BFE OFEN UFE SFOE

Research program
heat pumping technologies, cogeneration, refrigeration

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

APPENDIX

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HTA –> HOCHSCHULE FÜR TECHNIK+ARCHITEKTUR LUZERN
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Test guideline for the Test of Compact Units

Units with supply and return air ventilation, heat recovery and exhaust-air heat pumps for domestic hot water heating and/or heat delivery to a separate water-based system.

1 Introduction

This standard describes a testing programme for compact ventilation units with supply and return air as well as heat recovery. In addition, these units include heat pumps using the supply and/or exhaust air as a source of heat which is delivered to domestic hot water and/or other water-based systems.

This standard is consistent as far as possible with EN 13141.

2 Aims and scope

The aim of this standard is to define the scope, procedures and laboratory-equipment specifications which apply to the testing of compact ventilation units with heat pumps.

3 Applicability

This standard applies to the testing of compact ventilation units with heat pumps operating with volume air flows in the range 100 m^3/h to 1200 m^3/h.

3.1 Specifications of the testing equipment

The testing units must include the following equipment:

- Supply and return air fans, possibly an additional outside-air fan for the heat pump with corresponding drives. The operation of the fans must allow for adjustment of measuring points.
- Air-to-air heat exchanger (e.g. plate heat exchanger)
- Heat pump using the exhaust and/or outside air as its heat source.
- Electrical equipment:
  All electrical components must securely mounted and wired in conformity with electrical safety codes to an electrical panel. The testing unit must be capable of being connected to a standard power supply (230 V or 400 V). Suitable electrical connections must be provided to permit separate measurement of the electricity consumed by the compressor and possible electrical heating elements. All fans, compressors and heating elements must be capable of being operated independently.
- Control units with labelled controls.
- Filters
  The outside-air and return air openings must be equipped with at least a Class G3 coarse filter.
- Housing
  The components must be operational and mounted within a housing.
- Documentation
  Instructions for installation, commissioning and operation

In addition to the above items the testing units may also include the following equipment:

- Domestic-hot-water storage
  The domestic-hot-water storage can either be incorporated in the overall housing or set up next to the testing unit
- Bypass for summertime operation
- Electrical heating elements for water heating
- Thermal transfer to a water-based heating system

The tuning and adjustment of the controls (set-points, parameters) is to be undertaken and documented by the client.

4 Scope of the testing

The overall testing procedure consists of:

- Testing the leakage
- Testing the ventilation
- Testing for the filter-bypass-leakage*
- Conducting energy measurements
- Conducting acoustic tests
- Operation and maintenance
- Assessment of hygienic aspects*

*This test is optional and is only to be conducted if required by the client

One or more of the above tests may be omitted if corresponding test reports from equivalent certified testing laboratories are supplied.

5 Concepts and Definitions

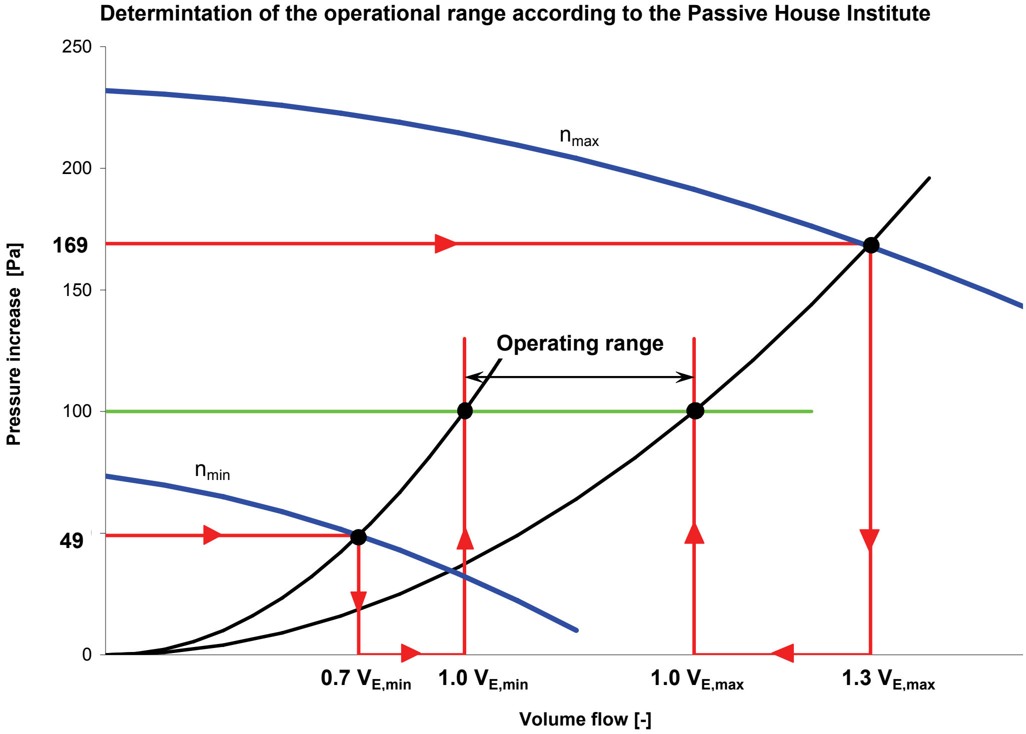
5.1 Definitions

Compact ventilation unit

Compact ventilation units are ventilating units providing supply and return air as well as heat recovery. In addition, such units are equipped with a heat pump coupled to the supply and/or exhaust, using the exhaust and/or the outside air as its heat source. The heat pump delivers heat to the supply air, to domestic hot water storage and/or to another water-based system.

Operating range

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



The maximum volume flow rate is determined at which the unit can deliver a pressure increase of 100 Pa $\times$ 1.3² = 169 Pa in its highest operating level at 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.

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

Operating point

An operating point is a particular state in which the respective temperatures and humidities of outside air and return air are understood to be constant.

Testing chamber

The testing chamber is the chamber in which the unit under test is installed.

If the unit is mounted in a heated area of the building, heat flows from the unit to the testing chamber contribute to the overall heating of the building. Correspondingly, heat flows in the reverse direction during the heating season represent losses.

Leakage

The leakage volume flow rate is the net flow resulting from external and internal leakages which result in warm sucked or blown into the outside air flow and/or supply air flows. Internal leakages result in considerable improved energy performance. Besides that, leaks needed to be regarded from the point-of-view of air hygiene.

Filter-bypass-leakage

Filter bypass leakage consists of air flow bypassing the filters.

Pressure difference

The pressure difference refers to the pressure difference between the outside-air and exhaust-air openings or between the return-air and exhaust-air openings.

As a rule it is assumed that the respective pressure increases across the supply and exhaust channels are the same. Otherwise differences between the two must be explicitly noted.

Temperature ratio (temperature change coefficient)

The temperature ratio is the temperature difference between the inlet and outlet of one of the air channels divided by the temperature difference between the inlets of the two channels.

Power consumption

The power consumption is the mean electrical power consumption of the unit at the main contacts during a specified time interval. It consists of:

- the power consumption of the fans
- the power consumption of all compressors and the power consumption for evaporation, excluding all additional heating devices not used for evaporation.
- the power consumption of all control and safety equipment in the unit.

Test voltage

The test voltage is the voltage at the mains supply during the testing procedure.

6 Test conditions

6.1 Determination of the number of measuring points and volume flows

If the ratio of the maximum to the minimum volume flow is larger than 1.6:1, the range between the minimum and maximum flow must be subdivided into subranges and within each subrange a nominal volume flow 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.

Determination according to the DIBt

The required number of subranges is determined by:

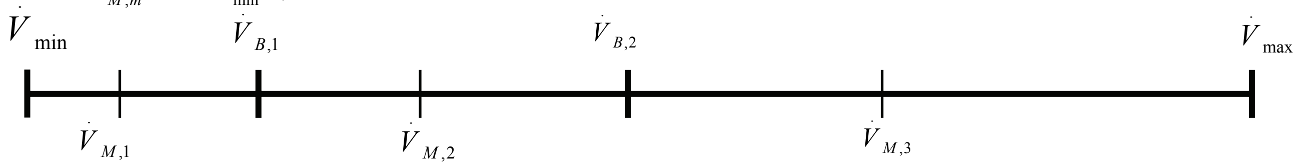
$$n \geq 2.13 \cdot \ln\left(\frac{\dot{V}_{\max}}{\dot{V}_{\min}}\right)$$

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

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

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

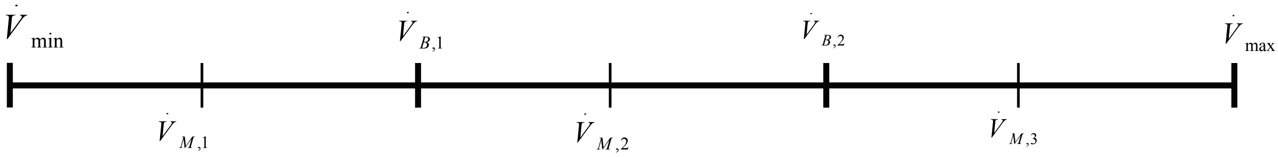
$$\dot{V}_{M,m} \geq \dot{V}_{\min} \cdot \sqrt[2n]{\left(\frac{\dot{V}_{\max}}{\dot{V}_{\min}}\right)^{(2m-1)}}$$



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 mean volume flow value.

Determination according 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 subdivides 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.



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

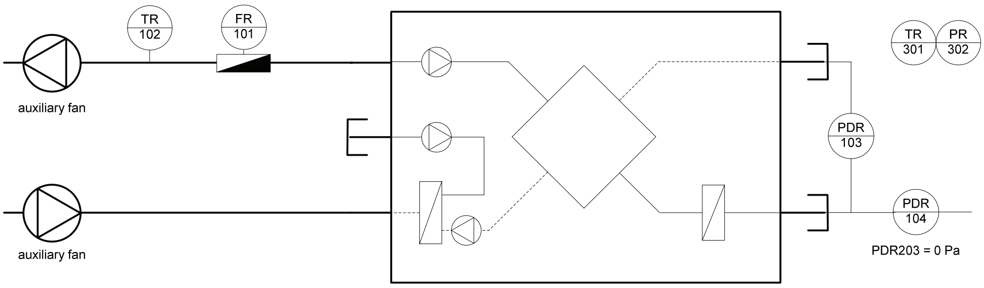
6.2 Testing for air leakage

Internal and external air leakages must be measured according to EN 308 and EN 13141-7. The test consists of creating a pressure difference between the inside and outside of the unit under test and determining the supply air flow (= leakage flow) required to maintain the difference.

Measuring points

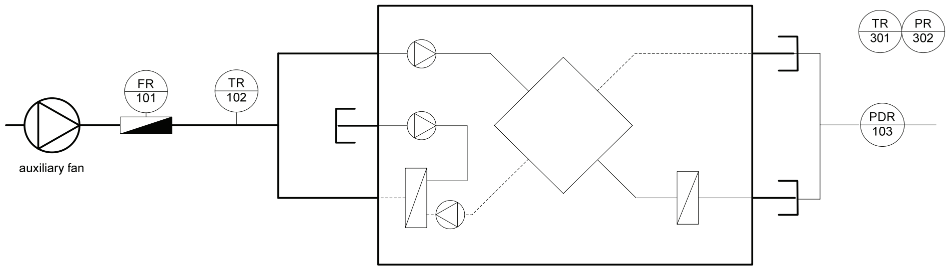
Measurements must be carried out for the following pressure differences:

Internal leakages: 50 Pa / 100 Pa / 150 Pa / 200 Pa



Item	Measured quantity	Instrument / Sensor	Measurement signal
FR101	air flow	bellows gas meter or turbine flow meter	momentum
TR102	leakage-air temperature	PT100	resistance measurement
PDR103	pressure difference between return / exhaust-air channel and outside / supply-air channel	pressure transducer	current or voltage
PDR104	pressure difference relative to surroundings (=0 Pa)	pressure transducer	current or voltage
TR301	temperature of surroundings	PT100	resistance measurement
PR302	air pressure	electronic barometer	voltage

External leakages: -100 Pa / -150 Pa / -200 Pa / -250 Pa / -300 Pa
+100 Pa / +150 Pa / +200 Pa / -+250 Pa / +300 Pa



Item	Measured quantity	Instrument / Sensor	Measurement signal
FR101	air flow	bellows gas meter or turbine flow meter	momentum
TR102	leakage-air temperature	PT100	resistance measurement
PDR103	pressure difference	pressure transducer	current or voltage
TR301	temperature of surroundings	PT100	resistance measurement
PR302	air pressure	electronic barometer	voltage

Measured quantities

external leakage volume flow rate	[m^3/h]
internal leakage volume flow rate	[m^3/h]
pressure difference	[Pa]
air temperature	[°C]
relative humidity	[%]
air pressure	[mbar]
time	[s]

Evaluation of data

The air volume flow rates of external leakages at excess and low pressure of 250 Pa must be specified, both as absolute measurements and as percentages of the maximum specified air flow rate of the unit.

The air volume flow rates of internal leakages at excess and low pressure of 100 Pa must be specified, both as absolute measurements and as percentages of the maximum specified air flow rate of the unit. In addition, the full set of measured external and internal leakages must be shown graphically as curves.

Remarks

During these measurements the fans of the unit under test must be switched off.

Variant: measurement of the internal return air leakage with tracer gas

In cases involving special types of heat recovery (e.g. rotors), special pressure conditions in the unit or at the request of the client, the internal leakage may be ascertained using tracer gas. This procedure determines the leakage mass flow from the return air to the supply air.

6.3 Filter-Bypass-Leakage

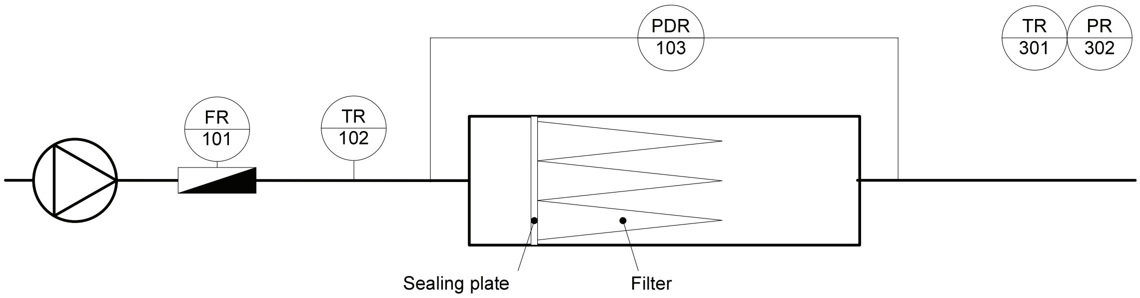
The reference values and classification of bypass leakage at the filters must be ascertained generally according to Paragraph 6 of EN 1886:1998, but with the following special alteration:

First, the pressure differences across the unit's filters as delivered must be measured at maximum volume flow (as specified in §6.1).

Then the reference pressure difference must be defined as double the measured quantity or 25 Pa, whichever is greater.

Measuring points

The measurements must be carried out at 5 pressure differences in the range from 50% to 150% of the reference pressure difference.



Item	Measured quantity	Instrument / Sensor	Measurement signal
FR101	air flow	bellows gas meter or turbine flow meter	momentum
TR102	leakage-air temperature	PT100	resistance measurement
PDR103	pressure difference across the filter or the sealing plate	pressure transducer	current or voltage
TR301	temperature of surroundings	PT100	resistance measurement
PR302	air pressure	electronic barometer	voltage

Measured quantities

bypass leakage volume flow rate	[m^3/h]
pressure difference	[Pa]
air temperature	[°C]
air pressure	[mbar]
time	[s]

Evaluation of data

The evaluation and classification of the data must be in accordance with Paragraph 6 of EN 1886:1998.

6.4 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. The air flow required for the measurement must first be fed through a settling chamber connected to the inlet of the unit being tested.

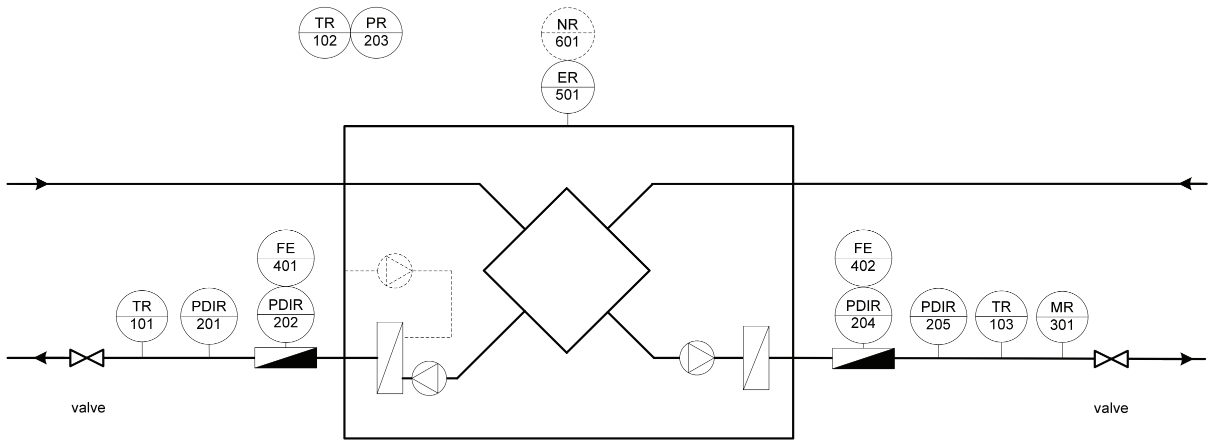
Measuring points

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 more than 30 Pa.

For units with **continuously variable-speed fans** the maximum and minimum volume flow rates must be established according to §5.1. 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 more than 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 according to §5.1. 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. In addition, two further characteristic curves must be recorded for the unit operating under constant volume flow-rate control, flow rates being established according to §5.1. Data points for each curve must be obtained within the stable operating range for pressure increments not more than 50 Pa. The volume flow rates between the minimum and maximum flows must be measured in accordance with §6.1

The points of the single characteristics are operated in a stable area in a distance of max. 50 Pa. In-between the lower and upper volume flow rate the volume flow rates are measured according to chap. 5.1.



Item	Measured quantity	Instrument / Sensor	Measurement signal
TR 101	air temperature	PT100	resistance measurement
TR 102	temperature of surroundings	PT100	resistance measurement
TR 103	air temperature	PT100	resistance measurement
PDIR 201	pressure difference (pressure increase)	pressure transducer	current or voltage
PDIR 202	pressure difference (pressure increase)	pressure transducer	current or voltage
PR 203	air pressure	pressure transducer	current or voltage
PDIR 204	pressure difference (pressure increase)	pressure transducer	current or voltage
PDIR 205	pressure difference (pressure increase)	pressure transducer	current or voltage
MR 301	relative humidity	Hygroclip	voltage
FE 401	volume flow rate	Varycontrol and pressure transducer	current or voltage
FE 402	volume flow rate	Varycontrol and pressure transducer	current or voltage
ER 501	electrical consumption	Infratek	
NR 601	fan speed		

Measured quantities

volume flow rate	[m^3/h]
pressure difference	[Pa]
air temperature	[°C]
relative humidity	[%]
fan speed	rpm
air pressure	[mbar]
electrical quantities	[W; V; A]

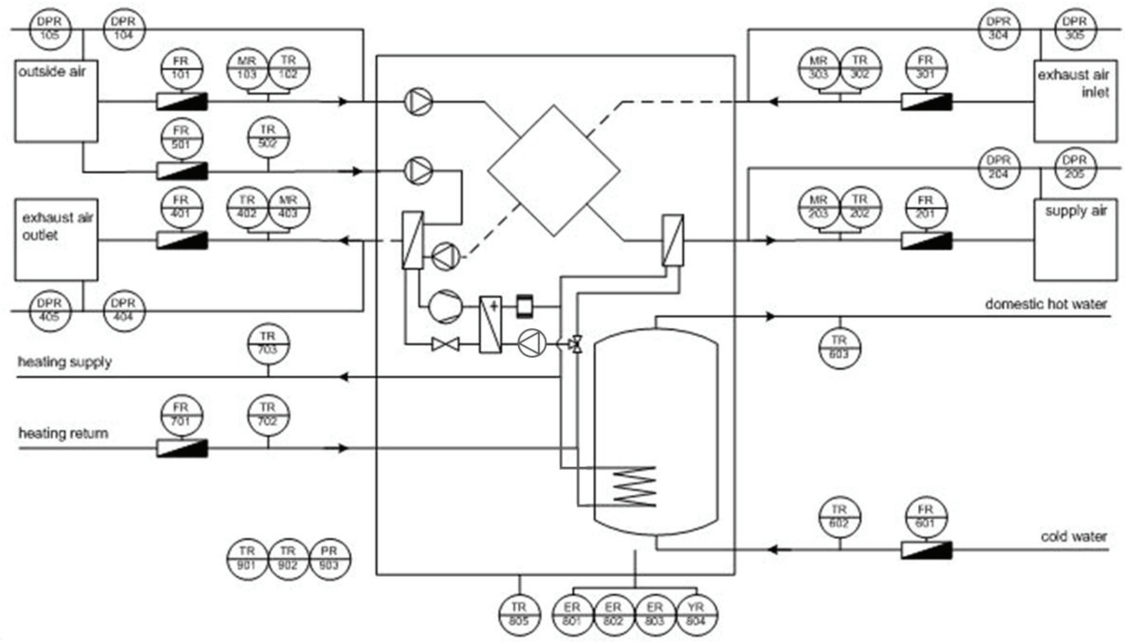
Evaluation of data

The measurements must be presented as characteristic curves in a plot of air volume flow rate vs. Pressure.

6.5 Thermodynamic tests

6.5.1 Thermal tests

Test points in accordance with EN 14511



Item	Measured quantity	Instrument / Sensor	Measurement signal
FR101	outside-air volume flow rate	Varycontrol und pressure transducer	current or voltage
TR102	outside-air temperature	5 PT100	resistance measurement
MR103	outside-air humidity	Hygroclip	voltage
DPR104	outside-air pressure drop before unit	pressure transducer	current or voltage
DPR105	outside-air pressure difference compensating chamber / surroundings	pressure transducer	current or voltage
FR201	supply-air volume flow rate	Varycontrol und pressure transducer	current or voltage
TR202	supply-air temperature	5 PT100	resistance measurement
MR203	supply-air humidity	Hygroclip	voltage
DPR204	supply-air pressure drop after unit	pressure transducer	current or voltage
DPR205	supply-air pressure difference compensating chamber / surroundings	pressure transducer	current or voltage
FR301	return-air volume flow rate	Varycontrol und pressure transducer	current or voltage
TR302	return-air temperature	5 PT100	resistance measurement
MR303	return-air humidity	Hygroclip	voltage

Item	Measured quantity	Instrument / Sensor	Measurement signal
DPR304	return-air pressure drop before unit	pressure transducer	current or voltage
DPR305	return-air pressure difference compensating chamber / surroundings	pressure transducer	current or voltage
FR401	exhaust-air volume flow rate	Varycontrol und pressure transducer	current or voltage
TR402	exhaust-air temperature	5 PT100	resistance measurement
MR403	exhaust-air humidity	Hygroclip	voltage
DPR404	exhaust-air pressure drop after unit	pressure transducer	current or voltage
DPR405	exhaust-air pressure difference compensating chamber / surroundings	pressure transducer	current or voltage
FR501	outside-air volume flow rate for HP	Varycontrol und pressure transducer	current or voltage
TR502	outside-air temperature for HP	5 PT100	resistance measurement
FR601	cold-water flow rate	MID magn./ind.	current or voltage
TR602	cold-water temperature	PT100	resistance measurement
TR603	domestic hot-water temperature	PT100	resistance measurement
FR701	flow rate of hot-water heating	MID magn./ind.	current or voltage
TR702	heating-return temperature	PT100	resistance measurement
TR703	heating-supply temperature	PT100	resistance measurement
ER801	power consumption of unit	signal converter	current or voltage
ER802	power consumption of HP	signal converter	current or voltage
ER803	monitoring of input voltage		
YR804	condensate removal		
TR805	surface temperature of unit	5 PT100	resistance measurement
TR901	Temperature of surrounding air	3 PT100	resistance measurement
TR902	surface temperature of walls	3 PT100	resistance measurement
PR903	air pressure	electronic barometer	voltage

	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 $\theta_{wb,in}$(°C)	Inlet dry bulb tem perature $\theta_{ea,in}$(°C)	Inlet wet bulb tem perature $\theta_{wb,ea,in}$(°C)	Inlet tempera-ture $\theta_{w,in}$(°C)	Outlet tempe-rature $\theta_{w,out}$(°C)	Outlet tempe rature $\theta_{w,,out}$(°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/kg$_{\text{dry air}}$ per outside temperature step based on the given rating point 20(12) in EN 14511

The testing is carried out for different volume flow rates dependent on the application determined in the air-flow testing for space heating.

variant: ground-to-air heat exchanger

	Outdoor air temperature (°C)			Exhaust air temperature (°C)		Heating		DHW	Comment
	Dry bulb outside tempera-ture θ_{oa} (°C)	Inlet wet bulb tem-perature $\theta_{wb,in}$ (°C)	Inlet tem-perature after ground-to-air-heat exchanger $\theta_{oa,in}$ (°C)	Inlet dry bulb tem-perature $\theta_{ea,in}$ (°C)	Inlet wet bulb tem-perature $\theta_{wb,ea,in}$ (°C)	Inlet tempe rature $\theta_{w,in}$ (°C)	Outlet tempe-rature $\theta_{w,out}$ (°C)	Outlet tempe-rature $\theta_{w,out}$ (°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/kg$_{\text{dry air}}$ per outside temperature step based on the given rating point 20(12) in EN 14511

The testing is carried out for different volume flow rates dependent on the application determined in the air-flow testing for space heating.

The testing procedure for thermodynamic testing is based essentially on the guidelines of the DIBt (June, 2001) and EN 308 with the following exceptions:

- The mass flows of the outside air and exhaust air are to be adjusted on the unit (in the absence of automatically controlled fans) in the context of the measurement precision required.
- The volume flows of not only the outdoor and return air but also of the exhaust and supply air are to be measured.

Variant: air conditions according to DIBt

	Measurement point 1	Measurement point 2	Measurement point 3
exhaust-air inlet temperature	21°C	21°C	21°C
exhaust-air inlet humidity	36% rel. humidity	46% rel. humidity	56% rel. humidity
outside-air temperature	-3°C	4°C	10°C
outside-air humidity	80% rel. humidity	80% rel. humidity	80% rel. humidity

6.5.2 Air volume flows

Variant 1

Measurements must be carried out on the air volume flows described in §6.1. The pressure difference required for the measurements (using external pressure) is **100 Pa in general**. The corresponding external pressure gradient must be equally distributed between the inlet and outlet of the tested unit (i.e. 50% each).

Variant 2

The volume flows are adjusted according the client's specifications. Nevertheless they must lie within the range defined in Variant 1.

6.5.3 Measured quantities

Quantity		outside air	supply air	return air	exhaust air	unit	surroundings
temperature	[°C]	x	x	x	x	-	x
relative humidity	[%]	x	x	x	x	-	-
volume flow rate	[m^3/h]	x	x	x	x	-	-
surface temperature	[°C]	-	-	-	-	x	x

pressure difference	[Pa]
air pressure	[Pa]
electrical quantities, HP	[W; V; A]
electrical quantities, unit	[W; V; A]

Evaluation of data

The following quantities must be ascertained and presented both in tables and in diagrams:

- temperature change coefficient
- heat recovery coefficient
- humidity change coefficient
- electrical efficiency ratio
- electro-thermal amplification factor

Remarks

The overall electrical power consumption of the unit (including controls) must be determined both during the thermodynamic tests and in standby mode.

If the unit is set up for operation in cold climatic conditions (i.e. anticipated outside temperatures of -10°C or lower), then an additional test according to EN 308 must be carried out.

6.5.4 Ventilation operation solely with heat recovery

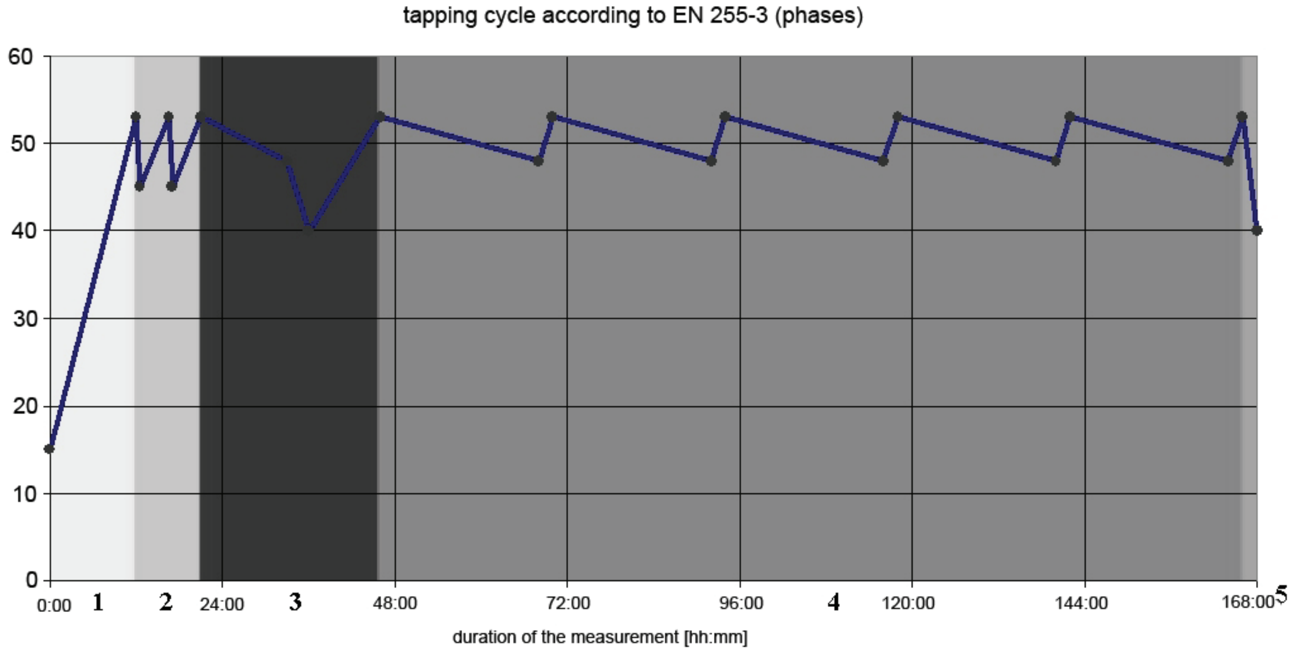
The heat pump and electrical heating elements must be switched off and the hot-water thermal storage should have an external temperature of ±5 K. The thermal measurement must be carried out for all of the prescribed temperatures and volume flows and must be carried out at constant air temperatures and humidities.

6.5.5 Heat pump operation for supply air heating

Heat must be stored in the hot-water thermal storage so as to ensure an operational temperature of 60°C. No hot water may be drawn from an external water-based system nor any thermal energy delivered to it. Any heating elements in the system must be switched off. The ventilation unit must be operated with only the heat pump running, and the heat pump must be operated solely to heat the supply air. The thermal measurements must be carried out for all of the prescribed temperatures and volume flows and must be carried out at constant air temperatures and humidities. In this connection the level of supply-air temperature is not to be limited.

6.5.6 Operation of the heat pump for domestic hot water

This test must be carried out according to EN 255-3



1 initial heating phase
2 determination of the coefficient of performance (COP) for providing domestic hot water
3 first stage of determining the domestic hot water temperature
4 determining the power consumption during the operational phase
5 second stage of determining the domestic hot water temperature - determination of the maximum deliverable quantity

Phase 1	**Pre-heating phase**	
Measured quantity	**Interval**	**Definition**
Duration of the initial heating phase until the thermostat switches the compressor off		
t_h	5 min	
Power consumption of the compressor during the heating phase		
P_{el}	5 min	
Calculated quantity		**Formula**
Pre-heating energy		
$\mathbf{W_{eh}}$		$P_{el} \cdot t_h$

Period 2 Determination of the performance coefficient (COP)

Measured quantity	**Interval**	**Definition**
domestic hot-water temperature		
θ_{wh}	max. 10s	
cold-water temperature		
θ_{wc}	max. 10s	
power consumption during the second re-heating phase		
P_{el}	max. 10s	
volume flow		
q_{wh}	max. 10s	$V_n \leq 400dm^3 \rightarrow q_{wh} = 0.2\ dm^3/s$: $V_n \geq 400dm^3 \rightarrow q_{wh} = 0.0005 \cdot V_n\ dm^3/s$
Duration of compressor operation during the cooling and re-heating phases		
t_t'	max. 10s	
Duration of compressor operation during the cooling and re-heating phases		
t_t	max. 10s	

→ **Drain tank to half of its nominal volume V_n**

Calculated quantity	**Formula**	
Energy consumption required by the compressor during the re-heating phase		
W_{et}	$P_{el} \cdot t_t$	
Thermal energy of the hot water removed when last draining the tank to half volume		
Q_t	$Q_t = \int_0^{t_t} \rho_{wh} \cdot c_{pw} \cdot q_{wh} \cdot (\vartheta_{wh} - \vartheta_{wc})\, dt$	
coefficient of performance COP of the providing domestic hot water		
COP_t	$COP_t = \frac{Q_t}{W_{et} - P_{es} \cdot t_t}$	P_{es} from Period 4

→ **The two values for Q_t (from t_t and t_t') may differ by at most 10%.**

Tolerances and required precision of the “tapping model according to EN 255-3

Deviation from set-point value

Measured quantity	**Individual measurement**	**Arithmetic mean**
domestic hot-water temperature	± 1K	± 1K
domestic hot-water flow	± 5%	± 10%
air temperature	± 0.3K	± 1K
air flow	± 5%	± 10%
room temperature	± 1K	± 2K

Measurement precision

Measured quantity	**Measurement uncertainty**	**Units**
temperatures (domestic hot water and air)	± 0.2K	[°C]
domestic hot-water flow	± 2%	[l/h]
air flow	± 5%	[l/h]
domestic hot-water thermal energy	± 5%	[kWh]
air pressure difference	± 5Pa ($\Delta p \leq$ 100Pa) ± 5% ($\Delta p >$ 100Pa)	[Pa]
voltage	± 0.5%	[V]
current	± 0.5%	[A]
electrical power	± 1%	[W]
electrical energy	± 1%	[kWh]
temperature of surroundings	± 1K	[°C]
time	± 10s	[s]

6.5.7 Operation of the heat pump of delivery of energy to a water-base system

The supply and return air channels must be in operation.

Heat must be stored in the hot-water thermal storage so as to ensure an operational temperature of 60°C. No hot water may be tapped and heat may be delivered to the supply air. Any heating elements in the system must be switched off. The ventilation unit must be operated with only the heat pump running, and the heat pump must be operated solely to heat the external system. The thermal measurements must be carried out for all of the prescribed temperatures and volume flows and must be carried out at constant air temperatures and humidities. The appropriate value for the water volume flow must be provided by the supplier and the supply temperature must be either 35° or 45°C.

6.5.8 Testing the frost-protection system (optional)

Starting with the thermodynamic measurement point the outside-air temperature must be lowered from -3°C (return air 21°C und 36 % rel. hum.) to max. –12°C. In this connection the rate of lowering the outside-air temperature should not exceed 1 K every 5 minutes.

To test whether the supplier's frost-protection strategy is sufficient, a cyclic test must be carried out.

Cyclic test

At outside-air temperature 4 K 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 should be recorded for a duration of at least 30 minutes, so as to include at least 3 switching cycles.

If at the end of the above time the recorded the data curves show insufficient evidence of repeated cyclic behaviour (e.g. constancy of exhaust temperature and pressure drop), the test must be continued for three more switching cycles.

At the conclusion of this test, the unit must continue to operate until the frost-protection device is again re-activated, at which point the unit must be immediately switched off and opened for visual inspection of the heat exchangers. Any evidence of condensation or icing should be noted, particularly when the measurement data are not convincingly cyclic. Any icing would clearly indicate that the frost-protection strategy provided is unsuitable for the unit.

This cyclic test must be carried out at the volume flow corresponding to one of the measurement points in the thermodynamic test and at a return relative air humidity of 36%. If the operational range of the unit results in two prescribed measurement points, then the lower of the two volume flows should be used for the cyclic test. In case of three measurement points the intermediate volume flow should be used.

6.6 Acoustic Tests

Acoustic tests must be carried out for sound emission through the housing of the unit under test and for sound emission into the connected ventilation ducts. Acoustic tests must be carried out in accordance with the following norms:

- Sound Power Determination using sound intensity methods
  ISO 9614-1, ISO 9614-2, ISO 9614-3,

- Determination of sound power radiated into a duct by fans - In-duct method
  EN 25136 translation of ISO 5136

- Air conditioners, heat pumps and dehumidifiers with electrically driven compressors.
  Measurement of airborne noise. Determination of the sound power level
  ENV 12102

For the acoustical measurements the test unit must be mounted in an air-conditioned acoustic chamber in accordance with the manufacturer's instructions. In the case of units with heat pumps the test must be carried in real operation – i.e. with conditioned air. The outsides of the air ducts must be covered in damping material in order to reduce the sound emitted from the ducts.

The measurements must be made as follows:

Sound source	**Volume flow**	**Pressure difference**
emission from unit	max. volume flow	100 Pa
exhaust & outside air ducts	max. volume flow	100 Pa
supply & return air ducts	min. & max. volume flow	70/100/130 Pa

Measurement points

The sound power values must be determined using the maximum air volume flow rate determined in §6.1. Both the sound emission through the housing and the sound emission of the fans into the ducts must be measured.

Measured quantities

linear sound intensity level (frequency band analysis)	[dB]
surface area to be scanned	[m^2]
volume flow rate	[m^3/h]
temperatures	[°C]
relative humidity	[%]
air pressure	[mbar]

Evaluation of data

The following measured data must be presented in tables and in diagrams:

- sound intensity level in octave bands and as overall value
- A-weighted sound intensity level in octave bands and as overall value

6.6.1 Tested operational modes

- - ventilation only (heat pump off)
- - ventilation and heat pump

6.7 Safety testing

In addition to checking any safety certificates and proofs of compliance provided, the following safety tests must be undertaken:

Mechanical safety (inspection)

- check of protection against touching moving parts
- check of sharp edges
- check whether the fans are suitable mounted and protected

Electrical safety (inspection)

- check of protection against direct or indirect contact
- check of the mechanical protection of the wiring
- check that the earthing is properly connected

6.8 Operation and service

The assessment, which must be conducted using a check list, evaluates the completeness and quality of the documentation.

Documentation

The technical documentation may be provided in draft form. The documentation must be provided at least in German or English.

Data sheets

- scope of possible use permissible temperatures, requirements regarding air quality (e.g. suited for kitchen return air) and water quality, …
- provisional tables and/or diagrams with flow data, thermal data, electrical data and acoustical data
- masses and weights
- specifications of the unit's construction (materials, surface finishing, …)
- thermal storage
  volume, materials, surface coatings, thermal insulation
- refrigerant
  type and quantity
- filter types and classes

Planning and installation manuals

– scale schematic diagram with labelling and specification of the connections

– specifications regarding setup (damping elements, requirements regarding the room, accessibility for service)

– instructions for commissioning

Operation manual and maintenance

– notes regarding safety

– description of the switches and controls and their function

– instructions regarding possible problems and troubleshooting procedures

– addresses of service and repair agencies

– specifications regarding disposal (agencies responsible, refrigerant, …)

– service plan (filter change, cleaning, heat exchangers, …)

– specification and sources for the commonest replacement items (filter, fuses)

Labelling on the unit

Easily visible and durable labelling, at least in German or French for:

– designation of all switches and controls

– supplier’s address

– specifications regarding filter change

– type designation plates for the unit as a whole as well as for the fans and compressor

– identification of materials relevant to questions of disposal

Checking the functions

– opening and closing the service hatches

– accessibility for filter change

– accessibility for checking and cleaning the heat exchangers, the fan drive wheels, condensation tray and the housing

6.9 Hygienic investigation

Air-hygiene investigations must be carried out based on the VDI 6022 standard (corresponds to SWKI 2003-5).

7 Test reports

Three different reports are required:

Test Report A

This report contains all test documentation. It is an integral part of the testing programme and may only be given to the client.

Test Report B

This report contains all test documentation as in Test Report A. In addition a report must be prepared based on the testing and approval criteria of the DIBt.

B RECALCULATION OF SOLAR ENERGY ON TILTED SURFACE

The following calculation scheme is taken from [B 1] and [B 2]. For the consideration of the components of the diffuse radiation an isotropic sky model is used, i.e. diffuse irradiation consists of only one component and is not differentiated in isotropic and circumsolar irradiation and horizon brightening.

Input data

- Geographical longitude of the site λ
- Geographical latitude of the site ϕ (north positive, south negative)
- Monthly global solar irradiation on the horizontal surface G_h
- Orientation of the collector α (collector azimuth => east negative, west positive)
- Inclination of the collector β (0° horizontal, 90° vertical)
- Month of the year

Output data

- Solar radiation on tilted surface

According to in chap. 4.1.4 of the main part of the report the solar irradiation on tilted surface is calculated by the equation

$$G_t = G_{bh} \cdot \frac{\cos\theta}{\cos\theta_z} + G_{dh} \cdot \frac{1+\cos\beta}{2} + G_h \cdot \rho \cdot \frac{1-\cos\beta}{2} \quad [-]$$

eq. B 1

where

G_t	total solar irradiation on tilted surface in the month	[W/m^2]
G_{bh}	direct irradiation on horizontal surface	[W/m^2]
G_{dh}	fraction of diffuse irradiation from sky	[W/m^2]
G_h	fraction of diffuse irradiation reflected from the ground	[W/m^2]
ρ	reflection coefficient of the ground	[-]
β	inclination of surface	[rad]
θ	incidence angle of inclined surface	[rad]
θ_z	zenith angle	[rad]

In the following, the different variables of the equation are determined.

The monthly radiation sum is recalculated to a daily sum by division by the number of days of the respective month. The calculation is done for the day 15 of each month, since this day is the average value for the change of the solar coordinates zenith angle and declination of the sun during the month.

With the geographical coordinates and the number of day 15 of the respective month in the year, the declination of the sun, e.g. the season of the year can be calculated by the empiric equation (taken from [B 2])

$$\sin\delta = 0.3978 \cdot \sin(x-77.51° + 1.92*\sin x)$$

eq. B 2

where

δ = declination of the sun (positive northern hemisphere, negative southern hemisphere) [°]

x = time of the year [°]

x specifies the time in the year and is calculated according to the following empiric equation

$$x = 0.9856*(n_{month}*30-15) - 2.72$$

eq. B 3

where

n_{month} = number of the month in the year (i.e. 1 corresponds to January)

With the declination, the hour of sunset can be calculated according to the equation

$$\cos \omega_s = -\tan\phi \cdot \tan \delta$$

eq. B 4

where

ω_s = hour angle of sunset [rad]

ϕ = geographical latitude [rad]

δ = declination angle [rad]

To evaluate the ratio between the solar radiation on the horizontal and the solar radiation on the tilted surface $R_b = \cos \theta/\cos \theta_z$ the hourly theoretical radiation based on the solar constant for the horizontal and the tilted surface are calculated. Therefore, the hourly zenith angle is calculated according to the equation

$$\cos \theta_z = \cos\phi \cdot \cos \delta \cdot \cos \omega + \sin \phi \cdot \sin \delta$$

eq. B 5

where

θ_z	zenith angle	[rad]
ϕ	geographical latitude	[rad]
ω	hour angle	[rad]
δ	declination of the sun	[rad]

and the solar azimuth angle is calculated according to the following equation

$$\cos |\gamma_s| = (\sin\phi \cdot \cos \theta_z - \cdot \sin \delta) /(\cos \phi \cdot \sin \theta_z)$$

eq. B 6

where

γ_s	azimuth angle of the sun	[rad]
θ_z	zenith angle	[rad]
ϕ	geographical latitude	[rad]
δ	declination of the sun	[rad]

The incidence angle can be calculated according to the equation

$$\cos \theta = \cos \theta_z \cos \beta + \sin \theta_z \cdot \sin \beta \cdot (\cos(\gamma - \gamma_s)$$

eq. B 7

where

θ	incidence angle of receiving surface (collector)	[rad]
θ_z	zenith angle	[rad]
β	inclination of surface	[rad]
γ	azimuth angle of the surface	[rad]
γ_s	azimuth angle of the sun	[rad]

The values are calculated in hourly resolution. The theoretical radiation on the horizontal is calculated with the average value of the solar constant G_o and the hourly incidence angle on the horizontal according to the equation

$$G_{bh} = G_0 \cdot \cos \theta_z$$

eq. B 8

where

G_{bh}	direct radiation on horizontal surface	[W/m^2]
G_0	average solar constant (1367 W/m^2)	[W/m^2]
θ_z	zenith angle	[rad]

and respectively the direct radiation on the tilted surface by the equation

$$G_{bt} = G_0 \cdot (\cos \theta / \cos \theta_z)$$

eq. B 9

where

G_{bt}	direct radiation on tilted surface	
G_0	average solar constant (1367 W/m^2)	[W/m^2]
θ	incidence angle of receiving surface (collector)	[rad]
θ_z	zenith angle	[rad]

With the daily sums of the respective radiation on horizontal and tilted surface the ratio R_b can be calculated by the equation

$$R_b = \frac{\sum_{day} G_{bh}}{\sum_{day} G_{bt}}$$

eq. B 10

where

R_b = ratio of solar radiation on horizontal and on tilted surface [-]

G_{bh} = hourly solar radiation on horizontal surface [W/m^2]

G_{bt} = hourly solar radiation on tilted surface [W/m^2]

For the evaluation of a clearness index, which describes the ratio between the measured solar radiation on horizontal and the theoretical solar radiation on horizontal based on the solar constant and the site, the global solar radiation on the horizontal can be calculated according to the equation

$$G_{h0} = 24/\pi \cdot (\cos \phi \cdot \cos \delta \cdot \sin\omega_s + \omega_s \cdot \sin \phi \cdot \sin\delta)$$

eq. B 11

where

G_h = global solar radiation on horizontal [W/m^2]

ω_s = hour angle of sunset [rad]

ϕ = geographical latitude [rad]

δ = declination [rad]

The clearness index is then calculated according to

$$K_T = \frac{G_h}{G_{h0}}$$

eq. B 12

where

K_T = clearness factor for monthly solar irradiation values [-]

G_h = measured global solar irradiation on horizontal [W/m^2]

G_{h0} = theoretical global solar irradiation on horizontal [W/m^2]

The ratio of diffuse to global on horizontal surface is given by the Erbs correlation dependent on the clearness factor and the hour angle of sunset according to the equation

For $\omega_s < 81.4°$ and $0.3 \le K_T \le 0.8$

$$\frac{G_{dh}}{G_h} = 1.391 - 3.56 \cdot K_T + 4.189 \cdot K_T^2 - 2.137 \cdot K_T^3$$

eq. B 13

For $\omega_s > 81.4°$ and $0.3 \le K_T \le 0.8$

$$\frac{G_{dh}}{G_h} = 1.311 - 3.022 \cdot K_T + 3.427 \cdot K_T^2 - 1.821 \cdot K_T^3$$

eq. B 14

where

K_T = clearness factor for monthly radiation values [-]

G_{dh} = diffuse solar radiation on horizontal [W/m^2]

G_h = measured global solar radiation on horizontal [W/m^2]

By subtracting the diffuse radiation on horizontal from the global, the direct radiation on horizontal can be calculated

$$G_{bh} = G_h - G_{dh}$$

eq. B 15

where

G_{bh} = direct solar radiation on horizontal [W/m^2]

G_{dh} = diffuse solar radiation on horizontal [W/m^2]

G_h = measured global solar radiation on horizontal [W/m^2]

References

[B 1] J. Duffie, W. Beckman: Solar engineering of thermal processes, Second edition, John Wiley & sons, inc, New York, 1991, US

[B 2] Deutscher Wetterdienst (DWD), Berechnung der Stundensummen der Sonnenstrahlung auf geneigte Ebenen bei wolkenlosem Himmel, Hamburg, 1990, DE

C COP CORRECTION WITH FIXED EXERGETIC EFFICIENCY

The method is outlined in [C 1]. The method assumes that the thermodynamic quality of the process stays constant over the whole operating range. Thermodynamic quality of a process can be expressed by the exergetic effiiciency as ratio between the real COP of the process and an ideal COP of the Carnot process. However, in real processes, the exergetic efficiency does not stay constant over the entire operating range, so the correction is only an approximation which shows good results near the standard test point. Accuracy deteriorates with increasing distance from the test point, and therefore the method is best suited for temperature correction where temperatures are not too far from the test point.

The exergetic efficiency can be calculated according to the equation

$$\eta_{ex} = \frac{COP}{COP_C} \qquad [-]$$

eq. C 1

where

η_{ex}	exergetic efficiency	[-]
COP	coefficient of performance	[-]
COP_C	Carnot COP	[-]

The Carnot COP for a heat pump is calculated according to the equation

$$COP_C = \frac{T_h}{T_h - T_c} \qquad [-]$$

eq. C 2

where

COP_C	Carnot COP	[-]
T_h	temperature on the hot heat pump process side (sink)	[K]
T_c	temperature on the cold heat pump process side (source)	[K]

Both source and sink temperature can be considered by this approach. The advantage of the method is that only one test point is needed. In case of testing according to EN 255-3, for instance, where only one test point is defined, an interpolation to correct the COP values for different source or sink temperature is not possible, but a correction with the fixed exergetic efficiency is still applicable.

The effective COP for a different source and/or sink temperature can be calculated according to the equation

$$COP_{op} = COP_{nom} \cdot \frac{COP_{C,op}}{COP_{C,nom}} \qquad [-]$$

eq. C 3

where

COP_{op}	COP at source and sink temperature in operation	[-]
COP_{nom}	COP at measured standard test point	[-]
$COP_{C,op}$	Carnot COP at source and sink temperature in operation	[-]
$COP_{C,nom}$	Carnot COP at measured standard test point	[-]

For electrically-driven heat pumps, the correction factor f_T to take into account the impact of different temperatures can be calculated according to the following equation

$$f_T = \frac{COP_{C,op}}{COP_{C,nom}} = \frac{T_{sk,out,op} \cdot (\theta_{sk,out,nom} - \theta_{src,in,nom})}{T_{sk,out,nom} \cdot (\theta_{sk,out,op} - \theta_{src,in,op})} \qquad [-]$$

eq. C 4

where

f_T	correction factor for temperature deviation from measured standard test point	[-]
$COP_{C,op}$	Carnot COP at source temperature in operation	[-]
$COP_{C,nom}$	Carnot COP at measured standard test point	[-]
$T_{sk,out,op}$	outlet temperature on sink side in operation	[K]
$T_{sk,out,nom}$	outlet temperature on sink side at measured standard test point	[K]
$T_{sk,in,nom}$	inlet temperature on sink side at measured standard test point	[K]
$\theta_{src,in,op}$	inlet temperature on source side in operation	[°C]
$\theta_{src,in,nom}$	inlet temperature on source side at measured standard test point	[°C]
$\theta_{sk,out,op}$	outlet temperature on sink side in operation	[°C]
$\theta_{sk,out,nom}$	outlet temperature on sink side at measured standard test point	[°C]
$\theta_{src,in,nom}$	inlet temperature on sink side at measured standard test point	[°C]

References

[C 1] T. Afjei et al: Low cost low temperature heating with heat pumps, phase 4: Technical Handbook, Final report of SFOE research project, Dec. 2000, CH

D DETAILED CALCULATION RESULTS FOR BUILDINGS TYPES

D.1 MINERGIE-P building with air-heating system

The air heating system has the characteristic given in Tab. 23 in the main part of the report.

Tab D. 1: Results of the calculation for planning data Zeiningen ***for air-heating system with constant supply temperature*** *(calculated values, not field monitoring)*

Air heating system			
	Space heating	**DHW**	**Total**
System boundary heat pump			
Energy need of the distribution system [kWh_{th}]	6344	2750	9094
Reduction of space heating energy requirement by heat recovery [kWh_{th}]	3689	-	3689
DHW storage losses [kWh_{th}]	-	461	461
Heat supplied by heat pump [kWh_{th}]	2621	3211	5832
Evaluation time [h]	4134	8760	-
Electrical energy to cover storage losses [kWh_{th}]	-	263	263
Total electricity heat pump [kWh_{el}]	1483	1563	3046
SPF-HP by weighting over bins [-]	**1.77**	**2.05**	**1.91**
System boundary SPF-G			
Heat supplied by the back-up heater [kWh_{th}]	34	0	34
Electrical back-up energy input [kWh_{el}]	36	0	36
Running time of the heat pump [h]	1992	2305	4297
Auxiliary electrical energy stand-by [kWh_{el}]	26	0 (included in EN 255-3)	26
Heat supplied to air-heating/DHW [kWh_{th}]	2655	3211	5866
Electrical energy “generator” [kWh_{el}]	1545	1563	3108
SPF-G [-]	**1.72**	**2.05**	**1.89**
System boundary SPF-S			
Electrical energy heat recovery (fans and control – only winter operation) [kWh_{el}]	378	0	378
Sink pump/storage loading pump [kWh_{el}]	0	0 (included in EN 255-3)	0
Total used heat [kWh_{th}]	6344	2750	9094
Electrical energy “system” [kWh_{el}]	1923	1563	3486
SPF-S [-]	**3.30**	**1.76**	**2.61**

[1)] recovered ventilation losses by the heat recovery not subtracted

*Tab D. 2: Results of the calculation for planning data Zeiningen for **air-heating system with ground-to-air heat exchanger***

Air heating system with ground-to-air heat exchanger			
	Space heating	**DHW**	**Total**
System boundary heat pump			
Energy need o the distribution system [kWh_{th}]	6344	2750	9094
Reduction of space heating energy requirement by heat recovery [kWh_{th}]	3931	-	3931
DHW storage losses [kWh_{th}]	-	461	461
Heat supplied by heat pump [kWh_{th}]	2410	3211	5621
Evaluation time [h]	4134	8760	-
Electrical energy to cover storage losses [kWh_{el}]	-	263	263
Total electricity heat pump [kWh_{el}]	1315	1535	2850
SPF-HP by weighting over bins [-]	**1.83**	**2.09**	**1.97**
System boundary SPF-G			
Heat supplied by the back-up heater [kWh_{th}]	3	0	3
Electrical back-up energy input [kWh_{el}]	3	-	3
Running time of the heat pump [h]	1743	2235	3978
Auxiliary electrical energy stand-by [kWh_{el}]	29	0 (included in EN 255-3)	29
Heat supplied to air-heating/DHW [kWh_{th}]	2413	3211	5624
Electrical energy “generator” [kWh_{el}]	1347	1535	2882
SPF-G [-]	**1.79**	**2.09**	**1.95**
System boundary SPF-S			
Electrical energy heat recovery (fans and control – only winter operation) [kWh_{el}]	425	-	425
Sink pump/storage loading pump [kWh_{el}]	-	0 (included in EN 255-3)	0
Total used heat [kWh_{th}]	6344	2750	9094
Electrical energy “system” [kWh_{el}]	1772	1535	3307
SPF-S [-]	**3.58**	**1.79**	**2.75**

1) recovered ventilation losses by the heat recovery not subtracted

D.2 MINERGIE building with floor - heating system

*Tab D. 3: Results of the calculation for planning data Zeiningen for **floor-heating system** (* recovered ventilation losses not subtracted)*

Floor heating system			
	Space heating	**DHW**	**Total**
System boundary heat pump			
Energy requirement [kWh_{th}]	6344	2750	9094
Reduction of space heating energy requirement by heat recovery [kWh_{th}]	3689		3689
DHW storage losses [kWh_{th}]		461	461
Heat supplied by heat pump [kWh_{th}]	2653	3211	5864
Evaluation time [h]	4134	8760	-
Electrical energy to cover storage losses [kWh_{el}]	-	263	263
Total electricity heat pump [kWh]	884	1563	2447
SPF-HP by weighting over bins [-]	**3.00**	**2.05**	**2.40**
System boundary SPF-G			
Heat supplied by the back-up heater [kWh_{th}]	2	0	2
Electrical back-up energy input [kWh_{el}]	2	0	2
Running time of the heat pump [h]	1678	2305	3983
Auxiliary electrical energy stand-by [kWh_{el}]	29	0 (included in EN 255-3)	29
Heat supplied to air-heating/DHW [kWh_{th}]	2655	3211	5866
Electrical energy “generator” [kWh_{el}]	915	1563	2478
SPF-G [-]	**2.90**	**2.05**	**2.37**
System boundary SPF-S			
Electrical energy heat recovery (fans and control – only winter operation) [kWh_{el}]	378	0	378
Sink pump/storage loading pump [kWh_{el}]	165	0 (included in EN 255-3)	165
Total used heat [kWh_{th}]	6344	2750	9094
Electrical energy “system” [kWh_{el}]	1458	1563	3021
SPF-S [-]	**4.35**	**1.76**	**3.01**

E DETAILED RESULTS GELTERKINDEN AND GRAFSTAL

E.1 Grafstal Space heating

Tab. E 1: Results of the calculation for the pilot plant Grafstal in heating mode

Comparison Grafstal Space heating			
	Monitoring (reference)	**Calculation**	**Difference**
System boundary heat pump			
Heating energy requirement [kWh]	10423	10423	Set input value
Losses of the distribution system	336	336	Set input value
Evaluation time (heating period)	5875[1)]	5580	-5%
Electric energy for heat pump [kWh]	2030	1956	-3.6%
SPF-HP by weighting over bins [-]	**5.29**	**5.33**	**0.8 %**
SPF-HP monitoring	**5.13**	**-**	**3.9 %**
System boundary SPF-G			
Running time of the heat pump [h]	2096	1870	-10.8 %
Energy for the source pump [kWh]	252	225	-10.7 %
Energy for control in stand-by [kWh]	53	52	-1.9 %
Sum of electricity input “generator” [kWh]	2335	2233	-4.4 %
SPF-G [-]	**4.46**	**4.67**	**4.7 %**
System boundary SPF-S			
Used heat for space heating	10087	10087	Set input value
Energy for the circulation pump [kWh]	256	243	-5.1 %
Sum of electricity input “system” [kWh]	2591	2519	-2.8 %
SPF-S [-]	**3.89**	**4.07**	**4.6 %**

[1)] calculated value

E.2 Grafstal DHW

Tab. E 2: Results of the calculation for the pilot plant Grafstal in domestic hot water mode with 4 bins and averaged supply temperature of the heat pump θ_{sfl}=48°C

Comparison Grafstal DHW			
	Monitoring (reference)	**Calculation**	**Difference**
System boundary heat pump			
Energy requirement [kWh]	1505	1505	Set input value
DHW storage losses [kWh]	368	368	Set input value
Losses of the distribution system [kWh]	394	394	Set input value
Total generated DHW energy [kWh]	2267	2267	-
Electric energy for heat pump [kWh]	640	629	-1.7 %
SPF-HP by weighting over bins [-]	**3.54**	**3.61**	**1.9 %**
SPF-HP monitoring	**3.58**		
System boundary SPF-G			
Stand-by electricity for control [kWh]	34	39	14.7 %
Running time of the heat pump [h]	481	411	-14.6 %
Energy for the source pump [kWh]	65	49	-24.6 %
Sum of electricity input “generator” [kWh]	738	717	-2.8 %
SPF-G [-]	**3.07**	**3.16**	**2.9 %**
System boundary SPF-S			
Tapped DHW energy	1505	1505	Set input value
Energy for the circulation pump [kWh]	21	18	-14.3 %
Sum of electricity input “system” [kWh]	759	735	-3.2 %
SPF-S [-]	**1.98**	**2.05**	**3.5 %**

E.3 Grafstal overall seasonal performance

Tab. E 3: Overall seasonal performance factors and comparison to monitored values of the pilot plant Grafstal equipped with a brine-to-water heat pump

B/W-HP Grafstal, ZH			
	Monitoring	**Calculation**	**Difference**
SPF-HP	4.75	4.91	3.4%
SPF-G	4.13	4.30	4.1%
SPF-S	3.46	3.61	4.3%

E.4 Frequency of the outdoor air temperature for Gelterkinden

Tab E 4: Evaluation of the outdoor air temperature based on on-site measurement at the pilot plant Gelterkinden, see chap. 6.1 in the main part of the report)

Evaluation of outdoor temperature Gelterkinden						
Bin **[°C]**	**Bin time** **[h]**	**cumulative bin time** **[h]**	**HDH 20** **[Kh]**	**Cumulative HDH 20/14** **[Kh]**	**Weighting factor SH 1 K bins** **[-]**	**Weighting factor DHW 1 K bins** **[-]**
-11	3	3	93	93	0.00	0.000
-10	7	10	210	303	0.00	0.001
-9	8	18	232	535	0.00	0.001
-8	12	30	336	871	0.00	0.001
-7	24	54	648	1519	0.01	0.003
-6	33	87	858	2377	0.01	0.004
-5	31	118	775	3152	0.01	0.004
-4	26	144	624	3776	0.01	0.003
-3	72	216	1656	5432	0.02	0.008
-2	**114**	**330**	**2508**	**7940**	0.03	0.013
-1	207	537	4347	12287	0.06	0.024
0	250	787	5000	17287	0.07	0.029
1	243	1030	4617	21904	0.06	0.028
2	306	1336	5508	27412	0.08	0.035
3	368	1704	6256	33668	0.09	0.042
4	**230**	**1934**	**3680**	**37348**	0.05	0.026
5	295	2229	4425	41773	0.06	0.034
6	331	2560	4634	46407	0.06	0.038
7	314	2874	4082	50489	0.06	0.036
8	293	3167	3516	54005	0.05	0.033
9	281	3448	3091	57096	0.04	0.032
10	376	3824	3760	60856	0.05	0.043
11	336	4160	3024	63880	0.04	0.038
12	373	4533	2984	66864	0.04	0.043
13	341	4874	2387	69251	0.03	0.039
14	**322**	**5196**	**1932**	**71183**	0.03	0.037
15	341	5537	1705	72888	0.02	0.039
16	426	5963	1704	74592		0.049
17	371	6334	1113	75705		0.042
18	393	6727	786	76491		0.045
19	376	7103	376	76867		0.043
20	**295**	**7398**	**0**	**76867**		0.034
21	291	7689	0	76867		0.033
22	246	7935	0	76867		0.028
23	171	8106	0	76867		0.020
24	163	8269	0	76867		0.019
25	100	8369	0	76867		0.011
26	103	8472	0	76867		0.012
27	74	8546	0	76867		0.008
28	67	8613	0	76867		0.008
29	56	8669	0	76867		0.006
30	42	8711	0	76867		0.005
31	26	8737	0	76867		0.003
32	8	8745	0	76867		0.001
33	11	8756	0	76867		0.001
34	2	8758	0	76867		0.000
35	2	8760	0	76867		0.000
					Σ = 1.00	**Σ = 1.00**

E.5 Gelterkinden Space heating

Tab. E 5: Results of the calculation for the pilot plant Gelterkinden in space heating mode

Comparison Gelterkinden Space heating			
	Monitoring (reference)	**Calculation**	**Difference**
System boundary heat pump			
Space heating energy requirement [kWh]	12014	12014	Set input value
Reduction of space heating energy requirement by heat recovery [kWh]	1120	1054	-5.9 %
Evaluation time (heating period)	5196	5196	-
Energy to be produced by the heat pump [kWh]	10757	10866	1.0%
Electrical energy heat pump [kWh]	2875	2974	3.4 %
SPF-HP by weighting over bins [-]	**3.66**	**3.66**	**0 %**
SPF-HP monitoring	**3.74**	**-**	**-2.1 %**
System boundary SPF-G			
Electrical back-up heat energy input [kWh_{th}]	137	92	-32.8 %
Electrical back-up energy use [kWh_{el}]	137	97	-29.2 %
Running time of the heat pump [h]	2496	2323	-6.9 %
Auxiliary electrical energy stand-by [kWh]	28	29	3.6%
Heat supplied to floor heating	10894	10960	0.6 %
Electrical energy “generator” [kWh]	3039	3095	1.8 %
SPF-G [-]	**3.58**	**3.54**	**-1.1 %**
System boundary SPF-S			
Electrical energy heat recovery (fans and control) [kWh]	296	296	0.0 %
Auxiliary energy for circulation pump	171	281 (125)[1)]	64.3% (-26.9%)[1)]
Heat energy "system"	12014	12014	-
Electrical energy “system” [kWh]	3507	3672 (3516)[1)]	4.7% (0.3 %)
SPF-S [-]	**3.43**	**3.27 (3.42)**	**-4.7% (-0.3)%**

[1)] Values in brackets refer to the case the sink pump runs through the whole heating period

E.6 Gelterkinden DHW

Tab. E 6: Results of the calculation for the pilot plant Gelterkinden in DHW mode

Comparison Gelterkinden Domestic Hot Water			
	Monitoring (reference)	**Calculation**	**Difference**
System boundary heat pump			
Domestic hot water energy [kWh]	1179	1179	Set input value
DHW storage losses [kWh]	1073	1073	Set input value
Evaluation time [h]	8760	8760	-
Electrical energy heat pump [kWh]	715	731	2.2 %
SPF-HP by weighting over bins [-]	**2.92**	**3.09**	**5.8 %**
SPF HP monitoring	**3.06**		**1.0 %**
System boundary SPF-G			
Electrical back-up energy input [kWh]	62	2	-96.8 %
Running time of the heat pump [h]	609	508	-16.6 %
Auxiliary stand-by electricity [kWh]	30	31	3.3 %
Heat energy "generator"	2252	2252	0.0 %
Electrical energy “generator” [kWh]	807	764	-5.3 %
SPF-G [-]	**2.79**	**2.95**	**5.7 %**
System boundary SPF-S			
Auxiliary energy storage pump [kWh]	20	17	-15.0 %
Electrical energy “system” [kWh]	827	781	-5.6 %
SPF-S [-]	**1.43**	**1.51**	**5.6 %**

E.7 Gelterkinden overall seasonal performance

Tab. E 7: Results of the calculation for the pilot plant Gelterkinden overall

HP compact unit Gelterkinden, BL			
	Monitoring (Reference)	**Calculation**	**Difference**
SPF-HP	3.66	3.54	-3.3%
SPF-G	3.42	3.42	0.0 %
SPF-S	3.06	2.96 (3.07)	-3.2% (0.3%)[1)]

[1)] Values in brackets refer to the case the sink pump runs through the whole heating period

F COMPARISON OF CALCULATION METHODS

F.1 Excel sheet of WPEsti V. 2.O default

Projekt:

MINERGIE / FWS / AWEL-En
WPesti.xll / V2.0 April 05 / HET

Gelterkinden design data and default values of WPEsti

Gebäudedaten				
Klimastation			Basel-Binningen	
Gebäudekategorie			EFH	
Energiebezugsfläche	EBF_0	m²	153	
Energiebezugsfläche korrigiert	EBF	m²	153	
Heizwärmebedarf nach SIA380/1	Q_h	MJ/m²a	155	6588 kWh
Transmissionswärmeverluste nach SIA 380/1	Q_T	MJ/m²a	270	
Lüftungswärmeverluste nach SIA 380/1	Q_V	MJ/m²a	40	
Heizleistungsbedarf SIA 384/201 bei -8°C Vorschlagswert	4.3	kW	4.1	
Warmwasserbedarf nach SIA380/1	Q_{ww}	MJ/m²a	63	2656 kWh
Warmwasser: Speicher- und Verteilverluste		%	25 %	

Wärmepumpen-Anlage			
Name und Typ der Wärmepumpe:	Stiebel Eltron LWZ 303 Sol		
Wärmequelle:	Luft - Wärmepumpe		
Einsatz (Heizung oder Warmwasser):	Heizung + Warmwasser		
Heizungsspeicher	ohne Heizungs - Speicher		
Betriebsweise der Wärmepumpen-Anlage:	mit elektrischer Zusatzheizung		
Steuerung des Elektro-Heizeinsatzes			
COP bei Normtemperatur 7 °C / 50°C (A7 / W50):		-	2.71
COP bei Normtemperatur -7 °C / 35°C (A-7 / W35):		-	2.9
Heizleistung bei Normtemperatur -7 °C (A-7 / W35):		kW	3.36
COP bei Normtemperatur +2 °C (A2 / W35):		-	3.27
Heizleistung bei Normtemperatur +2 °C (A2 / W35):		kW	4.24
COP bei Normtemperatur +7 °C (A7 / W35):		-	3.54
Heizleistung bei Normtemperatur +7 °C (A7 / W35):		kW	4.66
Temperaturerhöhung in der Wärmepumpe bei Normbedingungen	dT Nutzer	°C	5
Vorlauftemperatur der Heizung:	T VL	°C	30
Rücklauftemperatur der Heizung:	T RL	°C	25
elektrische Zusatzheizung Warmwasser:	Elektro-Einsatz im Parallelbetrieb		
garantierte Warmwassertemperatur ohne Elektorheizstab:		°C	52
Warmwassertemperatur mit Elektro - Nachwärmer Qww :		°C	52
WW-Speicher-Inhalt		Liter	200

Resultate					
Elektro-Anteil für die Heizung	ε =	3.4%	kWh =	233	
Elektro-Anteil für das Warmwasser	ε =	5.0%	kWh =	144	Verluste
Verluste im Heizbetrieb (Anfahren, Carterheizung, Speicher)		4 %	Ƞah =	96 %	264 kWh
Verluste im WW-Betrieb (Anfahren, Carterheizung, Speicher)		8 %	Ƞaw =	92 %	213 kWh
Laufzeit der Wärmepumpe			h / a	2'198	SPF-HP
Anteil und JAZ der Wärmepumpe für die Heizung	ε =	96.6%	JAZ_h =	3.88	4.04
Anteil und JAZ der Wärmepumpe für Warmwasser	ε =	95.0%	JAZ_{ww} =	2.40	2.61
Gewichtungsfaktor Heizung w_h:			-	0.71	
Gewichtungsfaktor Warmwasser w_{ww}:			-	0.29	
Jahresarbeitszahl Heizung + Warmwasser JAZh+ww:			-	3.30	3.48
Wärmeerzeugernutzungsgrad Heizung (incl. Back-up)				3.52	
Wärmeerzeugernutzungsgrad WW (incl. Back-up)				2.23	
Wärmeerzeugernutzungsgrad gesamt (incl. Back-up)				3.02	

F.2 Excel sheet of WPEsti V. 2.O Gelterkinden

Projekt:

MINERGIE / FWS / AWEL-En
WPesti.xls / VZD April 06 / HET

Monitoring Gelterkinden
Summenhäufigkeit Gelterkinden
Energien Gelterkinden

Gebäudedaten					
Klimastation				Basel-Binningen	
Gebäudekategorie				EFH	
Energiebezugsfläche		EBF_0	m^2	153	
Energiebezugsfläche korrigiert		EBF	m^2	153	
Heizwärmebedarf nach SIA380/1		Q_h	MJ/m^2a	256	10894 kWh
Transmissionswärmeverluste nach SIA380/1		Q_T	MJ/m^2a	270	
Lüftungswärmeverluste nach SIA380/1		Q_V	MJ/m^2a	40	
Heizleistungsbedarf SIA384/201 bei -8°C	Minimalwert:	4.8	kW	4.1	
Warmwasserbedarf nach SIA380/1		Q_{ww}	MJ/m^2a	53	2252 kWh
Warmwasser: Speicher- und Verteilverluste			%	91%	

Wärmepumpen-Anlage			
Name und Typ der Wärmepumpe:	Stiebel Eltron LWZ303 Sol		
Wärmequelle:	Luft - Wärmepumpe		
Einsatz (Heizung oder Warmwasser):	Heizung + Warmwasser		
Heizungsspeicher	ohne Heizungs - Speicher		
Betriebsweise der Wärmepumpen-Anlage:	mit elektrischer Zusatzheizung		
Steuerung des Elektro-Heizeinsatzes			
COP bei Normtemperatur 7 °C / 50°C (A7 / W50):		-	2.71
COP bei Normtemperatur -7 °C / 35°C (A-7 / W35):		-	2.9
Heizleistung bei Normtemperatur -7 °C (A-7 / W35):		kW	3.36
COP bei Normtemperatur +2 °C (A2 / W35):		-	3.27
Heizleistung bei Normtemperatur +2 °C (A2 / W35):		kW	4.24
COP bei Normtemperatur +7 °C (A7 / W35):		-	3.54
Heizleistung bei Normtemperatur +7 °C (A7 / W35):		kW	4.66
Temperaturerhöhung in der Wärmepumpe bei Normbedingungen	dT Nutzer	°C	5
Vorlauftemperatur der Heizung:	T VL	°C	30
Rücklauftemperatur der Heizung:	T RL	°C	25
elektrische Zusatzheizung Warmwasser:	Elektro-Einsatz im Parallelbetrieb		
garantierte Warmwassertemperatur ohne Elektorheizstab:		°C	46
Warmwassertemperatur mit Elektro - Nachwärmer Qww:		°C	44
WW-Speicher-Inhalt		Liter	200

Resultate					
Elektro-Anteil für die Heizung	ε =	1.1%	kWh =	128	
Elektro-Anteil für das Warmwasser	ε =	0.0%	kWh =	0	Verluste
Verluste im Heizbetrieb (Anfahren, Carterheizung, Speicher)		4%	Etah =	96%	436 kWh
Verluste im WW-Betrieb (Anfahren, Carterheizung, Speicher)		8%	Etaw =	92%	180 kWh
Laufzeit der Wärmepumpe			h / a	2'904	SPF-HP
Anteil und JAZ der Wärmepumpe für die Heizung	ε =	98.9%	JAZ_h =	3.93	4.09
Anteil und JAZ der Wärmepumpe für Warmwasser	ε =	100.0%	JAZ_{ww} =	2.71	2.95
Gewichtungsfaktor Heizung w_h:			-	0.83	
Gewichtungsfaktor Warmwasser w_{ww}:			-	0.17	
Jahresarbeitszahl Heizung + Warmwasser JAZh+ww:			-	3.65	3.83
Wärmeerzeugernutzungsgrad WNG Heizung (incl. Back-up)				3.80	
Wärmeerzeugernutzungsgrad WNG WW (incl. Back-up)				2.73	
Wärmeerzeugernutzungsgrad WNG gesamt (incl. Back-up)				3.56	

G CALCULATION OF THE GROUND-TO-AIR HEAT EXCHANGER

G.1 Outlet temperature of the ground-to-air heat exchanger

The simplified static heat transfer model for the calculation of the ground-to-air heat exchanger is taken from [G 1]. It has been validated with the ground-to-air heat exchanger at DLR, Köln in different depth of the pipes in the ground. The maximum deviations between the measurement and the calculation were 1.2 K. In the cooling application, the values exceeded the measured values while in the heating application the calculated values were lower than the measured temperatures. Thus, the calculation delivers a conservative estimation of the outlet temperature of the ground-to-air heat exchanger.

Heat input to the air flow in the ground-to-air heat exchanger

$$\dot{Q}_{oa} = \dot{V}_{oa} \cdot \rho_{oa} \cdot c_{p,oa} \cdot (\theta_{oa,in} - \theta_{oa,out}) \qquad [W]$$

eq. G 1

where

$\dot{Q}_{oa}$	heat flux to the air flow while passing the ground-to-air heat exchanger	[W]
$\dot{V}_{oa}$	volume flow rate of the air	[m^3/s]
ρ_{oa}	density of the outdoor air	[kg/m^3]
$c_{p,oa}$	heat capacity of the outdoor air	[J/(kgK)]
$\theta_{oa,in}$	temperature of the inlet air	[°C]
$\theta_{oa,out}$	temperature of the outlet air	[°C]

Heat flux between air and the inner wall of the pipe, where the inner wall temperature is assumed to be isothermal, becomes

$$\dot{Q}_{oa,winn} = \alpha_{oa} \cdot d_{inn} \cdot \pi \cdot l \cdot \frac{(\theta_{w,inn} - \theta_{oa,out}) - (\theta_{w,inn} - \theta_{oa,in})}{\ln \frac{(\theta_{w,inn} - \theta_{oa,out})}{(\theta_{w,inn} - \theta_{oa,in})}} \qquad [W]$$

eq. G2

where

$\dot{Q}_{oa,winn}$	heat flux between the air and the inner pipe wall	[W]
α_{oa}	heat transfer coefficient between inner pipe wall and the air flow	[W/(m^2K)]
d_{inn}	inner diameter of the pipe	[m]
l	length of the pipe	[m]
$\theta_{w,inn}$	temperature of the inner pipe wall	

The heat transfer coefficient α_{air} for the convective heat transfer in the pipe is determined by the Nusselt correlation according to Gnielinski which is valid for

Reynolds-Numbers $10^4 \le Re \le 10^6$, Prandtl numbers $0.6 \le Pr \le 1000$ and the ratio diameter to length of the pipe $d_i/l \le 1$

$$Nu = \frac{\alpha_{oa} \cdot d_{inn}}{\lambda_{oa}} = \frac{(\xi/8) \cdot Re \cdot Pr}{1 + 12.7 \cdot \sqrt{\xi/8} \cdot (Pr^{2/3} - 1)} \cdot \left[1 + (\frac{d_{inn}}{l})^{2/3}\right] \qquad [-]$$

eq. G3

with

$$\xi = (1.8 \cdot \log_{10} Re - 1.5)^{-2} \text{ and } Re = \frac{w_{oa} \cdot d_{inn}}{\nu_{oa}} \text{ and } Pr = \frac{\nu_{oa}}{a_{oa}}$$

eq. G4

where

w_{oa}	velocity of the air flow	[m/s]
ν_{oa}	kinematic viscosity of the outdoor air	[m^2/s]
a_{oa}	temperature conductivity of the outdoor air	[m/s^2]
ξ	resistance coefficient	[-]

Heat conduction between the pipe inner wall and the ground

$$\dot{Q}_g = \frac{1}{0.5 \cdot [\frac{1}{\lambda_p} \cdot \ln(\frac{d_{out}}{d_{inn}}) + \frac{1}{\lambda_g} \ln(\frac{d_g}{d_{out}})]} \cdot \pi \cdot l \cdot (\theta_{w,inn} - \theta_g) \qquad [W]$$

eq. G5

where

$\dot{Q}_g$	heat flux between the air and the inner pipe wall	[W]
d_{out}	outer diameter of the pipe	[m]
λ_p	heat conductivity of the pipe material	[W/(mK)]
λ_g	heat conductivity of the ground	[W/(mK)]
d_g	diameter of the considered ground layer around the pipe	[m]
θ_g	temperature of the undisturbed ground	[°C]

For d_g a value of 1.3 m for pipe diameters d_{out}<1 is set based on evaluation of measurements on the ground-to-air heat exchanger installed on the Solar Campus Juelich.

The three heat flux in eq. G 1, eq. G2 and eq. G5 are the identical

$$\dot{Q}_{oa} = \dot{Q}_{oa,winn} = \dot{Q}_g \qquad [W]$$

eq. G6

After setting $\dot{Q}_{oa}$ for $\dot{Q}_{oa,winn}$ in eq. G2 it can be rearranged to

$$\theta_{w,inn} = \frac{\theta_{oa,out} - \theta_{oa,in} \cdot \exp(A)}{1 - \exp(A)}, \text{ where } A = \frac{\pi \cdot l \cdot \alpha_{oa} \cdot d_{w,inn}}{c_{p,oa} \cdot \rho_{oa} \cdot \dot{V}_{oa}} \qquad [°C]$$

eq. G7

Introducing $\dot{Q}_{oa}$ for $\dot{Q}_g$ in eq. G5 and rearranging

$$\theta_{w,inn} = \frac{B \cdot c_{p,oa} \cdot \rho_{oa} \cdot \dot{V}_{oa} \cdot (\theta_{oa,in} - \theta_{oa,out})}{\pi \cdot l} + \theta_g, \text{ where } B = 0.5 \cdot [\frac{1}{\lambda_p} \cdot \ln(\frac{d_{out}}{d_{inn}}) + \frac{1}{\lambda_g} \ln(\frac{d_g}{d_{out}})] \cdot \qquad [°C]$$

eq. G8

Introducing eq. G7 in eq. G8 and rearranging delivers

$$\frac{\theta_{oa,out} - \theta_{oa,in} \cdot \exp(A)}{1-\exp(A)} = \frac{B \cdot c_{p,oa} \cdot \rho_{oa} \cdot \dot{V}_{oa} \cdot (\theta_{oa,in} - \theta_{oa,out})}{\pi \cdot l} + \theta_g \qquad [°C]$$

eq. G9

and resolving to the outlet temperature it follows

$$\theta_{oa,out} = \frac{\theta_{oa,in} \cdot (\pi \cdot l + B \cdot c_{p,oa} \cdot \rho_{oa} \cdot \dot{V}_{oa} \cdot (\exp(A)-1)) - \theta_g \cdot (1-\exp(A)) \cdot \pi \cdot l}{\pi \cdot l \cdot \exp(A) + B \cdot c_{p,oa} \cdot \rho_{oa} \cdot \dot{V}_{oa} \cdot (\exp(A)-1)} \qquad [°C]$$

eq. G10

With eq. G10 the outlet temperature can be calculated dependent on the ground temperature, the inlet temperature and parameters of the ground to air heat exchanger.

For the ground temperature a damped and phase shifted model based on monthly values of the outdoor air temperature.

The outdoor temperature is approximated as a cos-function with an amplitude a phase shift. The ground temperature is calculated damped depending on the depth of the ground layer considered and the ground thermal characteristic according to the equation

$$\theta_g(z,t) = \theta_{oa} + (\theta_{oa,max} - \theta_{oa,avg}) \cdot \exp(-z \cdot (\frac{\pi}{a_g \cdot T})^{1/2} \cdot \cos(\frac{2 \cdot \pi \cdot (t-\varphi)}{T} - z \cdot (\frac{\pi}{a_g \cdot T})^{1/2}) \qquad [°C]$$

where

$\theta_g(z,t)$	ground temperature dependent on time of the year and depth in the ground	[°C]
$\theta_{oa, max}$	maximal monthly averaged outdoor temperature of the site	[°C]
$\theta_{oa, avg}$	yearly averaged outdoor temperature of the site	[°C]
a_g	temperature conductivity of the ground	[m^2/h]
z	depth in the ground	[m]
t	time in the year	[h]
T	period of the cosine function (8760 hours)	[h]
φ	phase shift constant (840 hours)	[h]

Ground characteristic for different ground types are listed in the following table

Tab. G 1: Properties of different ground types

Ground properties				
Ground type	**Thermal conductivity [W/(m·K)]**	**Density [kg/m³]**	**Heat capacity [J/(kg·K)]**	**Temperature conductivity [m²/s]**
Dry sand	0.7	1500	922	$5.06\ 10^{-7}$
Moist sand	1.88	1500	1199	$10.45\ 10^{-7}$
Moist clay	1.45	1800	1339	$6.02\ 10^{-7}$
Saturated clay	2.9	1800	1591	$10.13\ 10^{-7}$

References

[G 1] Dibowski, G. et al: Luft-Erdwärmetauscher L-EWT Planungsleitfaden Teil 2, Benchmark überschlägiges Abschätzverfahren, AG Solar Schlussbericht, Sept. 2005, DE

H SIMPLIFIED ESTIMATION OF BACK-UP ENERGY

H.1 Alternate operating mode of the back-up heater

In alternate operating mode of the back-up heater, the heat pump generator is switched-off at the balance point temperature, and only the back-up heating supplies the full heat energy need below the balance point.

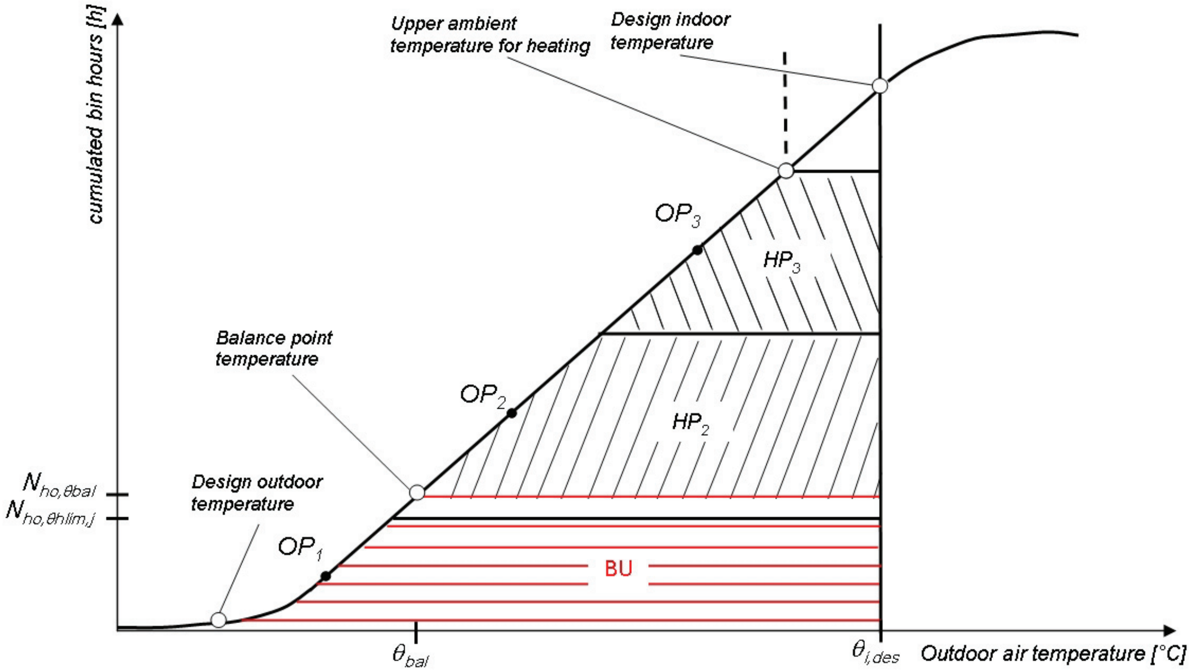


Fig. H 1– Fraction of back-up heat for alternate operating mode (HP – heat pump, OP – operating point, BU – back-up)

Fig. H 1 shows the areas under cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions. The area BU represents the energy fraction delivered by the back-up heater. The fraction of the back-up heat for alternate operation can be calculated by the equation eq. H 1, where index j denotes the lowest bin, j+1 the subsequent bin

For $\theta_{bal} > \theta_{hlim,j}$

$$k_{H,bu,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta hlim,j}}{DH_{H,\theta hlim,j} - DH_{H,\theta lim,j}} = \frac{DH_{H,\theta hlim,j}}{DH_{H,\theta hlim,j}} = 1 \quad [-]$$

eq. H 1

$$k_{H,bu,j+1} = \frac{A_{bu,j+1}}{A_{j+1}} = \frac{DH_{H,\theta bal} - DH_{H,\theta lim,j}}{DH_{H,\theta hlim,j+1} - DH_{H,\theta lim,j+1}} \quad [-]$$

eq. H 2

and for $\theta_{bal} < \theta_{hlim,j}$

$$k_{H,bu,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta bal}}{DH_{H,\theta hlim,j}} \quad [-]$$

eq. H 3

where

$k_{H,bu,j}$	fraction of space heating energy covered by the back-up heater in the lower bin j	(-)
$k_{H,bu,j+1}$	fraction of space heating energy covered by the back-up heater in subsequent bin j+1	(-)
$A_{bu,j}$	fraction of the total area BU in Fig. H 1 in bin j	(Kh)
$A_{bu,j+1}$	Fraction of total area BU in Fig. H 1 of the subsequent bin j+1	(Kh)
A_j	total area of bin j (between upper and lower temperature limit of bin j)	(Kh)
A_{j+1}	total bin area of the subsequent bin j+1	(Kh)
θ_{bal}	balance point temperature	(°C)
$\theta_{hlim,j}$	upper temperature limit of bin j	(°C)
$DH_{H,\theta bal}$	cumulative heating degree hours up to the balance point θ_{bal}	(Kh)
$DH_{H,\theta llim,j}$	cumulative heating degree hours up to lower temperature limit of bin j $\theta_{llim,j}$	(Kh)
$DH_{H,\theta hlim,j}$	cumulative heating degree hours up to upper temperature limit of bin j $\theta_{hlim,j}$	(Kh)

H.2 Parallel operating mode of the back-up heater

In parallel operating mode of the back-up heater, the heat pump generator is not switched-off at the balance point temperature, but contributes to cover the energy requirement. The back-up heater only supplies the part that the heat pump cannot deliver due to the lack of heating capacity.

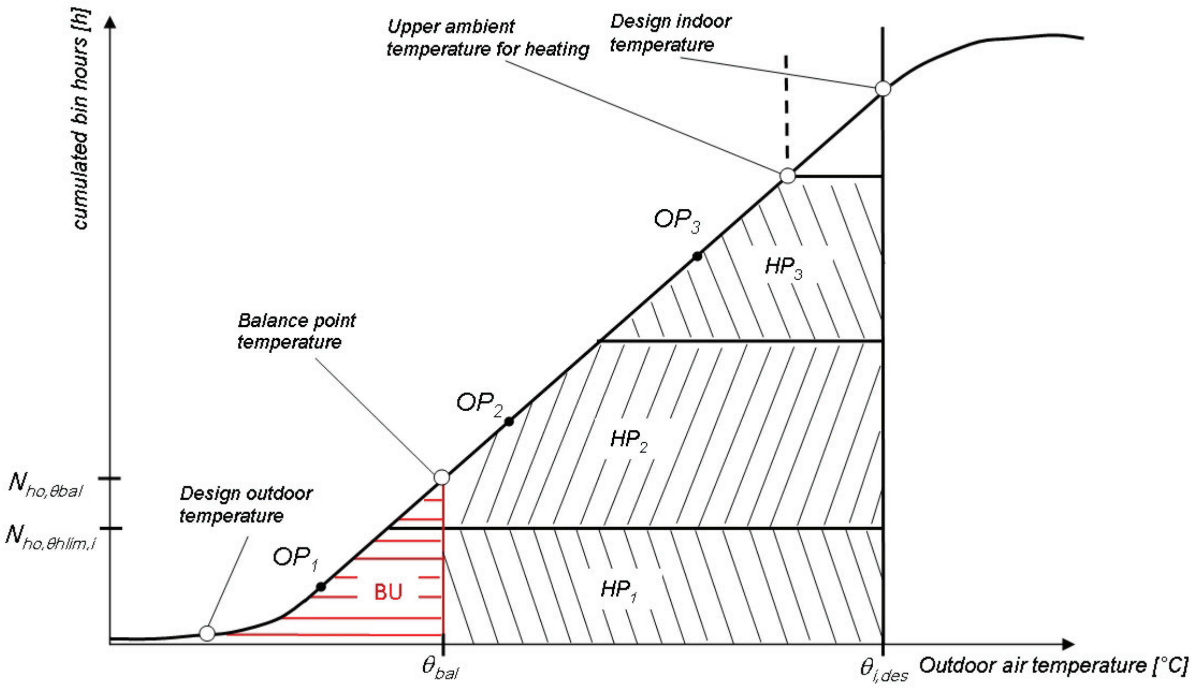


Fig. H 2 – Fraction of back-up heat for parallel operating mode (HP – heat pump, OP – operating point, BU – back-up)

Fig. H 2 shows the areas under cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions. The area BU approximates the energetic fraction delivered by the back-up heater. The fraction of the back-up heat for parallel operation can be approximated by the equation

For $\theta_{bal} > \theta_{hlim,j}$

$$k_{H,bu,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta hlim,j} - (\theta_{i,des} - \theta_{bal}) \cdot N_{ho,\theta hlim,j}}{DH_{H,\theta hlim,j} - DH_{H,\theta llim,j}} \quad [-]$$

eq. H 4

$$k_{H,bu,j+1} = \frac{A_{bu,j+1}}{A_{j+1}} = \frac{(DH_{H,\theta bal} - DH_{H,\theta llim,j+1}) - (\theta_{i,des} - \theta_{bal}) \cdot)(N_{ho,\theta bal} - N_{ho,\theta llim,j+1})}{DH_{H,\theta hlim,j+1} - DH_{H,\theta llim,j+1}} \quad [-]$$

eq. H 5

and for $\theta_{bal} < \theta_{hlim,j}$

$$k_{H,bu,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta bal} - (\theta_{i,des} - \theta_{bal}) \cdot N_{ho,\theta bal}}{DH_{H,\theta hlim,j} - DH_{H,\theta llim,j}} \quad [-]$$

eq. H 6

where

$k_{H,bu,j}$	fraction of space heating energy covered by the back-up heater in the lower bin j	(-)
$k_{H,bu,j+1}$	fraction of space heating energy covered by the back-up heater in subsequent bin j+1	(-)
$A_{bu,j}$	fraction of total area BU in Fig. H 2 in bin j	(Kh)
$A_{bu,j+1}$	fraction of total area BU in Fig. H 2 in subsequent bin j+1	(Kh)
A_j	total area of bin j (between upper and lower temperature limit of bin j)	(Kh)
A_{j+1}	total area of subsequent bin j+1 (between upper and lower temperature limit)	(Kh)
θ_{bal}	balance point temperature	(°C)
$\theta_{hlim,j}$	upper temperature limit of bin j	(°C)
$\theta_{i,des}$	indoor design temperature	(°C)
$N_{ho,\theta bal}$	cumulated number of hours up to the balance point temperature	(h)
$DH_{H,\theta bal}$	cumulative heating degree hours up to the balance point θ_{bal}	(Kh)
$DH_{H,\theta llim,j}$	cumulative heating degree hours up to the lower temperature limit $\theta_{llim,j}$	(Kh)
$DH_{H,\theta hlim,j}$	cumulative heating degree hours up to the upper temperature limit $\theta_{hlim,j}$	(Kh)

NOTE The vertical limit of A_{BU} in Fig. H 2 is an approximation, since the heating capacity of the heat pump is not constant and decreases with decreasing source temperature, thus the line is inclined to higher temperatures of the outdoor air. For high balance points and air-source heat pumps the inclination gets stronger and may lead to higher back-up fractions. However, the boundary condition for the running time given in eq. 20 in the main part of the report indicates if the approximation is not exact enough.

H.3 Partly parallel operating mode of the back-up heater

In partly parallel operating mode of the back-up heater, the heat pump generator is not switched-off at the balance point temperature and runs up to the low-temperature cut-out, where the heat pump is switched-off and only the back-up heater is operated to supply the total heating requirement.

Fig. H 3 shows the areas under the cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions.

The area BU represents the energy fraction delivered by the back-up heater. The fraction of the back-up heat for partly parallel operation can be approximated by the equation

For $\theta_{bal} > \theta_{hlim,j}$

$$k_{H,bu,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta hlim,j} - (\theta_{i,des} - \theta_{bal}) \cdot (N_{ho,\theta bal} - N_{ho,\theta ltc})}{DH_{H,\theta hlim,j} - DH_{H,\theta llim,j}} \quad [-]$$

eq. H 7

$$k_{H,bu,j+1} = \frac{A_{bu,j+1}}{A_{j+1}} = \frac{(DH_{H,\theta bal} - DH_{H,\theta llim,j+1}) - ((\theta_{i,des} - \theta_{bal}) \cdot (N_{ho,\theta bal} - N_{ho,\theta llim,j+1}))}{DH_{H,\theta hlim,j+1} - DH_{H,\theta llim,j+1}} \quad [-]$$

eq. H 8

and for $\theta_{bal} < \theta_{hlim,j}$

$$k_{H,bu,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta bal} - (\theta_{i,des} - \theta_{bal}) \cdot (N_{ho,\theta bal} - N_{ho,\theta ltc})}{DH_{H,\theta hlim,j}} \quad [-]$$

eq. H 9

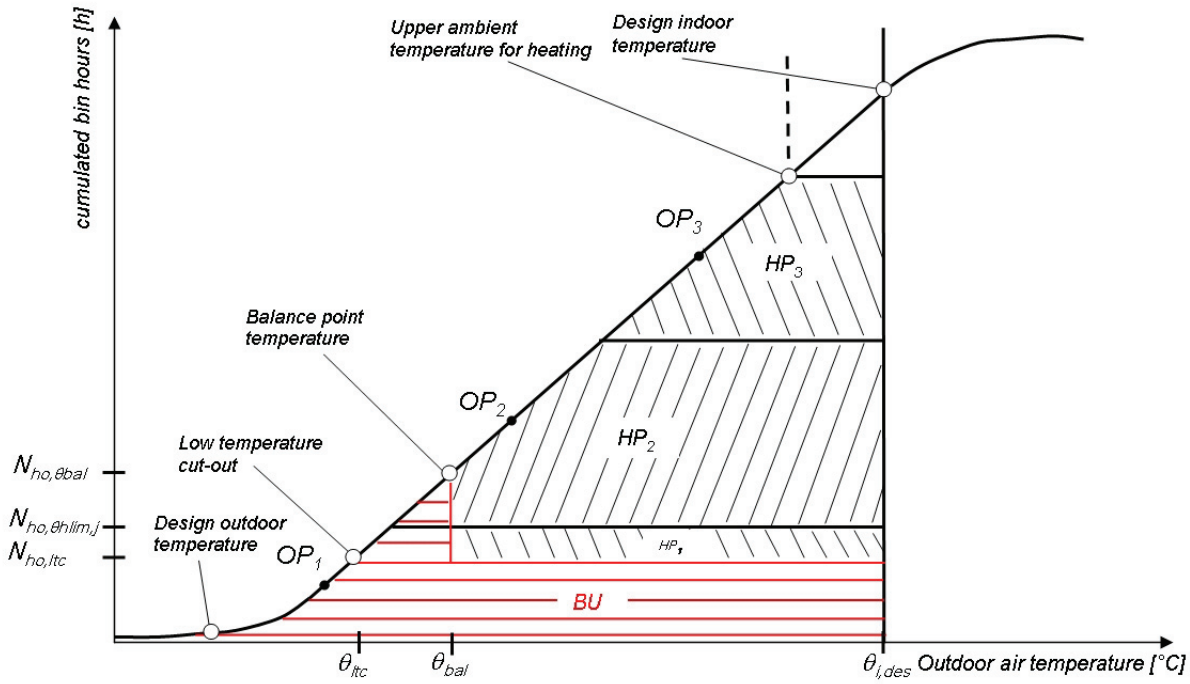


Fig. H 3 - Fraction of back-up heat for partly parallel operating mode (HP – heat pump, OP – operating point, BU – back-up)

where

$k_{H,bu,j}$	fraction of space heating energy covered by the back-up heater in lower bin j	(-)
$k_{H,bu,j+1}$	fraction of space heating energy covered by the back-up heater in subsequent bin j+1	(-)
$A_{bu,j}$	fraction of area BU in Fig. H 3 in bin j	(Kh)
A_j	total area of bin j (between upper and lower temperature limit of bin j)	(Kh)
θ_{bal}	balance point temperature	(°C)
$\theta_{i,des}$	indoor design temperature	(°C)
θ_{ltc}	low-temperature cut-out temperature	(°C)
$N_{ho,\theta bal}$	cumulative number of hours up to the balance point temperature	(h)

$N_{ho,\theta hlim,j}$	cumulative number of hours up to the upper temperature limit of bin j	(h)
$N_{ho,\theta ltc}$	cumulative number of hours up to the low-temperature cut-out	(h)
$DH_{H,\theta bal}$	cumulative heating degree hours up to the balance point temperature θ_{bal}	(Kh)
$DH_{H,\theta hlim,j}$	cumulative heating degree hours up to the upper temperature limit $\theta_{hlim,j}$	(Kh)
$DH_{H,\theta llim,j}$	cumulative heating degree hours up to the lower temperature limit $\theta_{llim,j}$	(Kh)