg/K} = 4.02\,K \qquad [K]$$

eq. 24

where

$\Delta\theta_{H,nom,j}$	nominal temperature spread in testing	[K]
$\phi_{H,hp,nom,j}$	nominal heating capacity of the heat pump in bin j	[W]
$\dot{m}_{H,nom}$	nominal massflow of the heat transfer medium	[kg/s]
c_w	specific heat capacity of water	[J/kg/K]

The correction of the COP characteristic is accomplished according to eq. 25 derived by Ehrbar[3] based on the fixed exergetic efficiency method, see Appendix C. eq. 25 is taken from [3].

$$COP_{H,j} = COP_{H,nom,j} \cdot \left[1 - \frac{\frac{\Delta\theta_{H,nom,j} - \Delta\theta_{H,op,j}}{2}}{\left\{ T_{H,sk,j} - \frac{\Delta\theta_{H,nom,j}}{2} + \Delta T_{H,sk,j} - (T_{H,src,j} - \Delta T_{H,src,j}) \right\}} \right] \qquad [-]$$

$$COP_{H,1} = 2.92 \cdot \left[1 - \frac{\frac{4.02\,K - 3.01\,K}{2}}{\left\{ (35°C + 273.15)K - \frac{4.02\,K}{2} + 4\,K - ((-7°C + 273.15)K - 15\,K) \right\}} \right] = 2.91 \cdot 0.99 = 2.89$$

eq. 25

where

$COP_{H,j}$	COP corrected for a different temperature spread in testing and operation	[-]
$COP_{H,nom,j}$	COP derived from standard testing (e.g. according to EN 14511)	[-]
$\Delta\theta_{H,nom,j}$	temperature spread on the condenser side due to standard test conditions	[K]
$\Delta\theta_{H,op,j}$	temperature spread on the condenser side in operation	[K]
$T_{H,sk,j}$	sink temperature	[°C]
$\Delta T_{H,sk,j}$	average temperature difference between heat transfer medium and refrigerant in condenser	[K]
$T_{H,src,j}$	source temperature	[°C]
$\Delta T_{H,src,j}$	temperature difference between heat transfer medium at the inlet of the evaporator and the refrigerant	[K]

According [3] the temperature difference in the condenser and evaporator between the heat transfer medium and the refrigerant can be approximated by ΔT_{sk}=ΔT_{src}=4 K for water-based components. In the case of air-based components ΔT_{sk}=ΔT_{src}=15 K is set.

The approach delivers the same values as the tabuled values in VDI 4650 [50] for an average temperature range, i.e. the test point A2/W35.

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Tab. 17: Temperature spreads for different conditions in testing and operation

Temperature spreads				
	Sink temperature	**Source temperature**		
		-7°C	**2°C**	**7°C**
Temperature spread in testing [K]	**35°C**	4.02	5.07	5.57
	50°C	2.85	3.72	4.72
Temperature spread in operation [K]	**35°C**	3.01	3.80	4.18
	50°C	2.13	2.79	3.54

The corrected COP characteristic for space heating operation has to be evaluated from the tested heat pump characteristic according to the test points described in chap. 3.3, i.e. it is interpolated for the respective source and supply temperature of the operating points. The supply temperature is evaluated by the heating characteristic curve. Tab. 17 gives the temperature spreads.
Tab. 18 gives the correction factor and the corrected COP characteristic for the temperature spread in testing and in operation.

Tab. 18: Correction of COP characteristic for different flow conditions in testing and operation

Correction factor for COP and corrected COP characteristic				
	Sink temperature	**Source temperature**		
		-7°C	**2°C**	**7°C**
Correction factor [-]	**35°C**	0.99	0.99	0.98
	50°C	0.99	0.99	0.99
Corrected COP [-]	**35°C**	2.88	3.23	3.48
	50°C	2.01	2.27	2.68

Tab. 19 shows the heat sink temperature corrected COP values. The supply temperature has been derived by the heating curve (see Fig. 49), which defines the set point supply temperature as a weighted average of 70% return temperature and 30% supply temperature. Therefore, the temperature spread of Tab. 17 has been used to evaluate the supply temperature according to the equation

$$\theta_{f,j} = \theta_{H,sp,j} + 0.7 \cdot \Delta\theta_{H,op,j} \quad [°C]$$

$$\theta_{f,1} = 28.5°C + 0.7 \cdot 3.01K = 30.6°C \quad [°C]$$

eq. 26

where

$\theta_{f,j}$	effective supply temperature	[°C]
$\theta_{H,sp,j}$	setpoint temperature according to heating curve in bin j	[°C]
$\Delta\theta_{H,op,j}$	temperature spread in bin j	[K]

Tab. 19: Resulting COP-values at the operation temperature conditions (flow temperature)

Resulting COP Values				
Space heating operation	**Source temperature**			
	-7°C	**2°C**	**7°C**	**20°C**
Supply temperature in operation [°C]	30.6°C	29.2°C	28.2°C	-
COP SH at supply temperature [-]	**3.13**	**3.60**	**3.85**	**-**
DWH operation	**Source temperature**			
	-7°C	**2°C**	**7°C**	**20°C**
Average DHW loading temperature [°C]	**46°C**			
Interpolated COP at average loading temp. [-]	**2.25**	**2.55**	**2.93**	**3.71**

The electricity consumption of the heat pump in bin j can be calculated according to the eq. 27 and added up over all bins.

$$E_{hp,in,j} = \sum_{i=1}^{nbins}\left(\frac{Q_{H,hp,j}}{COP_{H,j}} + \frac{Q_{W,hp,j} - Q_{W,st,ls,sb,j}}{COP_{W,i}} + P_{W,st,ls,sb} \cdot t_{op,tot}\right) \quad [J]$$

$$E_{hp,in,1} = \sum_{i=1}^{4}\left(\frac{1133kWh}{3.13} + \frac{85kWh}{2.25}\right) = 399kWh \quad [kWh]$$

eq. 27

where

$E_{hp,in,j}$	electricity consumption of the heat pump in bin j	[J]
$Q_{H,hp,j}$	produced heat by the heat pump for space heating bin j	[J]
$Q_{W,hp,j}$	produced heat for by the heat pump for DHW in bin j	[J]
$Q_{W,st,ls,sb,j}$	storage stand-by heat loss to the ambiance of the DHW storage in bin j	[J]
$COP_{H,j}$	COP for space heating at the operating point of bin j	[-]
$COP_{W,j}$	COP for DHW operation at the operating point of bin j	[-]
$P_{W,st,ls,sb}$	electrical power input to cover DHW storage losses according to EN 255-3	[W]
$t_{op,tot}$	total activation time of the compact unit for DHW	[s]

The COP of the DHW operation is only tested for one point. Therefore, the fixed exergetic efficiency method or the space heating characteristic given in Appendix C can be used to correct the COP in DHW mode for the source temperature, if required.
If no electrical power input to cover the storage losses according to EN 255-3 is available, the electricity in DHW operation can be calculated with the thermal DHW storage losses in the same way as space heating operation. Tab. 20 summarises the electrical energy input.

With the electricity input for the operation, seasonal performance factors can be calculated for different system boundaries and different operation modes.

Tab. 20: Electrical energy for heat pump, back-up, ventilation unit and auxiliary components

Electrical energy input					
	Bin 1	**Bin2**	**Bin3**	**Bin 4**	**Total**
Space heating mode					
Requirement of the heat pump for SH [kWh_{th}]	1135	4557	5178		10870
COP SH at operating point [W/W]	3.13	3.60	3.85		
Electrical energy use for heat pump SH [kWh_{el}]	363	1266	1345		2974
Heat energy produced by back-up SH [kWh_{th}]	92				92
Electrical back-up energy use for SH [kWh_{el}]	97				97
Electrical energy use stand-by SH [kWh_{el}] (Tab. 16)					29
Electrical energy use ventilation unit [kWh_{el}] (Tab. 16)					296
Electrical energy use sink pump [kWh_{el}] (Tab. 16)					281
Domestic hot water mode					
Requirement of the heat pump for DHW [kWh_{th}]	82	413	839	917	2251
COP at operating point, average supply temp. [W/W]	2.25	2.55	2.93	3.71	
Electrical energy use of heat pump DHW [kWh_{el}]	36	162	286	247	731
Heat energy produced by back-up DHW [kWh_{th}]	2				2
Electrical energy use back-up DHW [kWh_{el}]	2				2
Electrical energy use stand-by DHW [kWh_{el}] (Tab. 16)					31
Electrical energy use storage pump [kWh_{el}] (Tab. 16)					17

4.2.3 System boundaries and characteristic numbers

Fig. 17 shows the system boundaries and characteristic numbers used for the assessment of the compact unit.

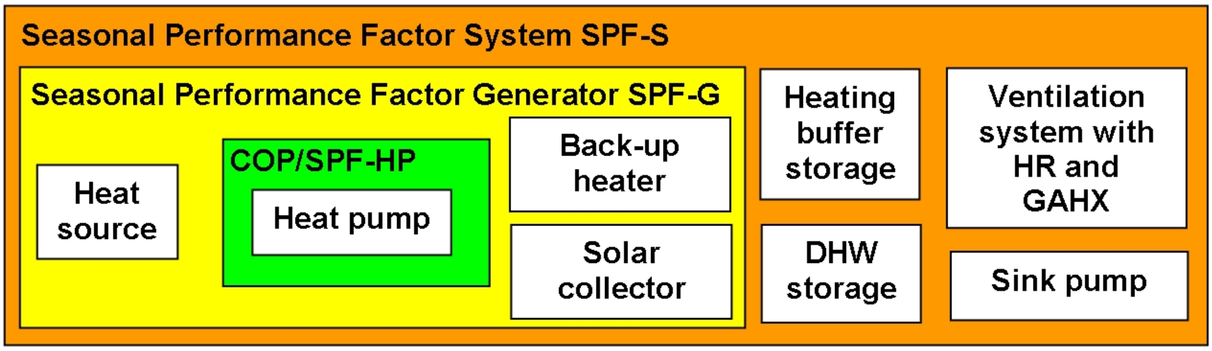


Fig. 17: System boundaries and characteristic numbers used

Seasonal performance factor heat pump SPF-HP

$$SPF-HP = \frac{Q_{H,hp} + Q_{W,hp}}{E_{HW,hp,cmp,in} + E_{HW,hp,fan,in}} \quad [-]$$

$$SPF-HP = \frac{10870\,kWh + 2251\,kWh}{2974\,kWh + 731\,kWh} = 3.5 \quad [-]$$

eq. 28

The SPF-HP is the ratio of the heat energy produced by the heat pump and the corresponding energy need of the heat pump compressor and source fans in case of the A/W heat pump. This system boundary corresponds to the system boundary for the COP.

Seasonal performance factor generator SPF-G (equivalent to efficiency of oil or gas fired boilers)

$$SPF-G = \frac{Q_{H,hp} + Q_{W,hp} + Q_{HW,bu} + Q_{HW,sol,out}}{E_{HW,hp,cmp,in} + E_{HW,hp,fan,in} + E_{HW,bu,in} + E_{HW,ctr,aux,in} + E_{W,aux,sol,in}} \quad [-]$$

$$SPF-G = \frac{10870\,kWh + 2251\,kWh + 92\,kWh + 2\,kWh}{2974\,kWh + 731\,kWh + 97\,kWh + 29\,kWh + 31\,kWh} = 3.4 \quad [-]$$

eq. 29

The SPF-G is the ratio of the heat energy produced by all generators and the corresponding electrical energy input to these generators.

Seasonal performance factor system SPF-SYS

$$SPF-S = \frac{Q_{H,dis,in} + Q_{W,dis,in} + Q_{H,ve,out}}{\sum E_{HWV,cu,in}} \quad [-]$$

where

$$Q_{H,ve,out} = (H_{V,sa} - H_{V,oa})$$

$$\sum E_{HWV,cu,in} = E_{H,hp,in} + E_{W,hp,in} + E_{HW,bu,in} + E_{H,aux,sk,in} + E_{W,aux,sk,in} + E_{HWV,ctr,aux,in} + E_{V,fan,in} + E_{W,aux,sol,in}$$

$$SPF-S = \frac{10960\,kWh + 1179\,kWh + 1054\,kWh}{2974\,kWh + 731\,kWh + 99\,kWh + 281\,kWh + 17\,kWh + 60\,kWh + 296\,kWh} = 3.0$$

eq. 30

The SPF-S is the ratio of the energy need of the distribution system for space heating and

DHW including the recovered heat and heat input to the ventilation system and the total electrical energy input to the single components of the compact unit. Since the recovered heat is included, the SPF-SYS can be higher than the SPF-G
where

SPF-HP	seasonal performance factor of the heat pump	[-]
SPF-G	seasonal performance factor generator	[-]
SPF-S	seasonal performance factor system	[-]
$Q_{H,hp}$	produced heat energy of the heat pump in space heating operation	[J]
$Q_{W,hp}$	produced heat energy of the heat pump in domestic hot water operation	[J]
$Q_{HW,bu}$	produced heat energy of the electrical back-up heater for SH and DHW	[J]
$Q_{HW,sol,out}$	produced heat energy by the solar system	[J]
$Q_{H,dis,in}$	heat energy need of the space heating distribution system	[J]
$Q_{W,dis,in}$	heat energy need of the DHW distribution system	[J]
$Q_{H,ve,out}$	heat energy for space heating supplied by the ventilation system	[J]
$H_{V,sa}$	enthalpy of the supply air	[J]
$H_{V,oa}$	enthalpy of the outdoor air	[J]
$E_{HWV,cu,in}$	electrical energy use of the compact unit for SH, DHW and ventilation	[J]
$E_{HW,hp,cmp,in}$	electrical energy use of the heat pump compressor for SH and DHW	[J]
$E_{HW,hp,fan,in}$	electrical energy use of the heat pump source fan for SH and DHW	[J]
$E_{HW,bu,in}$	electrical energy use of the electrical back-up heater for SH and DHW	[J]
$E_{HWV,ctr,aux,in}$	auxiliary electrical energy input for the control and stand-by operation	[J]
$E_{HWV,cu,in}$	total electrical energy use of the compact unit	[J]
$E_{H,aux,sk,in}$	auxiliary electrical energy use to the sink pump of the space heating system	[J]
$E_{W,aux,sk,in}$	auxiliary electrical energy use to the sink pump of the DHW system	[J]
$E_{W, aux,sol,in}$	auxiliary electrical energy use for the solar circuit pump	[J]
$E_{V,fan,in}$	electrical energy use of fans of the ventilation system	[J]

The energy amounts and the respective seasonal performance factors for the single operation modes and the overall seasonal performance for the compact unit are summarised in Tab. 21.

Tab. 21: Seasonal Performance Factors of the compact unit for space heating, DHW and overall

Seasonal performance factors for the compact unit	
SPF-HP space heating [-]	10870/2974=**3.7**
SPF-HP domestic hot water [-]	2251/731=**3.1**
SPF HP overall	(10870+2251)/(2974+731)=**3.5**
SPF-G space heating [-]	(10870+92)/(2974+97+29)=**3.5**
SPF-G domestic hot water [-]	(2251+2)/(731+31+2)=**3.0**
SPF-G overall [-]	(10870+2252)/(2974+97+29+731+31+2)=**3.4**
SPF-S space heating [-]	(10870+92+1054)/(2974+97+29+281+296)=**3.3**
SPF-S domestic hot water [-]	1179/(731+31+17+2)=**1.5**
SPF-S overall [-]	(12016+1179)/(2974+731+99+281+17+60+296)=**3.0**

4.2.4 Sensitivity analysis of the method

In order to characterise the impact of the parameters on the seasonal performance a sensitivity analysis has been performed. The method has been applied based on three bins at the test points of an air-to-water heat pump A-7/W35, A2/W35, A7/W35.

Tab. 22: Reference values for the sensitivity analysis of the bin method

Input data for sensitivity analysis	
Energy requirement	12014 kWh
Bin weighting factors	Lower: 0.11, Middle: 0.41, Upper: 0.48
COP characteristic	According to Tab. 10
Heating limit	14°C
Temperature change coefficient HRU	0.82
Heating curve	supply temperature at outdoor design -8°C: 30.6°C; supply temperature at heating limit 14°C: 28.2°C
Constant sink temperature	No heating curve, constant supply at 30.7°C
Nominal power sink pump	54 W
DHW-storage losses	123 W
SPF-S	3.27

The reference values of the parameters (corresponding to 0% deviation) based on the evaluation of the pilot plant in Gelterkinden are given in Tab. 22. Fig. 18 presents the sensitivity based on variation of the input parameters in a range of up to ±50%.

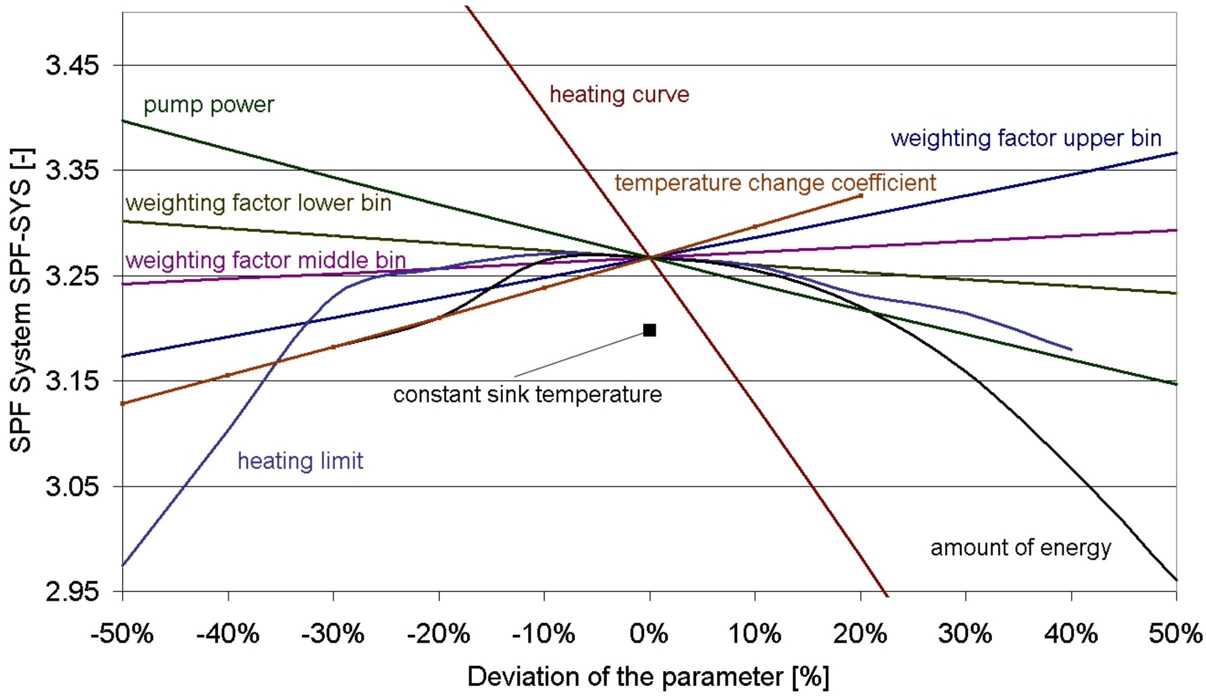


Fig. 18: Sensitivity of different parameters on the seasonal performance factor system SPF-SYS for a low-energy house (based on pilot plant Gelterkinden)

Even though the relative change in SPF also depends on the system configuration and parameters, a tendency can be derived by this particular case.
For the evaluation of the impact of the heating curve, the effective supply temperature has been elevated by the respective percentage. As can be easily seen, the **heating characteristic curve** has the largest impact on the SPF-SYS due to the change in sink temperature, which significantly changes the COP and the heating capacity of the heat pump. Since the design values were already quite low, only a deviation to the lower values of 10% was reasonable.
As linear interpolation of the COP characteristic is applied, a linear impact results.

The **constant sink temperature** of 30.7°C has a similar impact, even though the spread of the supply temperature at θ_{op}=-7 to θ_{op}=7 is only 2.4°C (acc. to Tab. 19) and thereby quite low due to the installed floor heating system. In a radiator heating system with a much stronger inclination of the heating curve and a higher supply temperature spread over the heating period, the changes would be much more significant.

In a range of ±20% the **energy need of the space heating distribution system** has only limited impact on the SPF-S, since in this range, the heat pump can cover most of the energy requirement, even though the given design is monoenergetic with a balance point of $\theta_{H,bal}$=-4°C (taking into account SH and DHW, see Tab. 15). For higher deviation, more and more back-up operation lowers the seasonal performance factor. For deviations to the lower end, the additional auxiliary consumption has a higher impact due to the lower total energy. The curve is almost identical to a higher temperature change coefficient, which increases the energy due to lower recovered ventilation losses.

The **temperature change coefficient** defines the amount of energy recovered by the heat recovery and thus reduces the energy to be produced by the heat pump. The impact is stronger than in case of the change of energy need, since the electro-thermal amplification of the heat recovery is normally higher than the heat pump seasonal performance, so it has a higher impact in the weighting of the SPF-SYS.

Weighting factors of the heat pump operation in the single bins have only limited impact on the SPF-SYS. This is due to the fact, that the spread of the COP values over the operation range is limited. The sum of the weighting factors is always one. If the respective weighting factor, e.g. of the middle bin, is lowered by 10%, the others are increased each half of the resulting change. The weighting factors for the upper and lower bin have a stronger impact, since the middle bin is related to an average COP value of the characteristic. The slightly higher impact of the upper bin is due to the fact that the weighting factor of the upper bin is higher, thus percental deviation is higher, as well. In fact, the lower bin has the strongest impact.

The **heating limit** basically changes the bin weighting factors, thus for a range of ±20%, the impact is limited, too. The slight oscillation is due to the fact, the temperature values have to be rounded to an integer value due to the resolution of the processing of the frequency of outdoor air temperatures in 1 K steps. The values of +50% is already higher than the indoor temperature. The stronger inclination to lower values is due to the increasing reduction of the upper bin and therefore increasing use of back-up energy.

The **pump power of the sink pump** has been chosen as parameter to characterise additional auxiliary consumption, since in many systems it still runs through the whole heating period for control reasons. Therefore, even though the power is small, the running time and thereby the energy can have quite an impact, so design values shall be kept as low as possible. In this case a 50% lower nominal pump power can increase the seasonal performance by about 4%. The same statement is valid for the heat recovery fans. For ground-to-water heat pumps special attention shall be paid to the design of the source pump – the fan for A/W heat pumps is usually included in the COP value - since the required power is normally higher than the sink pump and therefore there are bigger optimisation potentials.

By this evaluation it can be concluded, that in **DHW operation, the DHW temperature and the storage losses**, which may reach shares of above 50% of the DHW energy depending on the DHW consumption, have the biggest impact on the system seasonal performance, since changes in COP are restricted to changes in source temperature. A 50% reduction of the storage losses increases the SPF-SYS of the DHW operation by 30%.

Summarising, the dominating impact on the seasonal performance are the COP values and respectively the temperature conditions. Therefore, temperature conditions shall be taken into account in the calculation as detailed as possible.

4.3 Application for MINERGIE-P® building and different heating systems

In order to compare results of the calculation for different heating systems, a house according to the MINERGIE-P® standard (www.minergie.ch) has been calculated on the one hand with an air heating system and on the other hand with a floor heating system. Values of the MINERGIE-P® dwelling are based on design values of the pilot plant in Zeiningen, BL treated in chap. 6.2. Tabulated values according to the system boundaries depicted in Fig. 17 are given in Appendix D.

4.3.1 Input data for the calculation

The input data for the comparison are given in Tab. 23.

Tab. 23: Typical values of compact unit for the comparison of heating systems

Space heating mode				
Hourly values of outdoor temperature Basel-Binningen (evaluated from Meteonorm [18])				
Design indoor temperature [°C]	20			
Space heating energy need[1)] [kWh]	6344			
Design outdoor temperature [°C]	-8			
Range supply temperature floor heating [°C]	31 – 28			
Supply temperature air heating [°C]	50 (constant air heating)			
Upper temperature limit for heating [°C]	9			
Type of testing data	HRU/HP combined, total volume flow 250–300m^3/h			
HP characteristic SH	*A-7/W35*	*A2/W35*	*A7/W35*	
Heating capacity [kW] at range 107-163 m^3/h	1.4	1.5	1.7	
COP [-] at range 107-163 m^3/h	2.3	2.7	2.9	
	A-7/W45	*A2/W45*	*A7/W45*	
Heating capacity [kW] at range 107-163 m^3/h	1.2	1.4	1.5	
COP [-] at range 107-163 m^3/h	1.8	2.1	2.2	
Heat recovery characteristic	*w/o ground-air HX*	*with ground-air HX*		
Heat recovery efficiency at volume flow rate 107 - 163 m^3/h	0.8	0.83		
Volume flow rate heat recovery [m^3/h]	150			
total volume flow rate (HR and HP) [m^3/h]	300			
Auxiliary energy				
Nominal power heat recovery fans [W]	40			
Nominal power of the circulation pump [W]	40			
Additional electrical input (e.g. control) [W]	20			
Domestic hot water mode				
HP characteristic DHW (EN 255-3)				
DHW heat energy need[2)] [kWh]	2750			
DHW reference temperature [°C]*(EN 255-3)*	48			
Storage losses[3)] [kWh]/Electricity input [W]	460 / 30			
	A-7	*A2*	*A7*	*A20*
Heating capacity DHW [kW]	1.2	1.3	1.4	1.6
COP_t [-] (EN 255-3)	1.8	2.0	2.1	2.4

NOTE: [1)] recovered ventilation losses not subtracted; [2)] based on SIA 380/1 [3)]based on default values of Swiss energy regulation [24] for 200 l storage

4.3.2 MINERGIE-P® building with air-heating system

For the air heating system two cases have been considered, with and without an installed ground-to-air heat exchanger. As can be seen in Tab. 23 the supply temperature for the air heating system is set to a constant temperature of 50°C.

Fig. 19 shows the produced heat and the electrical energy fraction for an air heating system.

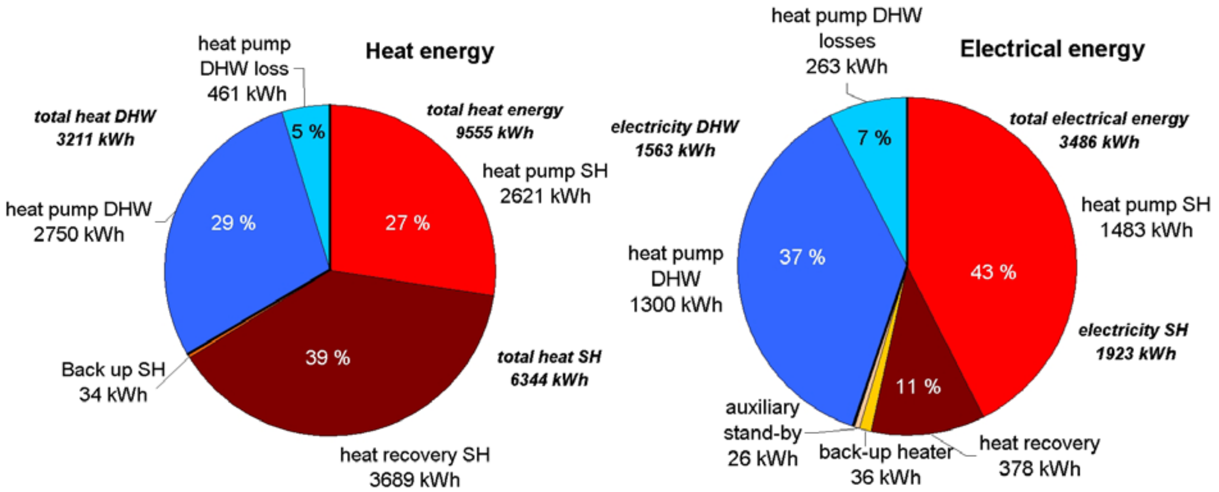


Fig. 19: Heat produced (left) and electrical energy input (right) for an air-heating system

Fig. 20 presents the same for an air heating system with an installed ground-to-air heat exchanger. The ground-to-air heat exchanger is designed to lift the inlet temperatures to the heat recovery at outdoor air temperature of θ_{oa}=-7°C to $\theta_{V,hru,in}$=2°C, at θ_{oa}=2°C to $\theta_{V,hru,in}$=5.5°C and at θ_{oa}=7°C to $\theta_{V,hru,in}$=8°C.

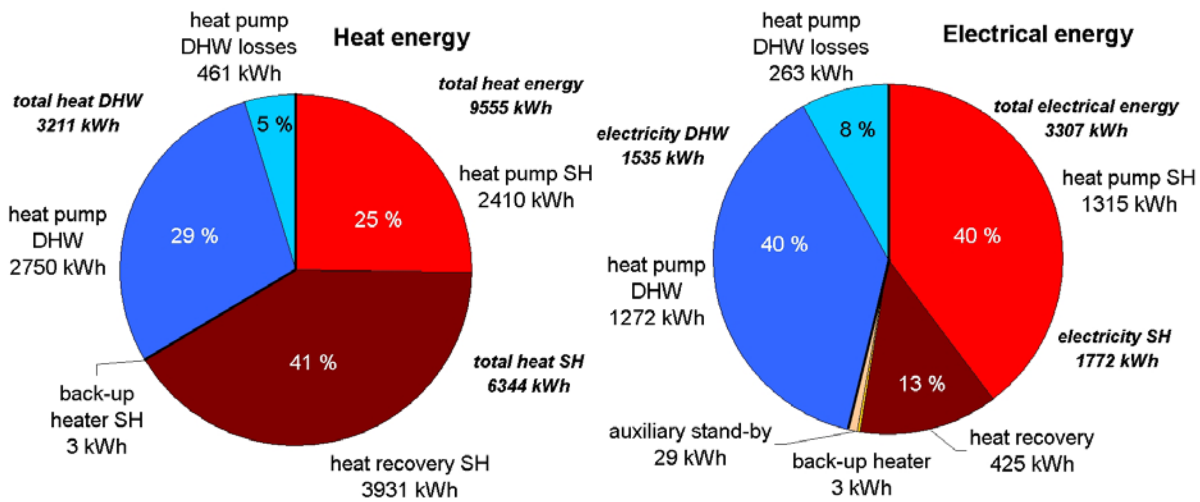


Fig. 20: Produced heat (left) and electrical energy (right) for air-heating system with GAHX

The temperature change coefficient increases from 0.8 to 0.83, see Tab. 23, due to the continuous operation without defrost interruption.

Therefore, the heat recovery operation results in a higher reduction of ventilation losses than in the case without GAHX. The heat pump operation is more efficient due to the higher inlet temperatures of the heat pump caused by the higher inlet temperature into the heat recovery and therefore, less back-up energy is needed. The electrical energy consumption of the heat pump is slightly reduced, on the one hand due to the lower heat requirement of the heat pump, on the other hand due to the better COP at higher inlet temperature.

On the other hand, the electrical energy consumption of the heat recovery increases due to the higher pressure drop of the inlet air duct with GAHX. This reduces the benefits of the GAHX. Altogether, the electrical energy input is decreased by about 5% or about 180 kWh.

4.3.3 MINERGIE-P® building with floor-heating system

The floor heating system has the advantage of a significantly lower temperature requirement for the heat pump. Thus, a much better heat pump performance can be expected. The floor heating system is operated at temperatures of 31°C to 28°C dependent on the outdoor temperatures.

Fig. 21 gives the heat energy and the electrical energy consumption. The heat production resembles the case of the air heating.
However, the electrical energy consumption shown in the right part of Fig. 21 differs considerably. The electrical energy input for the space heating is reduced by about 600 kWh. However, the floor heating requires a circulation pump which adds 165 kWh to the consumption. Due to higher heating capacity and thereby shorter running time, the stand-by energy increase a bit, as well. Summing-up, the electrical energy consumption decreases by 435 kWh.

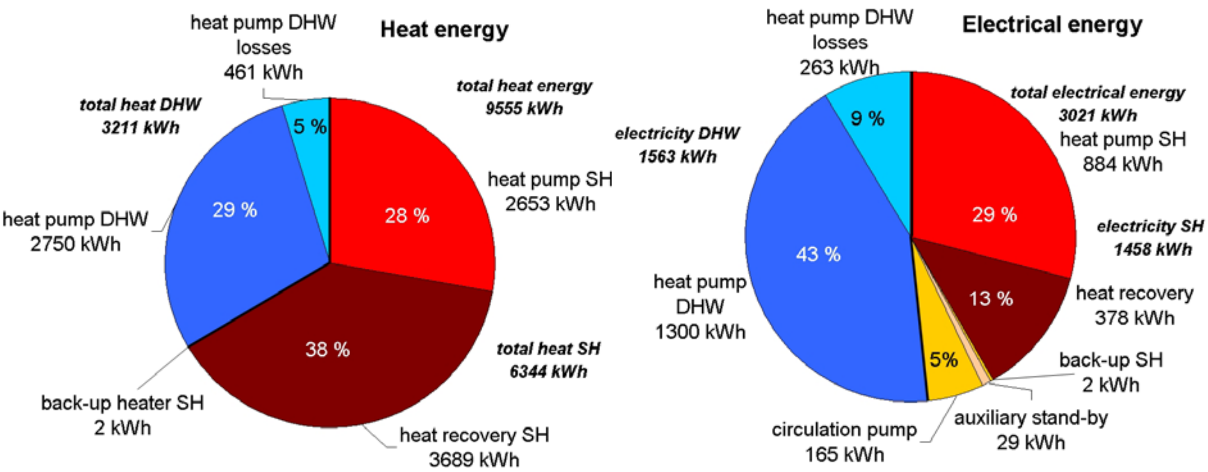


Fig. 21: Produced heat (left) and electrical energy input (right) for an floor-heating system

4.3.4 Discussion of the results

Fig. 22 gives a summary by a comparison of the seasonal performance of the generator SPF-G and seasonal performance factor system SPF-S.

As mentioned before the system boundary of the SPF-G does not include the heat recovery, so it characterises the heat pump and in case the back-up operation.

The ground-to-air heat exchanger increases the seasonal performance system moderately by about 0.15 due to the low weighting factor of the lower bin, where the ground-to-air heat exchanger delivers the highest temperature rise. On the other hand, the additional pressure drop of the GAHX causes increased ventilation power. The reduction of the ventilation losses does not increase much, since less heat is recovered by the heat recovery due to the higher inlet temperature. The benefit is mostly the higher inlet temperature for the heat pump and the continuous operation of the heat recovery without defrosting interruptions.

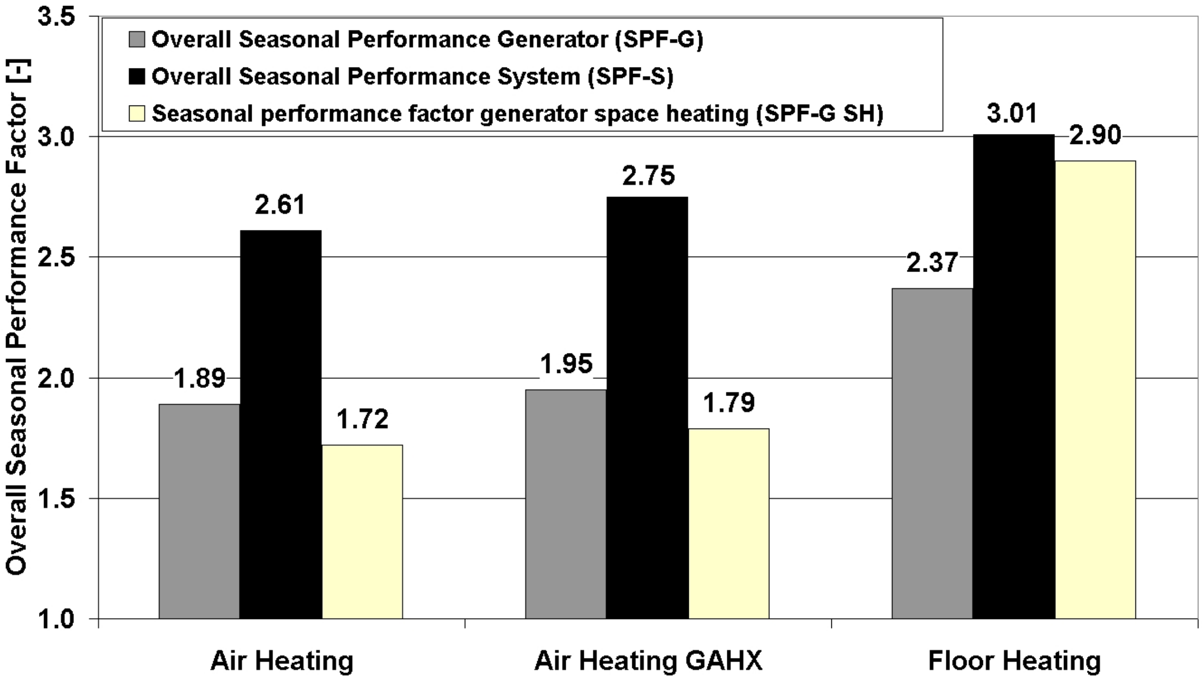


Fig. 22: Comparison of generator and system seasonal performance factor of different heating systems

The significant difference is between the air-heating and the floor-heating system due to the lower required temperature level. This can be seen in the SPF-G SH number, i.e. the SPF of the heat pump/back-up in space heating mode is notably increased. On the other hand, more auxiliary energy is needed due to an additional circulation pump compared to the case that for the air-heating system the condenser is directly in the supply air stream and no additional intermediate circuit is applied.

However, due to the higher DHW need in ultra-low energy houses, which can reach fractions around 50%, the difference is not so obvious as the temperature impact in the sensitivity analysis seen in chap. 4.2.4 for MINERGIE® houses. This can be seen in the SPF-G number, which includes the DHW operation, as well.

Actually, only an energy fraction of about 30% depends on the supply temperature, since also the ventilation does not depend on the supply temperature of the heating system, so both DHW and ventilation are not affected by the type space heating distribution system and the respective temperature requirements. So even for this extreme systems with high constant supply temperature of the air heating system and quite low variable supply temperature of the floor heating system, the difference for the overall system seasonal performance factor SPF-S is limited.

Summarising, air heating systems have a higher electrical energy consumption due to the required sink temperature level. By a ground-to-air heat exchanger, which can also be used for cooling in summer time, the seasonal performance can be increased, but does not reach the level of a floor heating system.

However, taking into account the costs, air heating systems may still be the preference in ultra-low energy houses, e.g. according to MINERGIE-P®, since investment costs of the floor heating distribution system are saved.

An in-depth comparison of air- and floor-heating systems in terms of exergy considerations is presented in the next chapter.

4.4 Exergy consideration

To characterise the efficiency and the losses of different heating systems an analysis with regard to the energy is limited, as only the amount of the energy flows is considered according to the first law of thermodynamics, but not the quality of the energy conversion in the sense of the second law of thermodynamic. This can be taken into account by an exergy analysis, which considers the thermodynamic losses caused by irreversibilities. Thus, the exergy seems well suited for an in-depth analysis of different heating systems in order to quantify the losses. The thermodynamic ideal is thereby always an fully reversible process.

4.4.1 Basics of an exergy analysis

Exergy is defined as the fraction of the energy that can be entirely transformed in each of the different energy forms. For instance, electrical energy is pure exergy, since it can be transformed entirely into other energy forms, e.g. heat or mechanical rotation energy. Heat, however, is not pure exergy, since only the fraction above an ambient temperature level can be arbitrarily transformed (exergy), while the rest is already in an equilibrium with the ambiance and cannot be transformed anymore (anergy). Thus, any energy flow can be decomposed in a fraction of exergy and anergy related to specified conditions of the ambiance (temperature, pressure). Only in case of the limit of an entirely reversible process, the exergy fraction stays constant. In all real processes, though, the fraction of exergy is converted to anergy during the transformation (exergy loss) due to irreversible state changes. So, the direction of natural processes is defined by the state change towards a lower exergy content and consequently, anergy cannot be transformed to exergy.

The most important natural mechanisms for exergy losses in the context treated here are heat transfer, which is irreversible, since heat flux will always take place from the hot temperature to cold temperature, and dissipation by friction (e.g. transformation of exergy in the form of mechanical energy to a fraction of mechanical energy and heat, where the anergy fraction of the heat characterises the exergy loss).

Applied to heat pump systems the energy conversion can be described by the two thermodynamic laws (*numeric values are taken from the sample for the compact unit with air heating in the next chapter, small deviations of the numeric values are due to rounding and averaging of the temperature curves in the heat exchangers*)

1. law of thermodynamics (energy balance):

$$\dot{Q}_c + P_{hp,cmp} = \dot{Q}_h \qquad [W]$$

$$689\,W + 512\,W = 1201\,W$$

eq. 31

i.e. the anergy of the source Q_c plus exergy supplied to the heat pump compressor $P_{hp,cmp}$ delivers the heat energy removed at the condenser Q_H. Q_c in eq. 31 is considered as pure anergy from the ambiance.

2. law of thermodynamics (in the form of an exergy balance):

$$P_{hp,cmp} = \eta_c(T_{amb} / T_h) \cdot \dot{Q}_h + \dot{E}_v = (1 - \frac{T_{amb}}{T_h}) \cdot \dot{Q}_h + \dot{E}_v = \dot{E}_h + \dot{E}_v \qquad [W]$$

$$P_{hp,cmp} = \eta_c(269\,K / 293\,K) \cdot 1201\,W + 477\,W = (1 - \frac{269\,K}{293\,K}) \cdot 1201\,W + 407\,W = 505\,W$$

eq. 32

i.e. exergy supplied to the compressor $P_{hp,cmp}$ equals the exergy removed at the condenser $\dot{E}_h$ and the exergy losses $\dot{E}_v$ of the process. In other words, all exergy losses in the different components of the heat pump and parts of the system periphery (e.g. the emission system) have to be supplied as exergy to the compressor of the heat pump.

The exergy content of the removed heat is equal to the maximum fraction of work that can be extracted from the heat flux under the respective ambient conditions and is therefore calculated by multiplication with the Carnot efficiency η_C, which is only dependent on the hot and the ambient temperature level.
The exergy content transported with a fluid flow, e.g. an air flow can be calculated by the equation

$$\dot{E}_a = \dot{m}_a \cdot (h_a - h_{amb} - T_{amb} \cdot (s_a - s_{amb})) = \dot{m}_a \cdot (c_a \cdot (T_a - T_{amb} - T_{amb} \cdot \ln(\frac{T_a}{T_{amb}})) - R_a \cdot T_{amb} \cdot \ln(\frac{p_a}{p_{amb}}))$$

eq. 33

where the second part of the equation refers to the approximation for ideal gases.
Exergy losses can be differentiated in internal exergy losses of the refrigerant cycle, which are basically due to non-isentropic compression and expansion and internal friction in the refrigerant cycle, and external exergy losses mainly due to heat transfer at finite temperature differences and friction of the source and sink fluid. The exergy loss due to the irreversible heat transfer depends on the temperature difference for the heat transfer, the temperature level and the exchanged heat flux and can be calculated by the equation

$$\dot{E}_{v,cond} = T_{amb} \cdot \frac{T_h - T_c}{T_h \cdot T_c} \cdot \dot{Q}_h \qquad [-]$$

$$\dot{E}_{v,cond} = 269\,K \cdot \frac{321K - 304\,K}{321K \cdot 304\,K} \cdot 1201\,W = 56\,W$$

eq. 34

Two characteristic numbers can be defined for the heat pump process. The exergetic efficiency characterises the thermodynamic quality of the process and is defined as used exergy output divided by the exergy input

$$\zeta = \frac{|\dot{E}_h|}{P_{hp,cmp}} = 1 - \frac{\dot{E}_{v,hp}}{P_{hp,cmp}} \qquad [-]$$

$$\zeta = \frac{256\,W}{512\,W} = 1 - \frac{256\,W}{512\,W} = 0.5$$

eq. 35

The thermodynamic ideal of an entirely reversible cycle can be expressed by the Carnot COP which is only dependent on the temperature levels between the hot and the cold process side. Thus, the ideal efficiency factor ε_{rev} can be expressed by a multiplication of the Carnot efficiency (thermodynamic ideal) with the used temperatures (i.e. indoor design temperature $\theta_{i,des}$=20°C (=$T_{i,des}$=293.15 K) and ambient temperature, here the outdoor design temperature θ_{amb}=-4°C (=T_{amb} =269.15 K)) with the exergetic efficiency. Thereby, a reversible heat transfer at the hot and ambient temperature level is implied. However, in real processes, a temperature difference for the irreversible heat transfer is required, so the real efficiency factor ε_{irr} is lower than the reversible. It is calculated with the effective temperature levels of the condensation and evaporation according to the equation

$$\varepsilon = \frac{|\dot{Q}_h|}{P_{hp,cmp}} = \frac{|\dot{Q}_h|}{\dot{E}_h} \cdot \zeta = \frac{|\dot{Q}_h|}{\dot{Q}_h \cdot \eta_c (T_c / T_h)} \cdot \zeta = COP_C \cdot \zeta = \frac{T_h}{T_h - T_c} \cdot \zeta \qquad [-]$$

$$\varepsilon_{rev} = \frac{1201\,W}{1201\,W \cdot \eta_c (269\,K / 293\,K)} \cdot 0.5 = \frac{293\,K}{293\,K - 269\,K} \cdot 0.5 = 6.1$$

$$\varepsilon_{irr} = \frac{1201\,W}{1201\,W \cdot \eta_c (269\,K / 293\,K)} \cdot 0.5 = \frac{321\,K}{321\,K - 252\,K} \cdot 0.5 = 2.3$$

eq. 36

where

$P_{hp,cmp}$	exergy input to the process as compressor power	[W]
$\dot{Q}_c$	anergy input as heat at the cold process side	[W]

$\dot{Q}_h$	heat rejected from the process at hot process side	[W]
$\eta_C(T_{amb}/T_h)$	Carnot efficiency between the hot and the ambient temperature level	[-]
$\dot{E}_v$	exergy loss of the process due to irreversibilities	[W]
$\dot{E}_h$	exergy rejected from the process on the hot process side	[W]
T_h	temperature of heat rejection at hot process side	[K]
T_c	temperature of heat input at cold process side	[K]
$\dot{E}_a$	exergy of the fluid flow, e.g. air flow	[W]
$\dot{m}_a$	mass flow rate of the fluid flow, e.g. air mass flow rate	[kg/s]
h_a	specific enthalpy of the fluid, e.g. enthalpy of the air	[J/kg]
h_{amb}	specific enthalpy of the ambiance	[J/kg]
s_a	specific entropy of the fluid flow, e.g. the air flow	[J/(kg·K)]
s_{amb}	specific entropy of the ambiance	[J/(kg·K)]
R_a	gas constant of the fluid, e.g. air	[J/(kg·K)]
p_a	pressure of the fluid flow, e.g. air pressure	[Pa]
p_{amb}	ambient pressure	[Pa]
T_{amb}	temperature of the ambiance	[K]
ε_{rev}	reversible efficiency factor of heat pump based on indoor and ambient temperature	[-]
ε_{irr}	irreversible efficiency factor of heat pump based on effective condensation and evaporation temperature	[-]
ζ	exergetic efficiency	[-]

From eq. 36, it follows that the two possibilities to improve the process are to maximize the Carnot COP (which practically means to minimize the temperature lift) and to maximize the exergetic efficiency (which means minimizing the internal exergetic losses in form of irreversibilities of the refrigerant process).

The maximization of the Carnot COP is mostly linked to the design of the heat exchangers and the system periphery (required temperature level of attached source and distribution system and temperature differences for the heat transport in the heat exchanger, see next chapter), which defines the condensation and evaporation temperature and thereby the Carnot COP of the heat pump operation.

The maximization of the exergetic efficiency is mostly linked to the internal refrigeration process. Means to reduce the internal exergetic losses are for instance an improved compression with higher isentropic efficiency or by intermediate cooling as in the case of EVI (Enhanced Vapour Injection), where by internal cooling, the compression is better approximated to the ideal of an reversible isothermic compression. Exergetic losses of the expansion can for instance be reduced by condensate subcooling, as it is applied in the LWZ 303 SOL by Stiebel Eltron. Since the inclination of the entropy curves in the fluid phase is steeper, a subcooling of the condensate leads to smaller entropy differences over the expansion valve and thus, entropy losses, which are calculated by the multiplication of the entropy difference times the ambient temperature, are reduced. By this reduction more heat can be decoupled at the same compressor power input, but also more heat has to be extracted by the evaporator, which may lead to a lower evaporation temperature, which reduces the benefits a bit. Due to dependencies of the compression and expansion on the temperature level, the exergetic efficiency is also dependent on the temperature levels.

4.4.2 Application to compact units

In the following exergy considerations are applied to a compact unit which uses only the exhaust air of the ventilation system as heat source for the heat pump. The focus is set on the external exergy losses of the heat pump linked to the temperature levels of the sink and source system. Therefore, an exergetic efficiency to characterise the internal losses of the refrigerant cycle is assumed constant at the different temperature levels at a value of ζ=0.5 to stress the changes in the external losses.

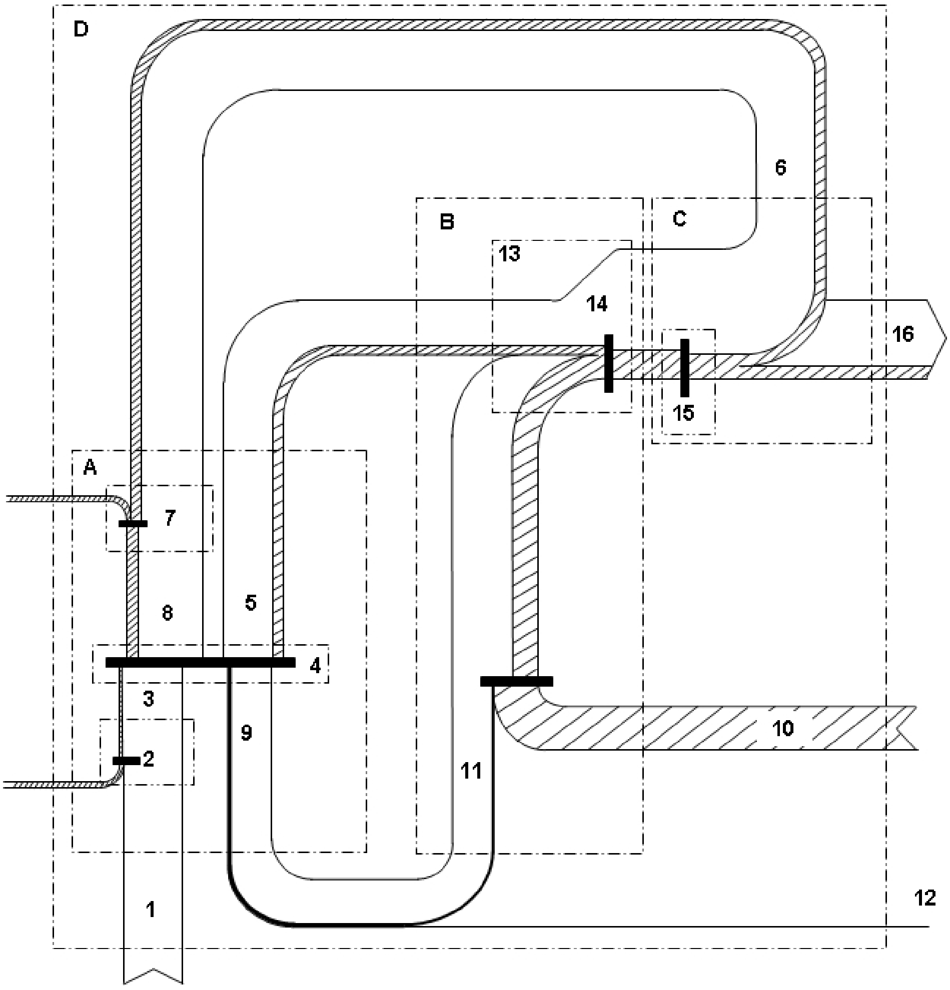
The analysis starts with an air-heating distribution system. Since the air flow rate should be limited to the hygienic necessary flow rate to reduce excessive electrical energy use for the transport of the air, air heating systems require higher inlet temperatures to supply the required amount of heat in comparison to hydronic systems.

The calculation of the system is performed in the following way: The outlet temperature of the supply air of the heat recovery unit defines the inlet temperature to the heat pump condenser, which has to transfer the heat to cover the load of the building. Thereby, the outlet temperature of the heat pump condenser is defined. The condenser is split in two separate parts, a condensing and a desuperheating part. By data of the refrigerant cycle, the share of heat decoupling in the desuperheating part is calculated, thereby determining the inlet temperature of the desuperheating part. The condensation temperature is defined by a pinch point above the air inlet temperature to the desuperheating part.

By a fixed condensation and evaporation temperature the Carnot COP of the cycle is calculated, which delivers by multiplication with the fixed exergetic efficiency the real effiiency factor ε_{irr} acc. to eq. 36 and thereby the electricity demand of the compressor. Acc. to eq. 31 the heat to be extracted by the evaporator is estimated. This heat has to be extracted from the exhaust air flow, which determines the air outlet temperature of the evaporator. The evaporation temperature is set by a pinch point below the outlet temperature of the air. Due to the dependency on the Carnot efficiency the determination of the evaporation temperature is iterative. The calculation is made for an ambient temperature of -4°C. Tab. 24 gives the input data of the components for the comparison.

Tab. 24: Data of the exergy analysis for an air- and hydronic floor-heating system

Boundary conditions	
Heat load for the heat pump [W]	1201
Ambient temperature [°C]	-4
Ambient pressure [Pa]	101325
Indoor design temperature [°C]	20
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 in the heat recovery heat exchanger, condenser and evaporator [Pa]	80
Pressure drop in the air distribution (each supply and return path) [Pa]	40
Constant fan power [W]	10
Pressure drop dependent fan power [W/Pa]	0.12
Heat pump	
Exergetic efficiency of the heat pump [-]	0.5
Pinch point condenser air heating [K]	5
Pinch point condenser floor heating [K]	2
Pinch point evaporator [K]	5
Air heating system	
Required temperature at ambient conditions [°C]	48
Condensation temperature air heating system [°C]	48
Evaporation temperature of the air heating system [°C]	-21
Floor heating system	
Required temperature at ambient conditions [°C]	29
Condensation temperature floor heating system [°C]	30
Evaporation temperature floor heating system [°C]	-23



	energy [W]	exergy [W]	anergy [W]	exergy loss [W]
A ventilation system				**78**
1 inlet outdoor air	454	0	454	
2 supply air fan	*34*	*34*	*0*	*28*
3 outdoor air after supply air fan	488	6	482	
4 heat recovery unit	*673*	*18*	*631*	*24*
5 supply air after heat recovery unit	1161	24	1137	
6 return air after room	1338	37	1301	
7 return air fan	*34*	*34*	*0*	*26*
8 return air after return air fan	1372	45	1327	
9 exhaust air after heat recovery	699	3	696	
B heat pump				**364**
10 compressor	*512*	*512*	*0*	*256*
11 evaporator	*689*	*-7*	*696*	*54*
12 exhaust air after evaporator	10	10	0	
13 condenser	*1201*	*139*	*1009*	*53*
14 supply air after condenser	2362	163	2199	
C emission/room				**41**
15 emission in the room	*2362*	*122*	*2240*	*41*
16 needs of the room	1201	98	1103	
C input to return air by room heating	177	13	164	
6 return air after room	1338	37	1301	

Fig. 23: Energy flow diagram of compact unit with fractions exergy and anergy

Fig. 23 shows the respective energy flows decomposed as exergy and anergy flows for the case of an air-heating system. The single components are shown as system boundaries, superordinate components like the ventilation system are denoted with capital letters, subcomponent like the heat recovery heat exchanger by numbers. The black bars symbolizes an exergy loss, where the exergy inlet is diminished and the anergy fraction grows. This can be best seen in the heat pump, where the supplied compressor power delivers an increased energy flow, but due to the internal exergy losses a reduced exergy fraction results.

As simplification the single components have been treated adiabatic, i.e the supplied electricity is recovered in the air flow as temperature and pressure rise (exergy) and the resulting exergy loss is calculated by balancing the electrical input and the exergy rise in the fluid acc. to eq. 33. In real components a heat loss occurs depending on the insulation which is composed of an anergy and, depending on the temperature level, an exergy fraction. Whether this heat loss is to be assessed as exergy loss depends on if the heat loss can be recovered for heating purposes or not.

The calculation follows the supply and return air flow through the components of the heat recovery unit and the heat pump, whereby energy, exergy and anergy transfer and exergy losses are calculated.

An example is given for the supply air flow through the heat recovery (lines 1-5 in the Table of Fig. 23) to illustrate, how the fractions are calculated. Starting with the energy of the outdoor air inlet (line 1) of 454 W, which is pure anergy, the first component is the supply air fan (line 2). The electricity input to the supply air fan is 34 W, which is pure exergy. The supplied electricity raises the energy of the supply air after the fans (line 3) by 34 W to 488 W, while the exergy fraction of the air flow is increased by 6 W due to pressure rise and a small temperature rise. The remaining 28 W are an exergy loss, which raises the anergy fraction to 482 W. In the next component, the heat exchanger of the heat recovery unit (line 4), a heat energy of 673 W is transferred by the return air, which consists of 18 W exergy and 631 W of anergy. However, due to the finite temperature difference an exergy loss of 24 W occurs in the heat recovery unit. The temperature conditions are defined by the temperature change coefficient which reflects the heat transfer characteristics of the ventilation heat recovery. The energy of 1161 W of the supply air flow after the heat recovery (line 5) thus consists of 24 W exergy and 1137 W anergy which results of the anergy transfer in the heat recovery of 631 W and the 24 W exergy loss. In this way the air flow of the supply and return air side are calculated component by component. The next component for the supply air flow is the condenser (line 13).

In air-tight low energy houses, a mechanical ventilation system is required to guarantee the hygienic necessary air exchange. Therefore, the fan power is required anyway. The heat recovery unit causes 24 W of exergy losses, but by the recovery of ventilation losses the exergy demand of the room is reduced by 84 W, so the heat recovery unit is a very useful means to reduce exergy demand of the room. In case of the air heating system the supply air flow has to provide the total heat load of the room. Since high supply temperatures are required due to the limited capacity of the air flow this implies a temperature rise in the condenser from the outlet temperature of the heat recovery of ~15°C to the required supply temperature of 48°C. Taking into account the desuperheating fraction, the condensation temperature is determined to 48°C. The exergy loss from the condensation temperature to the average temperature level of the air flow is accounted to the condenser.

During the emission of the heat in the building the temperature level and thereby the exergy of the air flow is reduced from the average condenser temperature level to room temperature level, while afterwards, the exergy and anergy needs of the room are transferred to the ambient temperature level by the heat losses of the building. The energy needs of the room also include the remaining ventilation losses, i.e. the fraction between the heat recovery outlet and the return air outlet. Therefore, this fraction is supplied back to the air flow.

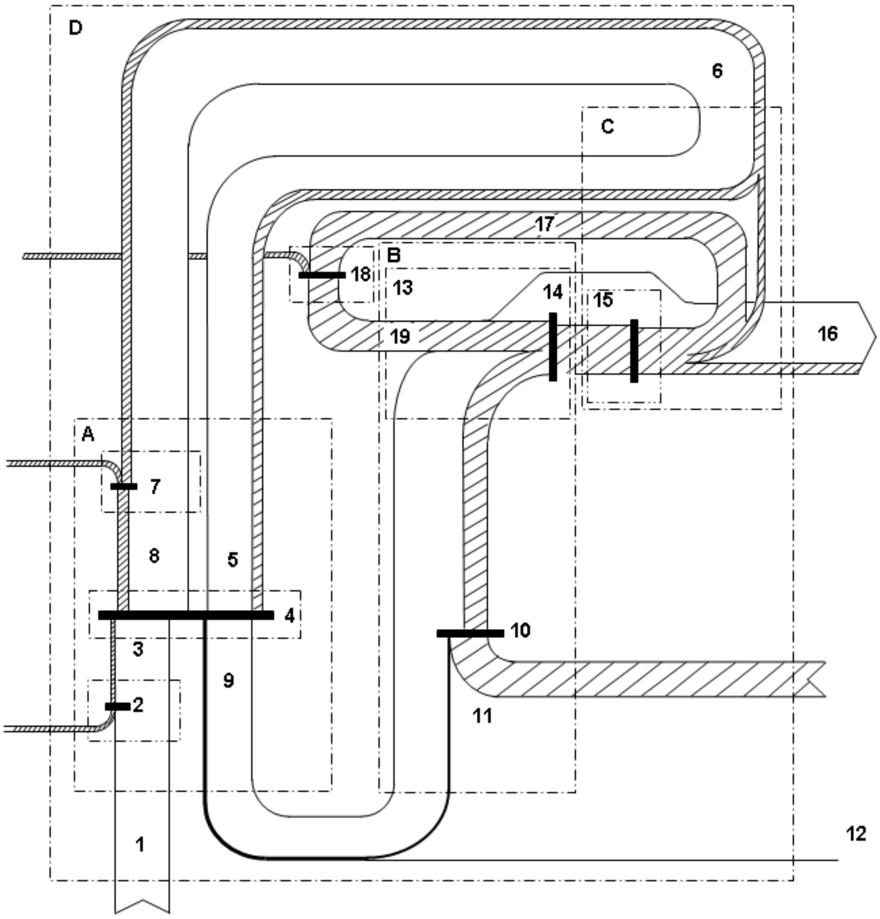
The exhaust air of the heat recovery is almost pure anergy. For reasons of clarity of the diagram the anergy inlet of the outdoor air flow has been scaled that no anergy leaves the system boundary. However, depending on the temperature level of the exhaust air after the evaporator in relation to the ambient temperature, a small exergy fraction may be extracted in the evaporator, which has to be supplied to the compressor together with the exergy losses in the evaporator. Therefore, no extra system boundary has been set for the evaporator. Due to the limited air flow, the evaporation temperature determined by the pinch point below the outlet temperature of the air flow results in $\theta_{hp,evap}$=-21°C to extract the necessary heat. For the heat pump a temperature lift of $\Delta\theta_{hp}$=69 K results. Considering the exergy losses, the highest losses in the components are on the supply side, where exergy losses of the condenser and the emission add up to 94 W.

Tab. 25 gives an overview of the air heating system by the balance of the building marked by the system boundary "D" in Fig. 23. Of the 512 W of exergy supplied to the compressor, only 98 W are required to cover the exergy demand of the room, while 416 W in heat pump and emission and 69 W in the heat recovery are transformed to anergy as exergy losses.

Tab. 25: Balance of the building with air-heating system according to Fig. 23

	energy [W]	**exergy [W]**	**anergy [W]**	**exergy loss [W]**
D building				
supplied				
1 inlet outdoor air	454	0	454	
A ventilation system	68	68	0	78
B heat pump	512	512	0	364
C input to return air in the room	177	13	164	
consumed				
C room/emission	1201	98	1103	41
12 exhaust air after evaporator	10	10	0	
Exergy loss/anergy gain		**485**	**485**	**483**
Balance	**0**	**0**	**0**	**2**

To illustrate the process improvement by the choice of a different emission system Fig. 24 shows the same energy flow diagram for a floor heating system at a design temperature of 30°C, so temperature requirements on the supply side are significantly reduced. The temperature spread temperature difference in the condenser due to the resulting condensation temperature of 30°C are very low, which results in a reduction of the exergy losses of the condenser by 81% of only 10 W instead of 53 W. For the emission in the room, the temperature difference is reduced to $\Delta\theta$=10 K resulting in a reduction of the exergy losses by 39% from 41 W to 25 W. On the other hand, a lower condensation temperature is required, so more energy in the form of anergy is extracted in the evaporator, which leads to a reduced evaporation temperature. The temperature lift for the heat pump is reduced to from $\Delta\theta$=69 K to $\Delta\theta$=53 K The reduced temperature spread leads to an improvement in the Carnot COP from COP_C=4.69 to COP_C=5.77. This yields a reduction of the necessary compressor power by 111 W and thereby avoids internal exergy losses of the refrigerant cycle of 21 %.
However, to circulate the heat transfer medium and overcome the pressure drop in the emission system, additional exergy input for the circulation pump of 40 W is required, while the fan power on the supply air side is slightly reduced due to the missing pressure drop of the condenser.



	energy [W]	exergy [W]	anergy [W]	exergy loss [W]
A ventilation system				**69**
1 inlet outdoor air	538	0	538	
2 supply air fan	*24*	*24*	*0*	*21*
3 outdoor air after supply air fan	562	3	559	
4 heat recovery unit	*690*	*21*	*649*	*20*
5 supply air after heat recovery unit	1252	24	1228	
6 return air after room	1429	37	1392	
7 return air fan	*34*	*34*	*0*	*28*
8 return air after return air fan	1463	43	1420	
9 exhaust air after heat recovery	773	2	771	
B heat pump				**273**
10 compressor	*402*	*402*	*0*	*201*
11 evaporator	*759*	*-12*	*771*	*62*
12 exhaust air after evaporator	14	14	0	
13 condenser	*1161*	*119*	*1032*	*10*
14 supply floor heating in the room	1600	522	1078	
C emission/room				**61**
15 emission in the room	*1600*	*497*	*1103*	*25*
16 needs of the room	1201	98	1103	
17 return floor heating from the room	399	399	0	
18 circulation pump	*40*	*40*	*0*	*36*
19 return floor heating after pump	439	403	36	
C exergy input in the room	177	13	164	

Fig. 24: Energy flow diagram of compact unit with floor heating, fractions exergy and anergy

Tab. 26 gives the balance of the building with floor heating emission system. An overall reduction of exergy input by 14% results despite an effective additional required exergy input of 30 W to the further auxiliary component (40 W pump minus 10 W diminished supply air fan).

Tab. 26: Balance of the building with floor-heating system according to Fig. 24

	energy [W]	**exergy [W]**	**anergy [W]**	**exergy loss [W]**
D building				
supplied				
1 inlet outdoor air	538	0	538	
A ventilation system	58	58	0	69
B heat pump	402	402	0	273
18 circulation pump	40	40	0	36
C input to return air in the room	177	13	164	
consumed				
C room/emission	1201	98	1103	25
12 exhaust air after evaporator	14	14	0	
exergy loss/anergy gain		**401**	**401**	**403**
balance	**0**	**0**	**0**	**2**

Regarding the remaining exergy losses further optimisation potential is on the source side, since due to the limited exhaust air volume flow rate, the temperature difference and thereby exergy losses in the evaporator can be reduced by an additional outdoor air flow or a preheating of the air by a ground-to-air heat exchanger, which both require further auxiliary energy. A ground-to-air heat exchanger uses with the ground a further anergy source, which in wintertime has a higher capacity than the outdoor air, and by the extension of the heat exchanger surface also reduces the exergy losses of the heat recovery, since temperature differences are diminished by the higher inlet temperature.

4.4.3 Summary

In this chapter an exergy analysis for one ambient temperature level of a heat pump compact unit with pure exhaust air use under the simplified assumptions of components without heat losses, constant internal exergetic efficiency of the heat pump process and negligence of defrosting operation has been performed in order to focus on the exergy losses caused by the requirements of the type of heat emission system. Starting with an air heating emission system, exergy losses on the sink side are dominant due to the high supply temperature requirements. As comparison a floor heating system with low supply temperature requirements is considered, which leads to a reduction of 14% of the exergy losses, although additional exergy input to auxiliaries is required and the evaporation temperature decreases due to the higher required anergy extraction in the evaporator. Further improvement for the floor heating system may thus be established on the source side, since restrictions in the air flow causes a high temperature spread and thereby high temperature differences in the evaporator.

Under the conditions of a fixed internal exergetic efficiency of the refrigerant cycle exergy losses are proportional to the temperature lift to be provided by the heat pump, which is defined by the attached source and sink systems. The temperature lift above the required indoor temperature and below the ambient temperature causes exergy losses due to irreversible heat transfer, which have to be supplied as exergy to the compressor.

Fig. 25 gives an overview of characteristic temperature ranges by common source and sink systems applied with compact units. Starting from the ambient air source followed by the exhaust air of the heat recovery unit, the temperature lift between evaporator and condenser defining the temperature lift in the heat pump to the emission in the room and finally to the building heat losses to the ambiance.

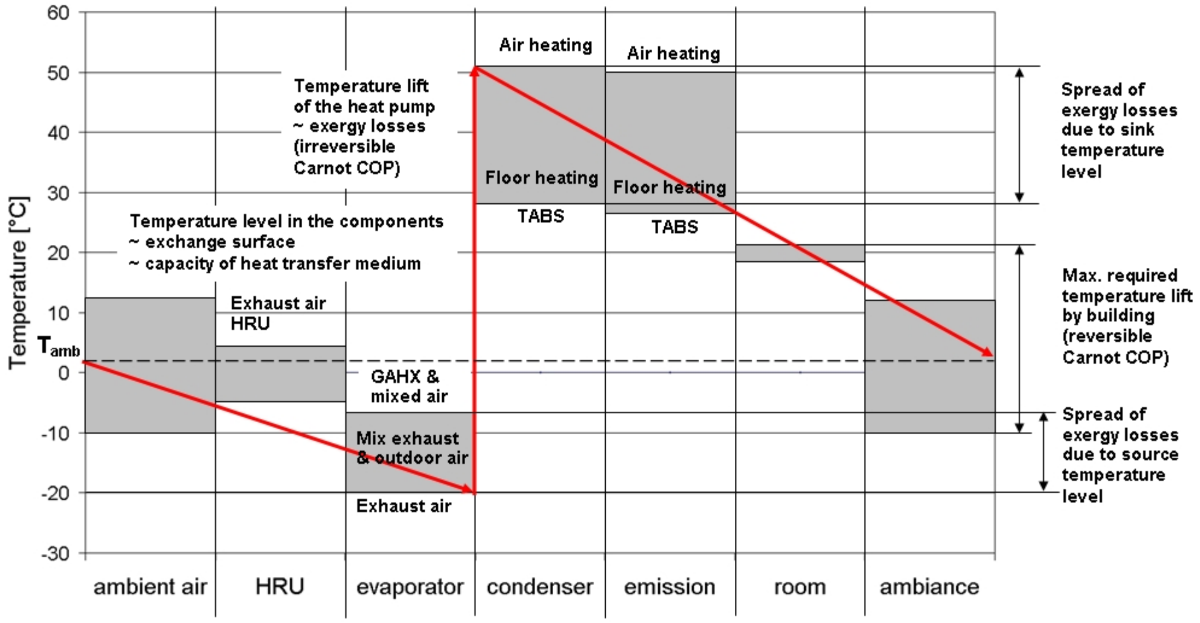


Fig. 25:Temperature levels for different configurations of the source and sink system of compact units (grey – temperature range, red line – temperature levels of the system from the outdoor air source to the building losses to the ambiance)

The temperature levels and linked to that the exergy losses depend on the one hand on the design of the heat exchangers, since a larger heat exchanger surface reduces the heat transfer resistance and leads 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 ventilation air as heating medium, for instance, high temperature spreads are required based to the limited heat transport capacity due to the volume flow rate and the heat capacity of the air flow. All limitations of the heat transfer due to surface or capacity restrictions of the heat transfer medium in the source and sink system thus has to be supplied as additional exergy to the heat pump due to the elevated temperature lift. This can been seen by the spread in the temperature lift between and air heating system and the TABS system on the sink side and the spread between the only exhaust air ventilation use or a preheating of a higher air flow on the source side.

The advantage of exergy analysis in comparison to an energy analysis is therefore, that the exergy losses, e.g. by irreversible heat transfer and pressure drop in the components, can be quantified and components can be optimised with regard to the reduction of irreversiblities.

However, the exergy consideration gives only the technical viewpoint, since in the next step exergetic improvements are assessed economically which in the end may result in a different assessment of the solutions, leading to the discipline of the overall optimisation with regard to energy and economic aspects called thermo- and exergoeconomics. Last but not least environmental and comfort issues have to be considered, as well.

5 VALIDATION OF THE CALCULATION METHOD

A validation of the calculation approach was carried out to prove the principle feasibility of the calculation method, to estimate the exactness of the method and to evaluate different calculation approaches with regard to the exactness of the results. Therefore, comparison to field data of monitored pilot plants was accomplished.

Besides these general objectives the following questions concerning single calculation approaches should be answered in particular:

- What impact has the number of bins on the exactness of the results (chapter 5.1.2)?
- Is it possible to directly subtract the yearly ventilation energy recovery from the space heating energy requirement using a fixed temperature change coefficient as it is currently the case in SIA 380/1 (optimisation) and in the calculation method of the PHPP (chap. 2.3.2)?
- Is it a viable solution to use the heating characteristic for the domestic hot water operation in alternate systems or is a test according to EN 255-3 required (chap 2.2.1.2)?
- How exact is the bin calculation with respect to the redistribution of energy according to heating degree hours (chap. 5.4)?
- What is the difference in the seasonal performance based on the monthly and an annual calculation (chap. 5.4.2)?

The validation has been carried out for a ground-source brine-to-water heat pump in the pilot plant Grafstal [15] to validate the FHNW-method outlined in chap. 2.3.4 and for the compact unit in Gelterkinden to validate the extension of the method of the ventilation system described in chap. 4.1.

For the pilot plant in Zeiningen validation has not been considered for the following reasons:

- Additional heating with wood that has not been monitored
- Prototype system where control has not been adapted
- System is undersized due to a higher real consumption of the house in comparison to the design value, so the system is not operating in his usual operation state
- No complete annual data available
- Insecurities about the heat pump characteristic for the application

In order to prove the principal feasibility of the method, information from the monitoring has been used for the validation:

- The source temperature for the B/W heat pump has been derived from the monitoring. It is usually not known and has to be determined e.g. by standard profiles depending on the site, by simulations or by other approximations. Thus, the exactness of the SPF values may decrease depending on how good assumptions for the source temperature are, although due to the damping of the ground, the range of ground temperatures is limited.
- Moreover, for the DHW operation, the average loading temperature of the monitoring has been used, which is usually not known, either. However, this is reflected in the EN 255-3 value, which was not available for used heat pump in the monitoring.
- Last but not least, storage and system losses have been taken from monitoring, which have to be assessed by the product and installation data. Consequences of this kind of approximations are considered at the pilot plant Gelterkinden in chap.5.2.3.

5.1 Low energy dwelling with B/W heat pump in Grafstal

The dwelling in Grafstal, canton Zurich, is equipped with a ground source heat pump with intermediate brine cycle. The basic design parameters are summarised in Fig. 26. The heat pump operates in alternate combined operation for space heating and domestic hot water production shown in Fig. 27 left. The system does not include a back-up heater. The field measurements took place from autumn 1998 until May 2000, the considered period for the validation is May 1999 to May 2000. A detailed description of the result of the field monitoring is given in [15].

Evaluation period: May 1999 – May 2000

Design heat load	4.3 kW
Outdoor design temperature ($\theta_{oa,des}$)	-11°C*
Indoor design temperature	22°C
Supply temperature at $\theta_{oa,des}$	34°C
Heating limit $\theta_{H,hlim}$	13°C
Supply temperature at heating limit $\theta_{H,hlim}$	30°C
Hot water design temperature $\theta_{hw,des}$	50°C
Hot water tapping	38 l/pers.

Fig. 26: Design parameter of the single family house in Grafstal, ZH (lower value due to light construction, $\theta_{oa,des}$ of the site is -8°C), based on measured values*

Tab. 27 gives the characteristic of the heating capacity and the COP of the installed heat pump based on EN 255-2. Since no testing according to EN 255-3 was available for the heat pump, this SH characteristic has also been used for the calculation of the DHW mode.

Tab. 27 Heat pump characteristic of the B/W- heat pump in Grafstal, ZH [15]

	B-5/W35	B0/W35	B5/W35	B-5/W50	B0/W50	B5/W50
Heating capacity [kWh]	4.2	4.9	5.6	3.9	4.6	5.3
COP [-]	4.0	4.6	5.2	2.5	2.9	3.4

Additional input values are given in Tab. 28.

Tab. 28: Input data (measured values) for the validation of Grafstal

Space heating mode	
Space heating energy requirement [kWh]	10423
Distribution losses [kWh]	336
Auxiliary energy	
Nominal power of source pump (averaged) [W]	121
Nominal power of the circulation pump [W]	44
Nominal power for control [W]	14
Domestic hot water mode	
Domestic hot water requirement [kWh]	1505
Design hot water temperature [°C]	50

To evaluate the heat pump characteristic, the dependency of the inlet brine temperature on the outdoor air temperature had to be estimated. Fig. 27 right gives a quadratic fit of the dependency based on hours with running source pump.

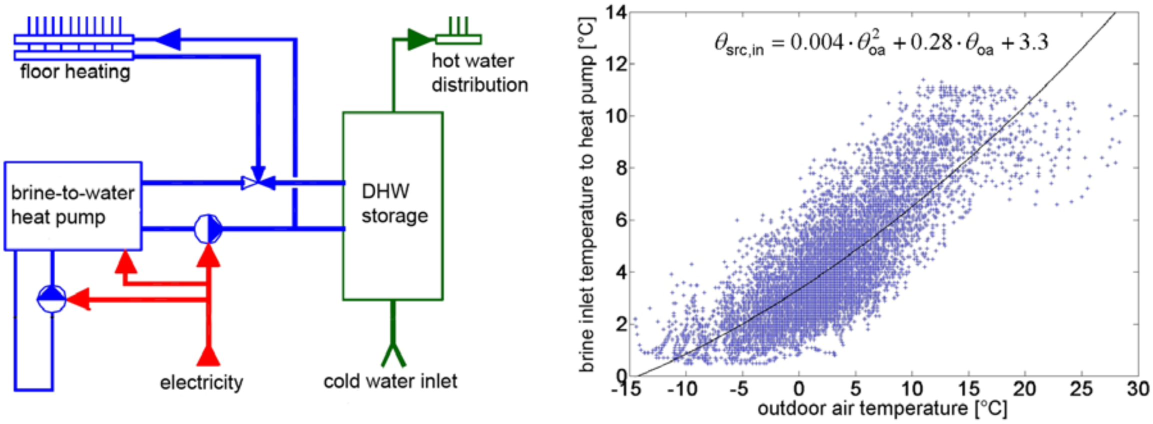


Fig. 27: System configuration of the pilot plant Grafstal (left) and dependency of the brine inlet temperature to the heat pump evaporator versus the outdoor air temperature (monitoring/polynomial fit)

5.1.1 Comparison of calculation and monitoring results

For the validation, three bins for the space heating operation and an additional bin for the DHW operation have been used. Since no EN 255-3 measurements were available for the used heat pump, the space heating characteristic has been evaluated for the DHW operation mode, too. To reflect the interaction between heat pump and DHW storage during the loading, the average supply temperature of the measurements has been set in the calculation.
As explained in the previous chap 2 and, chap. 4 the calculation according to the bin method is based on a redistribution of the totally produced energy for space heating and DHW to the bins. A comparison of the resulting energy fractions of the bins in the calculation based on the heating degree hours for space heating and the measured values in the monitoring are presented in Fig. 28 left. The weighting factors for the respective operation modes and the weighted seasonal performance factors are show in Fig. 28 right. Even though the weighting factors in the bins differ from each other, the weighted seasonal performance factors of the heat pump SPF-HP only show a small difference, i.e. the approach of the redistribution of the energy based on the heating degree hours and a heating limit for the space heating operation is a good approximation. For the DHW operation the assumption of a daily constant consumption might lead to higher deviation dependent on the real tappings, but in this case the approximation is good, too.

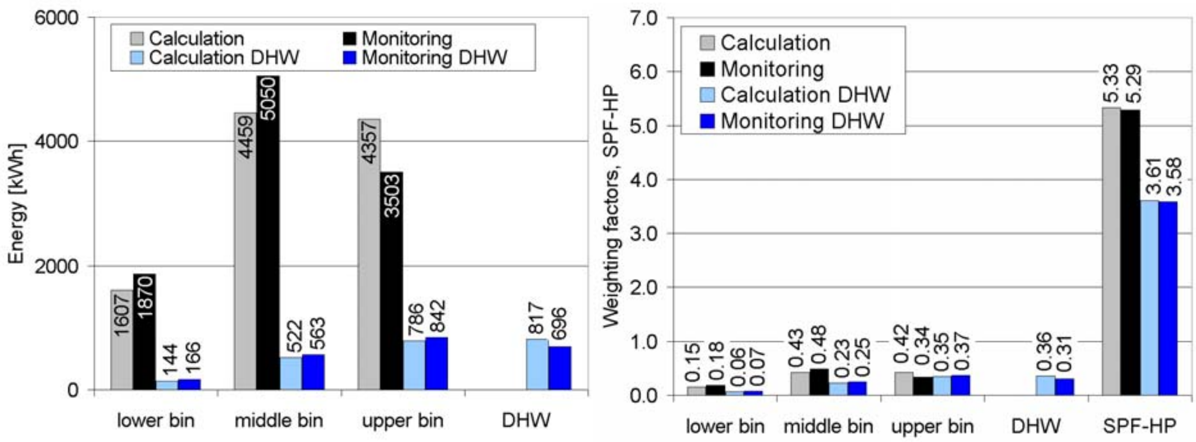


Fig. 28: Distribution of the space heating energy to the bins in the monitoring and calculation (left) and resulting weighting factors (right)

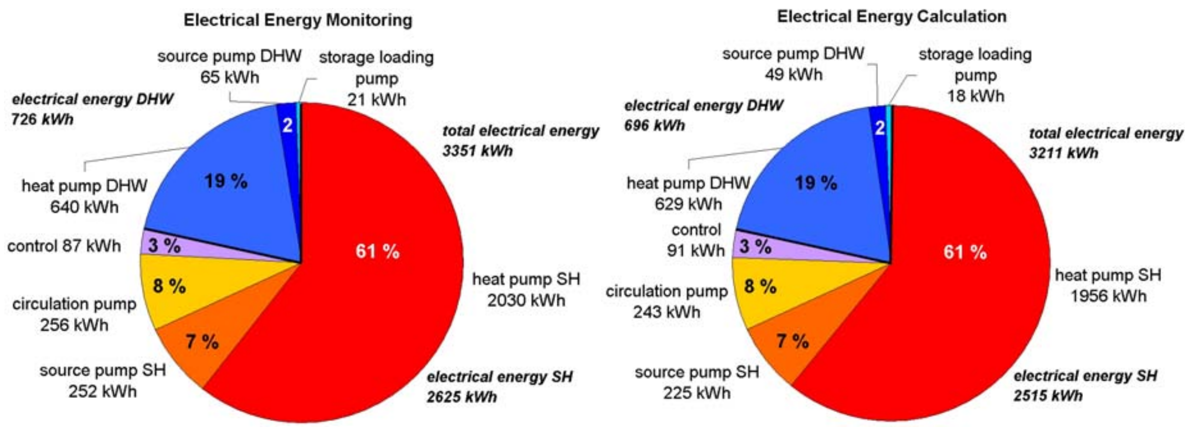


Fig. 29: Electrical energy consumption in monitoring (left) and calculation (right)

Fig. 29 presents the monitored and calculated electrical energy consumption. The real consumption of the heat pump for SH is with 2030 kWh slightly higher than the calculated consumption, leading to a seasonal performance for space heating of only 5.13 instead of 5.29 and a seasonal performance of 3.54 instead of 3.58 for DHW. The reason is the exactness of the COP measurements, which are in the range of 5%. On the other hand, the heat pump itself may have losses which are neglected in the calculation, since they are difficult to quantify unless extended testing, e.g. of cyclic operation, exists. Therefore, the real SPF-HP is lower than the calculated one.

The heating capacity according to the heat pump characteristic is higher than the monitored one, which can be seen on the difference in the energy consumption of the pumps. The source pump, for instance, consumes 27 kWh less in the calculation, which is due to a lower running time based on the higher heating capacity. For the same reason, the storage loading pump consumption is less in the calculation than in the monitoring. Differences in electrical energy input to controls are influenced by the running time, as well, as only the stand-by fraction of control is taken into account, while the fraction during operation is already included in the COP values (calculation) or the heat pump energy (monitoring).

Fig. 30 gives a summary of the comparison in form of seasonal performance factors for the different system boundaries, each as overall values and as values for the operation modes space heating and DHW. Detailed tabulated values are given in Appendix E. The percentage given in Fig. 30 is based on the detailed tabulated values in the Appendix.

Deviation between calculation and monitoring are in the range of 3%-5%. Since the COP values deviate to higher values, all calculated seasonal performance factors are higher than in the monitoring. The fraction of the produced energy for DHW is 17.8%, the fraction of the tapped DHW energy with the regard to the space heating energy need is 13%.

Lower values of the seasonal performance factor of the heat pump (SPF-HP) of the DHW operation are due to the higher required temperature level for DHW operation. Due to the lower numbers, the DHW operation shows a higher relative deviation than the space heating part. For the larger energy consumption for space heating operation, the overall SPF values are more influenced by the space heating mode.

The single impacts on the SPF values become transparent by comparing the three different performance factors based on the system boundaries in Fig. 17. The difference of the heat pump SPF-HP and the generator SPF-G are basically due to the energy consumption of the source pump and additional controls. For space heating operation, the difference of the SPF-G to the system SPF-S is mainly the consumption of the circulation pump. For DHW operation, however, the difference of the SPF-G and SPF-S is mainly due to the storage losses, since the SPF-G refers to produced energy and the SPF-S to used energy.

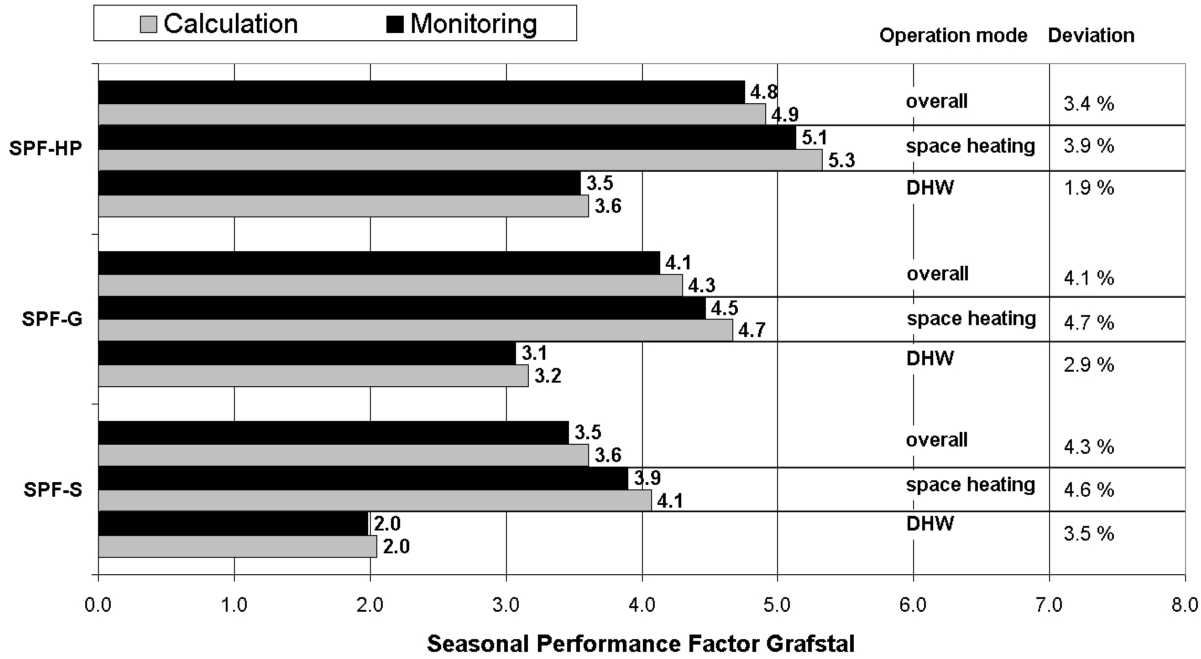


Fig. 30: Comparison of calculated and monitored seasonal performance factors for the different system boundaries in Grafstal (energy fraction of produced energy DHW 18%, tapped energy DHW 13%)

That means, the heat pump operation is still quite efficient at the higher supply temperatures of the domestic hot water with values of the SPF-G DHW above 3, and degradation to the SPF-S DHW of about 2 is mainly due to the system losses that have to be covered by the heat pump, as well.

The storage loading pump has only small impact in this case, since the running time for DHW operation is small due to the high heating capacity determined by the design heat load of the space heating.

Deviation between the calculation and the monitoring are in the range of the exactness of the COP measurements, which is the value that has the largest impact on the seasonal performance, see chap. 4.2.4, slightly increased by the difference in running time and the effect for the auxiliary energies. The results show that for alternate operating systems the space heating characteristic can be applied for the DHW operation, as well, if temperature conditions of the average loading temperature are not too far from the value in operation.

5.1.2 Calculation results with one bin

For ground coupled heat pumps, it is common to consider the COP-values at average testing conditions, i.e. at B0/W35 for floor heating and B0/W50 for radiator heating and DHW operation directly as seasonal performance.

Tab. 29 gives an overview of the variations carried out to test the feasibility of this approach. The 1 bin approach temperature corrected refers to both temperature correction for the source and sink temperature to the average operation conditions. The ground temperature has been set to the average ground temperature of the entire heating period and the sink temperature for space heating to an average of the heating curve, while for DHW the respective supply temperature during loading has been set. For the ground temperature corrected approach, only the ground temperature has been set to the average. For the nominal approach no temperature correction is applied, i.e. the COP is set to B0/W35 for space heating operation and to B0/W50 for DHW operation.

Tab. 29: Variations carried out to assess approaches for B/W-SPF calculations

Variants	Source inlet temperature [°C]				Sink outlet Temperature [°C]			
	SH			DH	SH			DHW
3 bins SH/4 bins DHW	1.5	3.9	5.5	10.3	34	32	31	48
1 Bin temperature corrected	4.2				32			48
1 Bin ground temperature corrected	4.2				35			50
1 Bin nominal	0				35			50

In Fig. 31 a comparison of 1 bin approaches with different applied corrections to the 3 bins for SH or 4 bins for DHW and monitoring results is depicted.

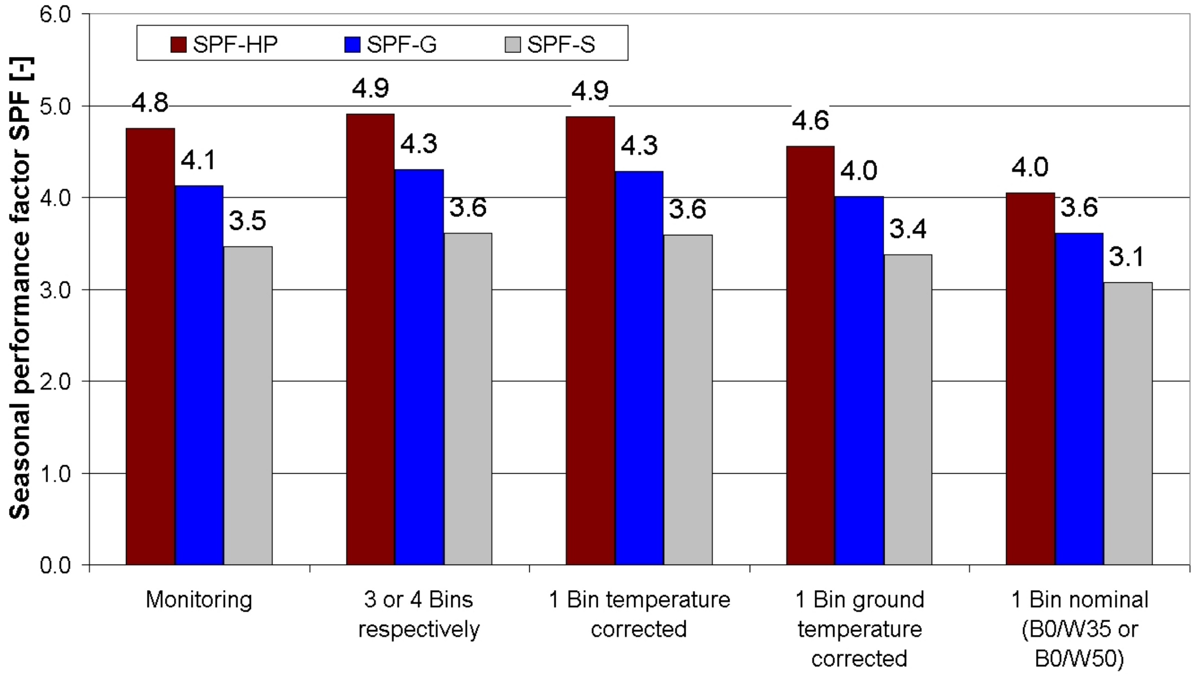


Fig. 31: Comparison of simplified 1 bin approaches for ground-coupled heat pumps

Fig. 31 confirms the assumption that with the adequate temperature correction, a 1 bin approach is feasible for ground-coupled (and equally for ground water) heat pumps, since both ground conditions and COP characteristic are approximately linear. Thus, an interpolation to more bins does not change the seasonal performance significantly, if the average temperature is assessed right. However, the problem is still an adequate approximation for average values or a distribution of the ground temperature dependent on the outdoor air temperature. On the other hand, if only the ground temperature is assessed right and no sink temperature correction is applied, the approximation of the real operation gets worse, and the pure COP values without temperature correction can only give a rough approximation of the seasonal performance, even though in this case the temperature correction is "only" 2 K for each space heating and DHW operation.

The comparison confirms that a 1 bin approach is feasible for B/W heat pumps, if the average source and sink operation temperatures are taken into account.

5.2 MINERGIE dwelling with HP compact unit in Gelterkinden

Detailed results of the field monitoring of the pilot plant in Gelterkinden are described in chap. 6.1. A detailed step-by-step calculation is given in chap. 4.2.2. In this chapter, a comparison of monitoring and calculation results is given. Tabulated values are given in Appendix E.4 and E.5.

5.2.1 Heat energies

Fig. 32 left depicts the comparison of the space heating energy recovered by the ventilation heat recovery and the floor heating system in the monitoring and calculation. The measured energies are evaluated for the respective outdoor temperature range in the bin, while the calculated energies are based on the heating degree hours i.e. the weighting factors in the right part of Fig. 32. Even though the weighting factors deviate slightly for the single bins, the

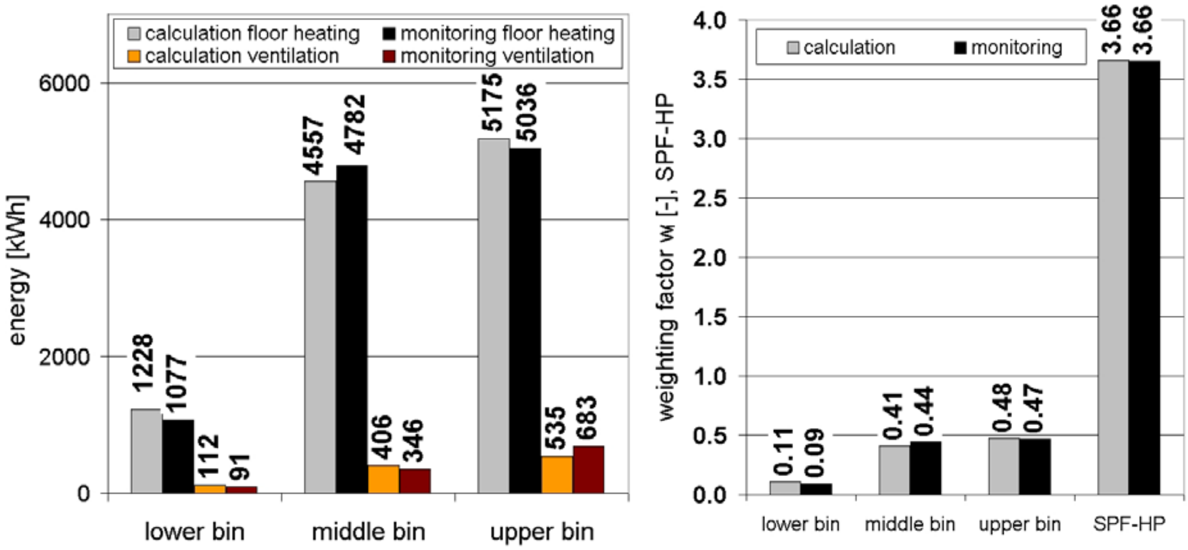


Fig. 32: Space heating energy and recovered ventilation energy (left) and respective weighting factors and the weighted SPF-HP from calculation and monitoring (right)

weighted seasonal performance factor of the heat pump SPF-HP is nearly the same. This confirms the results of the sensitivity analysis in chap. 4.2.4 that the performance factor is not very sensitive to the weighting factors and a redistribution based on the heating degree hours is a feasible approximation for the space heating energy.

For the deviation of the heat recovery, there are several reasons:

- In the monitoring a changing volume flow rate was found depending on the operation of the heat pump, thus in the lower bin with long running times of the heat pump, the volume flow is lower, while in the upper bin, it is higher. For the calculation, an average value of the whole heating period was used. Moreover, the temperature change coefficient is set constant in the bin, which is an average value, as well.
- The influence of the subcooler in the LWZ 303 SOL (see Fig. 48) may play a further role, since it is not taken into account explicitly in the calculation but just implicitly by a combined testing. However, monitoring results show, that concerning the supply temperature, the impact of the subcooler is small, since either the outdoor inlet air is preheated by the subcooler, which decreases the heat exchange in the heat recovery due to the higher inlet temperatures, or the heat recovery is more efficient in case of no subcooler operation, so the supply temperature is almost the same.
- In the lower bin, the difference of required back-up energy has a further impact.

Summarising, due to the approximations in the calculation the single energy values vary, in some cases up to about 200 kWh, but for the weighted SPF-HP, that determines the electrical energy input, the approximations are good, since the energy input is not very sensitive on weighting.

Fig. 33 shows the redistribution of the energy to the bins for the domestic hot water mode, in the monitoring evaluated by the real produced DHW energy, in the calculation based on the assumption of a constant daily profile, i.e. a redistribution according to the bin time.

The resulting weighting factors deviate more than in case of Grafstal, i.e. in the case of Gelterkinden the real tappings are not so time dependent as assumed by the model.
Concerning the total DHW energy the back-up operation found in the monitoring has an influence which is not considered in the calculated values. Therefore, the energy values deviate by the amount of back-up energy. If the weighting is performed on the real redistribution from the measurement a SPF-HP values of 2.92 results. In the real operation, however, the measured SPF-HP is 3.06 since COP-values are slightly different than in the characteristic.

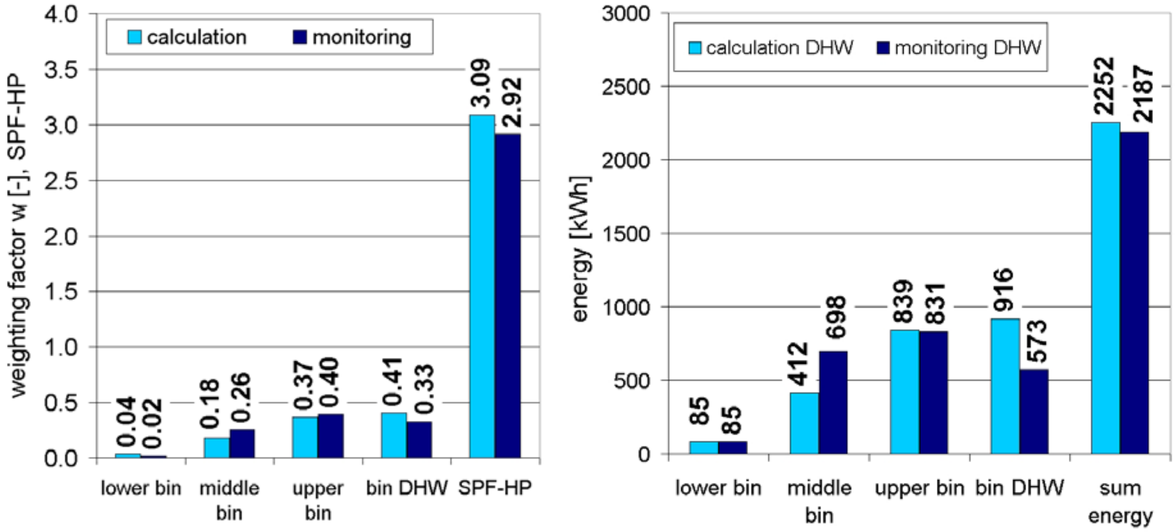


Fig. 33: Weighting factors and the weighted SPF-HP for DHW mode and produced thermal energy for DHW in the calculation and in the field monitoring

5.2.2 Electrical energies

Fig. 34 shows the electrical energy input to the different components of the compact unit in Gelterkinden in the monitoring and the calculation.

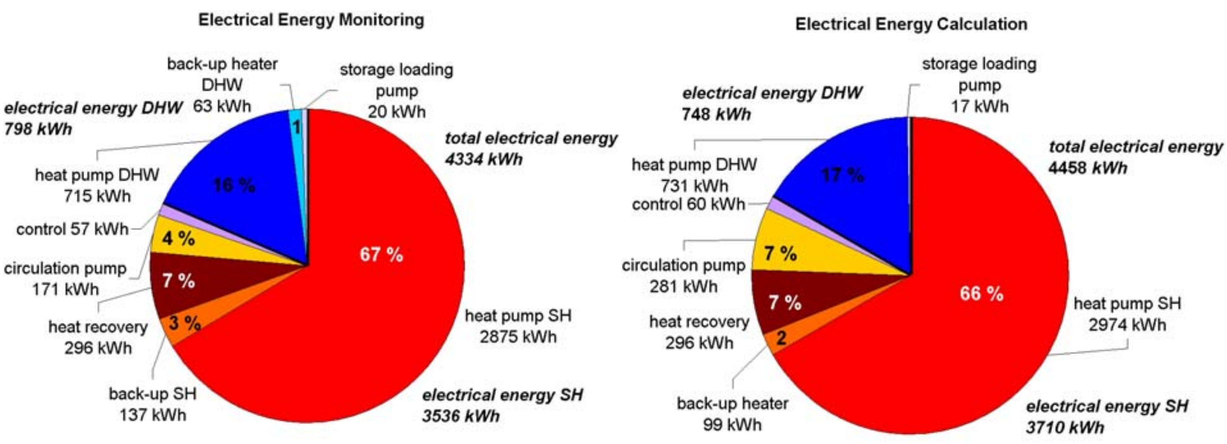


Fig. 34: Electrical energy consumption in monitoring and calculation

For the electrical energies the back-up and control for SH and DHW have been summarised. In general, the difference between the calculated and the monitored values is small, which is due to the fact, that the redistribution of the energies for the space heating is quite identical, and space heating energy need is 5 times the DHW energy need. Main differences in the space heating operation are the heat pump energy, the back-up energy and the energy for the circulation pump, for the DHW the operation of the heat pump and the back-up energy.

As shown in Fig. 33 the weighting itself delivers nearly the same seasonal performance of the heat pump. Thus, monitored **heat pump energy** may deviate due to slightly different COP values in the operation and possibly further heat pump losses due to cyclic operation, which are neglected in the calculation. In fact, the resulting SPF-HP in space heating mode of 3.74 in the monitoring is within the expected 5% exactness of the COP values, so further losses are negligible in this case.

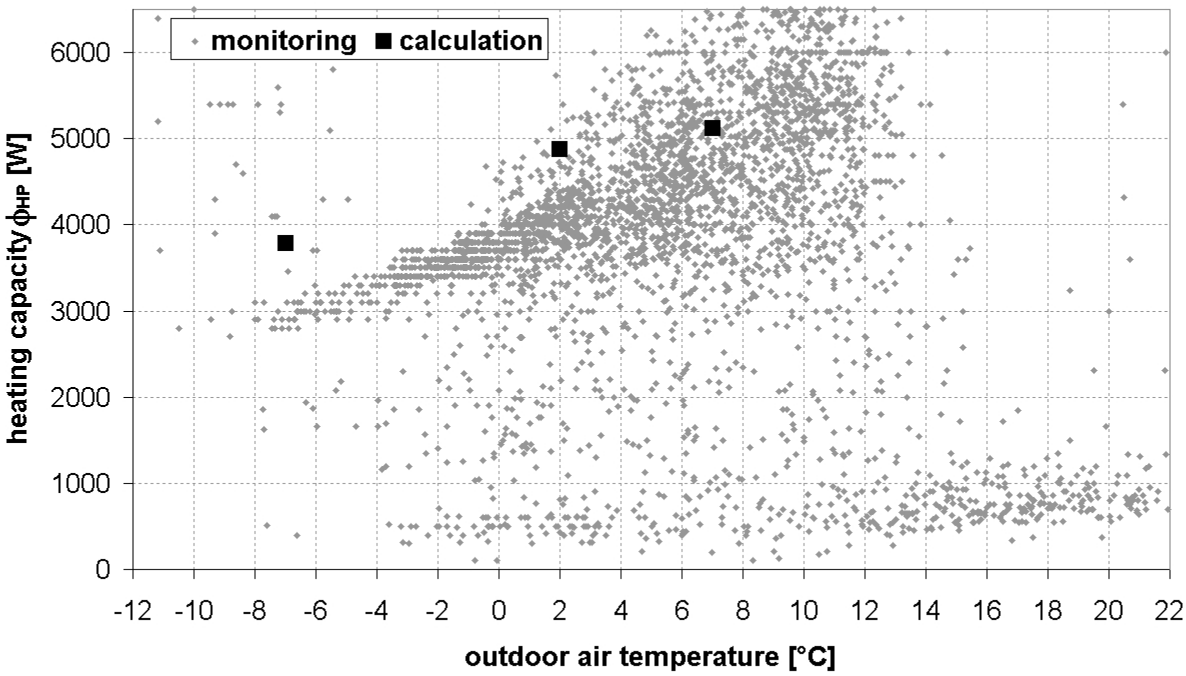


Fig. 35: Monitored heating capacity and points used in the calculation

Back-up energy deviates more, considering space heating and DHW operation together, since in space heating, the value is lower and DHW back-up is zero. The back-up energy has been evaluated by a detailed balance of the required running time as described in chap.4.2.2. However, in this case the reason for the deviation could be traced to a smaller monitored heating capacity in the lower temperature range as shown in Fig. 35. Therefore, more back-up energy is needed than calculated with a higher capacity. Moreover, about one third of the back-up energy for DHW operation is not due to a lack of capacity, but due to large tappings, where the back-up supports the reheating of the DHW after a time delay. This fraction is strongly dependent on tappings and control and cannot be estimated by the calculation method.

Difference in the **consumption of control** are due to different running times caused by a slight difference in the measured and the nominal power and by differences in calculated and monitored running time of the heat pump due to the different heating capacity.

Concerning the **circulation pump** the control is the reason for the big difference. In the calculation it has been assumed that the circulation pump runs through the whole heating period, while in reality the circulation pump is controlled dependent on the outdoor temperature. It is only switched on regularly for a short periods to get the return flow temperature evaluating thereby the heat load of the zone.

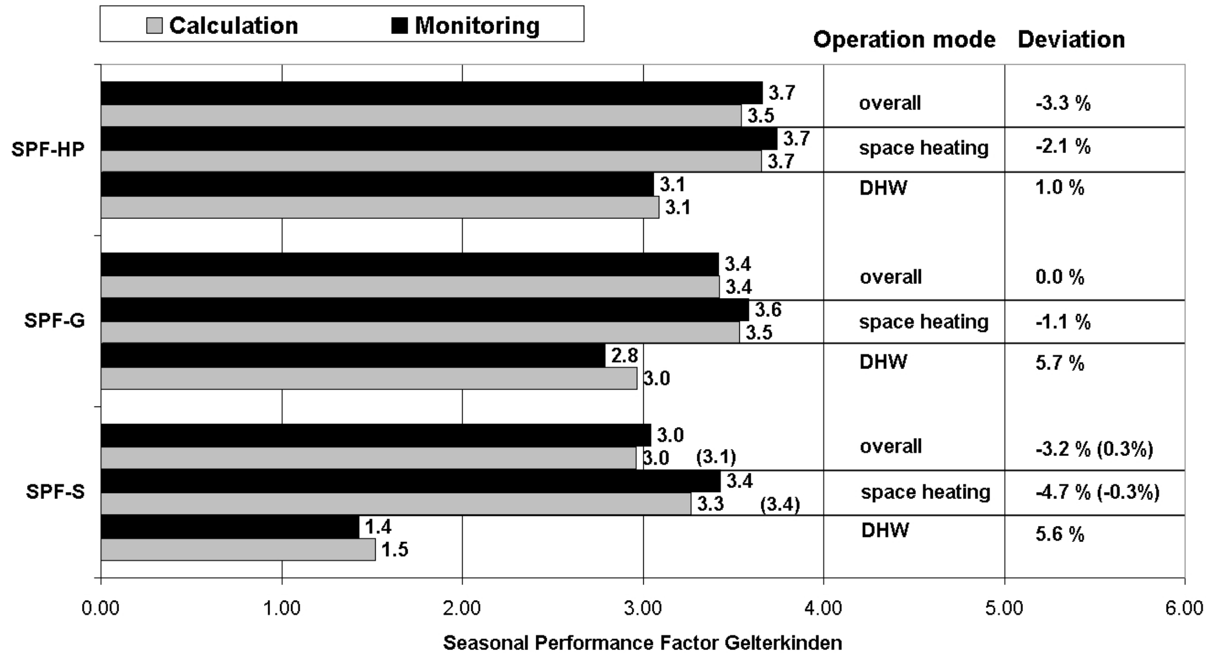


Fig. 36 Comparison of calculated and monitored seasonal performance factors for the different system boundaries in Gelterkinden

But this kind of special feature of a control strategy is seldomly known. Therefore, in the calculation only two defaults can be distinguished, an operation related to the running time of the heat pump and an operation during the entire heating period. The results with coupling of the sink pump to the heat pump operation are given in brackets in Fig. 36. The real consumption is somewhere in-between the two extreme values.
A comparison of the seasonal performance factors for the different operation modes and the different system boundaries derived by the monitoring and the calculation is depicted in Fig. 36. Percentage values are based on the detailed tabulated values given in Appendix E.

Concerning the SPF-HP, values are similar, since in space heating mode the weighting is quite the same and in DHW the lower heat production due to the back-up consumption in the monitoring levels out the difference in the redistribution.

Differences for the SPF-G are due to the different back-up energies and in the DHW mode due to the different energies in the bins caused by the time based redistribution.

For the SPF-S the ventilation heat recovery is considered, as well, so the difference is related to a larger amount of energy. The higher difference in space heating mode is caused by the higher consumption of the circulation pump. In case of a running time of the circulation pump linked to the heat pump operation, a smaller difference of 0.3% for the SPF-S SH and a difference of -1.0% for the SPF-S overall would result. Since the SPF-S is related to the energy need, the difference increases a bit in the DHW mode.

Summarizing, with the maximum deviation of about ±6%, the deviation is in the range of the COP measurement, thus approximations are valid to calculate the seasonal performance factor with the adequate exactness to be expected by the reliability of the input data. On the other hand, the different impacts on the seasonal performance can be evaluated.

5.2.3 Evaluation of further impacts

In this chapter the following variants for the pilot plant in Gelterkinden have been evaluated.

<u>Nominal values</u>

For the validation, parts of the monitored values have been used in order to evaluate, how the exactness of the method is related to the monitoring for the same input conditions. However, usually no monitoring exists, so nominal values of technical data sheets and ideal controller setting have to be used for the calculation. To evaluate the difference to the monitoring, the same energy requirements for space heating and DHW are used but the following nominal values have been set as input of the calculation

- nominal meteo data of the site (Basel-Binningen instead of on-site measurements in Gelterkinden, default heating limit MINERGIE® **12°C** instead of **14°C**)
- nominal volume flow rate of the ventilation system (60 m^3/h instead of effective 47 m^3/h)
- nominal pump powers (**65 W** instead of **54 W** for the circulation pump and **45 W** instead of **33 W** for the storage loading pump)
- nominal storage losses (**484 kWh** instead of **1073 kWh**)

<u>Reference DHW temperature</u>

The loading of the storage takes place at changing temperatures, resulting in a lower average temperature. This effect of the interaction of the storage and the heat pump is taken into account in the system testing of EN 255-3, see chap. 2.2.1.2. However, if no values according to EN 255-3 are available, the reference DHW temperature is often set.

<u>Impact of the heat recovery</u>

In many building regulations like the Swiss SIA 380/1, the ventilation heat recovery may be already considered in the calculation of the building, so values of the energy need for space heating might refer to already reduced ventilation losses, so that the recovered heat of the ventilation system is already included. Therefore, it is evaluated, what are the differences for the calculation.

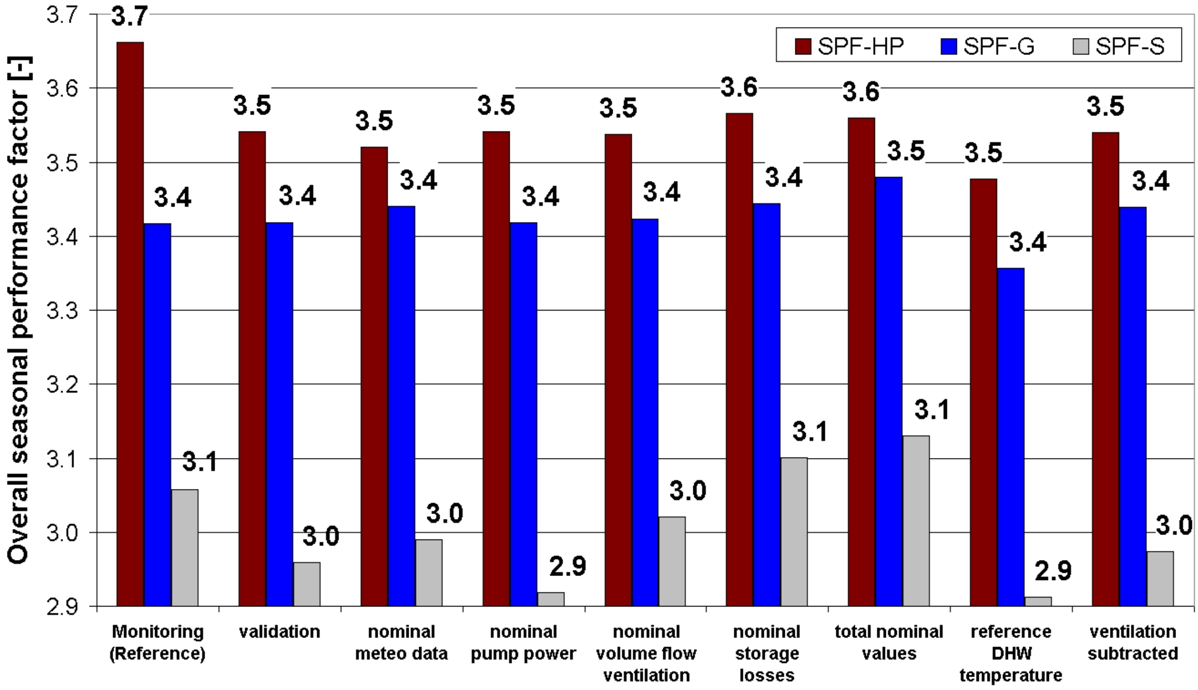


Fig. 37 Seasonal performance of the pilot plant Gelterkinden based on nominal input values and other impacts

Fig. 37 shows the results of the overall SPF for the different system boundaries and respective variants.

The weather data of Basel-Binningen and a standard heating limit for MINERGIE houses deliver more or less the same values, thus the climatic region can be reproduced quite good. Weighting factors for the space heating operation mode are thereby similar, and the impact on the SPF-HP is basically due to the time-based redistribution of the energy to the bins in the DHW mode. Nominal pump power is only contained in the SPF-S and lowers the SPF moderately. However, the impact basically depends on the running time, so for smaller capacity units with longer running time, the impact is stronger than in this case. The nominal air volume flow rate defines the amount of recovered heat and space heating energy to be covered by the heat pump. Normally, the volume flow rate changes the total energy needs and the used gains, as well, but for this evaluation, the effect has been neglected, since the change in volume flow rate is not so big. However, effects on the SPF-S as the only characteristic number, where the ventilation heat recovery is included, are due to the different weighting of the energy fractions as can be seen clearly. Marginal effects on the SPF-HP occur due to the change in amount of energy to be covered by the heat pump and the back-up. The largest impact in this consideration are the nominal storage losses, since monitored storage losses are about double of the nominal and mainly contribute to the low system seasonal performance of the DHW system.

If all nominal values are applied, the SPF-S changes by 5.7% compared to the validation, basically due to the impact of the changed pump power and volume flow rate of the heat recovery as well as the DHW storage losses.

The reference water temperature has an impact on all SPF values, since the SPF HP is influenced by the temperature level. In the SPF-S the impact seems small, but this is based on the weighting of the space heating and DHW energy. The SPF-S for the DHW operation changes by about 8%.

Concerning the subtraction of the ventilation energy, the effect on the SPF-numbers is marginal. This has been already seen in the sensitivity analysis in chap. 4.2.4.

Therefore, the calculation can be accomplished without explicitly taking into account the recovered heat of ventilation heat recovery in the calculation, if input data are available, where the recovered heat by the ventilation system is already taken into account, e.g. from SIA 380/1 optimization. However, electrical energy input for the ventilation and impact on the heat pump operation has to be taken into account.

With nominal values, the calculation results change, but stay in the same range compared to the monitoring values in the case of the pilot plant Gelterkinden.

Recovered ventilation losses do not have to be considered bin-wise, but can be subtracted in total from the space heating energy need or a reduced energy need can be given as input to the calculation, respectively.

5.3 Method in comparison to other existing methods

In this chapter the approach of the FHNW method is compared to the other existing methods for the calculation of compact units. The three methods presented in chap. 2.3 are contrasted in Tab. 3. The comparison has been made for Gelterkinden, since the calculation can be compared to the result of the field monitoring. For the PHPP, however, no values for the $COP_{HW,winter}$ and $COP_{W,summer}$ were available, so the method could not be performed for the pilot plant Gelterkinden. Excel sheet of the WPEsti calculation are given in Appendix F.

5.3.1 Comparison for the pilot plant Gelterkinden

Input data of the calculation are set according to the respective calculation method and given in Tab. 30. For the FHNW method, the calculation is given in more detail in chap. 4.2.2 and a detailed comparison to the field monitoring is given in chap. 5.2, so in this chapter, values of the FHNW method are depicted in the tables and figures, but only the results of WPEsti are discussed.

Tab. 30: Calculation results of the two methods for the pilot plant Gelterkinden

Input data and energies				
	WPEsti V 2.0 default	**WPEsti V 2.0 Gelterkinden**	**FHNW (see chap. 4.2.2)**	**Monitoring Field test**
Input				
Meteo data	Basel-Binningen	Gelterkinden	Gelterkinden	
Heating degree days [Kd]	3348	2732	2732	2732
COP characteristic [-]	A7/W35;A2/W35;A7/W35;A7/W50		acc. Tab. 10	
Temperatures SH [°C]	30/25		30.7 – 28.2	31 - 24
Temperatures DHW [°C]	52/52	46/44	46/46	42/48;50/55
Energies				
Net energy WHS [kWh]	6588	10894	10894	10894
Ventilation (design) [kWh]	1700	1700	1120	1120
Transmission (design) [kWh]	11475	11475	Not used	Not used
Total space heating [kWh]	6588	10894	12014	12014
DHW energy [kWh]	2125	1179	1179	1179
Storage loss DHW [kWh]	25 %=531	91% = 1073	1073	1073
Total DHW [kWh]	2656	2252	2252	2252

The Excel-Tool WPEsti V. 2.0 is evaluated for the standard settings (WPEsti V2.0 default), i.e the standard setting for the energy consumption (SH, DHW) according to the planning and the bins of Basel-Binningen evaluated by the approach of WPEsti are used. This reflects how the building would be considered based on the planning results for compliance with the MINERGIE® standard. For the DHW temperature, the value of 52°C has been set which corresponds to the value that is guaranteed by heat pump operation according to the data sheet for the MINERGIE® calculation [2]. In order to avoid higher back-up energy consumption, the reachable temperature by back-up operation has been set to 52°C, too.

According to the different calculation methods, the space heating requirement of the water heating system is set in WPEsti, while in the FHNW method, the measured energy need of the water heating and the ventilation system is used in the validation.

In case of WPEsti Gelterkinden, values have been adapted to the monitoring in Gelterkinden as far as possible. So, energy requirements for SH and DHW and the cumulative frequency and heating degree days according to the field monitoring in Gelterkinden are used. For the transmission losses of the building, the design values from the planning are kept, since these values could not be evaluated in the field monitoring. For the DHW operation the average loading temperature of the monitoring of 46°C is used.
Since no back-up energy due to a heat pump operation limit temperature has occurred in the monitoring, the reachable temperature with back-up operation has been set so low, that no back-up for DHW occurs anymore.
Tab. 31 shows the calculated weighting factors and back-up energy.
For the default values the back-up energy use is highest. This is due to the higher default temperatures of the DHW part, which delivers a fraction of 5% back-up energy use, if the same temperatures for heat pump and back-up are given as input.

Tab. 31: Weighting factors and back-up energy of the two methods

Weighting factors				
	WPEsti V 2.0 default	**WPEsti V 2.0 Gelterkinden**	**FHNW (see chap. 4.2.2)**	**Monitoring Field test**
Based on	Cum. Frequency WPEsti	Cum. Frequency Gelterkinden	Cum. Frequency Gelterkinden	Monitoring Gelterkinden
lower bin	0.07	0.04	0.11	0.09
middle bin	0.22	0.21	0.41	0.45
upper bin	0.71	0.75	0.48	0.46
Back-up energy				
space heating [kWh]	233	128	97	137
DHW [kWh]	144	-	2	62
Overall [kWh]	377	128	99	199
Hours in lower bin				
	855	330	330	330

For the space heating part, the back-up use is higher since the implemented weather data bin distribution which are corrected values based on Zurich and Davos, is used. This standard redistribution leads to more hours in the lower bin than in the on-site measurements in Gelterkinden, and therefore, there is a higher back-up energy use for space heating, as well. Fig. 38 shows the system boundaries used for the comparison based on the characterstic numbers applied in WPEsti extended by the characteristic including the back-up energy use.

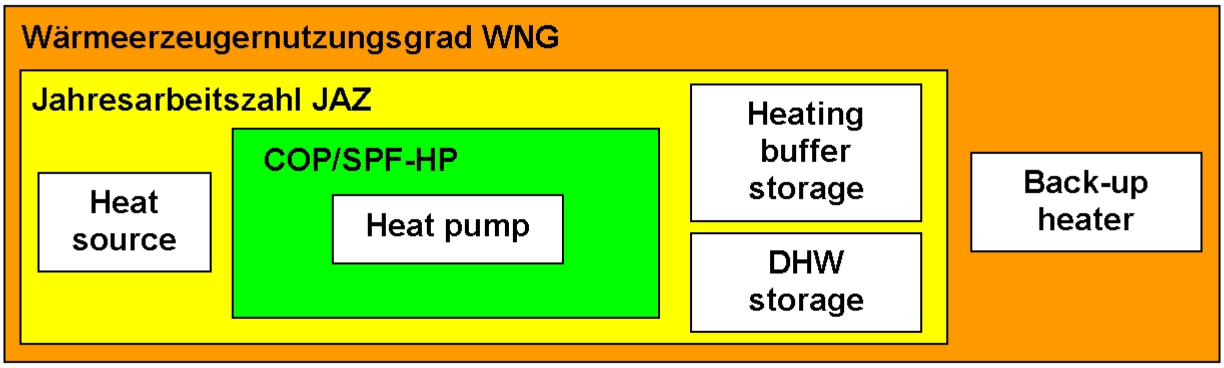


Fig. 38: System boundaries used for the comparison with WPEsti

Based on the calculation in WPEsti, the following characteristics have been used for the comparison.

$$JAZ = \frac{Q_{H,dis,in} + Q_{W,dis,in} - Q_{HW,bu}}{E_{H,hp,in} + E_{W,hp,in} + E_{HW,aux,src,in}} \quad [-]$$

eq. 37

$$WNG = \frac{Q_{H,dis,in} + Q_{W,dis,in}}{E_{H,hp,in} + E_{W,hp,in} + E_{HW,bu,in} + E_{HW,aux,src,in}} \quad [-]$$

eq. 38

where

$Q_{H,dis,in}$	Heat energy need of the space heating distribution system	[J]
$Q_{W,dis,in}$	Heat energy need of the domestic hot water distribution system	[J]
$E_{H,hp,in}$	Electrical energy use of the heat pump for space heating	[J]
$E_{W,hp,in}$	Electrical energy use of the heat pump for DHW	[J]
$E_{HW,bu,in}$	Electrical energy use of the electrical back-up heater for SH and DHW	[J]
$E_{HW,aux,src,in}$	Auxiliary energy use for the source system in the respective operation mode	[J]

The characteristic values SPF-HP corresponds to eq. 28.
Fig. 39 shows the resulting seasonal performance factors for the different calculation approaches, operation modes and system boundaries.

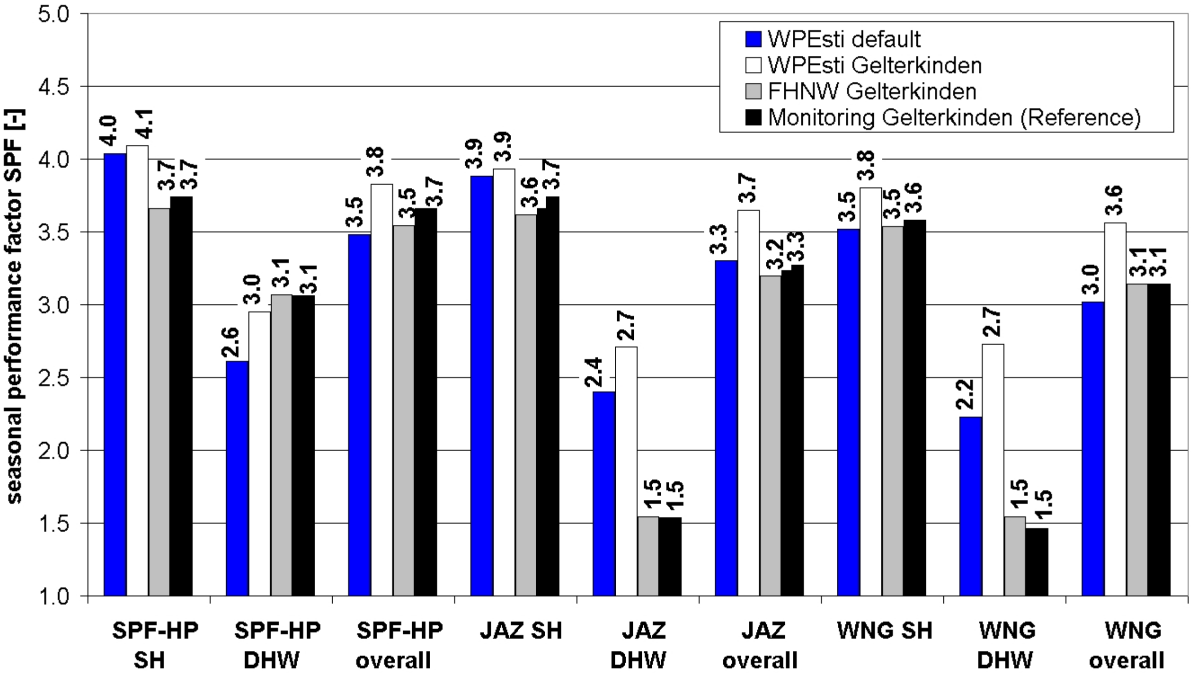


Fig. 39: Comparison of the SPF values for the pilot plant Gelterkinden

For the default energy consumption according to the SIA 380/1 planning the water consumption is much higher while the space heating consumption is lower than the monitored values.

The seasonal performance SPF-HP reflects the weighting of the COP characteristic. For WPEsti, this value is not an output of the Excel sheet. However, the value is the basis for the resulting JAZ. The SPF-HP in space heating operation is higher due to the weighting factors, which differ considerably from the monitoring for the middle and upper bin. The reason is that bin time is evaluated instead of the energy, so the higher building load in the lower and the middle bin is not reflected in the weighting factors. Even though the sensitivity to the weighting factors is moderate according to chap.4.2.4, the weighted SPF-HP is higher than in the monitoring.

For DHW operation the default case has a lower SPF-HP due to the higher water temperature. In case of the lower temperature of WPEsti Gelterkinden, the one bin approach delivers a slightly lower SPF-HP than in the monitoring. The overall SPF-HP is influenced by the lower DHW values, too.

The JAZ (from German **J**ahres**A**rbeits**Z**ahl) in WPEsti is defined by eq. 37. The difference to the SPF-G defined in this project is that the back-up system is not included and the JAZ is related to the energy need and not to produced energy. In WPEsti, the JAZ is derived by multiplication of the weighted SPF-HP with an efficiency based on the default losses given in Tab. 2 of the space heating and DHW mode, respectively. Thus, for the space heating mode without a heating buffer storage, the SPF-HP is reduced by 4%. The high difference in the DHW mode can be explained by the fact, that the DHW storage losses are not considered in this number, but just the default value of 4% acc. to Tab. 2, i.e. totally 8% for the DHW mode. Since the storage and system losses are considerable in DHW mode, in particular in this case of the low DHW consumption, the big difference results which also influences the overall seasonal performance. Concerning the WNG values, mainly the WPEsti default is affected, since the back-up energy use is highest in this case.

Differences between WPEsti and the calculation according to the FHNW method and the monitoring are mainly due to the calculation of weighting factors and the energy losses of the DHW storage. If these items are adapted in the calculation, the method should deliver a good estimation for systems with low supply temperatures.

5.4 Use of heating degree hours and upper temperature limit

The FHNW method uses heating degree hours and an upper temperature limit for heating as approximation for the energy redistribution. As described in chap. 2.3.1.1 the cumulative frequency and heating degree hours are related to a design heat load (e.g. according to SIA 384.201 [17] based on EN 12831 [31]) without taking into account solar and internal gains. However, the method does not use the design heat load to evaluate the energy consumption, but the annual energy need of the distribution system, which is based on the building energy according to SIA 380/1 [26] (currently updated based on EN ISO 13790 [21]). Thereby, the quantity of useful gains is taken into account correctly, but the redistribution to the bins is approximated by only taking into account the losses of the building since heating degree hours are only related to the outdoor temperature. The higher the fraction of used gains in low and ultra-low energy dwellings is, the rougher this approximation may get. In this chapter the impact of the redistribution of the energy need according to heating degree hours and a correction by an upper temperature limit for heating are discussed.

5.4.1 Impact of redistribution of the energy on the calculation results

Fig. 40 shows the redistribution of the annual energy need to 1 K bins according to a monthly and an annual calculation, evaluated for the planning data of Zeiningen performed with the SIA 380/1 software Thermo [28]. For the monthly calculation the bin distribution has been evaluated based on the monthly data sets and the monthly energy need according to SIA 380/1. For the redistribution the monthly net energy and the heating degree hours have also been used, so the same approximation is made for the month. However, all available information of the monthly calculation according to SIA 380/1 is used.

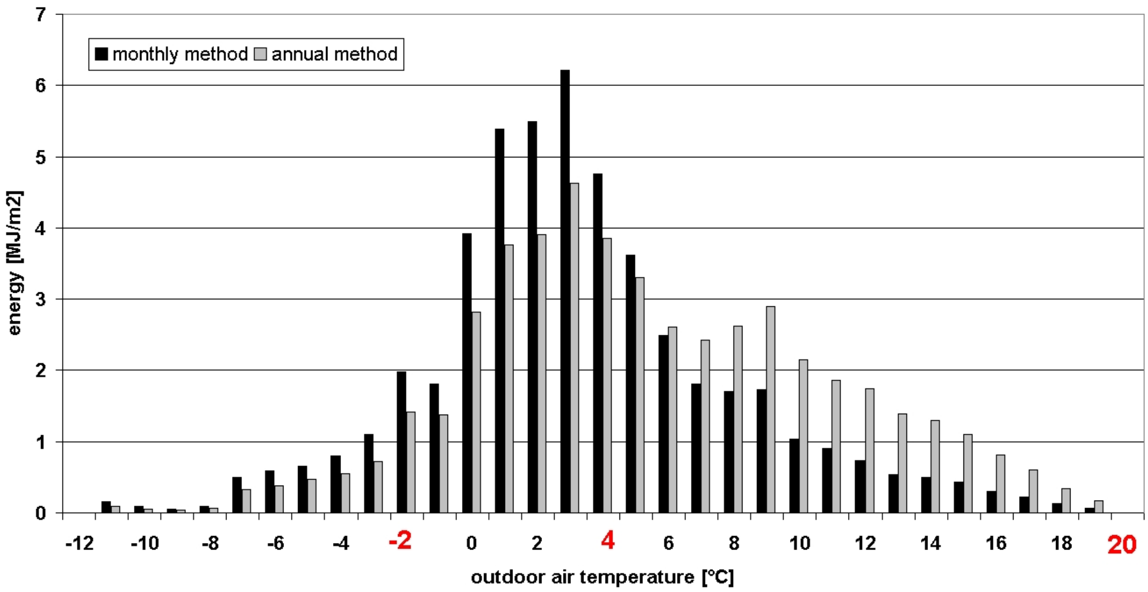


Fig. 40: Energy distribution according to monthly and annual 1 K bins without heating limit (bold red temperatures refer to the bin limits)

For the annual values, the monthly energy for the 1 K temperature bins is summed-up over all months.

For the annual method, the entire annual energy need for space heating is redistributed according to the annual heating degree hours. As shown in Fig. 40, an annual evaluation of the energy needs results in a shift to higher temperatures. While by the monthly evaluation, more energy is consumed in the lower outdoor air temperature range, the annual evaluation overestimates the energy need in the upper range, since the majority of solar gains are used in the upper bin due to the higher solar irradiation.

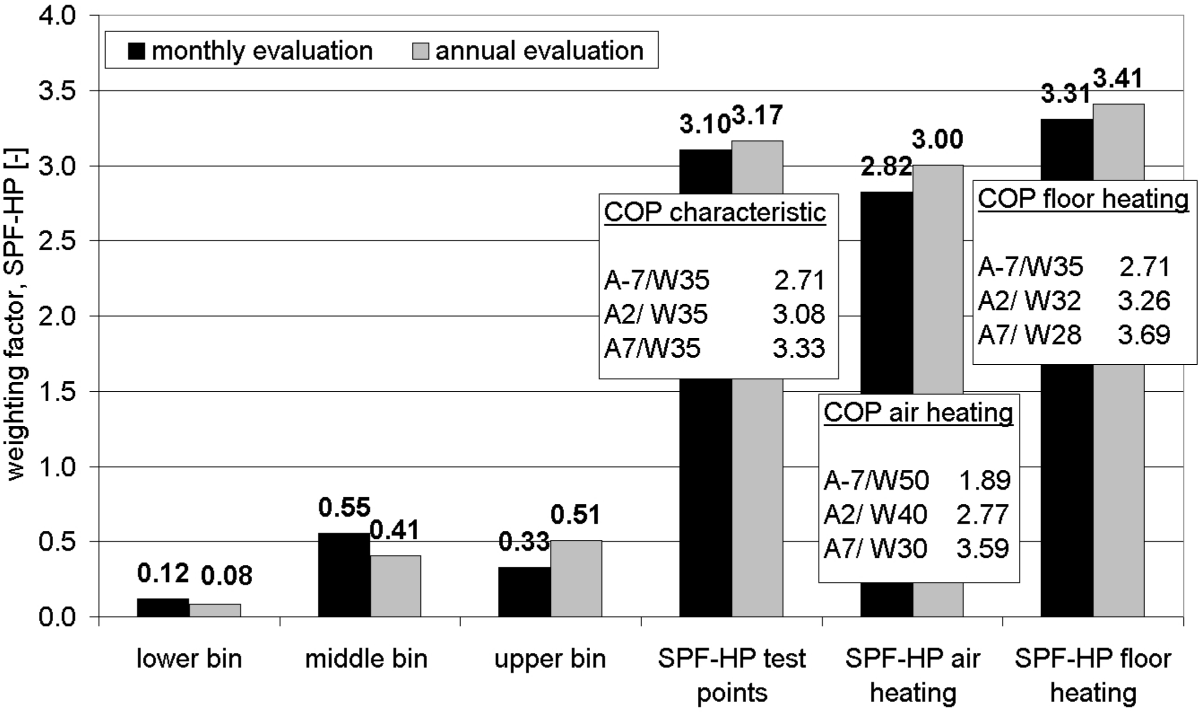


Fig. 41: Resulting weighting factors and weighted SPF-HP values for monthly and annual evaluation in climate Basel-Binningen

The impact of the different energy distributions of the monthly and annual approach for a typical 3-bin method with operating points at A-7, A2 and A7 is depicted in Fig. 41.

Even though the weighting factors differ considerably in the middle bin and the upper bin, the resulting weighted SPF-HP based on different COP characteristic only show a moderate variation depending on the temperature spread of the heating curve. Three different COP characteristics are depicted in the figure as well, referring to the test conditions, and sample heating curves of an air- and floor heating system.

At the extreme, an air heating system with decreasing supply temperature from 50°C to 35°C leads to a difference in the weighted performance factor of ΔSPF=0.18. This only moderate difference of the SPF-HP is due to the fact, that the spread of the COP characteristic is limited, as can be seen at the COP characteristic at constant flow temperature of 35°C, where the spread of COP values is about 0.65, which corresponds to the maximum deviation to be expected (difference for the weighting with 1 for each the lower or the upper bin). Even if this only refers to the direct impact of the COP characteristic on the SPF-HP and does not consider effects of the running time and back-up energy, it shows the tendency, since the sensitivity analysis in chap. 4.2.4 shows that the COP characteristic is the major impact.

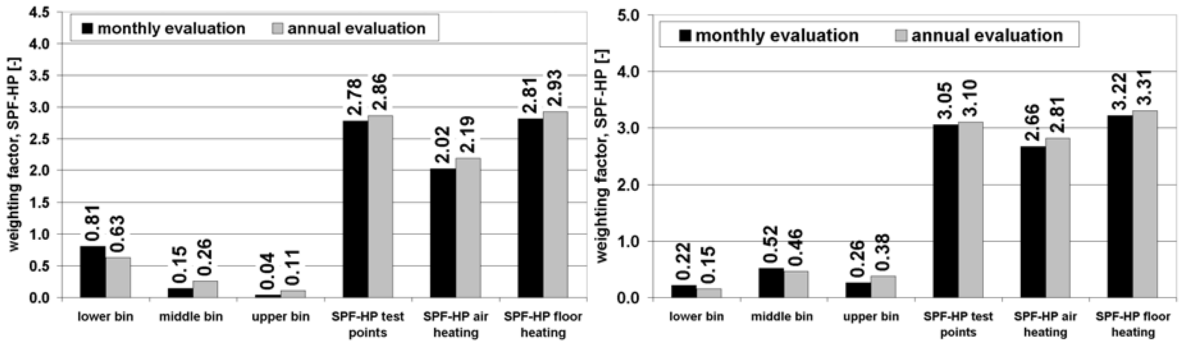


Fig. 42: Resulting weighting factors and weighted SPF-HP for monthly and annual evaluation of the meteorological station Weissfluhjoch (left) and Locarno (right)

Fig. 42 left shows the same diagram for the meteo station Weissfluhjoch, which is at an altitude of 2663 meters in the Swiss Alps. The building data have not been changed, so the building no longer complies with the MINERGIE-P®-standard in the meteorological conditions of the Weissfluhjoch. Due to the altitude, more hours of the year are in the lower bin. The annual evaluation underestimates this fraction, in the middle bin and upper bin, the annual method overestimates the fractions. Thus, the difference between the redistribution corresponds to the situation in case of Basel-Binningen, but fractions are shifted to the lower bins due to the cold temperatures down to –21°C. Differences in SPF-HP values, however, are not significantly different to the case Basel-Binningen.
As third site, the meteo station of Locarno-Monti with a milder climatic region of the Lago Maggiore, canton TI, with high temperatures and short winters is presented in Fig. 42 right. The tendencies are the same for these outdoor temperature conditions, the absolute differences in weighted SPF-HP are a bit lower than in the case of Basel-Binningen due to a smaller difference of the weighting factors in milder climate.
In conclusion, in the extreme case of ultra-low energy houses and depending on the climate, the weighted SPF-HP values can vary moderately up to about 0.2 depending on the heating curve, so a calculation on a monthly basis may lead to a higher exactness of the results. For low energy houses and conventionally insulated buildings, however, redistribution of solar gains according to the heating degree hours does only show a slight deviation, so resulting weighted SPF-HP values only vary slightly.

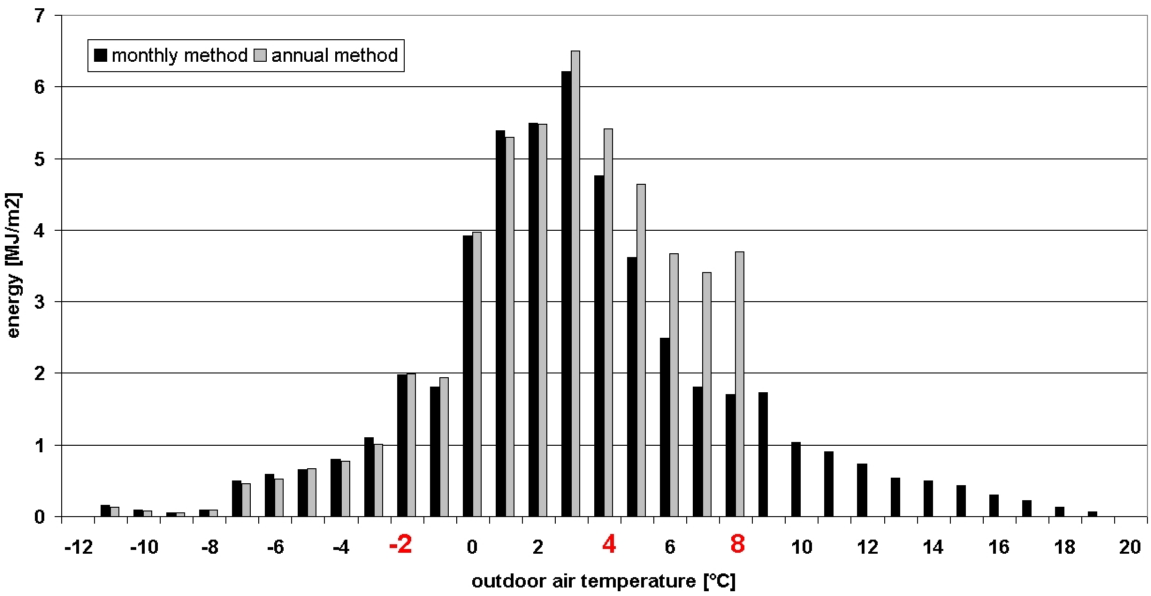


Fig. 43: Energy distribution according to monthly and annual 1 K bins with heating limit 8°C (bold red temperatures mark the bin limits)

Therefore, the approach for an annual method with a redistribution of the space heating requirement according to the heating degree hours is a feasible approach.
In Fig. 43 the energy redistribution of monthly and annual 1 K bins considering a heating limit of 8°C is presented. It can be seen, that in the lower temperature range, the redistribution fits quite well to the monthly energy distribution in the single bins.

However, from a temperature of about 3°C the temperatures start to deviate. But, concerning a 3-bin approach, the weighting factors, which are used in the calculation are well approximated by a heating limit, even though the total bin distribution cannot be reproduced. Moreover, based on these results, the annual calculation method can be improved with regard to the energy redistribution to the bin.

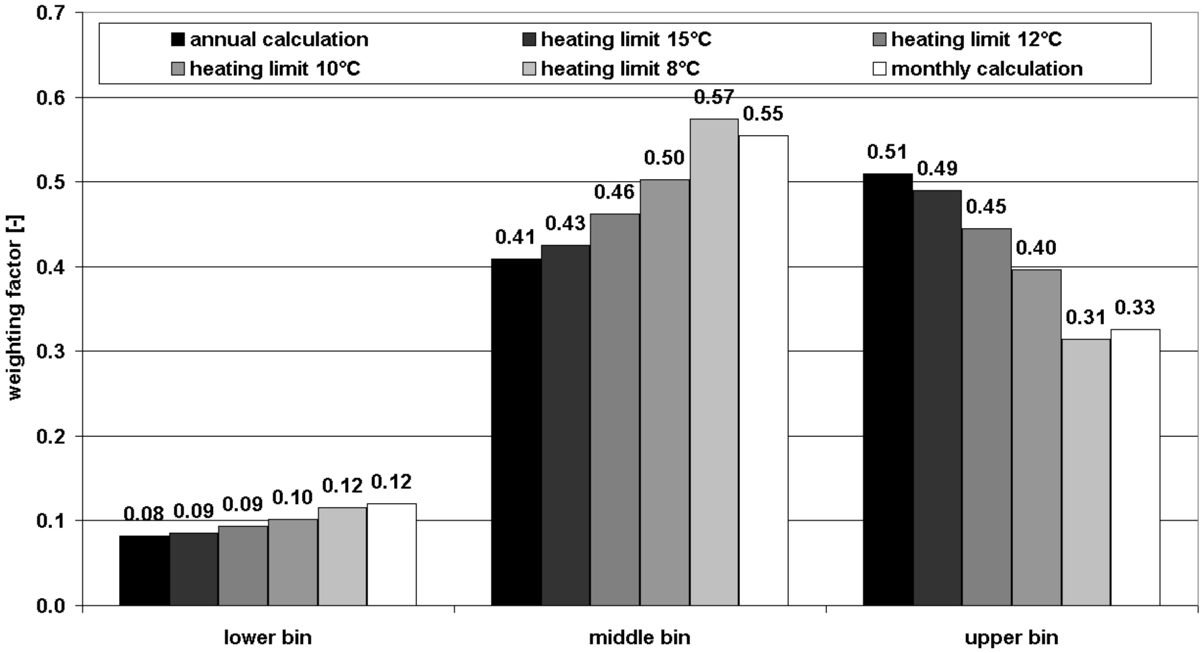


Fig. 44: Resulting weighting factors for a variation of the upper heating limit for Basel-Binningen

Fig. 44 shows the weighting factors of the three bin for a variation of the upper temperature limit for heating. It can be seen that high heating limits correspond to a redistribution of the energy to the bins that ressembles the annual. If the heating limit is lowered, the redistribution approaches the exacter monthly distribution. In this case the limit 8 is too low, since weighting factors supercede the monthly redistribution. This is clear, since the upper bin is more and more reduced with an effect on the area ratios under the cumulative frequency of the other two bins, as well. That also means that the correct monthly redistributions to the bins can be approximated by a variation of the upper temperature limit for heating. If information on the monthly energy is available, the redistribution of the energy can be fitted to the correct distribution by varying the upper temperature limit for heating. If no monthly energy values are available, the annual solar and internal gains according to EN 13790 can be considered and the heating limit can be adapted due to this value. If the used solar and internal gains are high, the heating limit has to be lowered, if the used solar and internal gains are low, a higher heating limit has to be chosen. Since the redistribution is not so sensitive, default heating limits for different buildings types can be defined, too, e.g. 17°C for the building stock, 14°C for the new buildings, 12°C for low energy houses and 9°C for ultra-low energy houses.

5.4.2 Comparison of monthly and annual calculation

To evaluate the difference in exactness between the annual calculation approach and the more detailed calculation based on monthly energy values, both calculations have been accomplished and compared for the design data of the pilot plant in Zeiningen. Based on the evaluation of the previous chapters Fig. 45 depicts the comparison of a monthly and annual calculation of the seasonal performance factor according to the different system boundaries, each for space heating and DHW operation mode and the overall values.

It is obvious that with an adapted temperature limit, which delivers nearly the same weighting factors as the monthly energy redistribution, the seasonal performance factors are quasi the same. That means the redistribution of the solar and internal gains can be approximated by the introduction of an adequate heating limit.

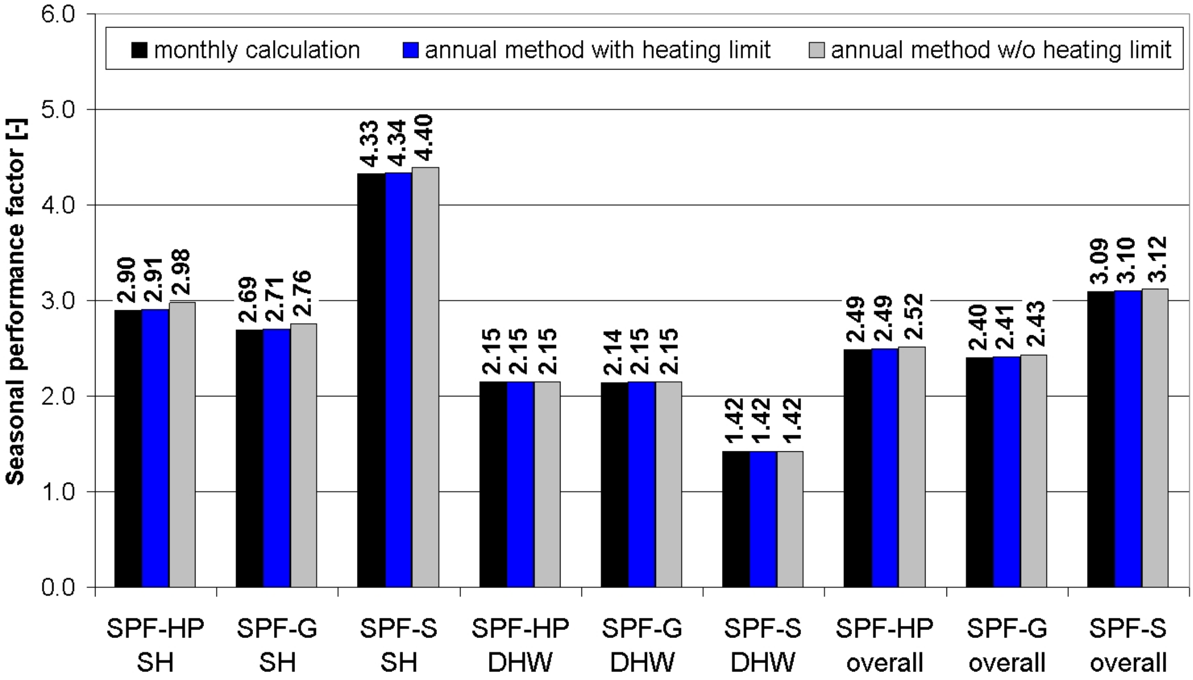


Fig. 45: Comparison of the SPF numbers of the annual and monthly calculation for the different system boundaries and operation modes

It is obvious that with an adapted temperature limit, which delivers nearly the same weighting factors as the monthly energy redistribution, the seasonal performance factors are quasi the same. That means the redistribution of the solar and internal gains can be approximated sufficiently by the introduction of an adequate heating limit.

Fig. 45 emphasizes that an annual calculation already gives a good estimation of the seasonal performance values, even though for this comparison, only little back-up energy was needed, in the annual calculation 2 kWh and in the monthly calculation 30 kWh due to small differences in the lower 1 K bins, so there could be a further impact of back-up energy due to the building design data. Of course, the DHW operation is not influenced by the heating limit since a time-based DHW consumption is assumed, which adds-up to the same values over the year. In case of no correction by a heating limit, differences are in the range as outlined in chap. 5.4.1.

6 PILOT PLANTS

Two pilot plants were evaluated to validate the calculation method on the one hand and to examine the operation of compact units on the other hand. In respect to the aimed building types of MINERGIE and MINERGIE-P two field objects were chosen. One MINERGIE-certified building in Gelterkinden with a compact unit LWZ 303 SOL of manufacturer Stiebel-Eltron and one MINERGIE-P certified building in Zeiningen with compact unit Vitotres 343 of manufacturer Viessmann. The realised measurements are described in the following chapters.

6.1 Pilot plant Gelterkinden

The field measurement of the compact unit LWZ 303 SOL of manufacturer Stiebel-Eltron in Gelterkinden/BL (s. Fig. 46) was carried out from the beginning of April 2004 until the end of July 2005.

Fig. 46: MINERGIE certified building in Gelterkinden (canton Basel-Land)

The compact unit integrates space heating, domestic hot water and ventilation with heat recovery using an air-to-water heat pump as heat source and an electric heater as backup heater (see Fig. 48).

Characterization of the building and boundary conditions for the measurements are:

- One-family house according to MINERGIE requirements in Gelterkinden/BL at south oriented hillside location with solid construction walls, 431m above sea level
- 2 storeys with flat roof and laterally placed basement outside insulation perimeter where the upper storey is named ground floor and the lower storey basement

- hydraulic floor heating and mechanical ventilation with heat recovery
- Energy reference area 153 m^2 (net living space 125 m^2, net volume 305 m^3)
- Heating energy demand according to SIA 380/1 = 157 MJ/m^2a ($DH_{H20/12}$=3348 Kd/a)
- Heating power demand according to SIA 384/2 = 4.1 kW (20 °C / -8 °C)
- design flow/return flow temperature of floor heating 30/25 °C
- Inhabitants are 2 adults, both employed, and 1 child in school
- Set hot water temperature was 45 °C until end of 11/2004 and has then been adjusted to 55 °C afterwards
- Volume flow of mechanical ventilation was set to 100 m^3/h (air change 0.33 per hour). Measurements showed an actual volume flow of 60 m^3/h (air change 0.2), that was suitable for the home owners.

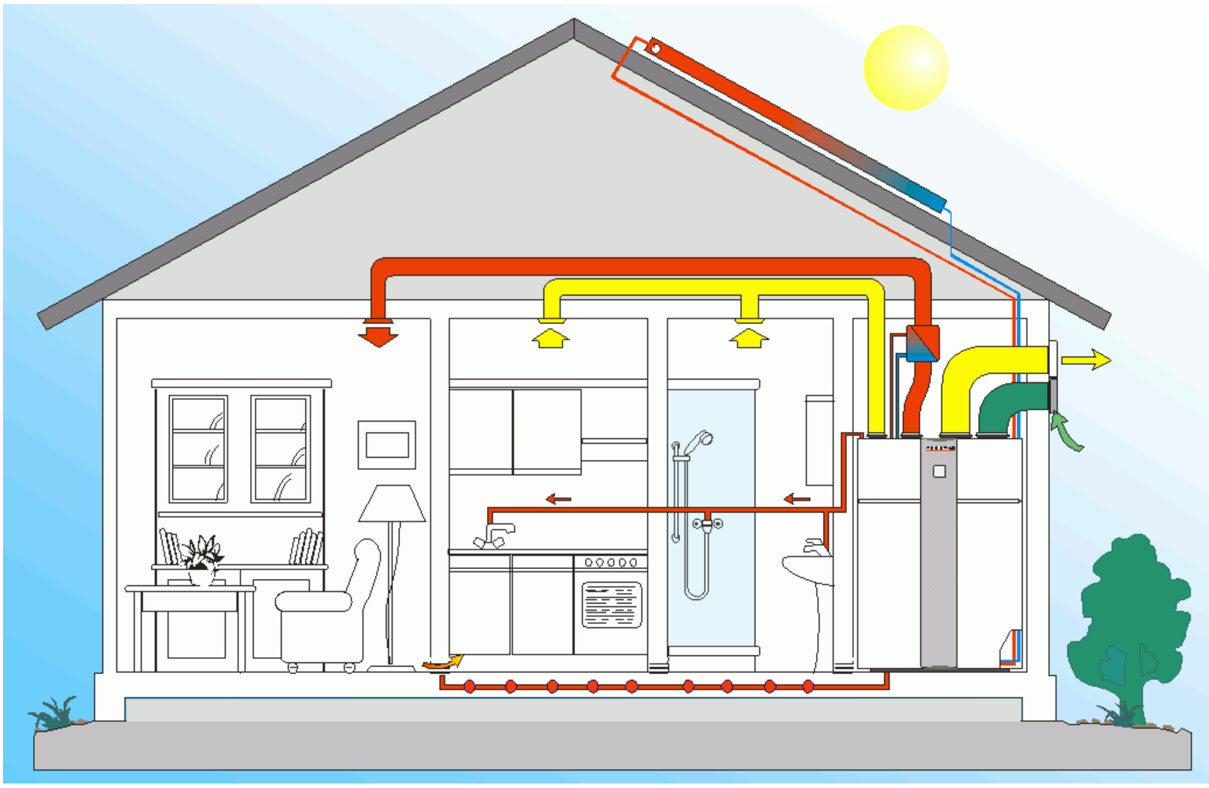


Fig. 47: Function principle of the compact unit in a house [Stiebel Eltron]

Fig. 48 shows the configuration of the compact unit with the following key features:

- Air/water heat pump — 4.2 kW_{th}, 1.3 kW_{el} (A2/W35)
- Electric heating element — 1.4 + 2.8 kW, i.e. a total of 4.2 kW
- Heat pump air volume flow range — < 1000 m^3/h
- Supply / exhaust air volume flow range — 80...230 m^3/h
- DHW-storage volume — 200 liters
- Option for solar heat system integration

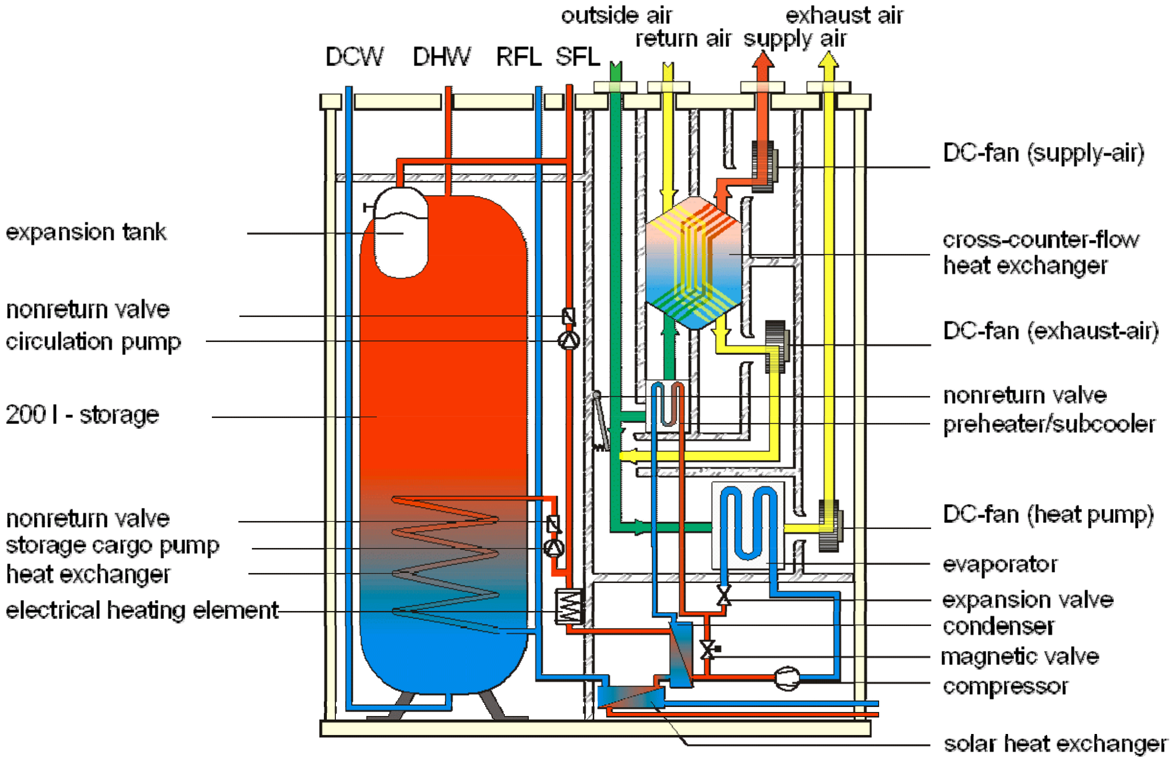


Fig. 48: Configuration of compact unit [Stiebel Eltron]

Few parameters of the pilot plant (s. Fig. 50), e.g. heating characteristic curve (s. Fig. 49), have been optimized to ensure a compact unit working in a well adapted mode for this building and to meet the owners requirements The applied parameters shown in Tab. 32 are the parameters that would also be adjusted by a competent installer during commissioning. The heating curve is given for a mainly return flow temperature controlled system considering 70% return flow and 30 % flow temperature.

Tab. 32: Parameters applied to compact unit:

Parameter	Standard setting	Applied setting
balance point in °C	5.0	-5.0
gradient of heating curve in K/K	0.4	0.3
base offset of heating curve in K	0.0	1.0
hysteresis of heating circuit desired flow temperature in K	4.0	1.0

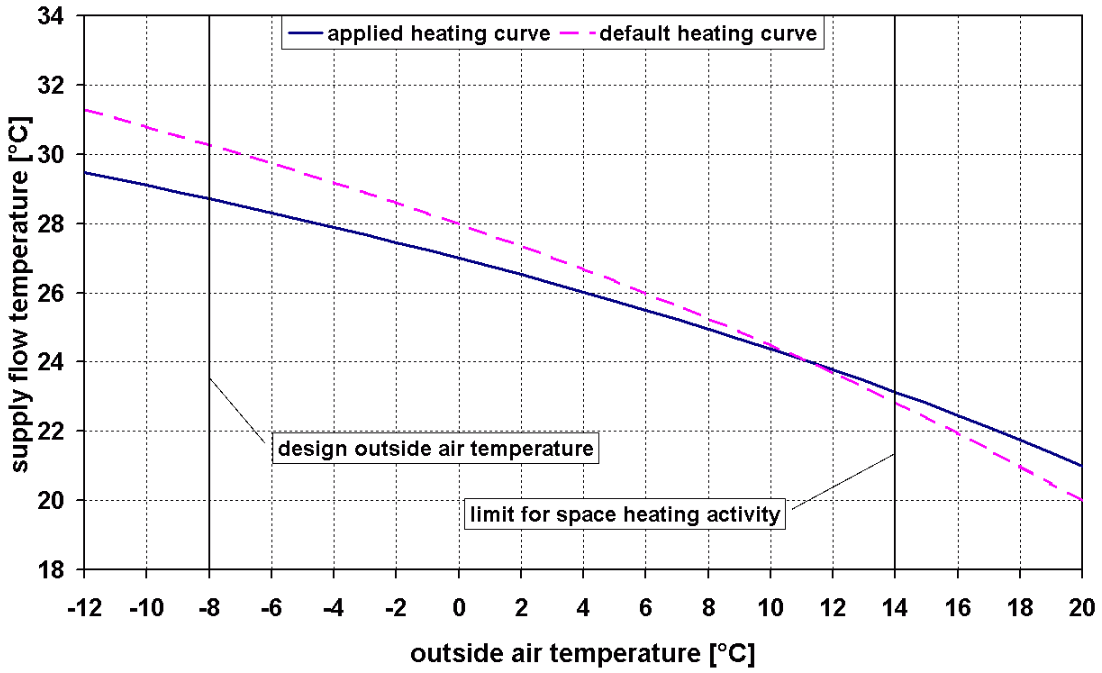
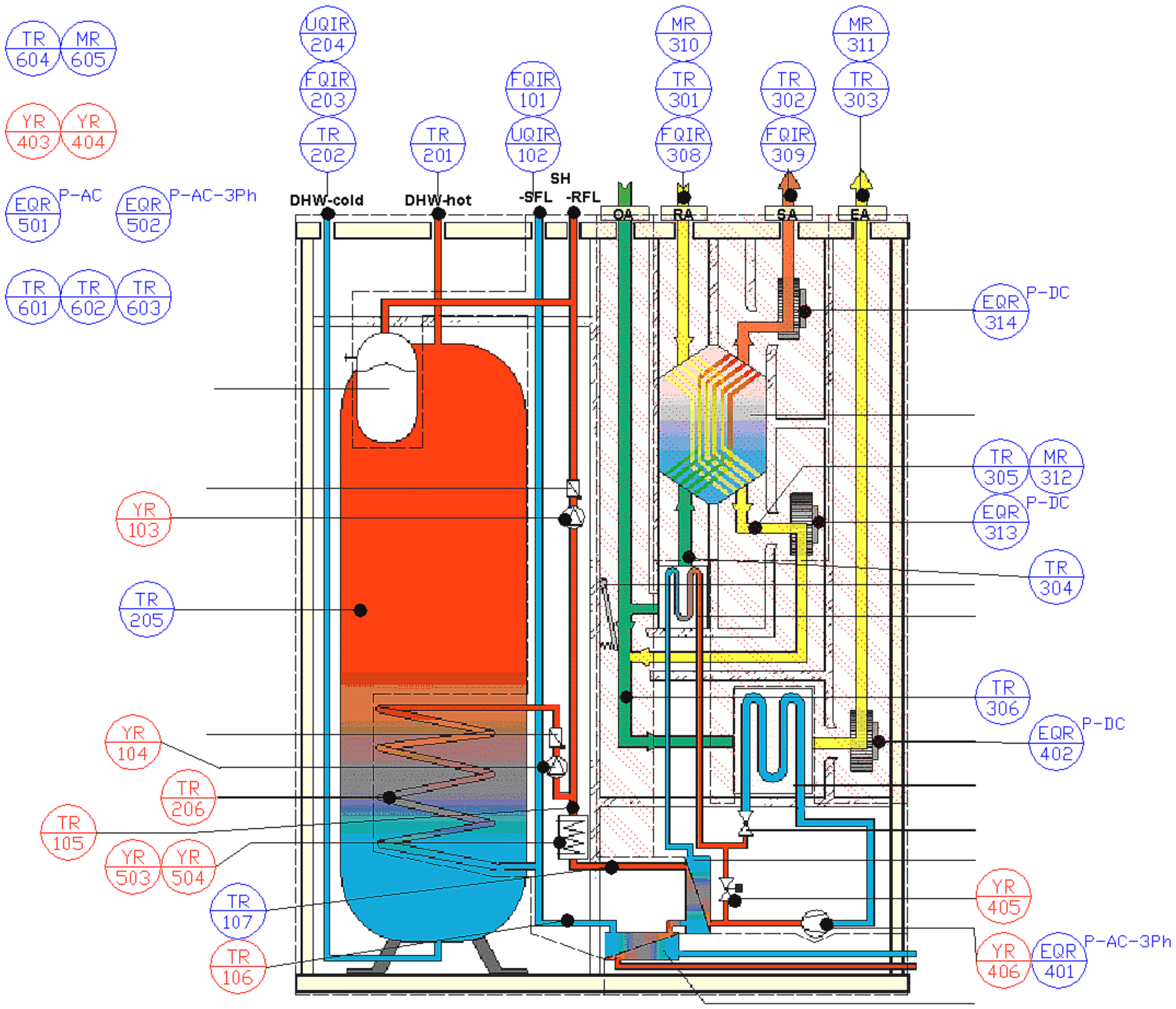


Fig. 49: space heating characteristic curve of field measurement object Gelterkinden/BL

Fig. 50: Installed compact unit in field object Gelterkinden/BL

6.1.1 Measurement scheme

The measurement points are set in a way that, as far as possible, energy balances for all component groups and operation modes can be carried out. The measurement points are shown in Fig. 51.



legend	
TR 100	temperature, registration
FQIR 100	massflow, registration of sum
MR 100	moisture, registration
UQIR 100	heat quantity, registration of sum
EQR 100 P-DC	el. power, registration of sum
EQR 100 P-AC	el. power, registration of sum
EQR 100 P-AC-3Ph	el. power, registration of sum
YR 100	state, registration

pos.	component
1xx	water heating system
2xx	domestic hot water
3xx	ventilation + HRV
4xx	heat pump
5xx	compact unit
6xx	building
7xx	solar heat system

Fig. 51: Measurement points for the pilot plant Gelterkinden (taken from [46])

A detailed description of the single points is given in Tab. 33. Measurements are partly recorded using the internal interface of the compact unit and partly realized with additional measurement devices and a data logging system. The data are transferred via modem connection to the Institute of Energy in Building in Muttenz and checked and evaluated there in a weekly schedule to avoid greater data losses at system faults.

Tab. 33 Description of measurands

Pos	symbol	unit	measurand	principle	accuracy
101	$\dot{V}_{WHS\_SH}$	l/pulse	volume flow WHS (960 l/h measured nominal flow)	flow counter	29 l/h
102	$\dot{Q}_{WHS\_SH\_NED}$	100Wh	heat quantity to SH	heat counter	3 Wh
103	Z_{WHS_PMP-SH}	1/0	state circulating pump SH	-/-	-/-
104	$Z_{WHS\_PMP-DHW}$	1/0	state loading pump DHW	-/-	-/-
105	θ_{WHS_SFL}	°C	temperature flow HP+ELH	PT100	0.3 K
106	θ_{WHS_RFL}	°C	temperature return flow	PT100	0.3 K
107	θ_{WHS_FL-HP}	°C	temperature flow HP	PT100	0.3 K
121	θ_{WHS_SH-SFL}	°C	temperature flow SH	PT100	0.3 K
122	θ_{WHS_SH-RFL}	°C	temperature return flow SH	PT100	0.3 K
123	$\theta_{WHS\_DHW-SFL}$	°C	temperature flow DHW storage heater	PT100	0.3 K
124	$\theta_{WHS\_DHW-RFL}$	°C	temperature return flow DHW storage heater	PT100	0.3 K
125	$\dot{V}_{WHS\_DHW}$	l/h	volume flow storage heater (720 l/h measured nominal flow)	state + nominal flow	42 l/h
201	θ_{DHW_COLD}	°C	temperature cold water	PT100	0.3 K
202	θ_{DHW_HOT}	°C	temperature hot water	PT100	0.3 K
203	$\dot{V}_{DHW}$	10 l	flow rate hot water	flow counter	0.5 l
204	$\dot{Q}_{DHW\_NED}$	100Wh /pulse	heat quantity hot water	heat counter	5 Wh
205	θ_{DHW_ST-t}	°C	temperature storage top	PT100	0.3 K
206	θ_{DHW_ST-m}	°C	temperature storage middle	PT100	0.3 K
301	θ_{V_RA}	°C	temperature return air	PT100	0.3 K
302	θ_{V_SA}	°C	temperature supply air	PT100	0.3 K
303	θ_{V_EA}	°C	temperature exhaust air	PT100	0.3 K
304	θ_{HRV_OA}	°C	temperature outside air before HRU	PT100	0.5 K
305	θ_{HRV_EA}	°C	temperature exhaust air after HRU	PT100	0.5 K
306	θ_{HP_MA}	°C	temperature mixed air HP	PT100	0.5 K
308	$\dot{V}_{V\_RA}$	m^3/h	flow rate return air	thermal anemometer	2 m^3/h
309	$\dot{V}_{V\_SA}$	m^3/h	flow rate supply air	thermal anemometer	2 m^3/h
310	φ_{V_RA}	%r.h.	rel. humidity return air	cap.Sensor	1.5%rF
311	φ_{V_EA}	%r.h.	rel. humidity exhaust air	cap.Sensor	1.5%rF
312	φ_{HRV_EA}	%r.h.	rel. humidity exhaust air after HRU	cap.Sensor	1.5%rF
313	$P_{V\_VE-RA\_ED}$	W	el. energy demand return air fan (15 W)	1)	1 W
314	$P_{V\_VE-SA\_ED}$	W	el. energy demand supply air fan (15 W)	1)	1 W
321	$\dot{V}_{V\_EA}$	m^3/h	volume flow outside air (650 m^3/h)	2)	13 m^3/h

Tab. 34 Description of measurands (continued)

Pos	symbol	unit	measurand	principle	accuracy
401	$P_{HP\_CMP\_GED}$	kWh	el. energy demand compressor	electric energy counter	1 W
402	$P_{HP\_VE\_GED}$	Wh	el. energy demand HP fan (90 W)	2)	2 W
403	Z_{CU_ERR-hp}	0/1	high pressure failure	-/-	-/-
404	Z_{CU_ERR-lp}	0/1	low pressure failure	-/-	-/-
405	Z_{HP_DV}	0/1	state defrost valve	-/-	-/-
406	Z_{HP_CMP}	0/1	state compressor	-/-	-/-
501	$P_{CU\_VE+CTRL\_ED}$	100Wh	el. energy demand fans & control	electric energy counter	1 Wh
502	P_{CU_GED}	10Wh	el. energy demand compact unit	electric energy counter	0.1 Wh
503	Z_{CU_ELH1}	0/1	state of el. heating element 1 (ELH1)	-/-	-/-
504	Z_{CU_ELH2}	0/1	state of el. heating element 2 (ELH2)	-/-	-/-
505	T_{CU}	°C	temperature inside compact unit	PT100	0.3 K
521	$P_{CU\_ELH\_GED}$	kW	energy demand electric heating elements (ELH1=1.4kW + ELH2=2.8 kW, i.e. a total of 4.2kW)	2)	0.2 kW
601	θ_{BLD_GF}	°C	room temperature ground floor	PT100	0.3 K
602	θ_{BLD_BA}	°C	room temperature basement	PT100	0.3 K
603	θ_{BLD_TR}	°C	temperature technical room	PT100	0.3 K
604	θ_{BLD_OA}	°C	temperature outside air	PT100	0.3 K
605	φ_{BLD_OA}	%r.h.	rel. humidity outside air	cap.Sensor	1.5%r.h.

1) Value is calculated from measured air flow and measured characteristic power consumption over volume flow
2) Value is assumed to be constant when active and therefore calculated with state of device and measurement of effective value in advance with clamp-on ammeter.

Verifications and Calibration

The supply and return air volume flow are continuously measured by a reference air velocity measurement combined with a transfer factor. The measurement of the volume flow and the velocity distribution for the transfer factor is carried out by a grid measurement in accordance with guideline VDI 2640 [48]. This means that the measuring section is divided into segments of equal areas, in which respective center random velocity measurements take place. The transfer factor from air velocity to volume flow is for the supply air 52 (m^3/h)/(m/s) and for the return air 56 (m^3/h)/(m/s).
The air volume flow of the heat pump was also determined by a grid measurement in accordance with guideline VDI 2640 and then assumed to be constant when the heat pump is active. Thus air flow energy calculations are evaluated with the state of the heat pump multiplied with the measured volume flow 650 m^3/h.
The power consumption of the HRV-fans was measured with special equipment in advance and is afterwards calculated on basis of the measured air velocity. Fig. 52 shows the measured points and the approximation with the cubic polynomial:

$$P_{V,sa,fan,in} = 8.4484 * v_{sa}^3 - 2.9923 * v_{sa}^2 + 3.7657 * v_{sa} \quad \text{eq. 39}$$

where

$P_{V,sa,fan,in}$	power input to the supply air side ventilation fan	[W]
v_{sa}	air velocity of supply air	[m/s]

The electric power consumption of the HP-fan is corresponding to the volume flow assumed to be constant with measured 90W.

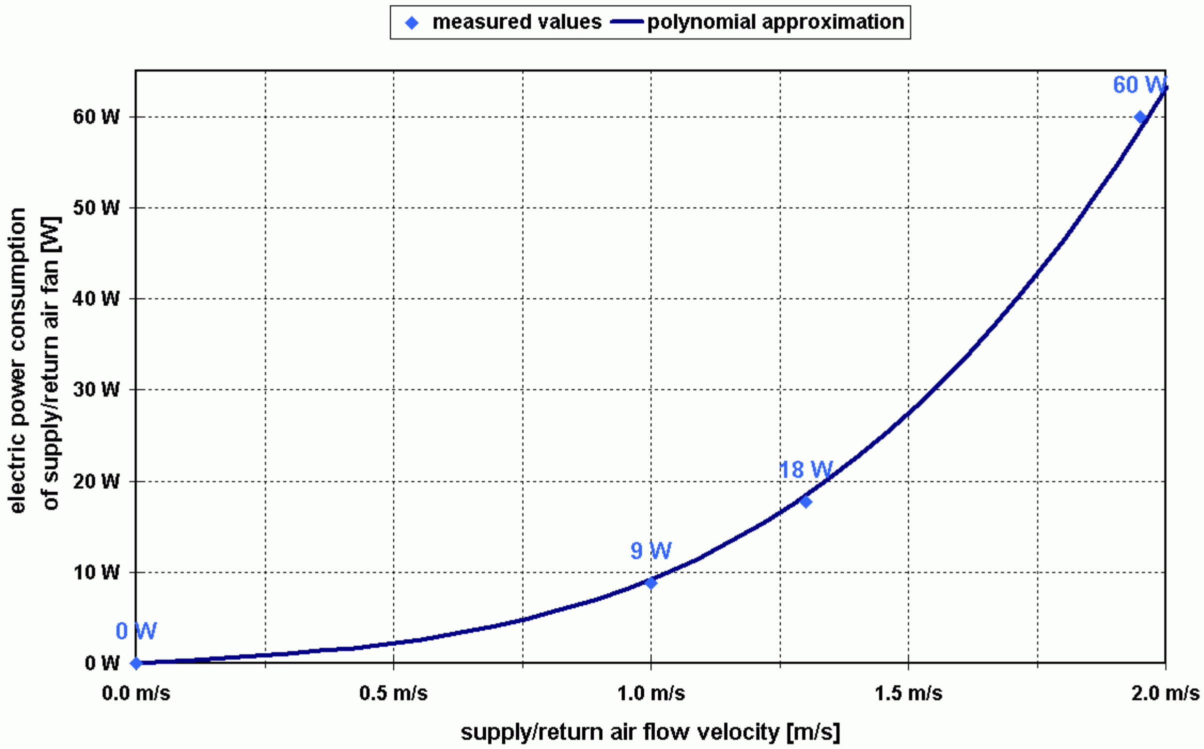


Fig. 52: Polynomial approximation of HRV-fan power consumption

The power consumption of the circulation pumps for space heating and domestic hot water loading were also measured explicitly and assumed to be constant when running. The power consumption of the space heating circulation pump working at stage 2 is measured with 54W where the data sheet shows 65W. The domestic hot water loading pump also working at stage 2 is measured with 33W where the data sheet shows 40W. In both cases the measured values are used for the evaluation of the measurement.

The mass flow of the space heating circulation pump is measured in the flow counter. The heat emission system works without thermostatic valves for single room control. Therefore the total mass flow of the space heating system is constant during operation with 960 l/h = 0.27 kg/s. For the evaluation the state of the space heating circulation pump is multiplied with the constant mass flow of 0.27 kg/s.

The mass flow of the DHW loading pump is unknown and has to be determined from a thermal energy balance measurement in advance. Therein the DHW loading pump runs with both electric heating elements active and the flow temperature and return flow temperature are recorded. The measurement at steady state conditions show the mass flow with

$$\dot{m}_{W,st} = \frac{P_{W,aux,sk,in}}{c_w * \Delta\theta_w} = \frac{4176W}{4179\frac{J}{kgK} * 5K} = 0.2\frac{kg}{s} = 12\frac{kg}{min} \qquad \text{[kg/s]}$$

eq. 40

where

$\dot{m}_{W,st}$	mass flow of the water for storage loading	[kg/s]
$P_{W,aux,sk,in}$	electrical power input to the storage loading pump	[W]
C_w	specific heat capacity of water	[J/(kg·K)]
$\Delta\theta_w$	temperature difference between flow and return temperature	[K]

For the evaluation the mass flow is again assumed to be constant during operation.

6.1.2 Key values to be determined (objective)

For a further comparison the following the characteristic values according to eq. 28 to eq. 30 have been evaluated. In contrary to the calculation method, for the pilot plant Gelterkinden the HP subcooler is explicitly taken into account, while for the calculation it is part of the characteristic.

- Electrothermal amplification factor ETV_{HRV} of air heat recovery system

$$ETV_{hru} = \frac{H_{V,sa} - H_{V,oa}}{E_{V,fans,in}}$$

eq. 41

  (can only be calculated when HP subcooler is not active and hence $\theta_{V,hru,in} = \theta_{oa}$)

- Temperature change coefficient of outdoor air Φ_{oa} of air heat recovery system

$$\Phi_{oa} = \frac{\theta_{V,sa} - \theta_{V,oa}}{\theta_{V,ra} - \theta_{V,oa}}$$

eq. 42

  (can only be calculated when HP subcooler is not active and hence $\theta_{V,hru,in} = \theta_{oa}$)

- Heat pump performance factor PF-HP (equivalent to COP)

$$PF - HP = \frac{Q_{H,hp} + Q_{W,hp} + Q_{H,ve,pre}}{E_{HW,hp,cmp,in} + E_{HW,hp,fan,in}}$$

eq. 43

  The PF-HP is the ratio between energy produced by the heat pump and corresponding energy demand in compressor and heat pump evaporator fan.

- Generator performance factor PF-G (equivalent to efficiency of oil or gas fired boilers)

$$PF - G = \frac{Q_{H,hp} + Q_{W,hp} + Q_{H,ve,pre} + Q_{HW,bu}}{E_{HW,hp,cmp,in} + E_{HW,hp,fan,in} + E_{HW,bu,in} + E_{HW,ctr,aux,in}}$$

eq. 44

  The PF-G is the ratio between energy produced by all heat generators and corresponding energy demand of these heat generators.

- System performance factor PF_{SYS}

$$PF - SYS = \frac{Q_{H,dis,in} + Q_{W,dis,in} + Q_{H,ve,out}}{\sum E_{HWV,in}}$$

eq. 45

$$Q_{H,ve,out} = (H_{V,sa} - H_{V,oa})$$

eq. 46

$$\sum E_{HWV,cu,in} = E_{HW,hp,in} + E_{HW,bu,in} + E_{H,aux,sk,in} + E_{W,aux,sk,in} + E_{HWV,ctr,aux,in} + E_{V,fans,in}$$

eq. 47

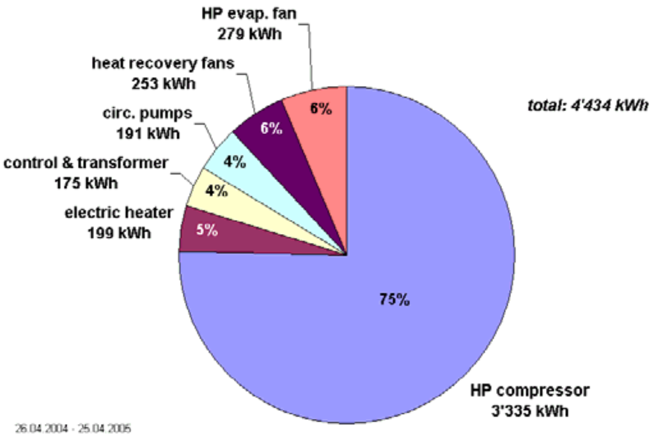
The PF-SYS is the ratio between net used energy including heat recovery and the total energy demand of the system. Due to the fact that heat recovery is included in the PF-SYS, the PF-SYS can be higher than the PF-G.

- Shares of electricity demand accounting for
  HP compressor, HP evap. fan, heat recovery fans / circ.pumps / control / electric heater
- Shares of net heat demand accounting for
  tap water / space heating / ventilation

NOTE: The PF-HP and the energy performance factor of the generator PF-G do not include the pressure drop in the condenser.

6.1.3 Results

The period from 26th April 2004 up to 25th April 2005 is chosen for evaluation out of the whole measured data. The time interval for the summer evaluation has been defined from the beginning of June to the end of September.



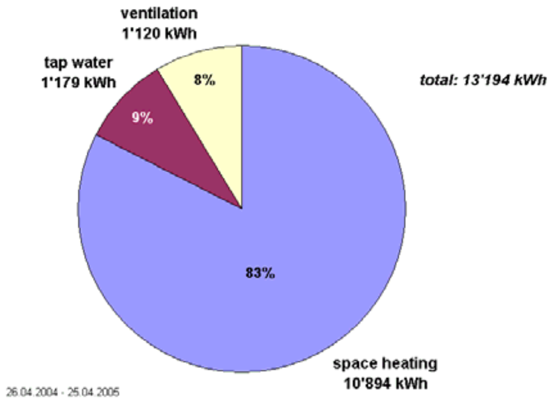


Fig. 53: left part: shares of electricity demand in year 2004/05
right part: shares of net heat demand in year 2004/05

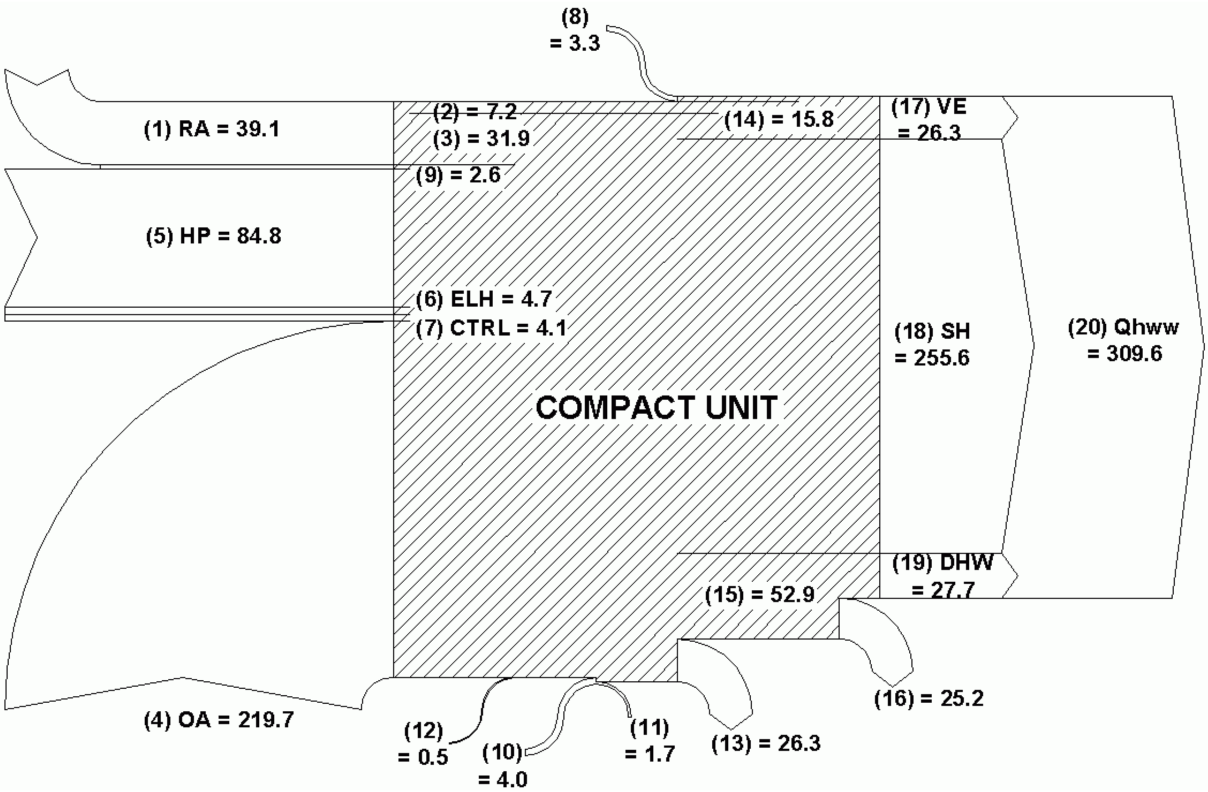
Fig. 53 left part shows that the heat pump with the evaporator fan uses 81% of the electricity demand in the year. The electric heater consumes with 5% of the electricity demand a small part. The main part of the heat pump and the minor part of the electric heater show that the compact unit is an appropriate choice for the building. Ventilation and heat recovery consume 6%, control and transformer 4% and circulation pumps 4%. Fig. 53 right part shows for the year a space heating (SH) energy demand of 10'894 kWh (256 MJ/m^{2}a) which is 60% higher than the calculated demand according to SIA380/1 with 6670 kWh. Contrary to this the domestic hot water (DHW) energy consumption is with 1'179 kWh very low (acc. SIA 380/1 [26]: 2125 kWh). The ventilation energy demand of 1'120 kWh lies in a normal range.

The causes for the high space heating energy demand are presumed in:

- Only small part of the optimistic solar gains was realized as internal heat gain because of frequently closed blinds. The additional heat demand is estimated with 40 MJ/m^{2}a which is the difference in two calculations of SIA380/1 one with open blinds and one with mainly closed blinds.
- The situation of the building at the inclination is strongly exposed to wind.
- The drying of the building is presumed not to be finished yet.

The air leakages were verified by a blower door test. The result is an air change rate n_{50} of 0.61 1/h, which is a very good result and nearly fulfills the MINERGIE-P requirements. A thermographic survey of the building envelope furthermore shows a uniform appearance of the building without avoidable thermal bridges.

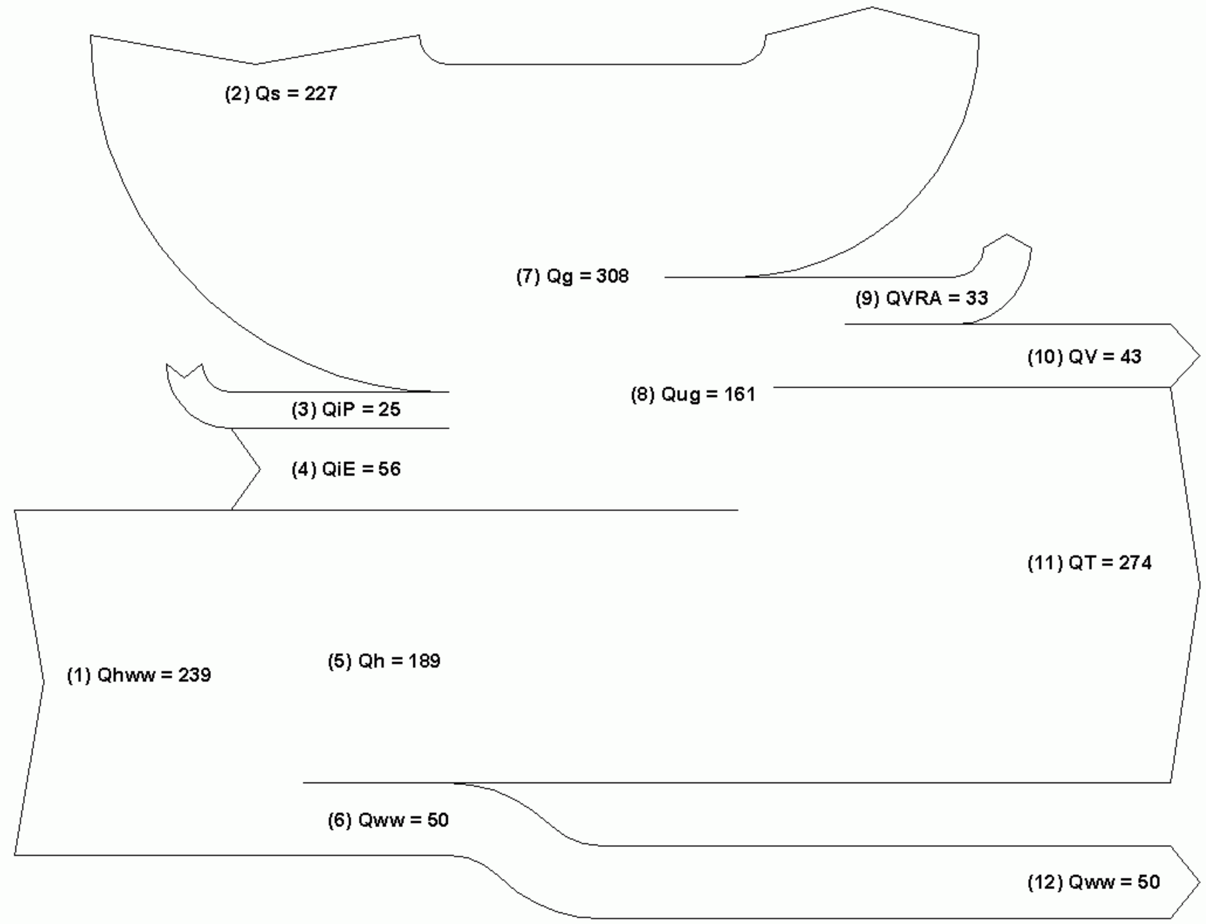
Energy flow chart of compact unit



No	Symbol	Description	Energy
1	$Q_{H,ve,ra,rvd}$	recovered heat from return air (RA) for SH	1666 kWh
2	$Q_{H,ve,hru,ra,rvd}$	heat from RA recovered in ventilation heat recovery for SH	308 kWh
3	$Q_{HW,hp,ra,in}$	heat from RA recovered by the heat pump for SH and DHW	1358 kWh
4	$Q_{HW,hp,oa,in}$	heat from outside air (OA) used by heat pump	9360 kWh
5	$E_{HW,hp,in}$	electricity consumption of heat pump (HP)	3614 kWh
6	$E_{HW,bu,in}$	electricity consumption of electrical back-up heater (ELH)	199 kWh
7	$E_{V,ctr,in}$	electricity consumption of control and transformer (CTRL)	175 kWh
8	$E_{V,fan,sa,in}$	electricity consumption of supply air fan	141 kWh
9	$E_{V,fan,ra,in}$	electricity consumption of return air fan	112 kWh
10	$E_{H,aux,sk,in}$	electricity consumption of space heating circulating pump	171 kWh
11	$Q_{H,aux,sk,ls,nrbl}$	non recoverable waste heat of space heating circulating pump	71 kWh
12	$E_{W,aux,sk,in}$	electricity consumption of domestic hot water loading pump	20 kWh
13	$Q_{HWV,cu,ls}$	heat losses of compact unit to technical room	1121 kWh
14	$Q_{H,ve,pre}$	generated heat of HP for ventilation by preheater	671 kWh
15	$Q_{W,gen}$	generated heat of HP and ELH for domestic hot water	2252 kWh
16	$Q_{W,st,ls,sb}$	heat losses of domestic hot water storage	1073 kWh
17	$Q_{H,ve,out}$	heat energy need by ventilation losses in mechanical ventilation	1120 kWh
18	$Q_{H,dis,in}$	heat energy need of the space heating distribution system	10894 kWh
19	$Q_{W,dis,in}$	heat energy need of the domestic hot water distribution system	1179 kWh
20	$Q_{HW,dis,in}$	heat energy need of the SH and DHW distribution system according to [26]	13194 kWh

Fig. 54: Energy flow chart of compact unit LWZ 303 SOL for the period April `04 to April 2005 with specific energy flows in [MJ/m^{2}a]

Energy flow chart of building Gelterkinden according to SIA380/1



No	Symbol	Description	Energy
1	Qhww	heat use for heating and hot water preparation	10184 kWh
2	Qs	passive solar gains	9673 kWh
3	QiP	metabolic heat	1065 kWh
4	QiE	heat from other appliances	2386 kWh
5	Qh	heat use	8054 kWh
6	Qww	heat for hot water preparation	2131 kWh
7	Qg	total gains	13124 kWh
8	Qug	useful gains	6860 kWh
9	Qvra	ventilation heat recovery	1406 kWh
10	QV	ventilation heat loss	1832 kWh
11	QT	transmission heat loss	11675 kWh
12	Qww	heat for hot water preparation	2131 kWh
13	Qhww	heat use for heating and hot water preparation	10184 kWh
14	Qs	passive solar gains	9673 kWh
15	QiP	metabolic heat	1065 kWh
16	QiE	heat from other appliances	2386 kWh
17	Qh	heat use	8054 kWh
18	Qww	heat for hot water preparation	2131 kWh
19	Qg	total gains	13124 kWh
20	Qug	useful gains	6860 kWh

Fig. 55: Energy flow chart of building Gelterkinden according to SIA380/1 [26] with specific energy flows in [MJ/m^{2}a]

Climate & Comfort

The evaluated period 2004/'05 is considering the heating degree days with 2'732 $Kd_{20/12}$ 18% warmer compared to the reference weather data set Basel-Binningen (SIA381/2 '88) with 3'348 $Kd_{20/12}$. The heating degree days at real conditions, i.e. the measured outside air temperature, the measured room air temperature (average in winter: 20.8°C) and a space heating activity limit of 14°C, account for 3'088 Kd. The warmer climate in the considered period is also visible in the average outside air temperatures shown in Tab. 35 for year, winter and summer period.

Tab. 35: Heating degree days&outside air temperature compared to reference weather data

	$DH_{H20/12}$ SIA381/2	$DH_{H20/12}$ measured	$DH_{H,\theta i/14}$ measured	$\theta_{bld,oa}$ SIA381/2	$\theta_{bld,oa}$ measured
	Kd	Kd	Kd	°C	°C
year	3348	2732	3088	9.4	11.4
winter	3259	2715	3050	5.6	7.2
summer	89	17	38	16.8	19.3

The room air temperatures are most of the time in the desired range of 20°C to 26°C (s. Fig. 56). Only at outside air temperatures below -8°C the room air temperature drops slightly below 20°C. At outside air temperatures over 26°C the temperature in the room stays in 50% of these times below 26°C.

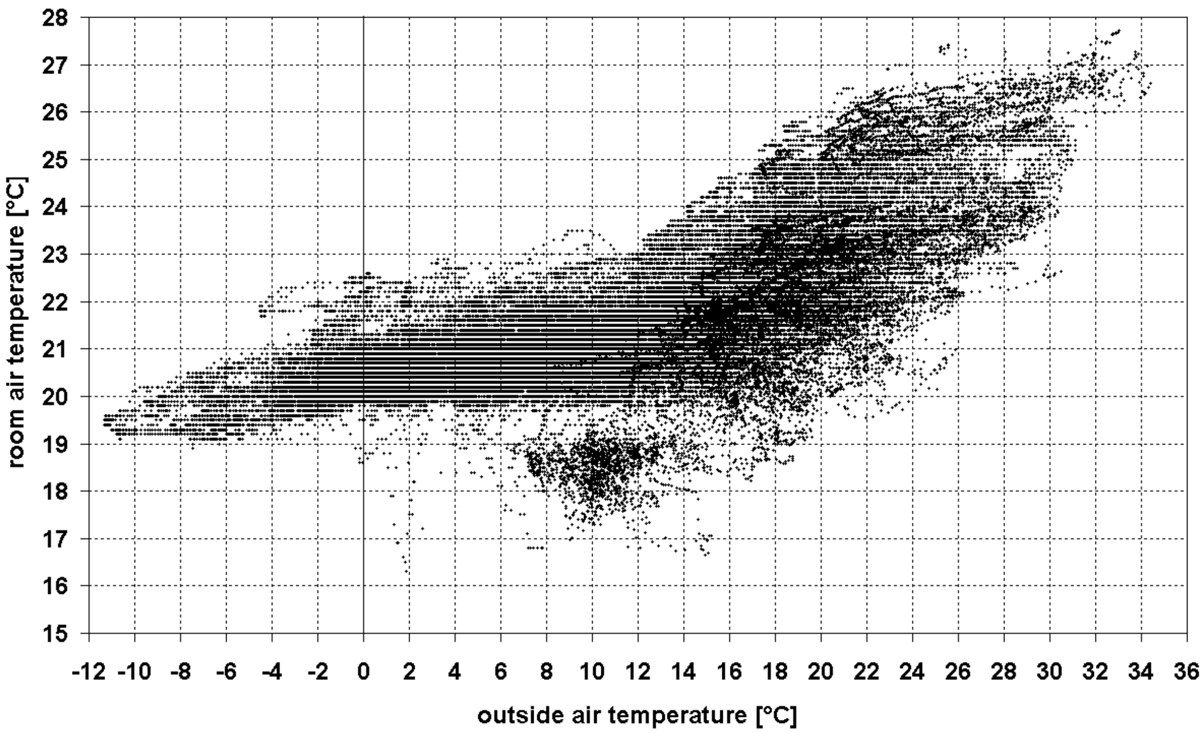


Fig. 56: Room air temperature versus outside air temperature

Further specific values of building and user behaviour out of measurement are:

- Space heating energy demand 256 MJ/m^2a (71 kWh/m^2a).
- Domestic hot water consumption 32 ltr/pers.d (50°C, 2.5 persons)
- Heat load 4 kW (-5°C) / 4.5 kW (-8°C)
  (estimated with average power over 6h in net space heating energy demand)

Efficiencies

Tab. 36: Key values of the pilot plant Gelterkinden for year, summer and winter period

HP compact unit Gelterkinden/BL	SPF-HP	SPF-G	SPF-SYS	ETV (weekly average)	Φ_{oa} (weekly average)
Summer	3.8	3.2	2.4	1.9..2.7	0.61..0.70
Winter	3.8	3.6	3.1	2.0..6.7	0.64..0.77
Year	3.8	3.6	3.1	1.9..6.7	0.61..0.77

Tab. 36 contains the characteristic key values for the year, summer and winter period. The seasonal system performance factor SPF-SYS shows a big difference from summer to winter period, caused by a significant influence of a low domestic hot water consumption and the low ventilation volume flow that lead to a small net energy requirement and a relatively bigger share of stand-by energy in summer period. The generator weekly performance factor SPF-G is higher in wintertime due to the high amount of produced energy and a lower average heat supply temperature. The summer period is set as time period, which covers the domestic hot water demand and a small, i.e. 230 kWh, of space heating energy demand justified by 38 Kd of heating degree days. The electro-thermal amplification factor ETV_{hru} and the temperature change coefficient Φ_{oa} are depicted as range of weekly average values, since there are especially in wintertime only few undisturbed values available where the heat recovery runs without heat pump. Both ETV_{hru} and Φ_{oa} cannot be calculated with running heat pump, since the subcooler influences the temperature of the heat recovery.

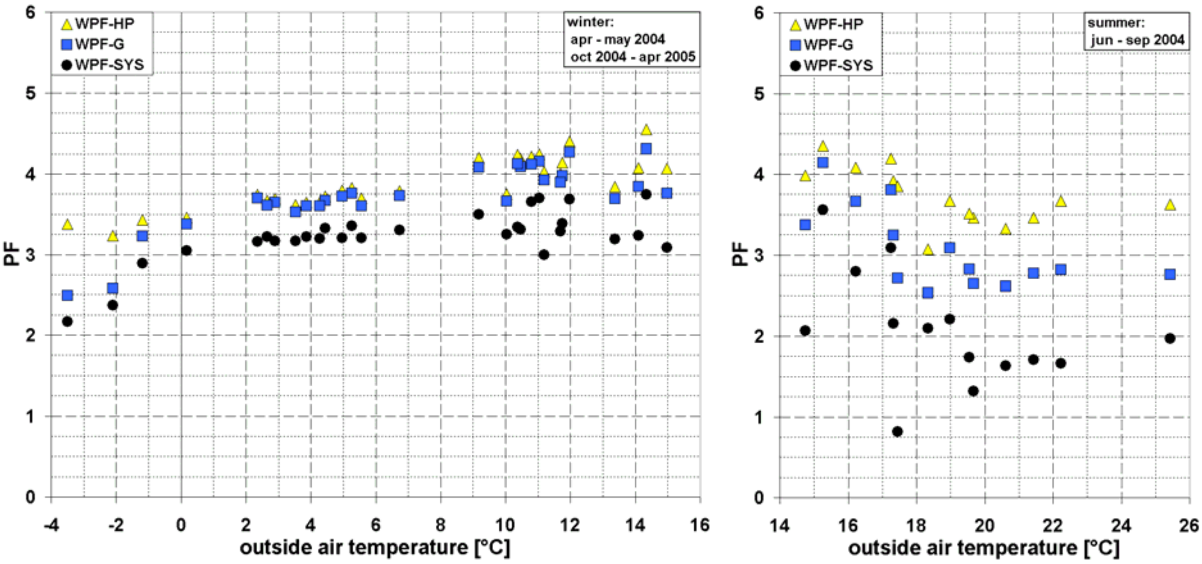


Fig. 57: Weekly Energy Performance Factors WPF for field measurement of Stiebel Eltron LWZ 303 SOL

The values for the WPF-HP presented in Fig. 57 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 Fig. 57 deteriorate with the outside air temperature. 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 they are with 509 kWh over 18 °C outside air temperature compared with 13194 kWh in total a small part of yearly energy requirement and thus of minor importance. The performance factors WPF-HP and WPF-G depicted in Fig. 57 include the generated heat in the subcooler for ventilation. This energy could not be evaluated without a detailed measurement like in this field object. Not considering the generated heat in the subcooler would decrease the WPF-HP and the WPF-G by about 0.2 or 5% (s. Tab. 37).

Tab. 37: Key values of the pilot plant Gelterkinden without subcooler energy

HP compact unit Gelterkinden/BL	SPF-HP	SPF-G	SPF-SYS
Summer	3.6	3.1	2.4
Winter	3.6	3.4	3.1
Year	3.6	3.4	3.1

Uncertainties of results

The central results of the field measurements are on the one hand the key values for this field object and on the other hand the hourly development of the measured values for validation. Measured values are always afflicted with an uncertainty of measurement. The uncertainties of the energy values, presented in Tab. 38, result from those of the measurement with the rules of the gaussian error propagation. The temperature change coefficient Φ_{OA} is depicted without uncertainty of leakages because they are unknown for this compact unit. The uncertainty of the key values SPF-HP, SPF-G and SPF-SYS, presented in Tab. 38, is calculated as simple error propagation based on the energy values.

The uncertainty of the temperature change coefficient of the ventilation heat recovery Φ_{oa} results from the uncertainty of the measurement and presumed leakages with the rules of the Gaussian error propagation. The leakages are presumed with 3% on the one hand between outside air and exhaust air and on the other hand between return air and supply air because of the two suction fans.

Tab. 38: Uncertainties of results

symbol	value	accuracy of measurement	relative accuracy of measurement
SPF-HP	3.8	0.3	7%
SPF-G	3.6	0.3	7%
SPF-SYS	3.1	0.1	4%
Φ_{oa}	0.77	0.03	4%
$Q_{H,dis,in}$	10'894 kWh	327 kWh	3%
$Q_{W,dis,in}$	1'179 kWh	35 kWh	3%
$Q_{H,ve,out}$	1'120 kWh	41 kWh	4%
$E_{cu,in}$	4'434 kWh	44 kWh	1%
$Q_{H,hp}$	2'252 kWh	137 kWh	6%
$Q_{H,ve,pre}$	671 kWh	94 kWh	14%
$E_{HW,hp,cmp,in}$	3'335 kWh	33 kWh	1%
$E_{hp,fan,in}$	279 kWh	7 kWh	2%
$E_{HW,bu,in}$	199 kWh	12 kWh	6%
$E_{HWV,cu,ctr,in}$	175 kWh	2 kWh	1%

6.1.3.1 *Winter and summer season*

Winter period

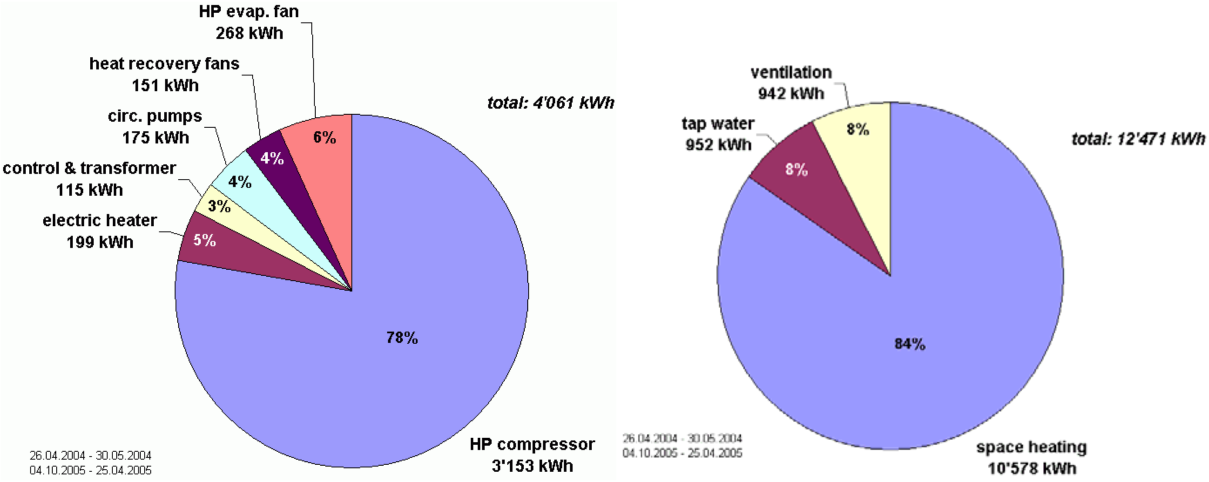


Fig. 58: left part: shares of electricity demand in winter 2004/05
right part: shares of net heat demand in winter 2004/05

Fig. 58 left part depicts that the heat pump with the evaporator fan uses 84% of the electricity consumption in wintertime. The electric heater consumes 5%, heat recovery fans 4%, control and transformer 3% and circulation pumps 5%. Fig. 58 right part shows still a low domestic hot water (DHW) energy consumption of 952 kWh for the winter period, although it is a bit higher compared to summer period. The space heating (SH) energy requirement of 10'578 kWh is comparatively high.

Summer period

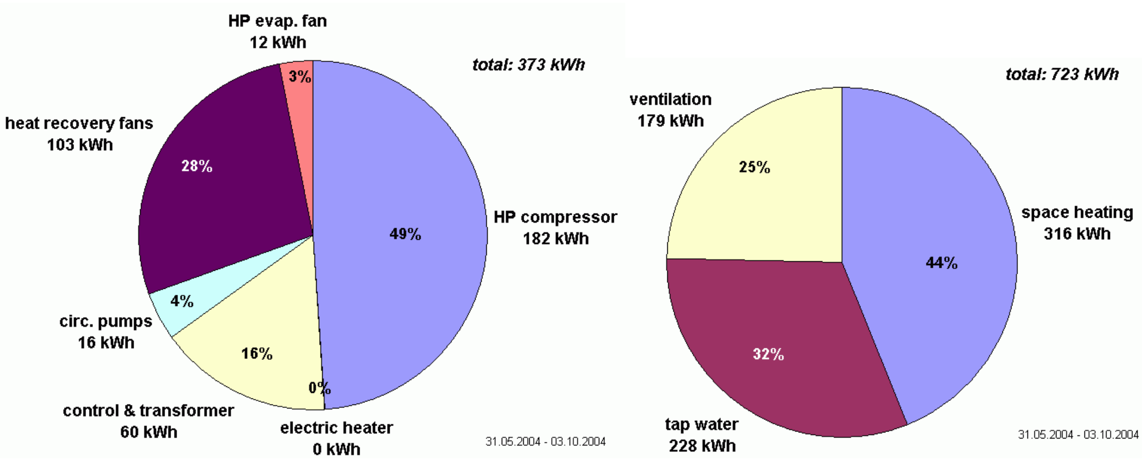


Fig. 59: left part: shares of electricity demand in summer 2004
right part: shares of net heat demand in summer 2004

Fig. 59 left part shows that the heat pump uses 52% of the electricity demand in summer.

6.1.3.2 *Seasonal*

Ventilation and heat recovery consume 28%, control and transformer 16% and circulation pumps 4%. Fig. 59 right part shows for the summer period a very low domestic hot water (DHW) energy consumption of 228 kWh (acc. SIA 380/1: 708 kWh) and an unexpected high space heating (SH) energy demand of 316 kWh, that is even higher than the DHW energy demand.

From a first view the share of the heat pump seems rather low and the consumption for control, transformer, fans and circulating pumps rather high, but the small total energy consumption for domestic hot water of 228 kWh in 18 weeks corresponds to the small energy consumption of the heat pump.

The control of the compact unit and the losses of the transformer used for control and ventilation fans consume 60 kWh in 18 weeks or a permanent 20 W which is a common value. The fans for the heat recovery (HRU), which run also during summer, consume 103 kWh. It has to be taken into consideration that the HRU was not active during 2 of 18 weeks. Thus the average power consumption during operation is 38 W for the measured 60 m^3/h. The specific power consumption for the ventilation is in this case 0.6 $W/(m^3/h)$. The electricity consumption of the two circulating pumps for SH and the DHW storage heating is quite small compared to their nominal power of 90 W for SH pump and 55 W for DHW pump due to the intermittent operation of both pumps yielding short running time.

The right part of Fig. 59 shows an unexpected result in regard to the net heat demand during summer period. The net heat demand for SH during summer was higher than for DHW. 134 kWh of the total 316 kWh SH heat supply would correspond to the measured heating degree days as net heat demand. For the remaining 182 kWh the explanation was specially analyzed and is described in chapter 6.1.4.

6.1.3.3 Typical days

The operation characteristic of the measured and evaluated compact unit, LWZ 303 SOL of manufacturer Stiebel-Eltron, will be described with by following figures and analyses for the topics domestic hot water production DHW (Fig. 60 & Fig. 61), space heating SH (Fig. 62), electricity consumption (Fig. 63 & Fig. 64) and ventilation heat recovery HRU (Fig. 65). The figures describe the behaviour of special selected days that represent a typical behaviour of the compact unit.

DHW

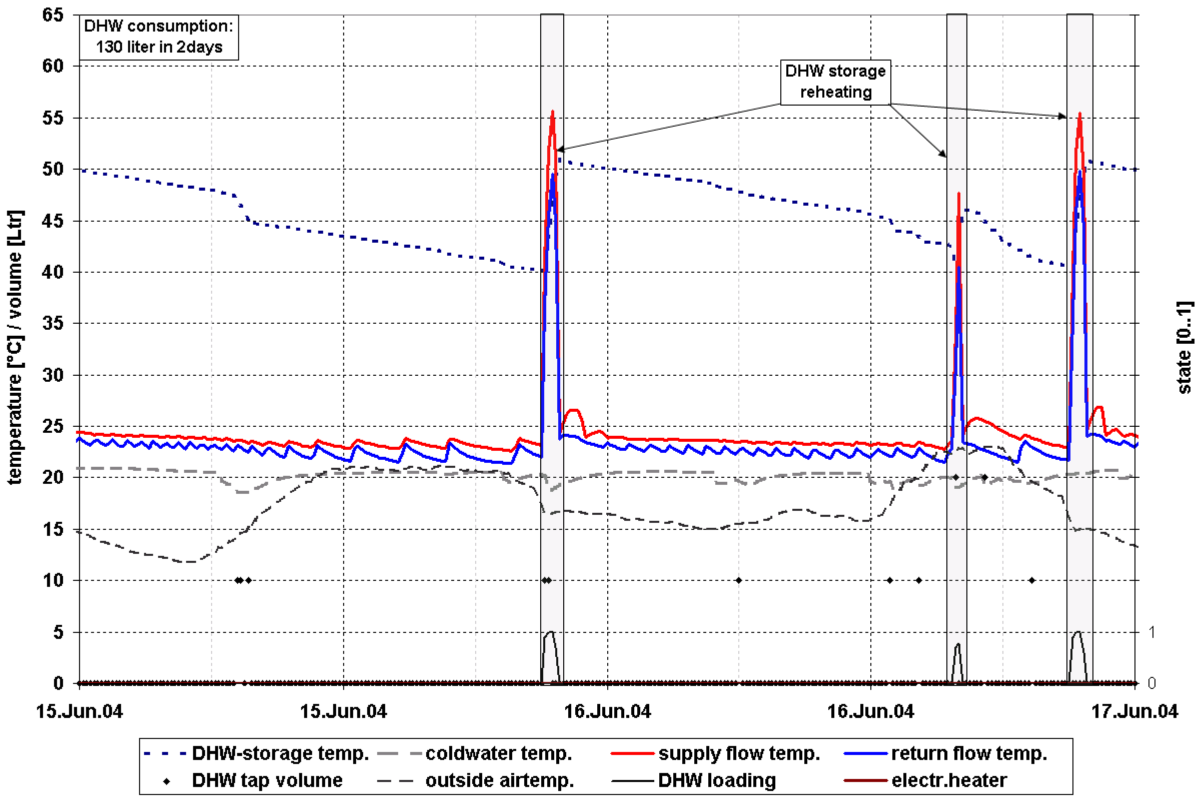


Fig. 60: Operating characteristic of domestic hot water production in summer time

The DHW storage reheating shown in Fig. 60 takes typically place one to three times a day. Every evening at 22:00 the storage is heated up to the set DHW temperature. The set temperature during the day can herein differ from the set temperature at 22:00. In standard setting the set hot water temperature during the day is 42°C and at 22:00 48°C. With this setting the cheaper electricity in night time shall be used to heat up the DHW storage. The home owners of the field object were completely satisfied with this setting. They recognized only a quite long time in the morning to get the first hot water at the tap. The average duration of DHW storage reheating is 15...35 minutes.

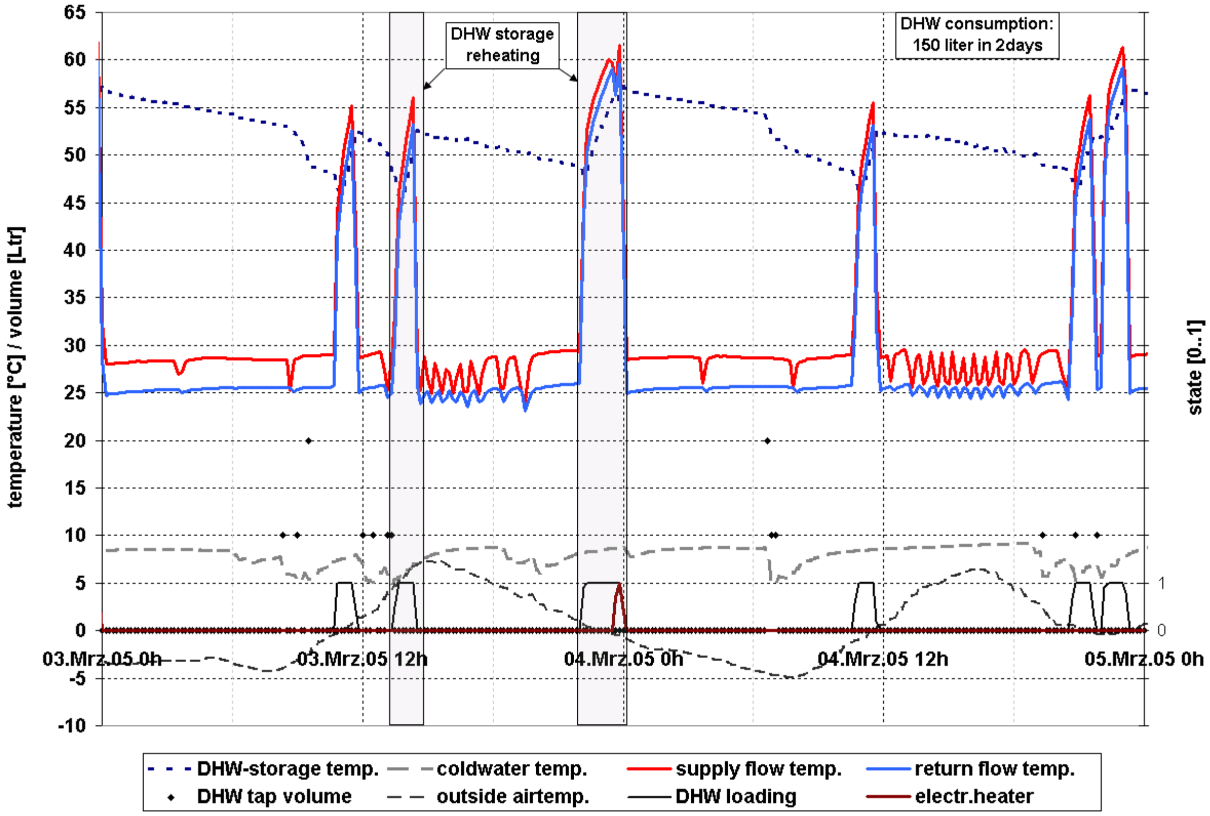


Fig. 61: Operating characteristic of domestic hot water production in winter time

The set hot water temperature was raised during the measurement period to get comparable conditions with other heat pump systems. It is set to 50°C during daytime and to the maximum possible 55°C for the 22:00 storage heating. In wintertime (s. Fig. 61) also the DHW storage reheating takes place 1...3 times a day. The average duration of DHW storage reheating is with 45...120 minutes significantly longer because of the decreasing heat pump power at lower outside air temperatures. If the DHW storage reheating takes more than 90 minutes then the electric heater supports the storage heating.

Space heating

The space heating circulation pump has an intermittent operation characteristic. The start frequency depends on the outside air temperature. At a maximum temperature the pump starts at least once a day and at a minimum temperature it operates continuously. In between the frequency of starts per day is linear interpolated. This reduces the electricity consumption of the space heating pump remarkably.

The three highlightings in Fig. 62 depict the operation mode of the heat pump for space heating at winter conditions. At outside air temperatures above 0°C the heat pump works intermittent to meet the heat demand of the building. Between 0°C and -5°C the heat pump can keep the room air temperature at the desired level. Below -5°C the power of the heat pump is too small and the electric heater supports the heat demand. Fig. 62 does not represent the typical space heating operation in case of the electric heating element because only 1% of the heat demand of the space heating is covered by the electric heating element; the main part is covered by the heat pump.

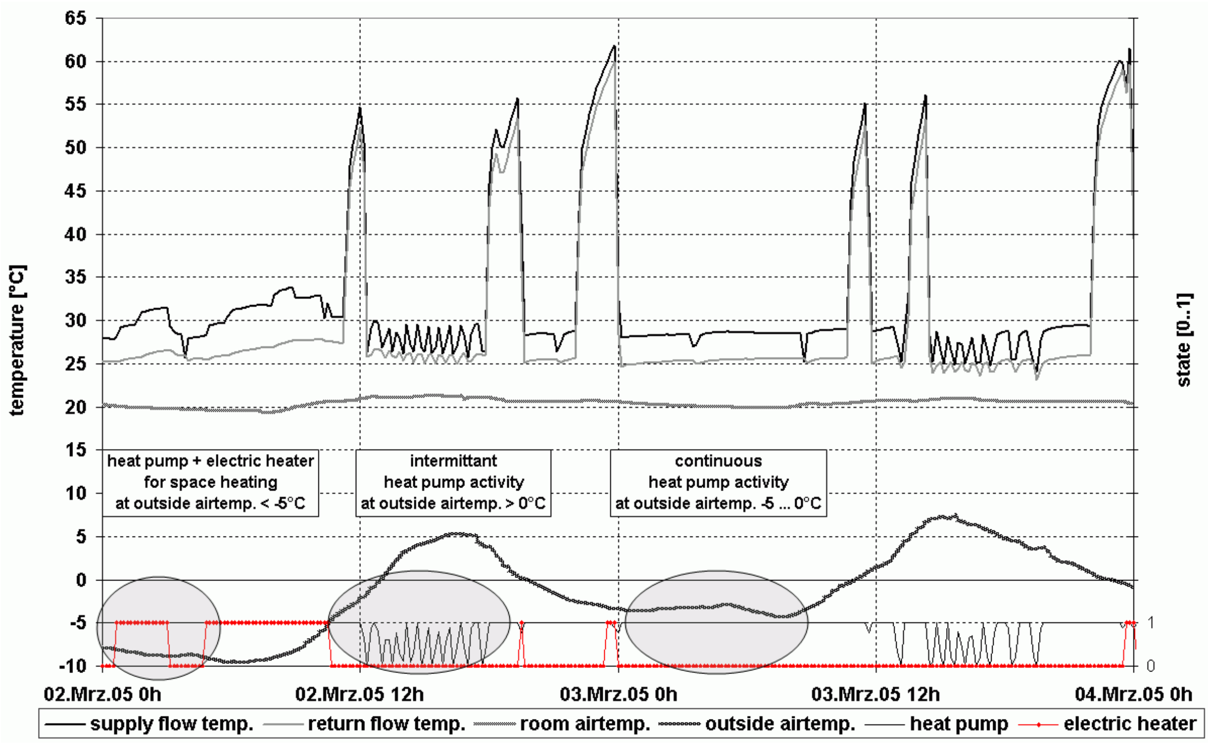


Fig. 62: Operating characteristic of water heating system in winter time

Electricity demand

The electricity demand of the compact unit consists in summertime (s. Fig. 63) predominantly of the basic demand which lies between 60...100 W and only few times a day of the consumption of the heat pump with 1.1...1.4 kW. The basic demand contains 10 W for control, 10 W losses of the transformer, about 36 W for ventilation and the average demand of the intermittent operation for the circulating pumps where the domestic hot water loading pump has 33 W during operation and the space heating circulation pump 54 W.

In wintertime the basic electricity demand is with 70...110 W higher because of a more frequent activity of the circulating pumps. The heat pump comprises the main electricity consumption. The operating characteristic of the electric heater can be seen in Fig. 64. For space heating the electric element operates in 3 stages ELH1 with 1.4 kW, ELH2 with 2.8 kW and both together with 4.2 kW. For domestic hot water production the two elements are always used in parallel.

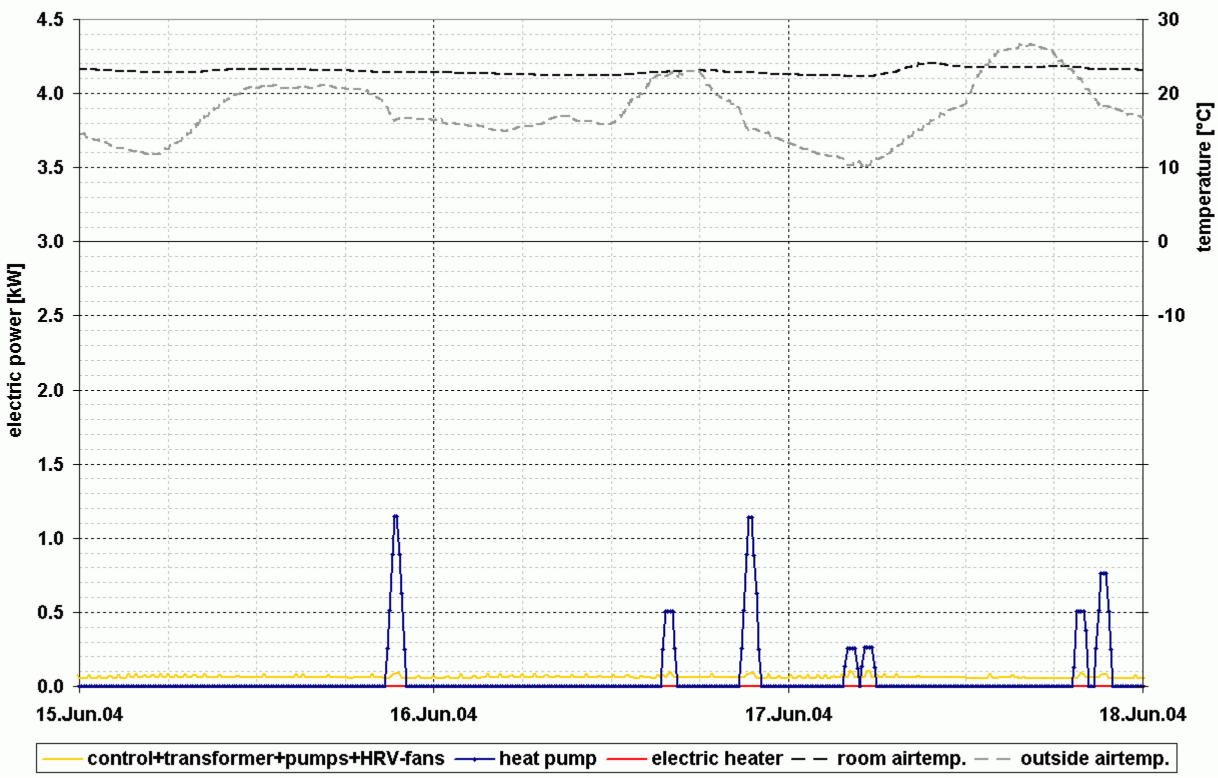


Fig. 63: Electricity demand of compact unit in summer time

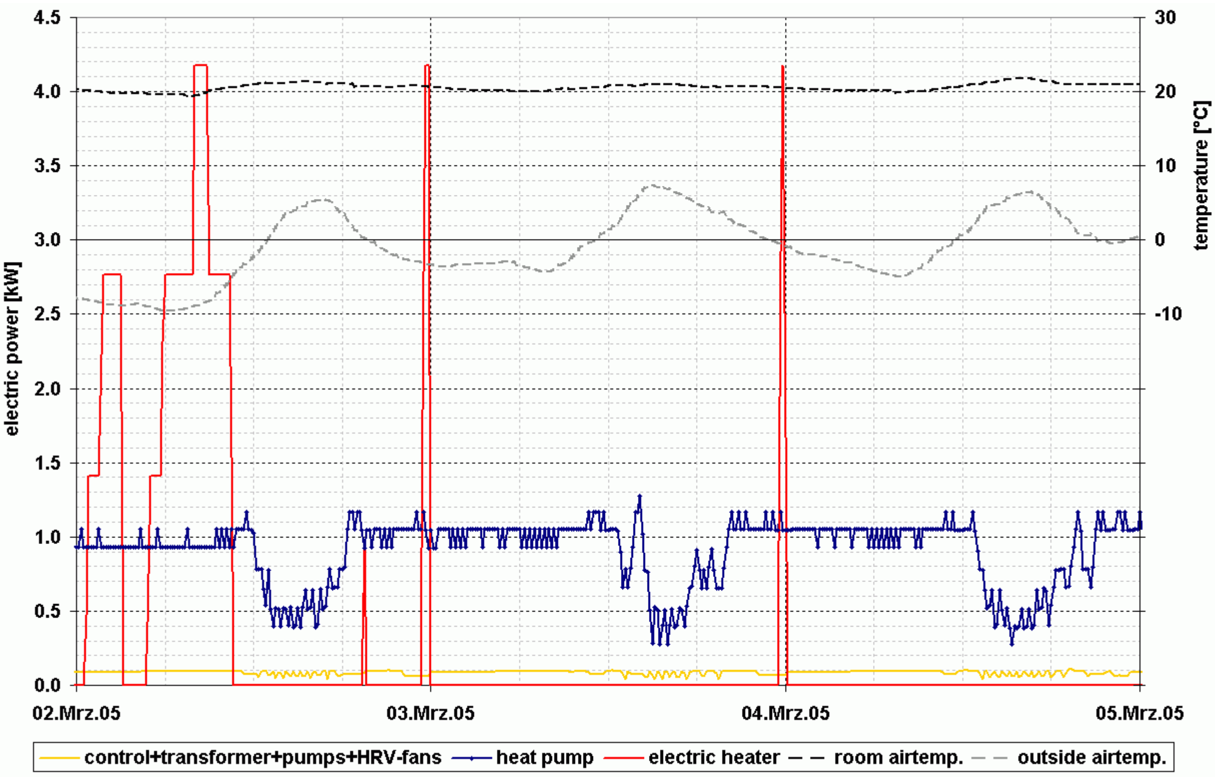


Fig. 64: Electricity demand of compact unit in wintertime

Ventilation heat recovery

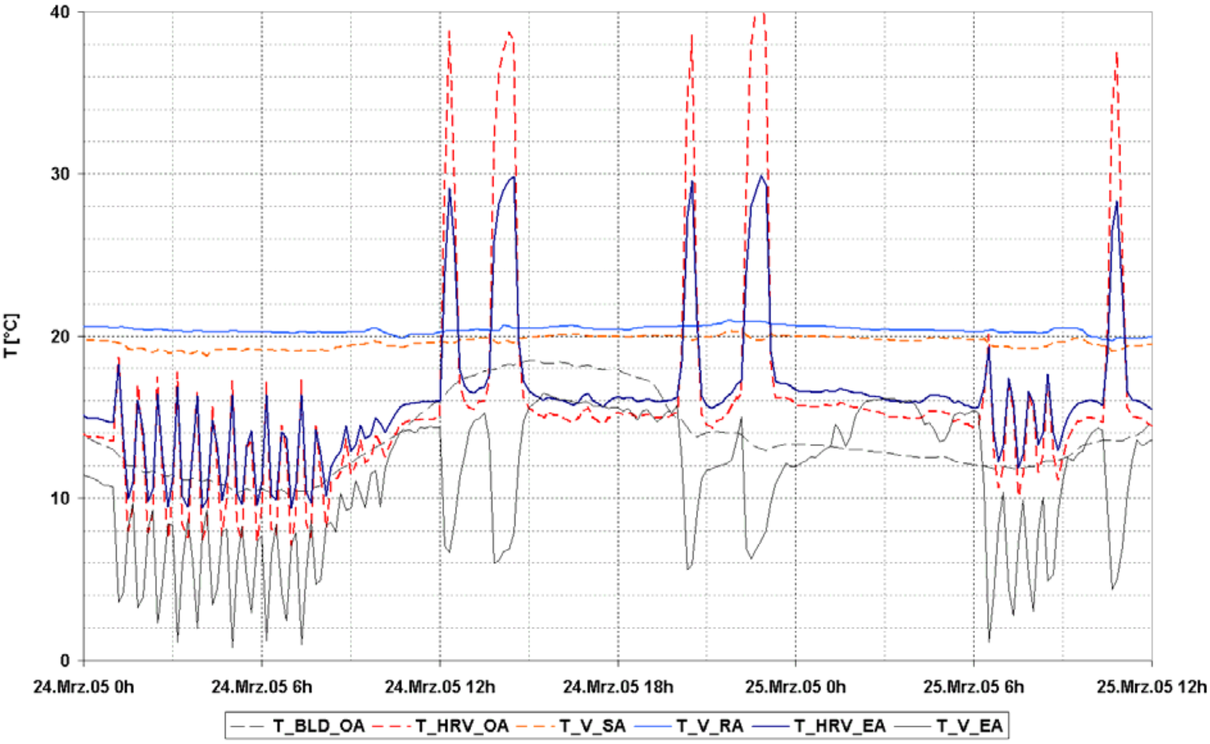


Fig. 65: Air flow temperatures at the ventilation heat recovery

A highly dynamic behaviour in the air temperatures at the ventilation heat recovery HRU of the compact unit is depicted in Fig. 65. On its way to the room the outside air passes first the subcooler of the heat pump and then the ventilation heat recovery. The return air passes first the HRU, then a possible mixture with outside air and at last the evaporator of the heat pump. When the heat pump is active, the return air is mixed with outside air before the evaporator and cooled down in the evaporator for energy gain in the heat pump and the outside air is preheated by the subcooler. Because of the effect of the subcooler the outside air temperature before the HRU can be lower and also higher than the return air temperature of the building. In the HRU the outside air and the return air adjust their temperatures with both ways of energy transfer, heating the outside air and also recooling the outside air. Thus at every start or stop of the heat pump the conditions of the HRU change in a wide range, especially for DHW production at high temperature level. In Fig. 65 the outside air temperature before the HRU reaches temperatures up to 40°C.

The key values for the evaluation of the heat recovery ETV_{hru} and Φ_{oa} can only be calculated in times of nearly constant conditions, which mean that the heat pump is not running and was not running for at least 15 minutes. With more heat pump activity at lower outside air temperatures the amount of periods with constant conditions decreases, thus for outside air temperatures below about 10°C almost no evaluation is possible. Fig. 66 shows the weekly average values of the periods with constant conditions.

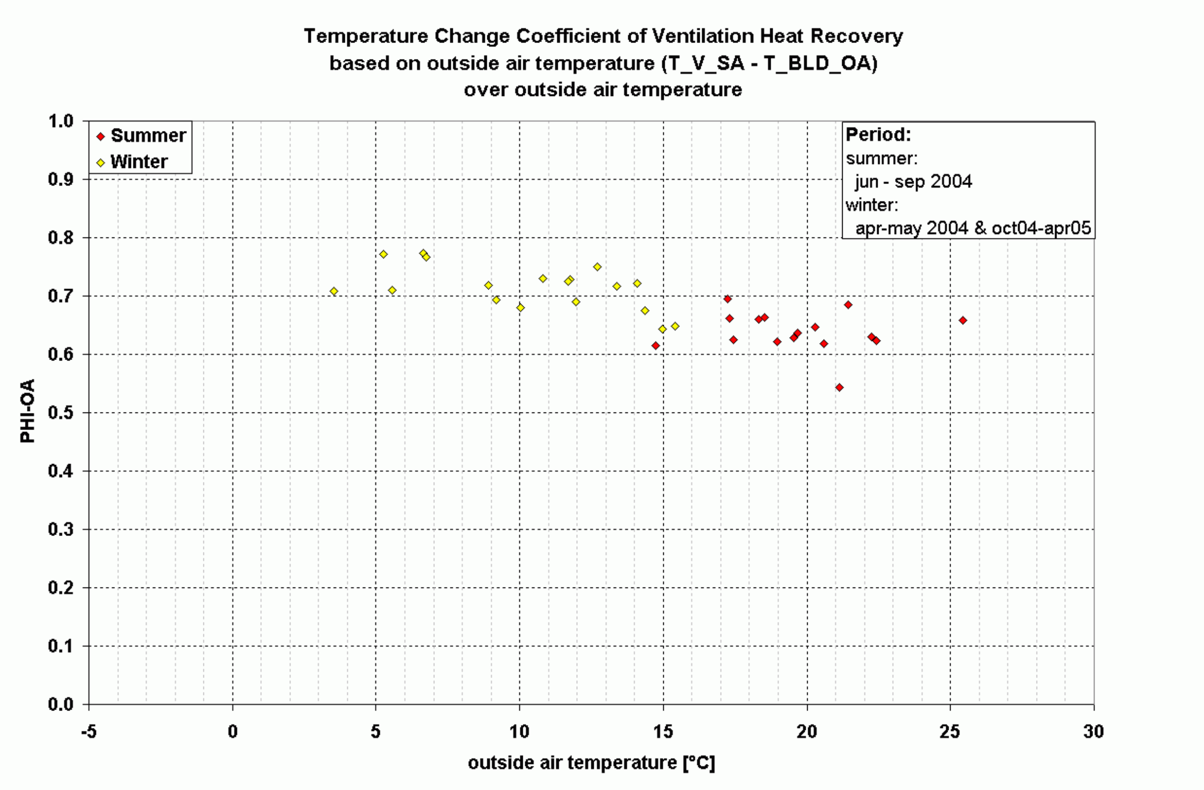


Fig. 66: Temperature change coefficient of ventilation heat recovery based on outside air temperature

6.1.3.4 General

Stand-by losses of storage

With the data of calendar week 40, when the building was unoccupied, the stand-by losses of the storage and the coefficient of performance of the heat pump (COP-HP) in DHW stand-by modus could be evaluated. Every evening at 10.00 p.m. the compact unit starts with DHW operation and heats the storage up to the set value, i.e. 45°C until end of November. There was no DHW extraction. The daily energy delivered to the storage ranges between 1.4 kWh/d and 1.9 kWh/d with an average of 1.7 kWh/d. Hence the average heat loss is 70 W. The ambient temperature in the technical room, where the unit is installed, ranges from 17.2°C to 19.6°C with an average 18.6°C. Temperature in the storage at the control sensor is between 41.3°C and 47.9°C with an average 44.2°C. The average temperature difference between storage and room is 25.6 K and the specific heat loss of the storage is 2.7 W/K. The COP-HP of the heat pump for DHW storage during stand-by operation ranges between 1.9 and 3.0 with an average of 2.4.

Working conditions of electric heater

Tab. 39 shows that the electric heater is active for space heating only at outside air temperatures between -5.1°C (balance point) and the minimum outside air temperature of -11.3°C. For domestic hot water production the electric heating element is active at outside air temperatures below 5°C. In the range between 0°C and 5°C the electric heating element is only active if at the same time a regular storage heating and hot tap water usage of more than 40 liters take place. Below 0°C the heat pump power is too small to heat up the storage in less than 90 minutes, the set time limit for heat pump single operation.

Tab. 39: Working conditions and energy demand of electric heater

	electric heater energy demand			outside air temperature when active		
	ELH1 (1.4 kW)	ELH2 (2.8 kW)	share of net heat demand	maximum	average	minimum
SH space heating	34.0 kWh	85.7 kWh	1%	-5.1°C	-8.2°C	-11.3°C
DHW domestic hot water	26.4 kWh	52.8 kWh	7%	5.1°C	-3.0°C	-7.8°C

Effect of subcooler and remaining potential for ventilation heat recovery

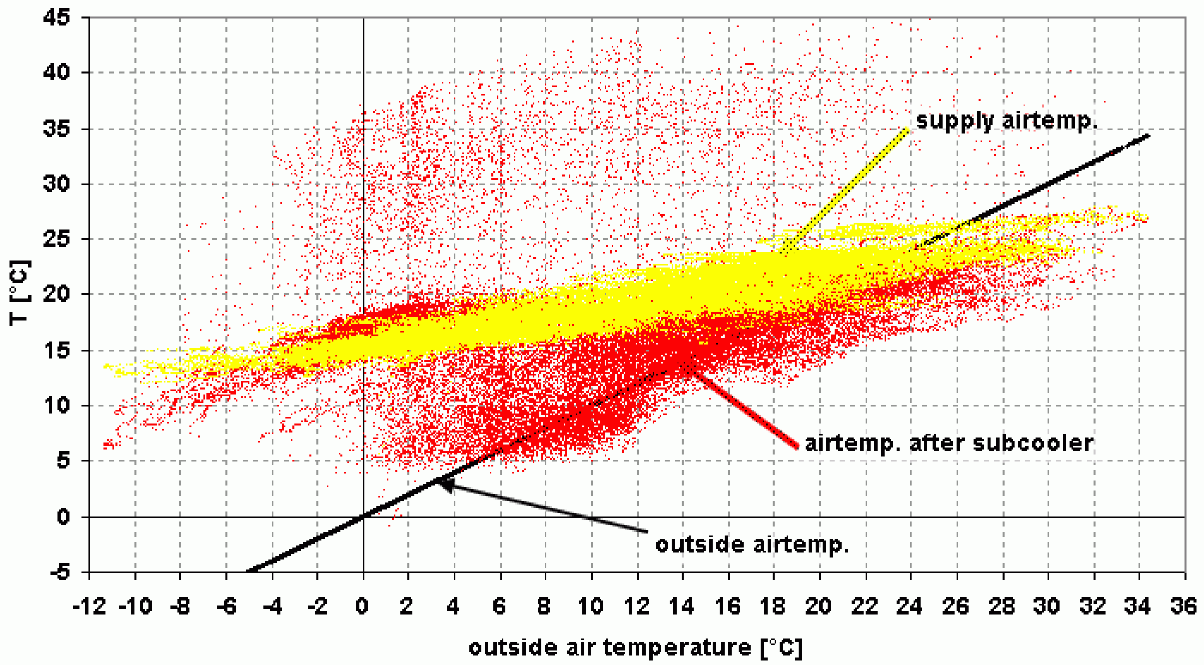


Fig. 67: Effect of subcooler for ventilation heat recovery

Fig. 67 shows the air temperature after the subcooler and furtheron the supply air temperature after the ventilation heat recovery HRU versus the outside air temperature. The subcooler assures a virtual outside air temperature for the HRU of at least 5°C. This works in this case with a supply air volume flow of average 50m^3/h in a range of 20...65 m^3/h down to an outside air temperature of -12°C. But even a bit under the upper limit for space heating of 14°C the subcooler has a remarkable influence on the ventilation temperatures. Between 6 and 14°C the subcooler on the one hand preheats the outside air mainly with 5...10 K, in maximum with 30 K. Basically the subcooler heats up the air flow if the heat pump is running. But capacitive effects in the subcooler and the HRU and an influence of the refrigerant also heat up the air flow without heat pump activity. This raises on the one hand the efficiency of the heat pump and prevents freezing in the HRU but on the other hand also reduces the energy gain in the HRU. The HRU comes especially at outside air temperatures between -5 and 6°C to the new function to reduce and to smooth the supply air temperature again. The smoothing of the supply air temperature in the HRU works according to Fig. 67 very good. There is no big variation in the supply air temperature.

Tab. 40: Influence of subcooler on outside air temperature difference over heat recovery

Φ_{oa} = 80%	**Outside air temp.**	**Air temp. after subcooler**	**Supply air temp.**	**Return air temp.**	**ΔT-HRV**
without subcooler	-5°C	-5°C	12°C	16°C	17K
only frost protection		0°C	13°C		13K
field object subcooler		12°C	15°C		3K

Tab. 40 shows three examples for the situation of the HRU in combination with an HP-subcooler where the outside air temperature and the return air temperature are identical for the three cases. The outside air temperature is chosen for a cold winter day. The return air temperature is the respective measured temperature at the entrance of the compact unit.

Without a subcooler the HRU would see the outside air temperature of -5°C and reach a supply air temperature of 12°C. The temperature difference over the HRU accounts for 17 K thus the heat gain is the highest but there will be defrost cycles.

With a subcooler achieving only frost protection the supply air temperature is with 13°C 1 K higher but the temperature difference over the HRU is only 13 K because of the preheating effect. This situation would be the case with the highest supply air volume flow of 170 m^3/h.

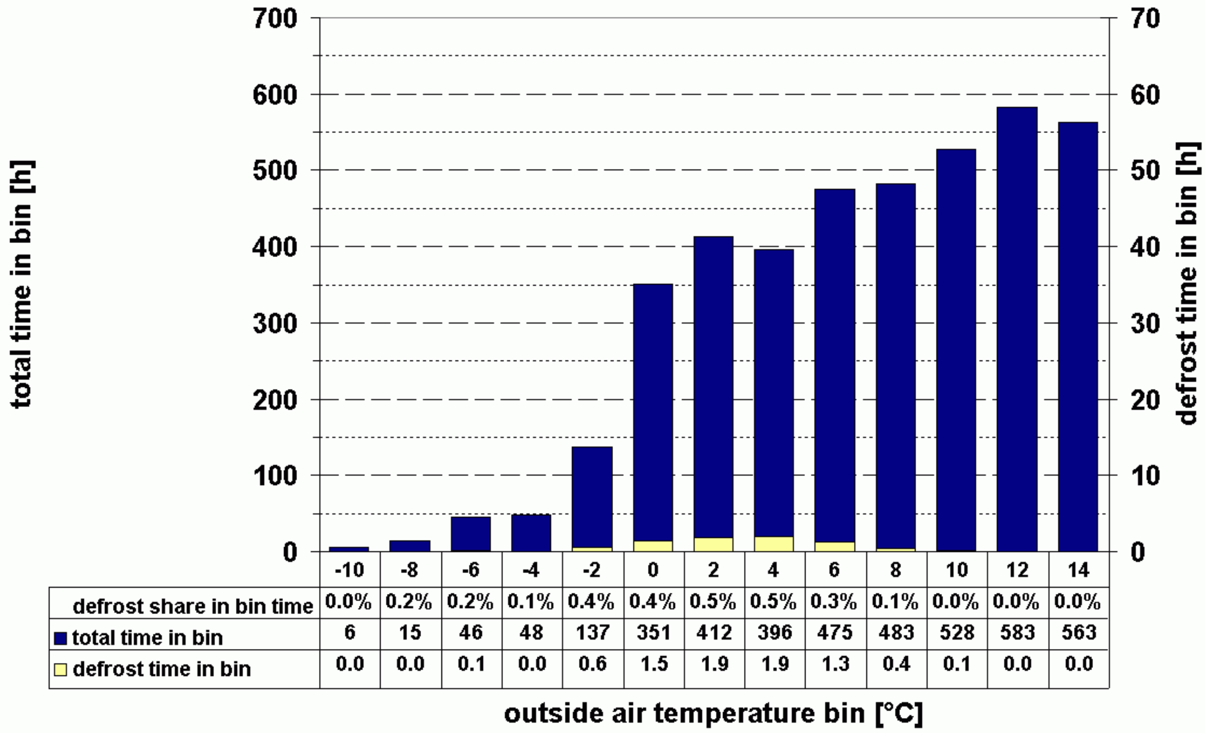
In the case of the field measurement now the outside air reaches 12°C after the subcooler / before the HRU. The supply air temperature is now the highest of these three cases with 15 °C but the heat gain in the HRU is with only 3 K the smallest.

These three examples show that on the one hand the subcooler prevents the HRU from frost but on the other hand reduces the heat gain in the HRU in the case of a small supply air volume flow. The total effect has to be assessed as a sound method to keep the HRU frost free in a cost and primary energy efficient way.

The return air temperature in Tab. 40 is the respective measured temperature at the entrance of the compact unit for an outside air temperature of -5 °C. The room air temperature at these conditions was measured with 20 °C which shows that over the ductwork 4 K are lost. At outside air temperatures around 15 °C and above room air temperature and return air temperature at compact unit are identical. In between the temperature difference over the ductwork rises with falling outside air temperatures.

Defrosting of heat pump evaporator

The evaluation of the defrost cycles of the heat pump evaporator show 386 defrost cycles in the evaluated year of average 1.5 minutes and maximum 4 minutes duration. Fig. 68 shows the defrost frequency related to outside air temperature bins. A remarkable fact is that the defrost frequency has it maximum a few Kelvin over 0°C and is relatively small a few Kelvin below 0°C.



	-10	-8	-6	-4	-2	0	2	4	6	8	10	12	14
defrost share in bin time	0.0%	0.2%	0.2%	0.1%	0.4%	0.4%	0.5%	0.5%	0.3%	0.1%	0.0%	0.0%	0.0%
total time in bin	6	15	46	48	137	351	412	396	475	483	528	583	563
defrost time in bin	0.0	0.0	0.1	0.0	0.6	1.5	1.9	1.9	1.3	0.4	0.1	0.0	0.0

Fig. 68 : Defrost time of HP-evaporator related to outside air temperature bins and bin time

Effect of heat pipe and working conditions

The heat pipe was active in 40 periods or 6h 45 min in the whole year at outside air temperatures between 0°C and -10°C. When the heat pipe function was active, always the compressor was running at the same time. The influence on the air temperature after the subcooler in this time is negligible.

6.1.4 Recommendations for further improvement

Energy supply to space heating in summer time

There are two reasons for the high space heating energy supply during summer period. On the one hand the space heating starts sometimes if the outside air temperature drops for a time of about three hours below the space heating limit of 14°C even if the room temperature is high enough. This could be avoided by expanding the damping of the outside air temperature measurement (parameter P77 in the control manual of the compact unit) from standard value 1h to a value that corresponds to the thermal time constant of the building. For the field object Gelterkinden we suggest 12h as parameter setting. On the other hand the measurement data show always an energy flow into the building as soon as the space heating circulating pump starts even without heat pump operation as shown in Fig. 69. The temperature difference between supply and return pipe rises to about 2 K during the off-period of the circulation pump. After a circulation pump running time of about 10 minutes and without heat pump operation, the temperature difference falls to 0.1…0.2 K and remains with this offset.

The remarkable fact is that there exists a temperature difference up to 2 K without heat pump activity.

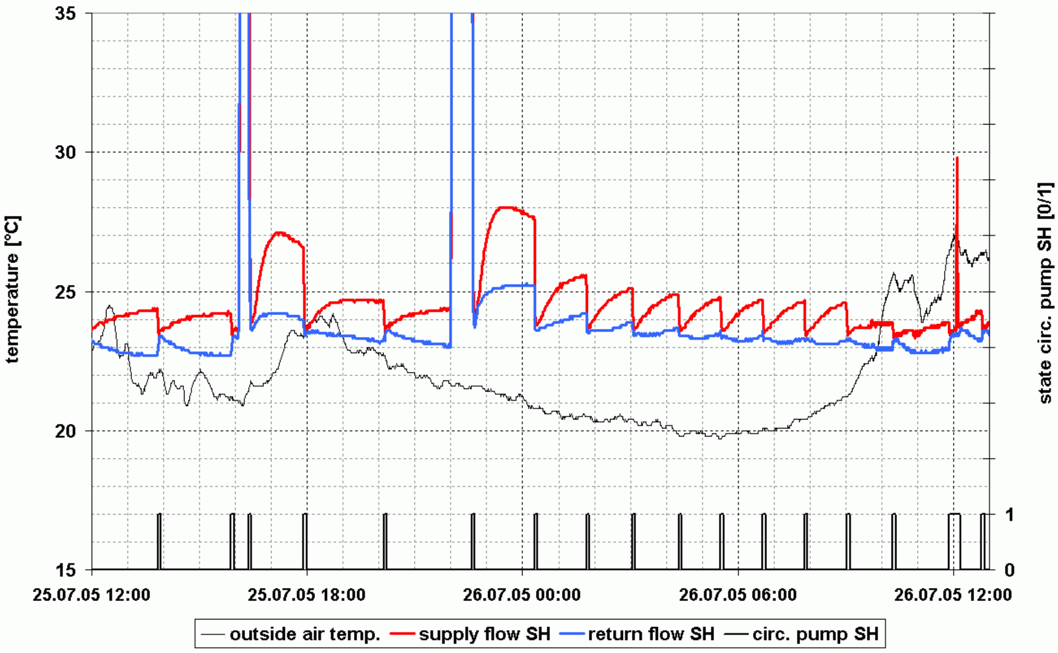


Fig. 69: space heating flow temperature characteristic at summerly SH heat supply

Default parameter sets

Additionally to the good control manual [47] and the valuable hints for an optimized operation in the manufacturer description [46] we suggest integrating the following hints:

1) The damping of the outside air temperature (P77) depends on the thermal mass and the related time constant of the building. For a light building set P77 = 1h, for a heavy building set P77 = 12h.

2) The hysteresis of heating circuit desired flow temperature (P21) should be set in reference to the capacity of the heat emission system and the controlled flow temperature (supply- or return flow). A radiator system for example could work with a greater hysteresis of 4 K like in the default parameter setting and a floor heating system usually allows a smaller hysteresis of 1..2 K.

6.2 Pilot plant Zeiningen

6.2.1 Description of the pilot plant

Tab. 41: Building specification and planning data

Plant concept	compact heating centre with: - compact unit with heat pump - hydraulic heat distribution - mechanical ventilation with heat recovery unit and pre-heating with heat pump - 250 litre hot-water tank
Building	detached house built to passive-house standard (MINERGIE-P certified)
Construction details	mixed construction: floor tiles, lower-floor ceilings and interior walls concrete, outer walls light construction Storeroom and work room outside insulated area
Location	4314 Zeiningen (BL)
Altitude	373 m above sea level
Heated living area	204 m^2
Heating energy requirements with MINERGIE-P standard values	52 MJ/m^2
Effective heating energy requirements	47 MJ/m^2
Max. specific heat output requirements (MINERGIE-P)	12.86 W/m^2
Air-water heat pump (manufacturer's specifications):	
Nominal power	1.5kW
Power delivery at L2/W35°C	2.6 kW
Auxiliary heating elements (3-stage) (measured values):	1.8 / 3.4 / 5.2 kW
Ventilation:	
Volume flow - adjustable range	70 to 250 m^3/h
- pre-set value	100 m^3/h
Heat recovery (measured value)	> 80%
Power consumption:	
Controls (devices on standby)	16 W
Fans	24 W at 100 m^3/h 40 W at 150 m^3/h 66 W at 200 m^3/h
Internal pump (Level 2) (see technical description)	68 W
External pump (floor-heating pump, Biral M10)	20 W

After a long search for a suitable object for field trials with the Viessmann Vitotres 343 compact heating centre, a choice of two homes belonging to the general contracting firm, Genesis Home AG became available. The project group decided in favour of the passive house in Zeiningen, constructed according to the standard Genesis concept for passive houses. This detached single family house is certified according to the Swiss MINERGIE-P® standard (MINERGIE 2005).
The measurement instrumentation could be started up in September, 2004. Initially, difficulties transferring data from the data logger to the PC via telephone lines repeatedly resulted in loss of data. From the end of October on, the data capture system functioned faultlessly. In a first phase a number of control parameters were optimized with regard to the house. A new control module, which takes not only the reference zone temperature but also the outside air temperature into account, was installed at the end of November. Validated data were obtained from December, 2004 on, and the measurements were concluded at the end of June, 2005.

Fig. 70: Pilot plant, Zeiningen

6.2.2 Description of the pilot plant

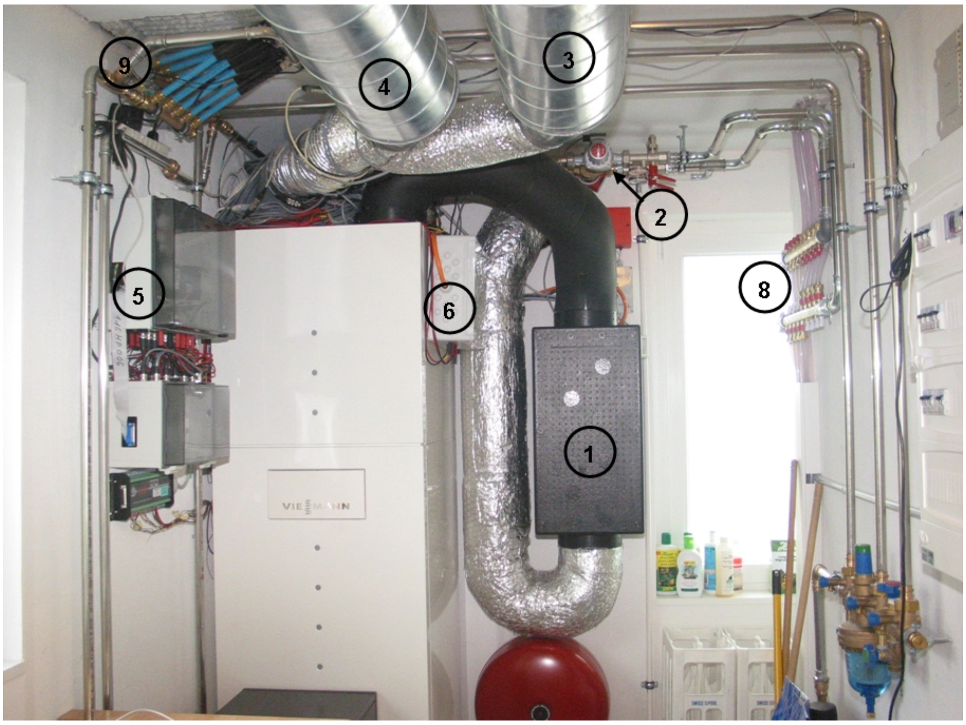


Fig. 71: Vitotres 343 in Zeiningen, (1) outside-air filter, (2) underfloor-heating pump, (3) supply air, (4) return air, (5) data logger, (6) electric power meter (for compact unit), (7) underfloor-heating manifold, (9) hot and cold water manifold

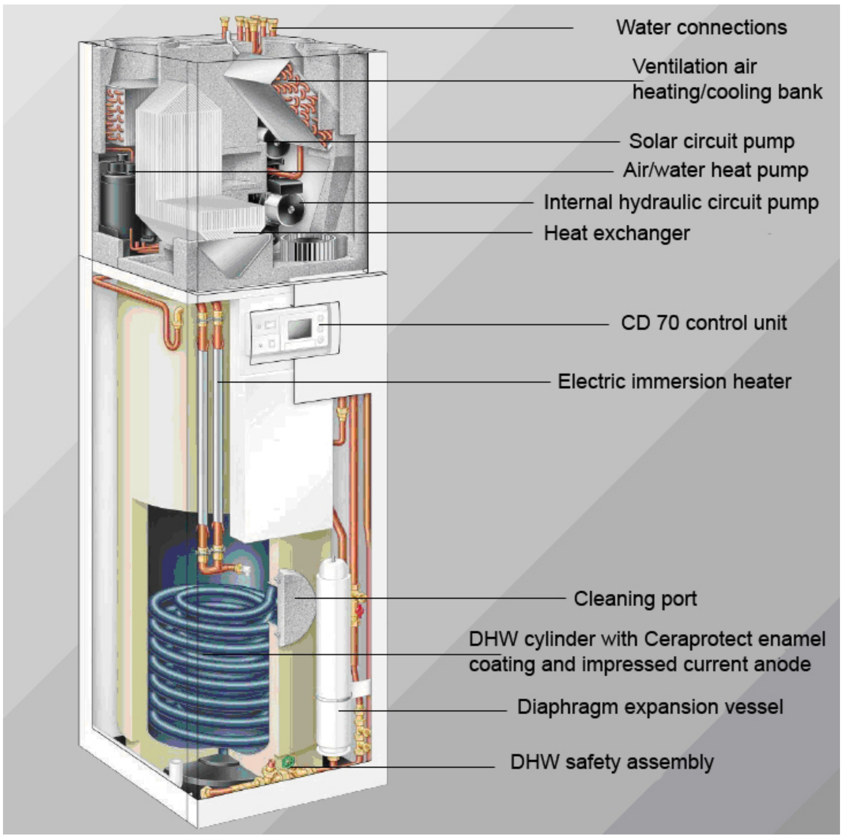
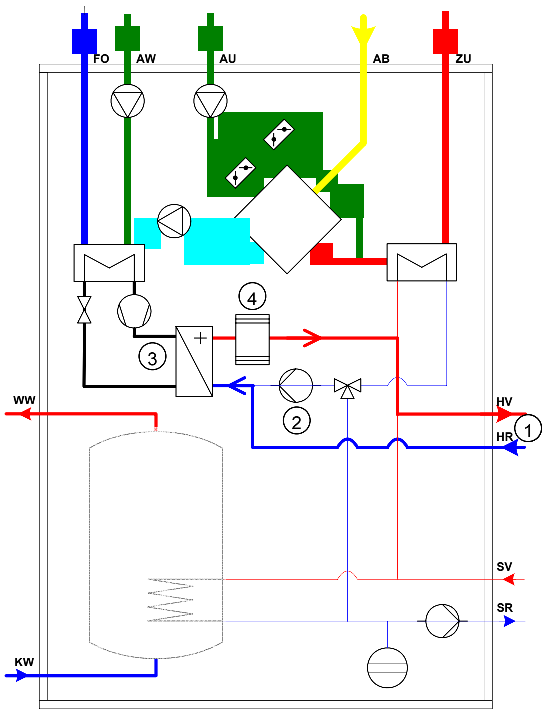


Fig. 72 Cutaway diagram of compact unit Vitotres 343 [Viessmann 9446 895 D 03/2003]

6.2.3 Operating principle

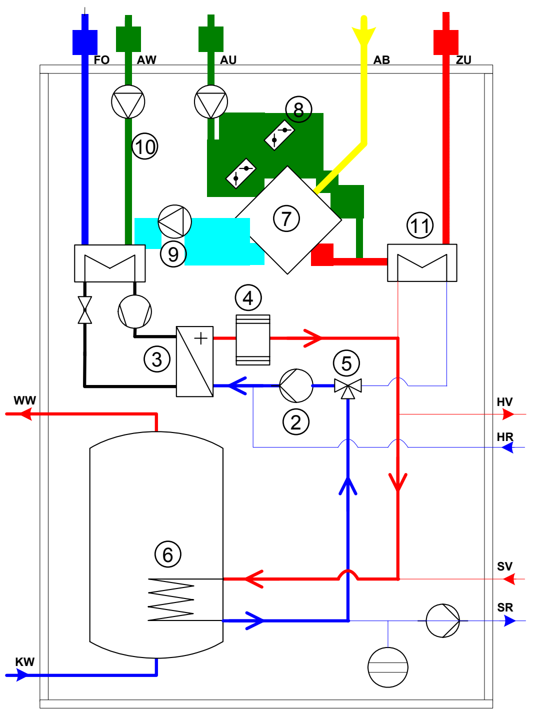
6.2.3.1 Hydraulic heating and ventilation



The external circuit (underfloor heating) is connected to the heating supply and return (1) of the unit. In heating mode the controller starts the underfloor-heating pump (external to the unit). Simultaneously the internal pump (2) is switched off. The carrier medium (Tyfocor LS) is heated by the heat-pump (3). If the output of the HP is insufficient, the electrical heating (4) is switched on as a supplementary source.

Fig. 73: Schematic diagram of hydraulic heating and ventilation

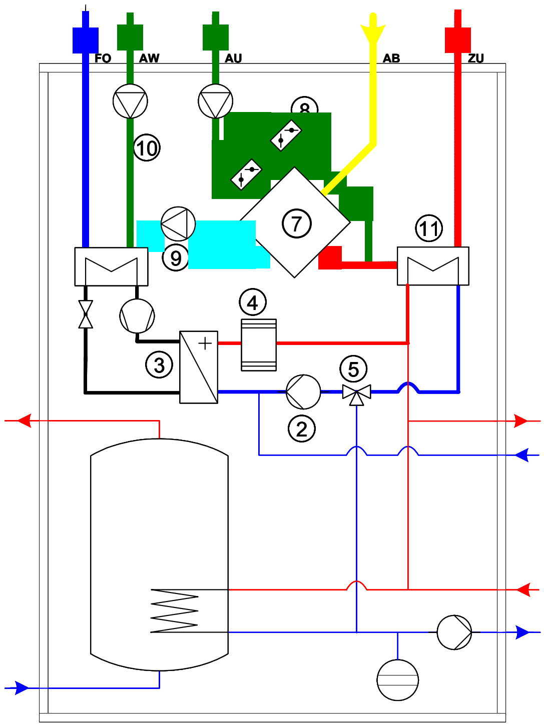
6.2.3.2 Domestic hot water and ventilation



The control scheme ensures that domestic hot-water heating has priority over space heating. The requirement to heat the domestic water is regulated by the temperature of the storage tank. For domestic hot-water heating the internal pump (2) is switched on and the external pump (for the underfloor heating) switched off. The three-way bypass valve (5) delivers the carrier medium from the heat pump (3) to the storage tank and the supply temperature is raised to the required value. In the event of increased demand, the electrical heating (4) can be used to heat the domestic water.

Fig. 74: Schematic diagram of domestic hot-water heating and ventilation

6.2.3.3 *Air heating and ventilation*



The compact heating centre includes a central domestic ventilation unit with heat recovery (7). If the house temperatures are very high, the heat recovery unit (HRU) is circumvented using the "summer bypass" (8). The heat pump (3) uses the residual heat of the exhaust air after the HRU (9). In order to attain the required volume flow at the evaporator, outside air (10) is added to the exhaust air. The supply air after the HRU can be heated via the supply-air heating coil (11).

Fig. 75: Schematic diagram of air heating and ventilation

6.2.4 Measurement details

6.2.4.1 *Measurement uncertainties*

The precisions of the various measured quantities are summarized in Tab. 42.

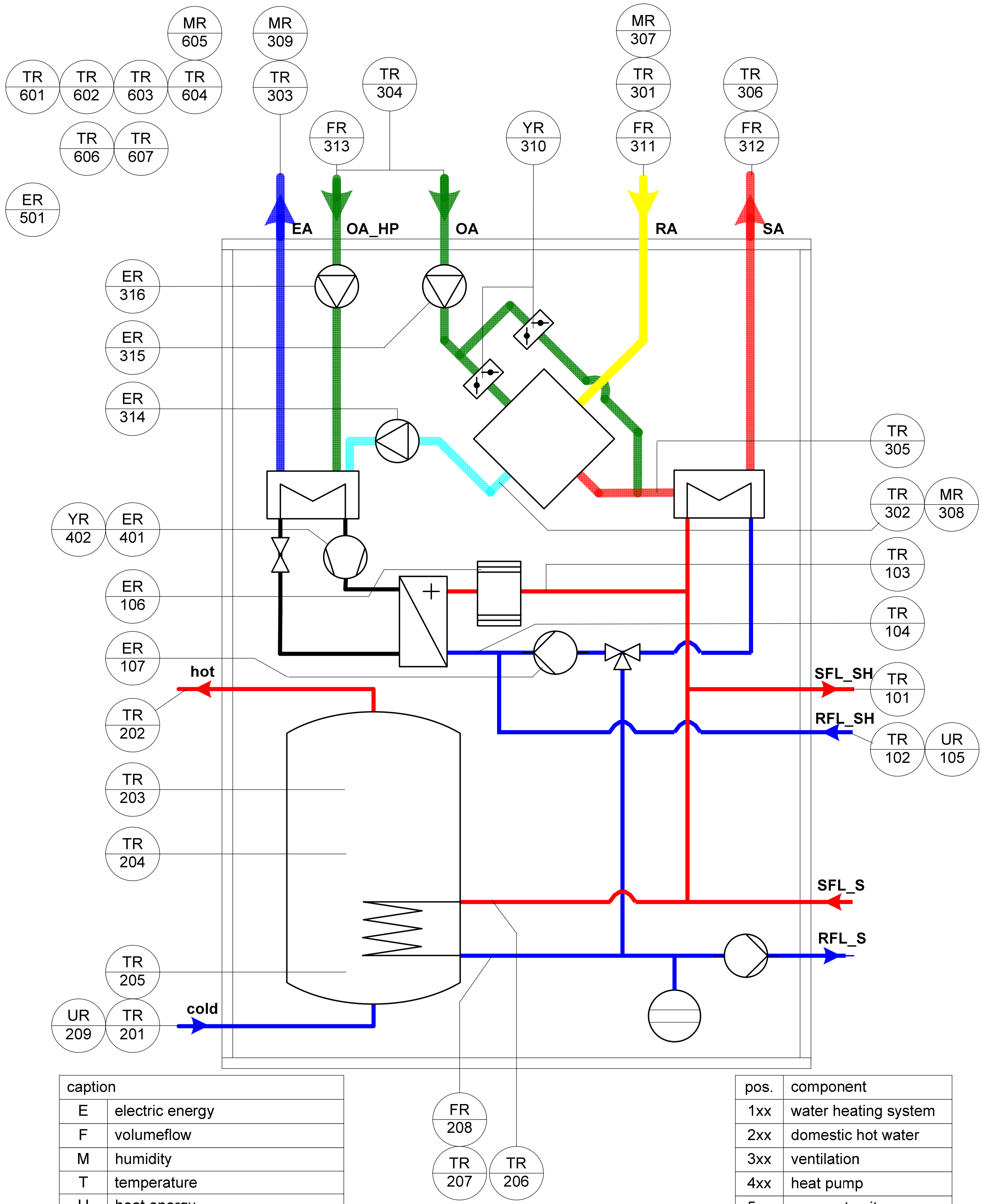
For the computed quantities, the uncertainties were calculated to the Gauss error propagation rule (mean errors). The uncertainties of the various parameters are given in chap. 6.2.6 in connection with the results. The remaining quantities in any case do not exceed the following values:

Tab. 42: Precision of measurement

heat output to domestic hot water	12%
net energy demand of ventilation	23%
heat output to air reheater [1)]	93%
enthalpy flow	8%

1) The output of the supply-air heating coil is calculated as the difference between the respective enthalpy currents of the supply air leaving the compact heating centre and the supply air leaving the heat recovery unit.

6.2.4.2 *Measurement scheme*



caption	
E	electric energy
F	volumeflow
M	humidity
T	temperature
U	heat energy
Y	state
R	recording (second letter)

pos.	component
1xx	water heating system
2xx	domestic hot water
3xx	ventilation
4xx	heat pump
5xx	compact unit
6xx	building

Fig. 76: Measurement scheme

6.2.4.3 Measurement values

Tab. 43: Designation of measurands

pos.	variables	unit	Description	principle	accuracy
101	$\theta_{H,f}$	°C	temperature space heating supply flow	PT100	± 0.5 K
102	$\theta_{H,r}$	°C	temperature space heating return flow	PT100	± 0.5 K
103	$\theta_{H,i,f}$	°C	temp. of internal hydraulic circuit supply flow from HP	PT100	± 0.5 K
104	$\theta_{H,i,r}$	°C	Temp. of internal hydraulic circuit return flow from HP	PT100	± 0.5 K
105	$Q_{H,dis,in}$	kWh	net energy demand space heating	heat meter	± 1.5%
106	E_{buin}	kWh	electric energy el. heating element	Relay[1)]	± 3%
107	$E_{H,pmp,sk}$	kWh	electric energy water heating system circulating pump	Relay[1)]	± 3%
201	$\theta_{W,cw}$	°C	temperature cold water	PT100	± 0.5 K
202	$\theta_{W,hw}$	°C	temperature hot water	PT100	± 0.5 K
203	$\theta_{W,st,\theta hlim}$	°C	temperature storage top	thermocouple	± 0.5 K
204	$\theta_{W,st,\theta avg}$	°C	temperature storage middle	thermocouple	± 0.5 K
205	$\theta_{W,st,\theta llim}$	°C	temperature storage bottom	thermocouple	± 0.5 K
206	$\theta_{W,i,f}$	°C	temp. of int.hydraulic circuit supply flow to tank	PT100	± 0.5 K
207	$\theta_{W,i,r}$	°C	temp. of int. hydraulic circuit return flow from DHW tank	PT100	± 0.5 K
208	$\dot{V}_{W}$		volume flow water heating system for domestic hot water	Relay[1)]	± 2%
209	$Q_{W,dis,in}$	kWh	heat quantity hot water	heat meter	± 1.5%
301	$\theta_{hru,ra,in}$	°C	temperature return air heat recovery ventilation inlet	PT100	± 0.5 K
302	$\theta_{hru,ra,out}$	°C	temperature return air heat recovery ventilation outlet	PT100	± 0.5 K
303	$\theta_{V,ea}$	°C	temperature exhaust air	PT100	± 0.5 K
304	$\theta_{hru,oa,in}$	°C	temperature outdoor air heat recovery ventilation inlet	PT100	± 0.5 K
305	$\theta_{hru,oa,out}$	°C	temperature outdoor air heat recovery ventilation outlet	PT100	± 0.5 K
306	$\theta_{V,sa}$	°C	temperature supply air	PT100	± 0.5 K
307	$\varphi_{hru,ra,in}$	%r	rel. humidity return air heat recovery ventilation inlet	humidity sensor	± 2.5%
308	$\varphi_{hru,ra,out}$	%	rel. humidity return air heat recovery ventilation outlet	humidity sensor	± 2.5%
309	$\varphi_{V,ea}$	%	rel. humidity exhaust air	humidity sensor	± 2.5%v
310	$Z_{hru,summer}$	0/1	state summer bypass valve	relay[1)]	± 1%
311	$\dot{V}_{V,ra}$	m^3/h	volume flow return air	anemometer	± 5%
312	$\dot{V}_{V,sa}$	m^3/h	volume flow supply air	anemometer	± 5%
313	$\dot{V}_{V,oa,hp}$	m^3/h	volume flow outside air for heat pump	relay[1)]	± 5%
314	$E_{V,ra,fan,in,}$	kWh	electric energy ventilator return air	relay[1)]	± 5%
315	$E_{V,oa,fan,in}$	kWh	electric energy ventilator outdoor air	relay[1)]	± 5%
316	$E_{hp,oa,fan,in}$	kWh	electric energy ventilator outdoor air for heat pump	relay[1)]	± 5%
401	$E_{hp,cmp}$	kWh	electric energy compressor	electricity meter	class 2
402	$t_{hp,op,cmp}$	h	running time compressor	relay[1)]	± 2%
501	$E_{cu,in}$	kWh	electric energy compact unit	electricity meter	class 2
601	θ_{tr}	°C	temperature technical room	PT100:	± 0.5 K
602	θ_{gf}	°C	temperature ground floor	PT100	± 0.5 K
603	θ_{ff}	°C	temperature first floor	PT100	± 0.5 K
604	θ_{oa}	°C	temperature outside	PT100	± 0.5 K
605	φ_{oa}	%r.h.	rel. humidity outside air	humidity sensor	± 2.5%

[1)] while in operation, a voltage of 24 V is sent to the data logger via a relay.

Energy flows

Generated heat water heating system

$$Q_{HW,hp} = \dot{V}_{HW} \cdot \rho_w \cdot c_w \cdot (\theta_{HW,f} - \theta_{HW,r})$$

Generated heat air reheater

$$Q_{H,ve,re} = \dot{H}_{V,sa} - \dot{H}_{hru,oa,out}$$

Net heat demand ventilation

$$Q_V = \dot{H}_{V,sa} - \dot{H}_{hru,oa,in}$$

Enthalpy flow return air heat recovery ventilation inlet

$$\dot{H}_{hru,ra,in} = \dot{V}_{V,ra} \cdot \rho_a(\theta_{hru,ra,in}) \cdot c_{p,oa}(\theta_{hru,ra,in}, \varphi_{hru,ra,in}) \cdot T_{hru,ra,in}$$

Enthalpy flow return air heat recovery ventilation outlet

$$\dot{H}_{hru,ra,out} = \dot{V}_{V,ra} \cdot \rho_a(\theta_{hru,ra,in}) \cdot c_{p,a}(\theta_{hru,ra,out}, \varphi_{hru,ra,out}) \cdot T_{hru,ra,out}$$

Enthalpy flow outside air heat recovery ventilation inlet

$$\dot{H}_{hru,oa,in} = \dot{V}_{V,sa} \cdot \rho_a(\theta_{V,sa}) \cdot c_{p,a}(\theta_{oa}, \varphi_{oa}) \cdot T_{oa}$$

Enthalpy flow outside air heat recovery ventilation outlet

$$\dot{H}_{hru,oa,out} = \dot{V}_{V,sa} \cdot \rho_a(\theta_{V,sa}) \cdot c_{p,a}(\theta_{hru,oa,out}, \varphi_{oa}) \cdot T_{hru,oa,out}$$

Enthalpy flow supply air

$$\dot{H}_{V,sa} = \dot{V}_{V,sa} \cdot \rho_a(\theta_{V,sa}) \cdot c_{p,a}(\theta_{V,sa}, \varphi_{oa}) \cdot T_{V,sa}$$

Enthalpy flow exhaust air

$$\dot{H}_{V,ea} = \left(\dot{V}_{V,hp,oa} \cdot \rho_a(\theta_{oa}) + \dot{V}_{V,ra} \cdot \rho_a(\theta_{hru,ra,in})\right) \cdot c_{p,a}(\theta_{V,ea}, \varphi_{V,ea}) \cdot T_{V,ea}$$

Enthalpy flow outside air for heat pump

$$\dot{H}_{V,hp,oa} = \dot{V}_{V,hp,oa} \cdot \rho_a(\theta_{oa}) \cdot c_{p,a}(\theta_{oa}, \varphi_{oa}) \cdot T_{oa}$$

Enthalpy flow mixed air for heat pump

$$\dot{H}_{V,ma,hp} = \dot{H}_{V,oa,hp} + \dot{H}_{hru,ra,out}$$

6.2.5 Key values to be determined (objective)

The same key values as in the pilot plant Gelterkinden are determined based on the characteristic numbers in eq. 26 to eq. 28. Instead of an air preheater, however, in this system configuration, an air reheater is used, so the terms for the air preheater are replaced by the air reheater. Moreover, due to the reheater the ETV_{hru} is calculated as balance of the inlet and outlet enthalpy flow of the HRU.

6.2.6 Results of the measurement period

Tab. 44: Partial energy consumptions at a mean room temperature in winter of 22°C for a heated living area of 204 m²

		Dec. 2004	Jan. 2005	Feb. 2005	Mar. 2005	April 2005	May 2005	June 2005	total
		kWh	kWh	kWh	kWh	kWh	kWh	kWh	MJ/m²
$Q_{H,hp}$	generated heat space heating	1006	849	1002	590	173	25	14	64.5
$Q_{W,hp}$	generated heat domestic hot water	245	254	203	245	221	204	161	27.0
$Q_{H,ve,re}$	generated heat air reheater	79	116	58	48	30	28	20	6.7
$Q_{W,dis,in}$	net heat demand hot water	164	156	136	154	138	144	112	17.7
Q_V	net heat demand Ventilation	378	384	283	146	53	-68	-174	21.9[1)]
$E_{cu,in}$	electric energy compact unit	800	648	889	502	152	121	118	56.9
$E_{hp,cmp,in}$	electric energy compressor heat pump	285	292	242	189	104	56	63	21.7
$E_{bu,in}$	electric energy el. heating element	407	251	553	226	2	6	2	25.5
$E_{W,sk,pmp}$	electric energy water heating system circulating pump	7.5	9.0	6.1	7.8	6.6	5.5	4.4	0.8
$E_{hru,fan,in}$	electric energy ventilator ventilation (=$E_{V,ra,fan,in}$+$E_{V,oa,fan,in}$)	46	44	25	21	19	27	27	3.7
$E_{hp,oa,fan,in}$	electric energy ventilator outside air for heat pump	44	49	57	47	21	9	6	4.1
$E_{H,sk,pmp}$	electric energy space heating circulation pump	11.8	11.2	11.0	11.6	11.1	10.6	6.9	1.3
$E_{cu,ctrl,in}$	Electric energy compact unit control	11.9	11.9	10.8	11.9	11.5	11.9	11.5	1.4
$H_{hru,ra,in}$	Enthalpy return air heat recovery ventilation inlet	980	897	570	551	539	720	740	88.0
$H_{hru,ra,out}$	Enthalpy of return air heat recovery ventilation outlet	728	707	396	488	542	859	976	82.8
$H_{hru,oa,in}$	Enthalpy of outdoor air heat recovery ventilation inlet	539	520	283	465	573	1065	1360	84.7
$H_{hru,oa,out}$	Enthalpy of outdoor air heat recovery ventilation outlet	839	789	507	563	596	969	1166	95.7
$H_{V,sa}$	Enthalpy of supply air	917	904	566	612	626	997	1185	102.3
$H_{V,ea}$	Enthalpy of exhaust air	652	642	425	668	746	1043	1155	93.9
$H_{hp,oa}$	Enthalpy of outdoor air for heat pump	647	630	586	650	438	285	269	61.8
$H_{hp,ma,in}$	Enthalpy of mixed air for heat pump	1375	1337	982	1137	980	1144	1246	144.5

[1)] disregarding energy utilized for ventilation in summer [cooling energy]

6.2.7 Results based on Seasons

Corresponding to the heating season 2004/2005 validated data are available for measurements from December, 2004 to April, 2005. The months May and June, 2005 are designated as "summer".

6.2.7.1 Energy consumption in winter

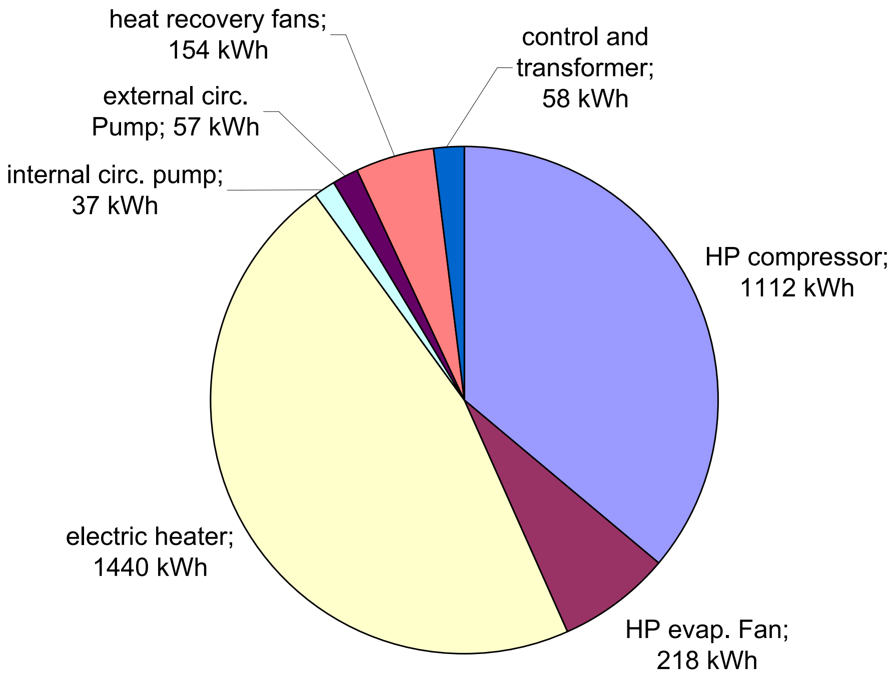


Fig. 77: Electrical energy consumption during the heating season Dec., 2004 – April, 2005

Of the total electrical consumption 47% was used for electric heating, 36% for the heat-pump compressor, 7% for the evaporator fan of the heat- pump, 5% for the ventilation and heat-recovery fans, 2% for the unit's controls and 3% for the pumps. In this connection the energy consumption of the external pump (circulation pump for underfloor heating, Biral M10) amounted to only 1.8% of the total, thanks to the fact that its power consumption of 20 W is extremely low.

The notably high electrical consumption of the electric heating suggests that the heat-pump output is too low. Possible causes are the high heating demand of the building and presumably suboptimal adjustment of the unit's control system, since the control has been designed for the use with air-heating systems and not for the application with thermally activated building systems, which has for instance an impact on the reaction time of the emission system to changes of the room temperature. The control has meanwhile been improved for the application with thermally activated building systems, see chap. 6.1.3.4.

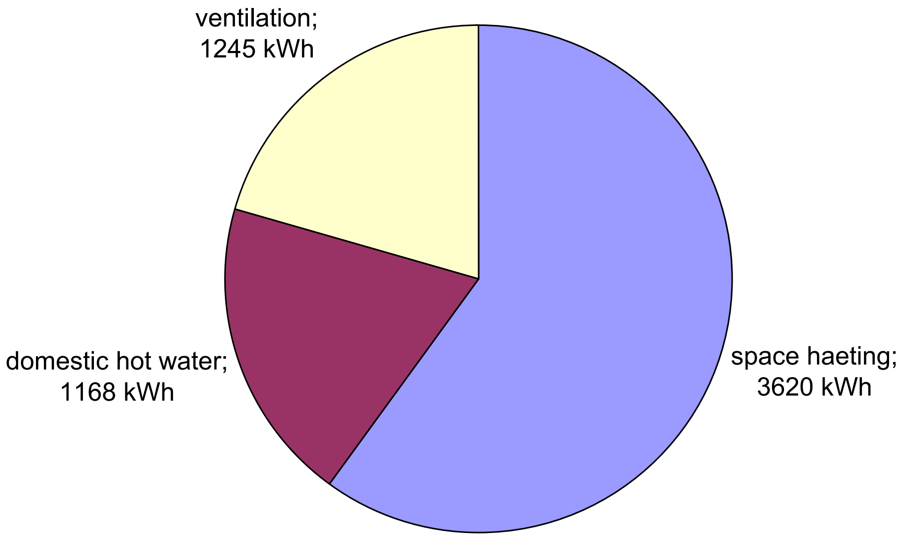


Fig. 78: Distribution of useful heat during the heating season Dec., 2004 – April, 2005

Space heating represents the largest single component of the overall heating demand, claiming 60% of total figure. The other two components, ventilation and domestic hot water, are considerably less, each being 20% of the total. Of the 1245 kWh provided via ventilation 79% was produced by heat recovery and the other 21% by the supply-air heating coil.

The demand domestic hot water is within a normal range. Nevertheless, just for the 5 winter months the measured output for the space heating was 3620 kWh, which exceeds the calculated annual space-heating demand of 2962 kWh/a by 22%. It should be noted, however, that heating demand is generally higher in the first year while the new building is still drying out. In addition, the residents discovered leaks around the windows. In response the windows were freshly adjusted after the end of the heating season. By reducing ventilation losses, this corrective measure will have a positive effect on heating demand.

6.2.7.2 Key quantities for the heating season

Electrothermal amplification factor ETV_{hru}

$$ETV_{hru} = \frac{H_{H,hru,oa,out} - H_{H,hru,oa,in}}{E_{V,fans,in}} = \frac{3294 - 2380}{154} = \underline{\underline{5.9 \pm 1.7}}$$

Heat pump seasonal performance factor SPF-HP

$$SPF-HP = \frac{Q_{H,gen} + Q_{W,gen} + Q_{H,ve,re} - Q_{HW,bu}}{E_{HW,hp,cmp,in} + E_{HW,hp,fan,in}} = \frac{3620 + 1168 + 331 - 1440}{1112 + 218} = \underline{\underline{2.8 \pm 0.3}}$$

Generator seasonal performance factor SPF-G

$$SPF-G = \frac{Q_{H,gen} + Q_{W,gen} + Q_{H,ve,re}}{E_{cu,aux,tot,in} - E_{H,aux,sk,in} - E_{V,fans,in}} = \frac{3620 + 1168 + 331}{2991 - 37 - 154} = \underline{\underline{1.82 \pm 0.13}}$$

System seasonal performance factor SPF-SYS

$$SPF-SYS = \frac{Q_{H,dis,in} + Q_{W,dis,in} + Q_{H,ve,out}}{E_{cu,aux,tot,in} + E_{H,aux,sk,in}} = \frac{3620 + 748 + 1245}{2991 + 57} = \underline{\underline{1.8 \pm 0.1}}$$

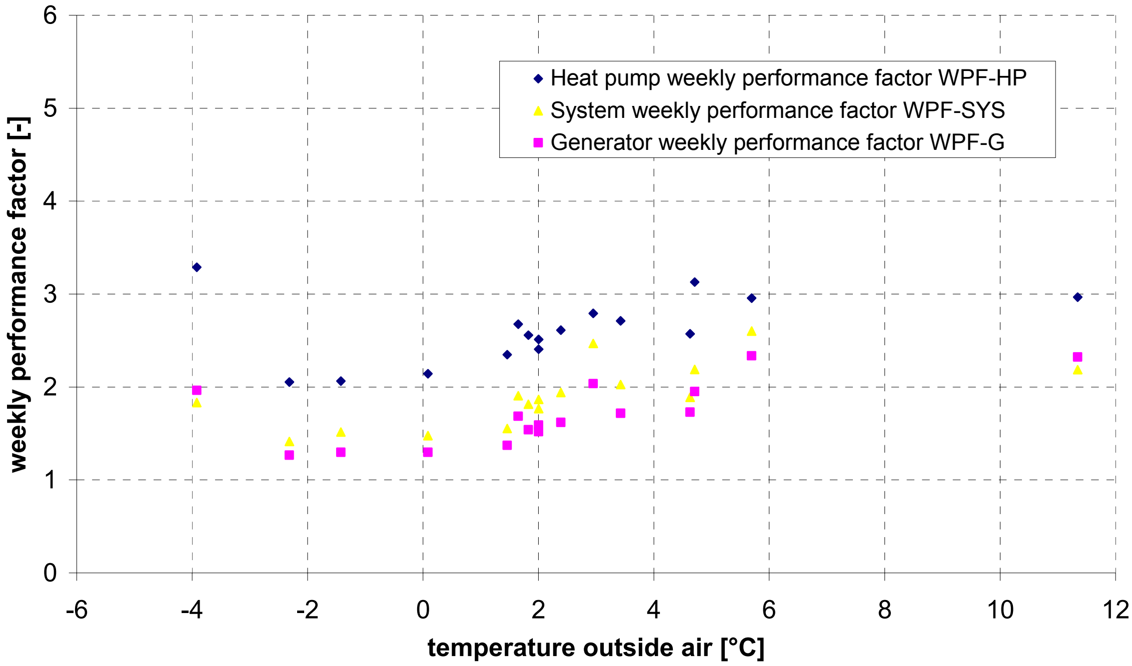


Fig. 79: Weekly performance factor during the heating season

6.2.7.3 Energy consumption in summer

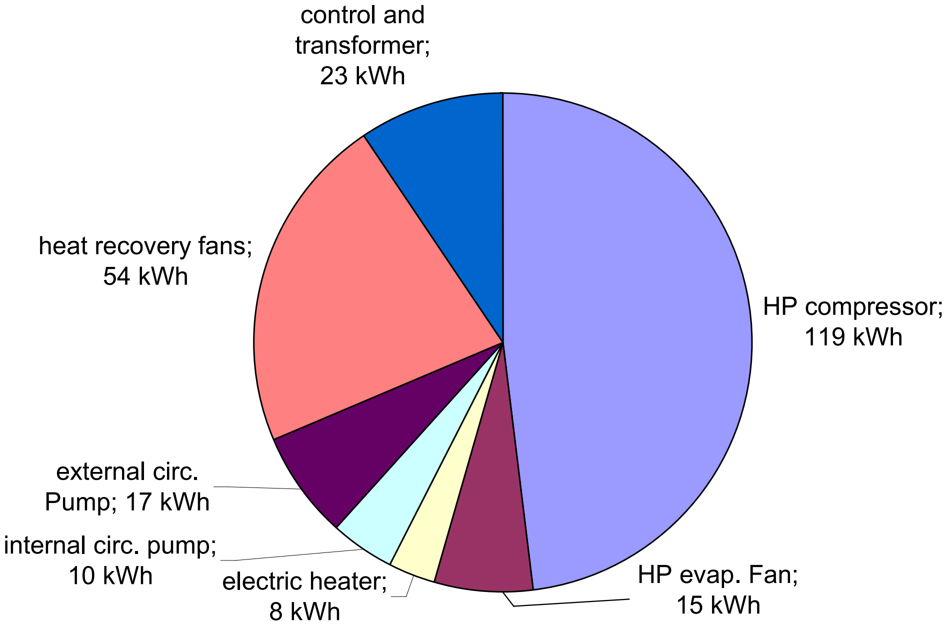


Fig. 80: Electrical energy consumption, May – June, 2005

Of the total electrical energy the heat-pump compressor consumes 48%, the evaporator fan 6%, the pumps 11% and the control systems 9%. The relative consumption of the ventilation and heat-recovery fans increased to 22%. Although the demand for space heating was non-existent, the electrical heating was sporadically switched without apparent cause at the start of hot-water tank heating, and consumed 8 kWh or 3% of the overall consumption.

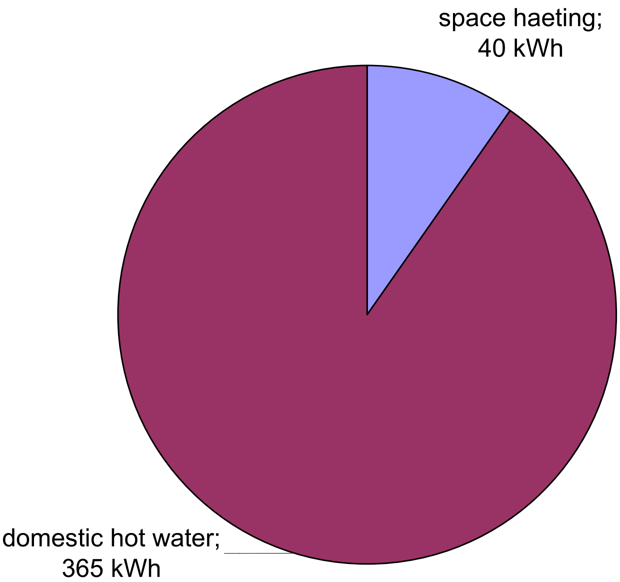


Fig. 81: Distribution of useful heat, May – June, 2005

Surprisingly, even during the warm months heat delivery for space heating was measured which amounted to 9% of the overall demand. This can be explained by the fact that the space heating was not switched off until the end of June. Since the heating control also takes the outdoor temperature into account, the heating valve would be opened whenever the outside-air temperature fell below approx. 5°C. Furthermore, during the heating of domestic hot water, creepage of the hydraulic circulation results in some output of heat to the space heating.

In contrast, a negative heat demand of -242 kWh was ascertained for the ventilation – i.e. the outdoor air was cooled by the heat recovery unit.

6.2.7.4 Key quantities for the summer months

Heat pump seasonal performance factor SPF-HP

$$\mathrm{SPF-HP} = \frac{Q_{H,gen} + Q_{W,gen} + Q_{H,ve,re} - Q_{HW,bu}}{E_{hp,cmp,in} + E_{hp,fan,in}} = \frac{40 + 365 + 48 - 8}{119 + 15} = \underline{\underline{3.3 \pm 0.5}}$$

Generator seasonal performance factor SPF-G

$$\mathrm{SPF-G} = \frac{Q_{H,gen} + Q_{W,gen} + Q_{H,ve,re}}{E_{cu,aux,tot,in} - E_{H,aux,sk,in} - E_{V,fans,in}} = \frac{40 + 365 + 48}{239 - 10 - 54} = \underline{\underline{2.6 \pm 0.4}}$$

System seasonal performance factor SPF-SYS

$$\mathrm{SPF-SYS} = \frac{Q_{H,dis,in} + Q_{W,dis,in} + Q_{H,ve,out}}{E_{cu,aux,tot,in} + E_{H,aux,sk,in}} = \frac{40 + 256 + 0}{239 + 17} = \underline{\underline{1.16 \pm 0.02}}$$

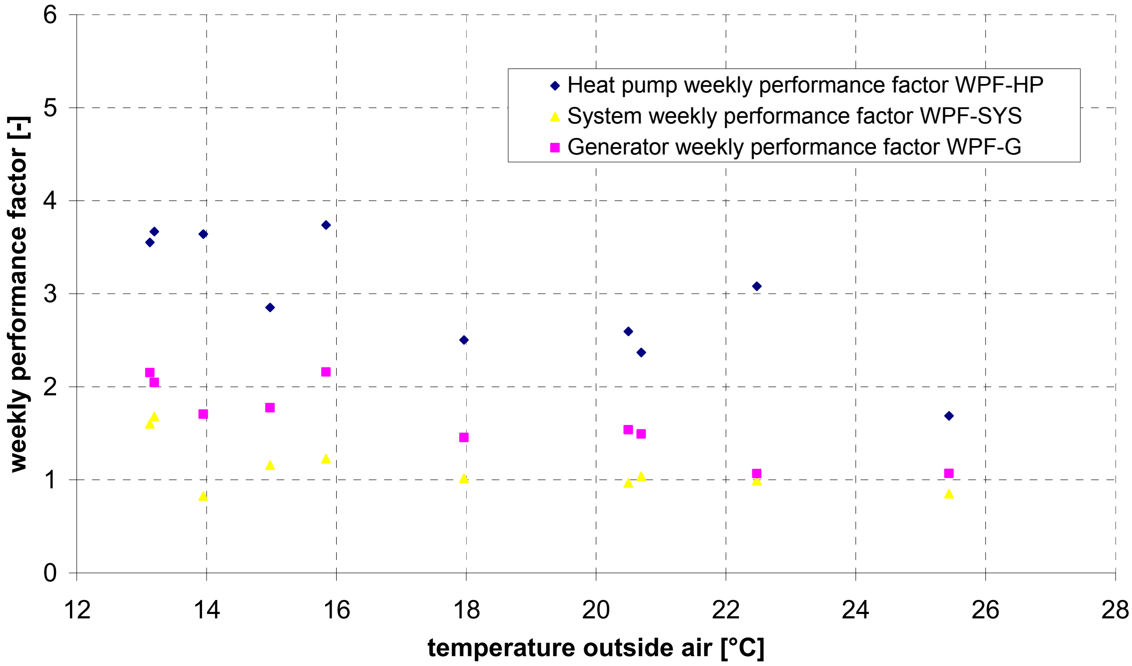


Fig. 82: Weekly performance factor, May – June, 2005

6.2.7.5 Outdoor-air temperatures

The measured heating degree days from December, 2004 to April, 2005, in terms of a base temperature of 14°C and the effective room temperature, amount to 2600 Kd. For the months of May and June a further 152 Kd were measured.

The climatic weather conditions during the measurement period Dec., 2004 – April, 2005 could be summarized, if anything, as being on the warm side. The measured $DH_{H20/12}$ (2192 Kd) are 11% below the long-term average value for Basel-Binningen (2462 Kd) [1].

Tab. 45: Average outside-air temperatures during the measurement period.

Month	Mean [°C]	Min. [°C]	Max. [°C]
December, 2004	2.2	-7.6	10.8
January, 2005	2.2	-10.7	14.2
February, 2005	1.0	-8.3	13.0
March, 2005	7.0	-12.9	24.3
April, 2005	11.1	-0.7	30.1
May, 2005	15.5	3.9	35.6
June, 2005	20.6	6.0	36.5

6.2.7.6 Room-air temperatures

Tab. 46 shows the room-air temperatures during the heating season. The utility room is the room in which the compact heating centre is located.

Tab. 46: Mean room-air temperatures during the measurement period

Month	Utility room [°C]	Ground floor [°C]	Upper floor [°C]
December, 2004	22.2	21.7	21.4
January, 2005	22.3	22.2	22.1
February, 2005	22.9	22.6	22.3
March, 2005	23.3	22.4	22.4
April, 2005	22.8	21.4	22.0
May, 2005	24.3	21.4	21.9
June, 2005	26.9	22.5	22.6

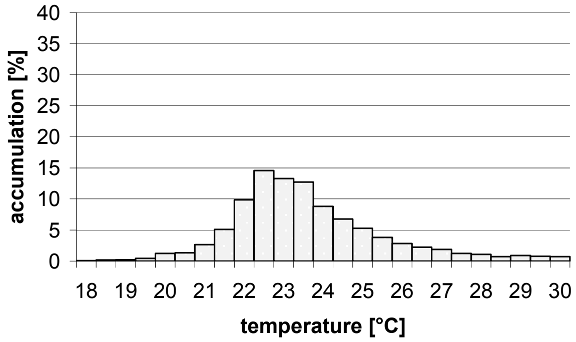


Fig. 83: Statistical distributions of utility-room temperatures

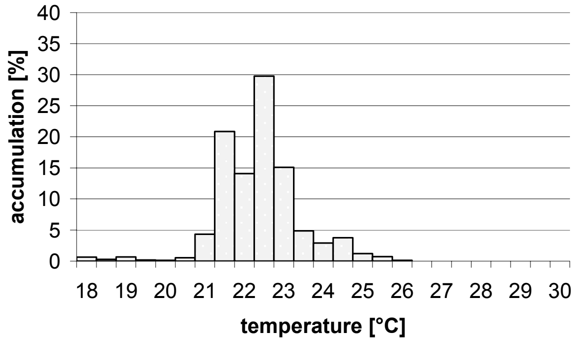


Fig. 84: Statistical distribution of ground-floor temperatures

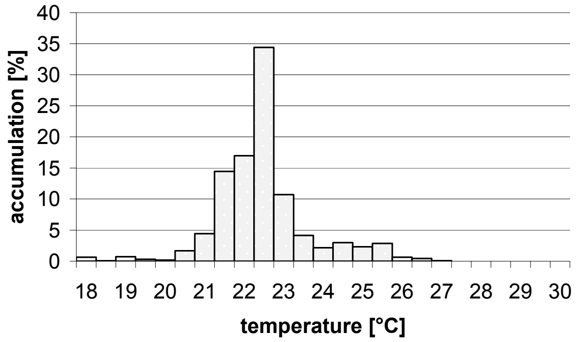


Fig. 85: Statistical distributions of upper-floor temperatures

6.2.8 Typical days

6.2.8.1 Water heating

During heating operation the measured supply temperatures of the space heating lie in the range 25°C – 30°C with a temperature difference of 3 to 5 K with respect to the return. During heating of the domestic hot water tank the space-heating pump is switched off and the internal circulation pump switched on, while the supply temperature delivered to the water heater is increased to 52°C. The space heating is controlled primarily by the temperature in a reference room, although the outside-air temperature is also taken into consideration.

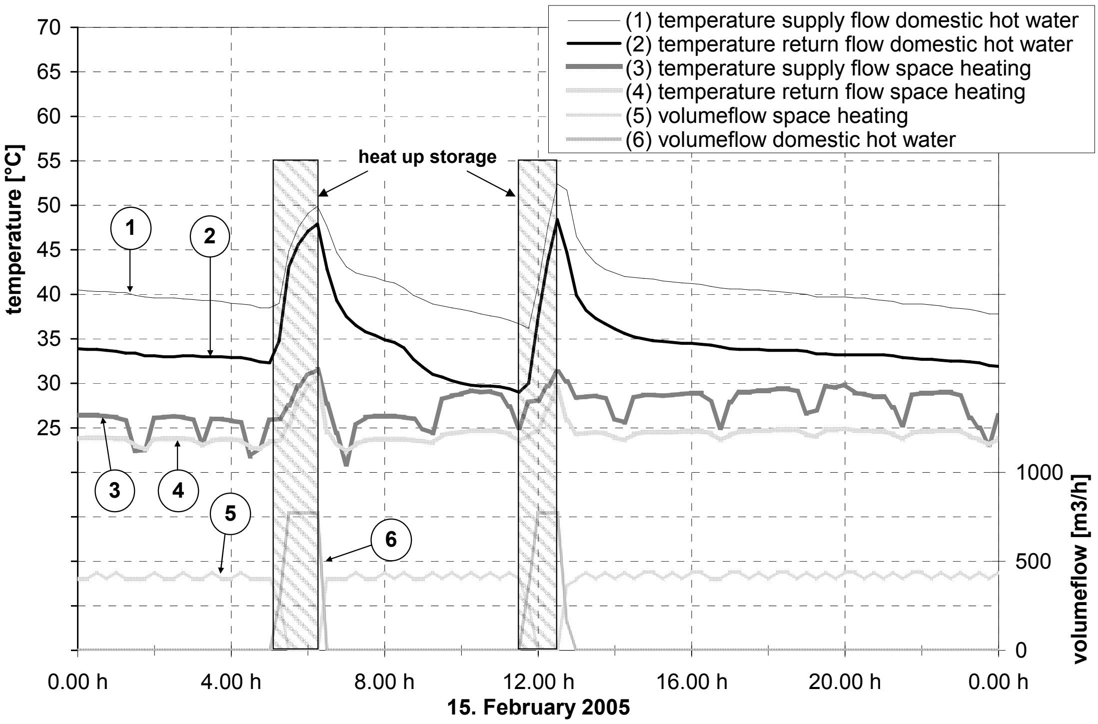


Fig. 86: Typical daily temperature curves of the water heating

In order to cope with the expected heating demand on colder days, with temperatures below 5°C, the heating must be supplemented by the electric heating. For temperatures down to approx. –5°C the first stage (1.7 kW) is sufficient. Beyond that, the second stage (3.4 kW) may sometimes have to be used. On warm days, when no heat is required for the space heating, it can happen that the electric heating gets switched when a domestic hot-water heating phase begins (without apparent reason).

The mean power consumption for all auxiliary devices such as pumps, fans and control systems is approx. 170 W during the heating season and approx. 70 W in summer.

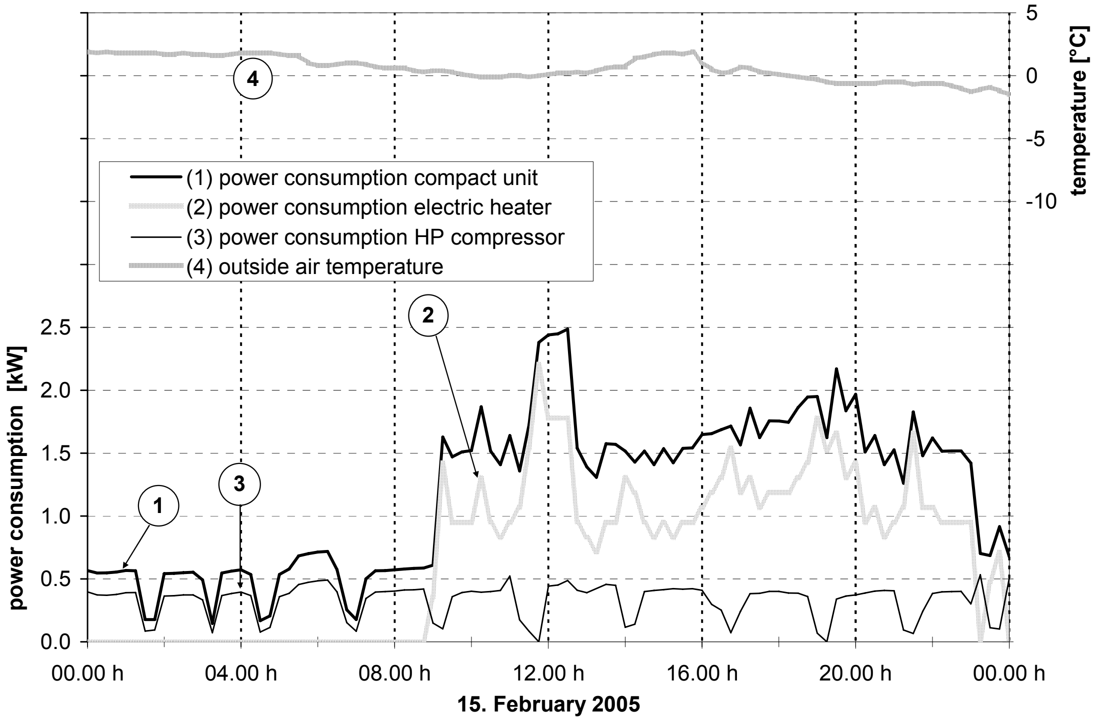


Fig. 87: Power consumption curves for the compact heating centre, the electrical heating and the compressor during a cold day (mean outside-air temperature 0.5°C)

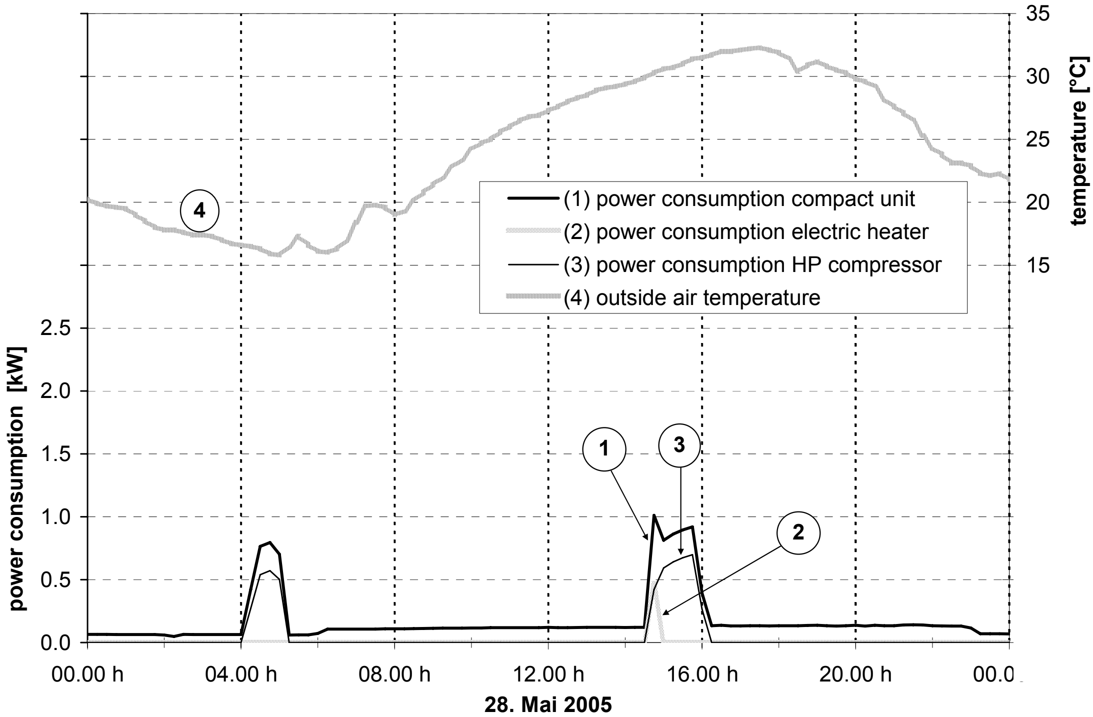


Fig. 88: Power consumption curves for the compact heating centre, the electrical heating and the compressor during a warm day (mean outside-air temperature 24°C)

6.2.8.2 Domestic hot water

The heating of the domestic hot-water always starts at a temperature of 45°C in the upper layer of the storage tank, and continues until it reaches approx. 52°C (at the beginning of the data collection the cut-off threshold was approx. 48°C). On the average the hot-water tank is heated up twice daily.

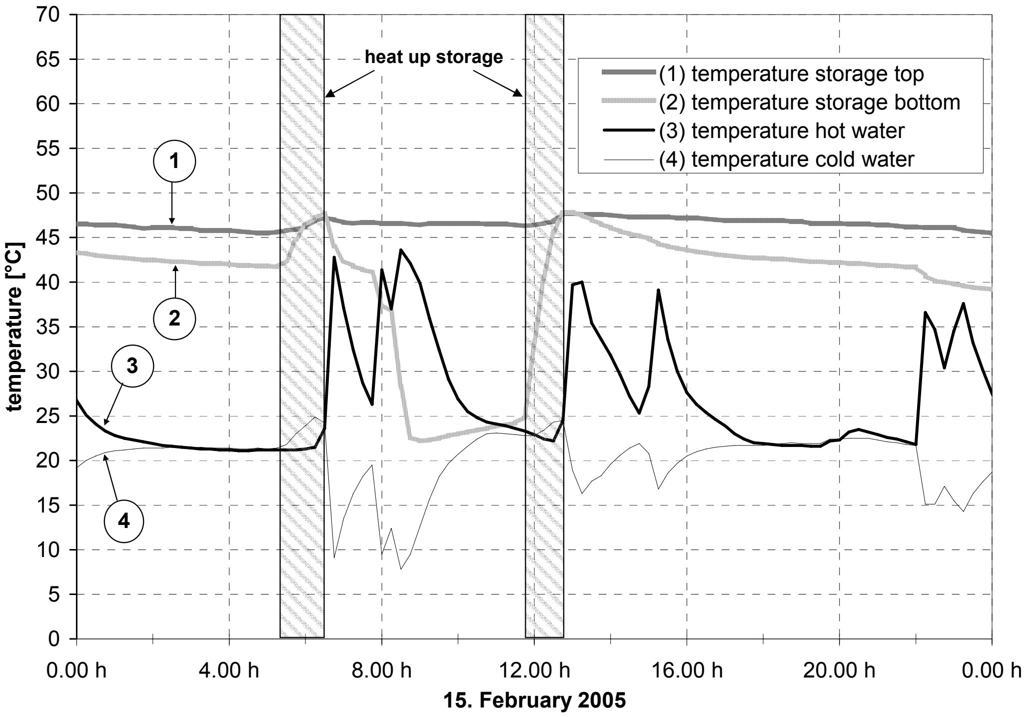


Fig. 89: Typical daily temperature curves for the storage tank, hot water and cold water

6.2.8.3 Ventilation

During the heating season the HRU attains an average temperature change coefficient of 82%. Up to the middle of February, 2005 the supply-air volume flow was set at about 150 m^3/h. Because acoustic problems were noticeable with the unit (whose cause could not be established up to the end of data collection), the volume flow was reduced on a trial basis to about 100 m^3/h. The noise problem was not significantly reduced as result of this measure. However, because the residents did not experience any reduction of air quality during this test, the lower volume flow was kept.

The sudden drop in temperature of the exhaust air indicates operation of the heat pump. On cold days, when the heat pump is continuously in operation, frost must be removed approx. every 1 ¾ hours. During the defrosting phase the exhaust air temperature rises significantly above the value of the air in front of the evaporator.

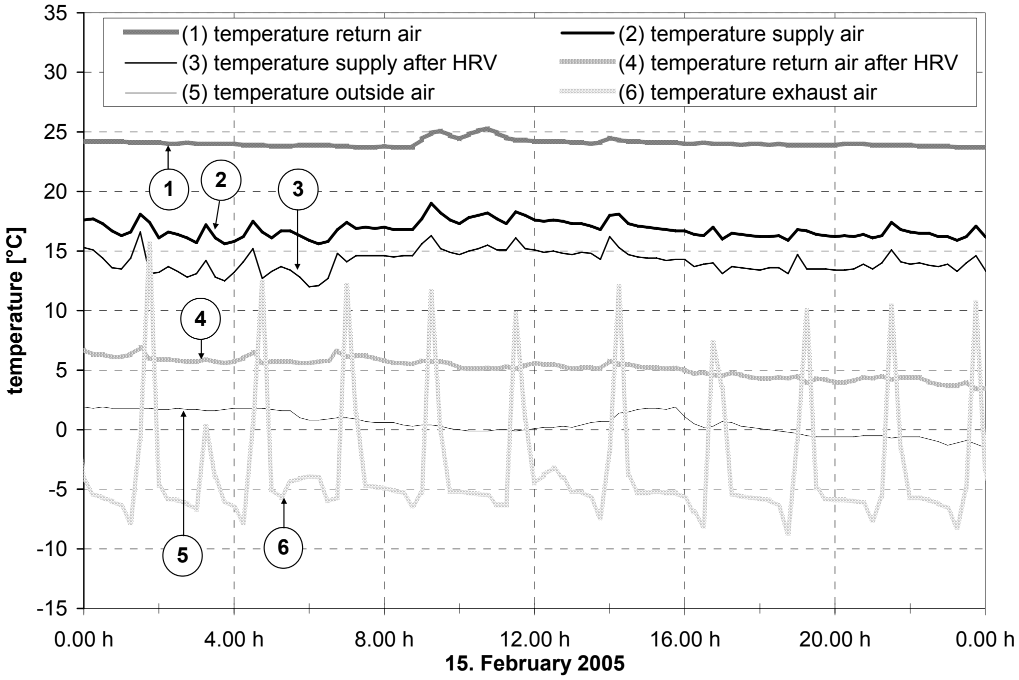


Fig. 90: Ventilation temperature curves during a cold day

On hot days the outside air can be cooled thanks to the HRU. This is evident in the following figure: the extract-air temperature is comparatively constant at approx. 25°C throughout the day while the outside-air heats up to over 32°C by the late afternoon. In the HRU at this point the outside air is cooled by the exhaust air down to approx. 27°C (supply-air temperature).

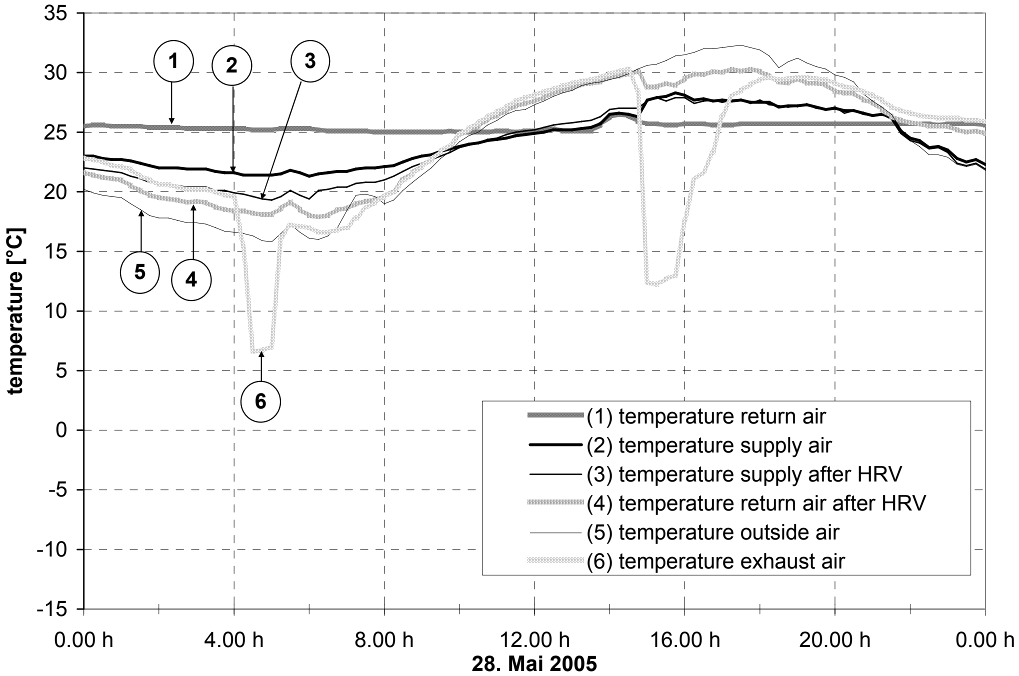


Fig. 91: Ventilation temperature curves during a warm day

6.2.9 Recommendations for further improvement

The output of the heat pump is too low for the building, even allowing for a reduced heating demand in the coming heating seasons, which may be expected for the following reasons:

- In the first heating season the heating demand in generally higher, because the building is drying out.
- The residents noticed some air leakage around the windows, which resulted in increased ventilation losses. This shortcoming was only rectified towards the end of the heating season.

According to the residents the unit sporadically produces a kind of “rumbling”, which can sometimes be quite loud. The noise was mostly noticed during the early morning hours. Unfortunately, until the end of the project none of the qualified specialists (project team, staff from Viessmann (Schweiz) AG actually heard the noise. In spite of all kinds of adjustments to the unit the rumbling could not be reproduced and consequently its cause could not be identified.

7 CONCLUSIONS

In this project the FHNW method for the calculation of the seasonal performance of heat pump systems for combined space heating and domestic hot water production has been extended to cover compact units. The calculation is based on a temperature class approach (bin method) and a component characteristics of a test procedure, which has been developed in parallel, so consistent testing and calculation for compact units is provided. The proposed method has been validated by comparison to field data of two pilot plants, one ground source heat pump and one compact unit, of which year-round measurements in hourly resolution were available.

The comparison shows a deviation between the measured and calculated values in the range of ±6% for the overall seasonal performance factor, which is in the range of the exactness of the component characteristics derived by the testing and of the field monitoring. Further evaluation performed in the frame of IEA HPP Annex 28 showed the same deviation range.

The basic impact on the seasonal performance factor is the temperature level, since the heat pump is the core component of the system and COP and heating capacity are mainly influenced by the source and sink temperature. So, temperature conditions over the operation range are to be taken into account attentively. If the heat pump characteristic and the temperature conditions are mostly linear in parts or even constant, only few bins are required in the calculation.

The redistribution of the energy to the bins has only limited impact on the resulting performance factor, since the COP characteristic has a limited range. Therefore, the redistribution of used solar and internal gains is not so sensitive. The applied approximation of a redistribution by the heating degree hours and a heating limit delivered good agreement with monitoring results as well as for the planning data of an ultra-low energy house. Approximation for the DHW consumption by a daily profile was also feasible for the calculation of the pilot plants. For the same reason, a monthly approach only delivered slightly different performance factors, so an annual calculation gives already a good estimation.

For the same reason recovered ventilation energy is not very sensitive, either. Therefore, the calculation of ventilation heat recovery can be separated from the calculation of the compact unit by directly subtracting the recovered ventilation energy from the space heating requirement, as it is done in some building standards. Notably the electrical energy use for the ventilation has to be taken into account. Moreover, the characteristic of the heat pump may depend on the volume flow rate. And, depending on the system configuration and the used test results, an impact of the ventilation and further attached systems like ground-to-air heat exchangers on the heat pump inlet temperature may have to be considered. Therefore, parameters of the ventilation system are needed for the calculation of the compact unit. Summarising, main attention should be paid to temperature level and characteristic of the heat pump, while for the other impacts, simplifications are possible.

Existing methods for the calculation are rather related to certain types of compact units due to the underlying approximations. The Tool WPEsti [3] used in the MINERGIE® calculation concentrates on the calculation of heat pumps with low temperature distribution, which are common for this building sector, while the PHPP Excel sheet [1] refers to the classical passive house configuration of air heating systems with higher supply temperatures. The PHPP calculation is dependent on adequate test results, since temperature level is not considered and only one test point for each winter and summer season is used.

The FHNW method has been designed to cover most common system configurations on the market and additionally attached systems like ground-to-air heat exchangers or solar collectors, which have a further influence on the overall performance. Thereby, more input data are required, but the impact of the different components integrated in the compact unit can be evaluated. However, control can only be approximately considered which is a main difference to simulations, and further validation of newly integrated models is a future task.

Last but not least comparison of different heating systems for has been performed. Air-heating systems require high supply temperatures which may have a strong impact on the seasonal performance of the heat pump depending on the COP characteristic, and therefore floor heating systems with low supply temperature level down to temperatures below 30°C deliver a better seasonal performance. However, depending on the energy shares for space heating, DHW and recovered ventilation heat, the impact on the overall performance is limited. Thus, as expected, floor heating system have advantages with regard to the electrical energy consumption, i.e. the operating costs, while air heating systems have advantages concerning investment costs. Therefore, air heating systems may still be the preference in ultra-low energy buildings to cut investment costs.
Concerning the testing, no common European test standard for compact units exists. In the time of the project a new test rig at the HLKS test centre, HTA Lucerne, for the comprehensive testing of ventilation systems including heat pump compact units has been designed, installed and commissioned. In particular detailed acoustic measurements of the ventilation systems and heat pump compact units can be performed, since the noise immission is also an important feature of the system in low- and ultra low energy houses, which may be installed inside the thermal insulation perimeter.
In the frame of this project, the test guideline for the testing based on existing European standards for the single components ventilation, heat pump and storage has been developed. The test procedure has been developed to deliver the relevant component characteristics to perform the calculation. It is based on existing standards for the single components of the compact unit, the ventilation, the heat pump system and the storage. In order to cover the most common systems on the market, a black box testing of the entired unit is applied. To consider the interdependencies of the components in the testing a combined testing of three operation modes ventilation-only, combined ventilation and space heating by the heat pump as well as combined ventilation and DHW production by the heat pump is performed. With regard to the calculation method the heat pump has been considered as core component of the system. Therefore, contrary to the testing of DIBt in Germany, test points of the European heat pump test standards EN 14511 for the space heating operation and EN 255-3 for the domestic hot water operation have been incorporated.
In order to gather experience with the real behaviour of pilot plants detailed field monitoring of two compact units has been performed. One pilot plant is installed in a single family house according to the MINERGIE® standard in Gelterkinden (canton Baselland, CH), using a compact unit adapted to the power range of MINERGIE® houses with hydronic distribution by a floor heating system. The other pilot plant is a single family house in Zeiningen (canton Aargau, CH) according to the MINERGIE-P® standard with a compact unit designed for ultra low energy houses. The compact unit is originally designed for air heating distribution, but has been modified to a prototype application with hydronic heat emission by thermally-activated building structures for this project.
The pilot plant in Gelterkinden delivered the results that in wintertime, 78% of the electrical energy is used by the heat pump, 5% by the electrical back-up heating and 17% for auxiliaries including the ventilation fans and circulation pump of the heating system. The resulting overall Seasonal Performance Factor (SPF-HP) of the heat pump is 3.8, while the System Seasonal Performance Factor SPF-S related to the energy need of the space heating and DHW distribution varied between 2.4 in summertime and 3.1 in wintertime at a measurement uncertainty of 4-7%.
The field monitoring of the pilot plant in Zeiningen yielded, that in wintertime 36% of the electrical energy was used by the heat pump, 47% by the electrical back-up heater and 17% by auxiliary components. The SPF-HP of the heat pump was 2.8 while the SPF-S is 1.8 due to the higher fraction of electrical back-up energy. Reasons for the higher back-up energy use was the low heating capacity of the heat pump of 1.5 kW related to a building design heat load of 2.5 kW.

8 FUTURE WORK

Heat pumps and heat pump compact units have recently become very popular heating systems, especially in low and ultra-low energy houses and developments of the components and systems are in progress.

In the frame of this project the calculation method for the seasonal performance has been compared to field results of a ground-coupled heat pump system and a compact unit. However, the systems investigated in this project worked on a hydronic distribution system with floor heating with low supply temperatures, while compact units with air heating distribution systems are quite common in ultra-low energy houses. Therefore, it would be useful to compare the calculation with a compact unit with air heating system. Moreover, effects of condensation are not treated in detail, which may become more important in units with moisture transfer. Actually, a further extension of the functionality of compact units to humidification in wintertime and additional cooling operation in summertime is in progress and partly already available on the market.

Furthermore, the calculation method has also been extended for the calculation of a ground-to-air heat exchanger and a solar collector, since systems on the market often incorporate these components. However, approaches have not been tested or validated so far, so further evaluation of the exactness of the models is a future task.

Hybrid cooling operation, e.g. by a ground-to-air heat exchanger, will be treated in a follow-up project in the frame of the IEA HPP Annex 32 entitled "Economical heating and cooling systems in low energy houses".

Concerning the testing, still no common European test procedure for heat pump compact units exists, yet, even though the market share of the systems is increasing with increasing number of low and ultra-low energy houses. However, recently, the European Committee for standardization CEN initiated a work item to the development of a common standard which will be supported by the results of this project. In the standardisation process it presently seems possible, that both the CEN ventilation and heat pump working groups will be involved in the process, so a standard with uniform testing seems possible.

Furthermore, in the frame of the certification of compliance with the MINERGIE® standard, heat pump compact units, in particular with air heating function, presently cannot be adequately considered. The results of this project for the testing and calculation can be the basis for the extension of the MINERGIE® calculation. Manufacturers of typical passive house air heating units already showed interest in a more detailed calculation as it is presently implemented in the MINERGIE® calculation. Objects for accompanying field monitoring have been envisaged in connection with manufacturer contact. Possibly in the frame of this work further evaluation can be accomplished.

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10 NOMENCLATURE

The nomenclature is based on the nomenclature used in the frame of the EPBD [13]

10.1 Definitions

10.1.1 Building

Building
Construction as a whole, including its envelope and all technical building systems, for which energy is used to condition the indoor climate, to provide domestic hot water and illumination and other services related to the use of the building

Building services
Services provided by technical building systems and by appliances to provide indoor climate conditions (by space heating, space cooling, ventilation, humidification, dehumidification), domestic hot water, illumination levels (by lighting) and other services related to the use of the building

Other services
Services supplied by energy consuming appliances (e.g. for cooking)

Technical building system
Technical equipment for heating, cooling, ventilation, domestic hot water, lighting and electricity production composed by sub-systems

NOTE 1 A technical building system can refer to one or to several building services (e.g. heating system, heating and DHW system).

10.1.2 Technical building systems

Energy need
Heat to be delivered to (space heating) or extracted from (space cooling) a conditioned space to maintain the intended temperature during a given period of time, latent heat in the water vapour to be delivered to (humification) or extracted from (dehumidification) a conditioned space by a technical building system to maintain a specified minimum or maximum humidity within the space and heat to be delivered to the needed amount of domestic hot water to raise its temperature from the cold network temperature to the prefixed delivery temperature at the delivery point not taking into account the technical building thermal systems

Energy use for space heating or domestic hot water
Energy input to the heating or hot water system to satisfy the energy need for heating, cooling (including dehumidification) or hot water respectively.

Energy use for ventilation
Electrical energy input to the ventilation system for air transport and heat recovery (not including the energy input for preheating the air)

Auxiliary energy
electrical energy used by technical building systems for heating, cooling, ventilation and/or domestic water to support energy transformation to satisfy energy needs

NOTE 1 This includes energy for fans, pumps, electronics, etc. Electrical energy input to a ventilation system for air transport and heat recovery is not considered as auxiliary energy, but as energy use for ventilation

Ventilation heat recovery
heat recovered from the exhaust air to reduce the ventilation heat transfer

Solar irradiation
incident solar heat per area over a given period

Solar heat gain
heat provided by solar radiation entering, directly or indirectly (after absorption in building elements), into the building through windows, opaque walls and roofs, or passive solar devices such as sun-spaces, transparent insulation and solar walls

Useful heat gains
proportion of internal and solar heat gains that contribute to reducing the energy need for heating

Gain utilisation factor
factor reducing the total monthly or seasonal heat gains to obtain the resulting reduction of the energy need for heating

Heat balance ratio
monthly or seasonal heat gains divided by the monthly or seasonal heat transfer

Balance point temperature
temperature at which the heat pump heating capacity and the building heat load are equal

Bin
A statistical temperature class (sometimes a class interval) for the outdoor air temperature, with the class limits expressed in a temperature unit

Seasonal performance factor SPF
The ratio of the total annual energy delivered to the distribution system for space heating and/or domestic hot water to the total annual electrical energy use incl. auxiliary energy.

Simultaneous operation
Simultaneous production of heat energy for the space heating and domestic hot water system by a heat generator with double service, e.g. by refrigerant desuperheating or condensate subcooling

Cut-out period
time period in which the electricity supply to the heat pump is interrupted by the supplying utility.

Alternate operation
production of heat energy for the space heating and domestic hot water system by a heat generator with double service by switching the heat generator either to the domestic hot water operation or the space heating operation

Frequency
the (statistical) frequency of an event is the number of times the event occurred in the sample. The frequencies are often graphically represented in histograms. In the frame of this report the frequency of the outdoor air temperature is evaluated based on a sample of hourly-averaged data for one year.

Cumulative frequency
frequency of the outdoor air temperature cumulated over all 1 K bins

10.2 Variables

$\dot{V}$	Volume flow rate	m^3/s
$\dot{E}$	Exergy flow	W
$\dot{m}$	Mass flow rate	kg/s
$\dot{H}$	Enthalpy flow	W
$\dot{Q}$	Heat flow	W
θ	Celcius temperature	°C
ϕ	Heat loss, heat power, heating capacity	W
ρ	Density	kg/m^3
β	Inclination angle	rad
η	Efficiency	-
ζ	Exergetic efficiency	-
Φ	Temperature change coefficient	-
Δ	Difference	-
β	Inclination angle of the receiving surface	rad
ε	Efficiency factor	-
φ	Relative humidity	%
ρ_g	Ground reflectance	-
θ_i	Incidence angle	rad
θ_z	Zenith angle	rad
A	Area, surface	m^2
c	Specific heat capacity	J/(kgK)
COP	Coefficient of Performance	-
DD	Degree day	Kd
DH	Degree hour	Kh
E	Electrical energy, Exergy	J
ETV	Electro-thermal amplification	-
G	Solar irradiation	W/m^2
H	Enthalpy	J
h	Specific enthalpy	J/kg
IAM	Incidence Angle Modifier	-
k	Fraction, coefficient	-
m	Exponent	
N	Number of quantity	
P	Electrical power	W
Q	Quantity of heat	J
SPF	Seasonal Performance Factor	-
R	Gas constant	J/(kg·K)
s	Specific entropy	J/(kg·K)
T	Thermodynamic temperature	K
t	Time	s
U	Length or surface specific heat loss coefficient	W/m or W/m^2
w	Weighting factor	-
x	Water content, absolute humidity	g/kg
X	Dimensionless Loss-load ratio	-
Y	Dimensionless solar-load ratio	-
Z	Binary state	-

10.3 Subscripts

Level 1		Level 2		Level 3		Level 4	
Which type of energy use		***Building without technical systems***		***Balance item, component***		***Balance item***	
H	Heating	nd	Need	ls	Losses	rbl	Recoverable
W	DHW	ve	Ventilation transfer	aux	Auxiliary	rvd	Recovered
V	Ventilation	tr	Transmission transfer	in	Input	nrbl	Non-recoverable
XY	Combination of H, W, V			out	Output	nrvd	Non-recovered
				des	design		
Tot	Total			i	indoor		
Which quantity		***Technical building system***		***Qualifier (where used)***		***Qualifier (which type)***	
ho	hours	dis	(hydronic) distribution	j	bin j	wb	Wet bulb
month	month	st	Storage	w	water	max	maximum
		ctr	Control	hw	Hot water	min	minimum
		gen	Generation (HP and BU)	cw	Cold water	llim	Lower limit
				winter	During winter	bh	beam horizontal
		Component		summer	During summer	dh	Diffuse horizontal
		bld	building	sk	Sink	p	at constant pressure
		cu	Compact unit	src	Source	hlim	upper limit
		hp	Heat pump	f	Flow	h	horizontal
		hru	Heat recovery unit	r	return	t	tilted
		bu	Back-up heater	loop	Collector loop	E	Energy reference
		oa	Outdoor air (flow)	c	Cold side	int	internal
		sa	Supply air (flow)	h	Hot side	co	Cut-out time
		ra	Return air (flow)	a	air	1+x	moist air
		ea	Exhaust air (flow)			nom	nominal
		ma	Mixed air (flow)			op	operation
		evap	Evaporator			C	Carnot
		cmp	Compressor			sb	Stand-by
		cond	Condenser			eff	effective
		pre	Preheater			avg	average
		re	reheater			tot	total
		fan	Fan			cap	lack of capacity
		pmp	Pump			l	length specific
		col	Solar Collector			bal	Balance point
		tr	Technical room			sol	solar
		gf	Ground floor			sp	Setpoint
		ff	First floor			irr	Irreversible
		pipe	piping			rev	reversible
						amb	ambient

Levels, that do not apply, are skipped

10.4 Nomenclature from other sources

Nomenclature from EN 255-3 [7]

Symbol	Description	-
COP_t	COP for the extraction of hot water according to EN 255-3	
P_{es}	electrical energy input to cover storage losses	W
t_h	heating-up time (phase 1)	h and min
V_{max}	Maximum volume of water in a single tapping	dm^3
W_{eh}	Heating up energy (phase 1)	kWh
θ_{wr}	Hot water reference temperature	°C

Nomenclature from SIA 380/1 [26]

Symbol	Description
Qhww	heat use for heating and hot water preparation
Qs	passive solar gains
QiP	metabolic heat
QiE	heat from other appliances
Qh	heat use
Qww	heat for hot water preparation
Qg	total gains
Qug	useful gains
Qvra	ventilation heat recovery
QV	ventilation heat loss
QT	transmission heat loss
Qww	heat for hot water preparation
Qhww	heat use for heating and hot water preparation
Qs	passive solar gains
QiP	metabolic heat
QiE	heat from other appliances
Qh	heat use
Qww	heat for hot water preparation
Qg	total gains
Qug	useful gains

Nomenclature from DIBt

Symbol	Description	-
η'_{WRG}	Heat recovery efficiency acc. to DIBt	
$\dot{V}_{M,m}$	Nominal interval volume flow rate	m^3/h

10.5 Abbreviations

A/W - Air-to-Water
ARI – Air conditioning and Refrigeration Institute
ASHRAE – American Society of Heating, Refrigerating and Air conditioning Engineers Inc.
AWEL - Amt für Wasser, Energie und Luft (http://www.awel.ch)
B/W - Brine-to-Water
COP - Coefficient of Performance
CU - Compact unit
DHW - Domestic Hot Water
DIBt - Deutsches Institut für Bautechnik (http://www.dibt.de)
DRY – Design Reference Year
EN - Euronorm
EPBD - EU Directive on the Energy Performance of Buildings
FhG - Fraunhofer Gesellschaft
FHNW - Fachhochschule Nordwestschweiz
FWS - Fordergemeinschaft Wärmepumpen Schweiz (http://www.fws.ch)
GAHX - Ground-to-air heat exchanger
HDD – Heating Degree Days
HDH – Heating Degree Hours
HP - Heat pump
HR - Heat recovery
HTA - Hochschule für Technik und Architektur Luzern
ISE - Institut für Solare Energiesysteme
OP - Operating point
RFL Return flow
PHPP - Passivhaus Projektierungs Paket (http://www.phi.de)
S - Solar energy
SFL Supply flow
SFOE - Swiss Federal Office of Energy
SH - Space heating
SIA - Schweizer Ingenieur- und Architektenverband (http://www.sia.ch)
SPF - Seasonal performance factor
TRY - Test Reference Year (see above)
TUEV - Technischer Überwachungsverein
WHS - Water-based space heating distribution system

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