Technology Collaboration Programme
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Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)

Annex 32

Economical heating and cooling systems for low energy houses

Final Report – Part 3

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Preface

This project was carried out within the Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP), which is a Technology Collaboration Programme within the International Energy Agency, IEA.

The IEA
The IEA was established in 1974 within the framework of the Organization for Economic Cooperation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster cooperation among the IEA participating countries to increase energy security through energy conservation, development of alternative energy sources, new energy technology and research and development (R&D). This is achieved, in part, through a programme of energy technology and R&D collaboration, currently within the framework of nearly 40 Technology Collaboration Programmes.

The Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)
The Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) forms the legal basis for the implementing agreement for a programme of research, development, demonstration, and promotion of heat pumping technologies. Signatories of the TCP are either governments or organizations designated by their respective governments to conduct programmes in the field of energy conservation.

Under the TCP, collaborative tasks, or “Annexes”, in the field of heat pumps are undertaken. These tasks are conducted on a cost-sharing and/or task-sharing basis by the participating countries. An Annex is in general coordinated by one country which acts as the Operating Agent (manager). Annexes have specific topics and work plans and operate for a specified period, usually several years. The objectives vary from information exchange to the development and implementation of technology. This report presents the results of one Annex.

The Programme is governed by an Executive Committee, which monitors existing projects and identifies new areas where collaborative effort may be beneficial.

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A central role within the HPT TCP is played by the Heat Pump Centre (HPC).

Consistent with the overall objective of the HPT TCP, the HPC seeks to accelerate the implementation of heat pump technologies and thereby optimize the use of energy resources for the benefit of the environment. This is achieved by offering a worldwide information service to support all those who can play a part in the implementation of heat pumping technology including researchers, engineers, manufacturers, installers, equipment users, and energy policy makers in utilities, government offices and other organizations. Activities of the HPC include the production of a Magazine with an additional newsletter 3 times per year, the HPT TCP webpage, the organization of workshops, an inquiry service and a promotion programme. The HPC also publishes selected results from other Annexes, and this publication is one result of this activity.

For further information about the Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) and for inquiries on heat pump issues in general contact the Heat Pump Centre at the following address:

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Final report Annex 32 part 3

IEA HPP Annex 32

Economical heating and cooling systems for low energy houses

Schweizerische Eidgenossenschaft
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Confederazione Svizzera
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Swiss Federal Office of Energy SFOE

Field monitoring

Results of field tests of heat pump systems in low energy houses

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The work presented here is a contribution to the Annex 32 in the Heat Pump Programme (HPP) Implementing Agreement of the International Energy Agency (IEA)

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Summary

Due to the increasing market shares of heat pumps in many countries, the quality management for heat pumps gets more important in order to guarantee well-functioning heat pump systems with good seasonal performance in real operation.
The report gives summarising comparative results of the performed field monitoring projects both for marketable heat pumps and for newly developed system concepts. As basis, an analysis of existing field tests performed in different countries is presented.
Extensive field tests accomplished in IEA HPP Annex 32 of above 100 heat pumps installed in low energy houses in Germany and about 10 heat pumps in Austria confirm that the average performance of air-source heat pumps are in the range of about 2.8 and brine-to-water heat pumps in the range of 4.
That means that virtually all measured heat pump systems in low energy house fulfil the criteria of the European RES Directive to be considered as renewable energies, which is currently a minimum Seasonal Performance Factor of 2.63.
Furthermore, an environmental evaluation in terms of primary energy and CO_2-eq-emission showed in many cases substantial savings compared to a high performance fossile fuel system with a condensing boiler of 96% efficiency based on applied national factors.
Concerning the system configurations, field results confirmed that modular systems with a rather complex hydronic configuration often do not reach the expected performance. Therefore, the system configuration shall be chosen carefully. On the other hand, highly integrated heat pumps with well-adjusted components and adapted controls may reach high performance, as the field monitoring of two ground-coupled heat pump compact units for the function of space heating, DHW, ventilation and passive space cooling in Austria confirms. In this respect, two field monitoring projects performed in Switzerland including a passive cooling function yielded a good performance factor of the ground-coupled residential passive cooling mode in the range of 8 with still potential for further improvement. Since the cooling function can be easily integrated in common system layouts for ground-coupled heat pumps, it seems an efficient system extension to overcome possible future increasing cooling need due to climate change.
Despite this generally good performance in the field operation, also malfunctions and optimisation potentials have been encountered. Average temperatures of the system confirm that performance of the heat pump systems still have performance improvements by lowering the temperature lift and improve design and system layout. Therefore, basic design considerations as conclusions of the presented and former field monitoring projects are summarised and amended by design considerations for highly integrated heat pumps with natural refrigerants.
The summarising results presented in this report are complemented by a more detailed description of single well-performing systems on 4-page Best Practice Sheets, which can be downloaded from the Annex 32 Website at http://www.annex32.net, which are linked in this report.
Upcoming field tests of prototype developments could partly not be covered in the timeframe of IEA HPP Annex 32. However, an outline of the field monitoring projects to begin soon have been documented as system concept sheets, also available on the website.

Contents

PREFACE

Introduction to IEA HPP Annex 32

Since the mid of the nineties low energy buildings with a significantly reduced energy consumption down to ultra-low energy standard (typical space heating energy need of 15 kWh/(m^{2}a)) or even net zero energy consumption (on an annual basis by an integration of on-site renewable energy systems) have been realised.
These building concepts recently show strong market growth in different European countries. Many governments address the spread of low energy buildings as a major strategy to reach climate protection targets according to the Kyoto protocol. Heat pump markets are growing in many countries as well.
Low energy buildings have significantly different load characteristics compared to conventional existing buildings. This requires adapted system solutions to entirely use energy-efficiency potentials for the remaining energy needs.
Integrated heat pump solutions have favourable features for the use in low energy houses. The main advantages are the potential for internal heat recovery and simultaneous operation to cover different building needs at the same time as well as installation space and cost benefits. This leads to a significantly improved system performance in an adequate capacity range to reduce primary energy consumption and cut CO_2-emissions and costs.
However, in many countries, no adequate system solutions are available on the market or energy performance of available and newly-introduced low energy house technologies are not yet approved by field experience. Therefore, system development and field approval of functionality and real-world operational performance of the systems are needed. These are the main working areas of IEA HPP Annex 32.

Main objectives of IEA HPP Annex 32

The main objectives of the IEA HPP Annex 32 are the further development and field monitoring of integrated heat pump systems for the use in low energy buildings, leading to the following objectives:

- To characterise the state-of-the-art in the different participating countries
- To assess and compare the energy performance of different system solutions for the residential low energy house sector
- To develop and lab-test new system solutions of integrated heat pumps in the low-energy-house capacity range including the use of natural refrigerants
- To accomplished field tests of new developments and marketable systems and to document best-practice examples
- To disseminate the results

Results of the IEA HPP Annex 32

The results of IEA HPP Annex 32 comprise:

- Overview of market system solutions of integrated heat pumps for low energy houses
- Design recommendations of the standard system solutions
- New system developments as prototypes including lab-test and simulation results
- Documentation of field monitoring results of new and marketable systems
- Dissemination of results by a website, workshop presentation and reports

1 INTRODUCTION

This report gives a summary of results from various field-monitoring projects of heat pump systems in low energy houses accomplished in the frame of IEA HPP Annex 32.
Since the mid of the nineties, low energy houses have been widely introduced mainly in the central European countries Austria, Germany and Switzerland and are currently becoming the standard building practice. Due to the changing energy needs, adequate systems solutions are required for low energy houses. However, since system solutions for low energy houses have been introduced into the market recently, energy performance of available and newly-introduced low energy house technologies are not approved by field experience, yet. Therefore, system development and field approval of functionality and real-world operational performance of the systems have been addressed by several field-monitoring projects in IEA HPP Annex 32.
The motivations for field monitoring projects have mainly been:

- Approval of expected functionality (in particular for new developments)
- Characterisation of the operation (e.g. in terms of reachable comfort, energy performance, operation limits and control strategies)
- Identification of optimisation potentials
- Comparison of different system solutions
- Documentation und dissemination of lessons-learned und Best-Practice systems

In this report, results of the different field test projects are summarised. For 16 single systems which have shown a good operation and a high performance in the field-monitoring, Best Practice Sheets have been prepared, where the monitoring results of particular systems are shown in detail. Further information about the systems is contained in the country reports referenced in the Best Practice Sheets, which can be downloaded from the website of IEA HPP Annex 32 at http://www.annex32.net. An overview of the Best Practice Sheets is included in Appendix A. This report is structured as follows:
Chapter 2 gives an overview about existing field test results and national projects involved in field testing.
Chapter 3 summarises field monitoring results for ground-coupled heat pumps and in chapter 4, field monitoring results for air-source heat pumps are discussed.
In chapter 5 the heat pump performance in low energy houses compared to retrofitted buildings and environmental aspects are treated. Chapter 6 addresses weak points with regard to functionality and typical system failures found in the field tests and gives design recommendations and optimisation potentials derived from the field experiences.
In Chapter 7 some conclusions are drawn.
In particular the field test of prototype systems developed in Annex 32 could not be entirely covered in the time frame of Annex 32. An outline of the system concept and the ongoing or starting field tests is given in system concepts sheets, which can be downloaded on the website, as well. A list of the documented system is given in Appendix A.2.

2 OVERVIEW OF FIELD TESTS

2.1 Overview of existing field tests

There are some experiences with the real-world operation of ground-source and air-source heat pumps in standard houses in the range of an energy consumption of 100 kWh/(m^2a) and higher. Former field test results cover systems which are installed in common residential buildings of that time. In the following existing field test results are summarised.

2.1.1 Field monitoring analysis of heat pumps (FAWA) in Switzerland (1996-2003)

System boundary:
The system boundary comprises the heat pump and source system as well as the storage and storage loading pump, i.e. the storage losses are taken into account in the seasonal performance factor (SPF). On the other hand, a back-up heating and the distribution pumps are not included.

Outline and results:
In Switzerland the field monitoring project "FAWA" (FeldAnalyse Wärmepumpen-Anlagen) was performed in the period 1996–2003. Different to other field monitoring projects the heat pumps have not been monitored the entire year, but shorter periods were used to derive the seasonal performance factor by a statistical evaluation.
236 systems were included in the project, of which 221 heat pumps systems are evaluated in Erb, Hubacher and Ehrbar (2004), of which each about 45% were brine-to-water (B/W) heat pumps with borehole heat exchanger (BHX) and air-to-water (A/W) systems. Design of about 90% of the the ground-coupled system was monovalent. In total the results represent 1.3 Mio. operating hours or 740 operation years, respectively. 60% of the systems are installed in new buildings, of which 92% are equipped with floor heating. 40% are installed in existing buildings, which had been retrofitted. 53% of the retrofitted are equipped with floor heating systems, partly with additional radiators. The average space heating consumptions of all buildings is 75 kWh/(m^2·a) with a standard deviation of 40%. Average space heating temperatures are with 41 °C for the floor heating systems in new building quite high, and only show a difference of 5 K or 46° C in the retrofitted buildings.

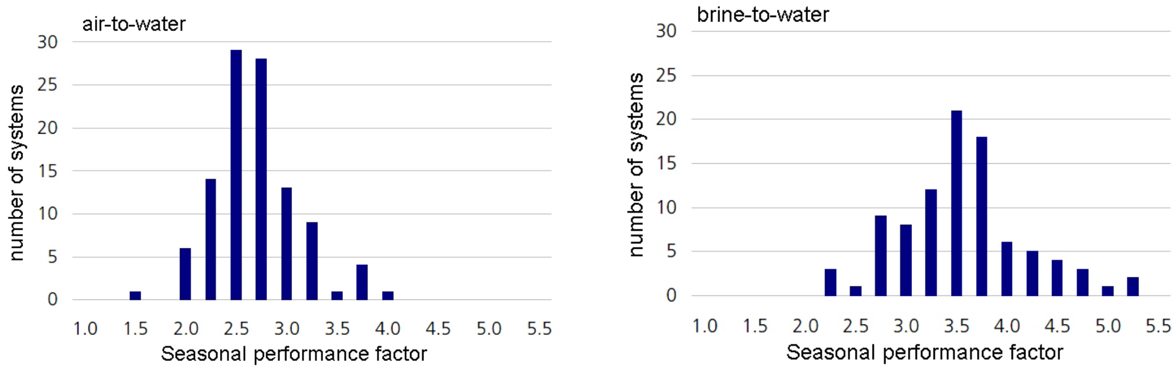


Fig. 1: Frequency of climate corrected nSPF 2 of all A/W systems (left) and all B/W systems (Erb, Hubacher and Ehrbar, 2004)

The average seasonal performance factor of the ground-coupled heat pumps (brine-to-water) was 3.5. Average values are higher than for air-to-water systems, which yield an average SPF of 2.7. However, values for brine-to-water heat pumps show more scatter than

values for air-to-water due to the very distinct capacity of the BHX. SPF values of both systems have improved since 1994/95 by about 15%. If all system in Switzerland are considered (59% A/W and 41% B/W) the increase is 20% from 2.5 to 3.0. This increase reflects the better performance of the heat pump units itself, which was also found in the national heat pump testing at WPZ (www.wpz.ch). Moreover, the study states that the seasonal performance factor did not deteriorate during the years of the study, i.e. in particular no decrease of the ground temperature in the first years of operation was detected.
Besides the seasonal performance factors several other evaluations have been performed. It was found that many installed heat pumps were oversized and buffer storages are often used where not necessary. Better design can results in the same thermal comfort and efficiency, but costs can be reduced by 20%. Concerning the DHW operation, the best performance was found by integration of internal heat exchangers in the parallel storage. Best control results were achieved in systems with additional consideration of the indoor air temperature, which reduces running costs without decreasing the comfort level. The design recommendations derived from the field monitoring results are discussed in chap. 7.2 in connection to the results of the field tests performed in Annex 32.

2.1.2 Best-Practice systems in Switzerland (SFOE project "Bestanlagen") (1996-2008)

System boundary:
The system boundary comprises the heat pump and source system as well as the storage and storage loading pump. A back-up heating and the distribution pumps are not included, as in the FAWA project in 2.1.2.

Outline and results:
Officially, the FAWA project was concluded in 2004 after a 7-year measurement period. However, for some of the FAWA systems the monitoring still continues in order to get a long-term evaluation of the heat pump systems, mainly to answer the following questions:

- Does the seasonal performance factor deteriorate with longer operation times of the heat pump?
- What is the availability of the heat pump systems with increasing running time, i.e. are system failures increasing with an augmentation of the running time?
- What are the costs for maintenance and repairing the system?

Therefore, for about 80 well-performing B/W and about 50 well-performing A/W heat pump systems, a continuous monitoring starting in 1996 was evaluated.

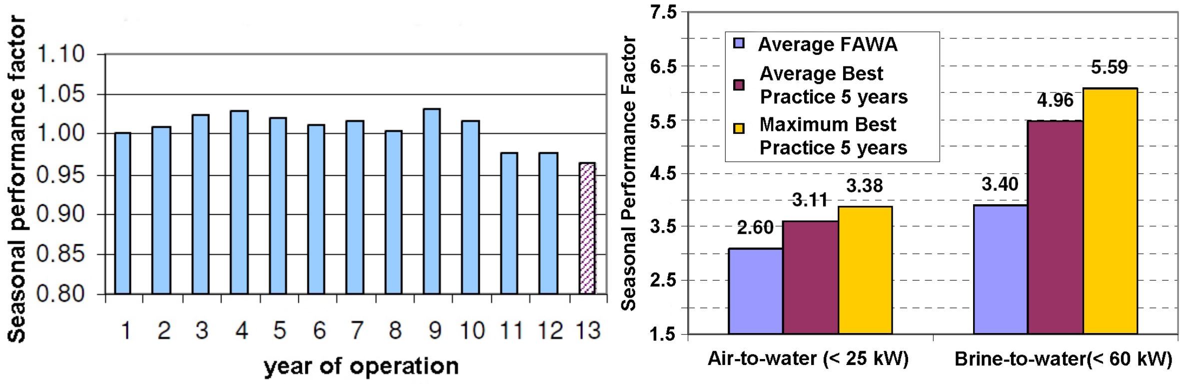


Fig. 2: Change in SPF of 132 systems (left) and comparison of the all FAWA average with average and best values of the Best-Practice Systems (Hubacher et al., 2008)

Fig. 2 left presents the results of the selected 132 Best Practice system for air-to-water and brine-to-water systems. Results show that during the 12-13 years of operation, no significant deterioration of the seasonal performance factor was detected. After about 12 years of op-

eration the systems still have a seasonal performance factor of 97% of the value in the first year of operation, a deviation, which is within the deviation of the seasonal performance factor from year to year. Fig. 2 right shows a comparison to the average results of the FAWA systems averaged over a 5-year period. Both for the air-source and the ground-source systems it is confirmed that systems selected for the continuous monitoring also outperform the systems averaged over a longer time period. The Best Systems reach quite high SPF numbers of 3.4 for the air-source and 5.6 for the ground-source systems and thereby significant differences to the average in the FAWA project.
Fig. 3 left shows the availability and maintenance cost. The evalulation of 1.6 million operating hours shows that the availability of the system is very well and reaches a value of 99 %. Costs for maintenance are very moderate with about 12 €/a as average of 61 systems. Fewer than 10 systems/year had to be repaired, which caused costs of about 50 €/year on average.

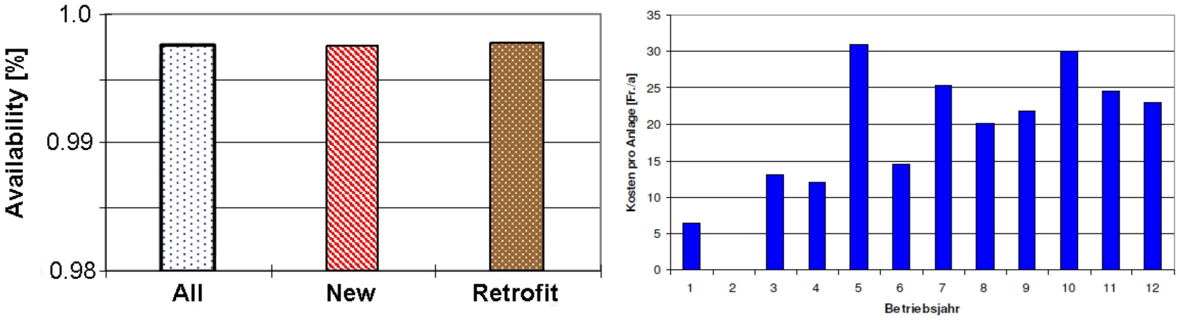


Fig. 3: Availability of systems distinguished by type of building (left) and average cost for maintainance during the period of 61 systems (taken from Hubacher et al., 2008)

Fig. 4 left shows a comparison of the SPF for ground-source heat pumps with borehole heat exchangers dependent on the used source fluid. If the borehole heat exchanger is designed in such way that temperatures above 0 °C can be guaranteed during the whole year, the borehole can be operated with water. Significantly higher SPF values of 0.6 are reached with water as source fluid. This is on the one hand due to the better heat capacity of water and on the other hand due to the lower viscosity, which decreased the auxiliary expenditure.

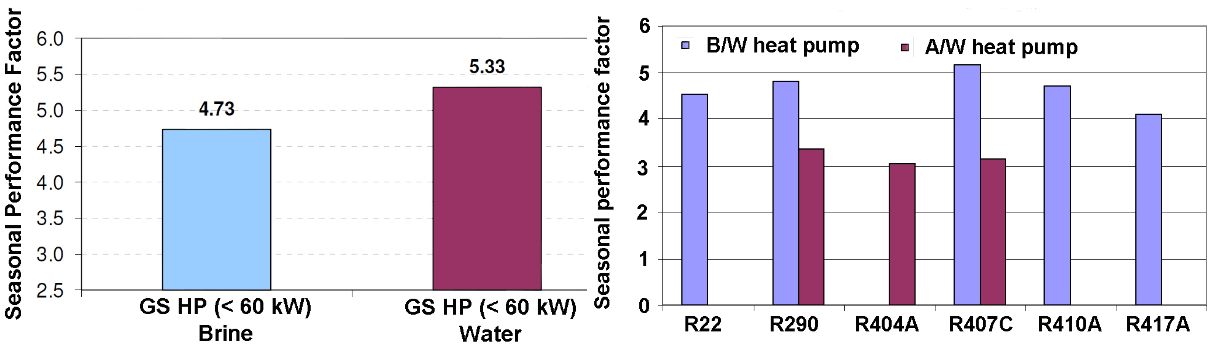


Fig. 4: Comparison of SPF of ground-source systems with brine and water as source fluid (left) and comparison of the average of the FAWA systems with average value and best value of the FAWA Best-Practice Systems (taken from Hubacher et al., 2008)

However, a generally longer design of the borehole heat exchanger to guarantee the higher source temperature may also have an impact. Fig. 4 presents a comparison of SPF values based on the used refrigerant and confirms that the reached SPF values also depend on the refrigerant used. For instance, both air-to-water and brine-to-water heat pumps reach a high performance factor for propane (R290), which additionally is a natural refrigerant with negligible Global Warming Potential (GWP), but has the drawback of flammability and therefore may require enhanced security measures.

2.1.3 E.ON field test (2001-2003)

System boundary:

The system boundary comprises the heat pump and source system as well as a back-up heating. The distribution pumps as well as the storage and storage loading pump are not included, i.e. the performance is based on produced heat of the heat pump and storage losses are not included in the seasonal performance factor.

Outline and results:

The German utility E.ON Energie AG, Munich performed a field test of 29 mostly ground-coupled heat pumps for space heating and DHW during the two heating periods 2001/02 and 2002/03. Due to installation errors and other problems encountered during the field tests, only results of 21 systems are presented in Ewert (2005).

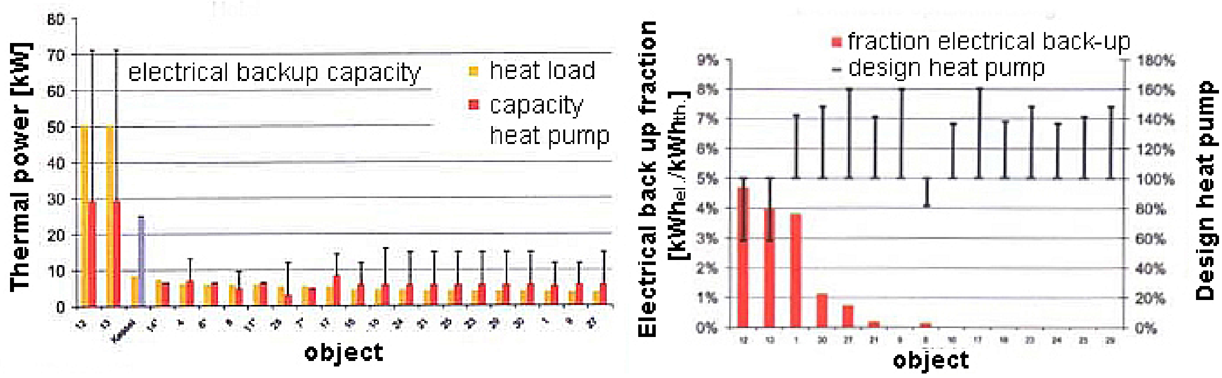


Fig. 5: Design of the ground-coupled heat pumps (left) and direct electrical back-up use (right) measured in the field test by E.ON Energie AG (taken from Ewert (2005))

Fig. 5 gives the design of the evaluated systems (left) and the fraction of used direct electrical back-up energy.

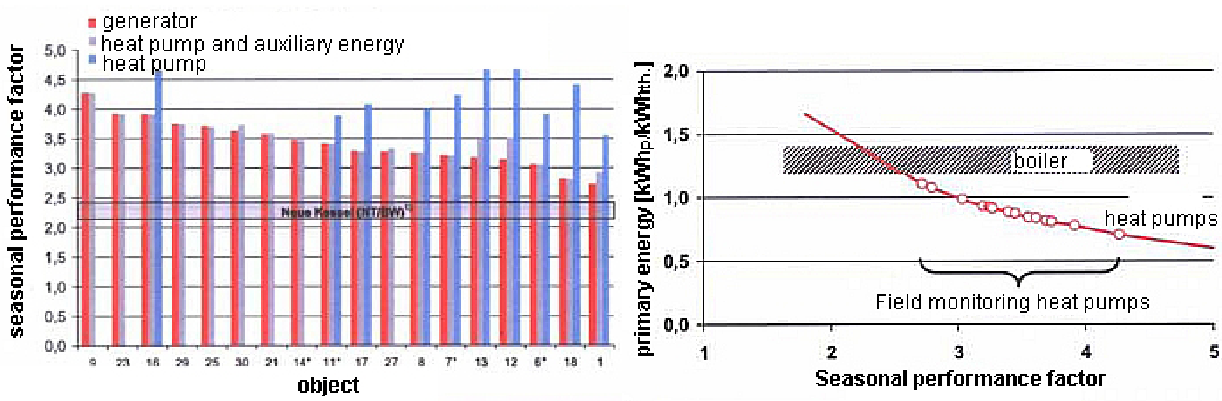


Fig. 6: Seasonal performance factors of the ground-coupled heat pumps (left) and primary energy factor compared to gas boilers (right) measured in the field test by E.ON Energie AG (taken from Ewert (2005))

Most of the heat pumps are designed monovalently, partly up to 160% higher than the design heat load. In these systems, no back-up heat was measured. On the other hand, heat pumps only designed to 60% of the design heat load had a maximum back-up fraction lower than 5%, a heat pump designed to 80% of the design heat load had negligible back-up use. The seasonal performance factor and the primary energy savings are depicted in Fig. 6.

The seasonal performance factor generator (SPF-G) incl. the back-up heating and the source pump ranges between 2.8 and 4.2. Primary energy expenditure factors (primary energy factor divided by efficiency or SPF) of new boilers are in the range of 1.2-1.4, while the heat pumps reach primary energy expenditure factors between 1.1 and 0.7, as shown in Fig. 6 right.

2.1.4 Lokale Agenda 21 Lahr Phase 1, Germany (2006-2008)

System boundary:

Two system boundaries are applied. The first system boundary is related to the "generator" performance and comprises the heat pump and source system as well as a back-up heating. The second boundary is related to the "system" performance and comprises additionally the distribution pumps as well as the storage and storage loading pump, i.e. storage losses are subtracted from the produced heat of the heat pump.

Outline and results:

The group “Lokale Agenda 21” of the City of Lahr, Baden-Württemberg, Germany performed a field test of 33 heat pumps for two entire years 2006 until 2008.

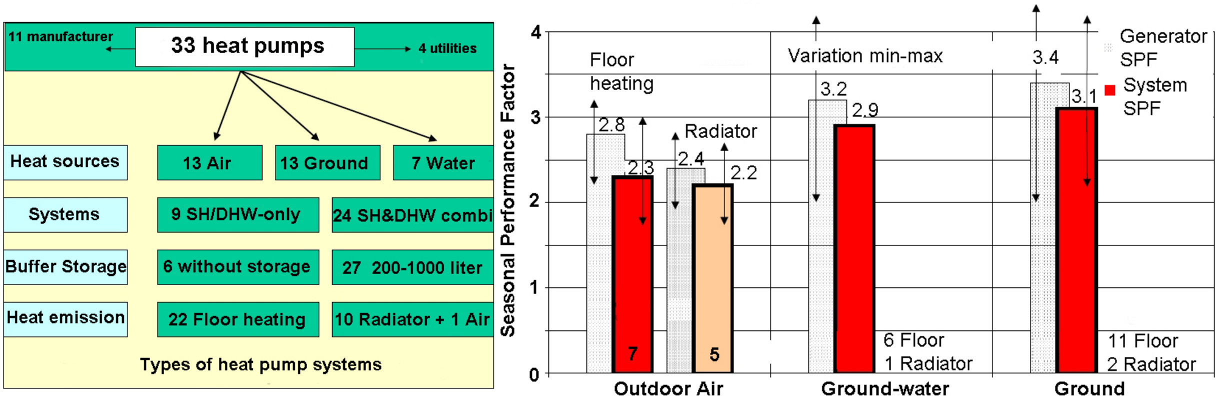


Fig. 7: Overview of field tested heat pumps (left) and seasonal performance factors for "generator" and "system" boundary of different heat sources and emission systems of the field test of Lokale Agenda 21 in Lahr, Germany (translated from Auer and Schrote, 2008)

Fig. 7 gives a summary of seasonal performance factors for different heat sources (x-axis), different emission systems and the system boundaries generator and system. The 13 measured ground-coupled heat pumps performed best and reached an average SPF-G of 3.4 and an SPF-S of 3.1, however, with a large range of measured seasonal performance from 2-4.4 of the generator performance factor and 2.3 – 3.1 for the system seasonal performance factor. 11 of the ground-source heat pumps used vertical borehole heat exchanger, while 2 used horizontal collectors. Concerning the emission system, 11 of the ground-source heat pump worked on floor heating systems and only 2 with radiators. Further information is found in the final report (Auer and Schrote, 2008).

2.1.5 Field test Enbau:Monitor of low energy office buildings in Germany (FhG-ISE)

System boundary:

The system boundary comprises the heat pump and source system as well as the primary pump in case of a heating buffer storage. The pumps of the distribution system and the storage are not included, i.e. storage losses are not included in the SPF.

Outline and results:

The Fraunhofer Institute of Solar Energy system (FhG-ISE) as well as the Universities of Karlsruhe and Wuppertal perform a long-term monitoring of highly-efficient office buildings in Germany. The target value for the office buildings is an overall primary energy consumption below 100 kWh/(m^2·a).

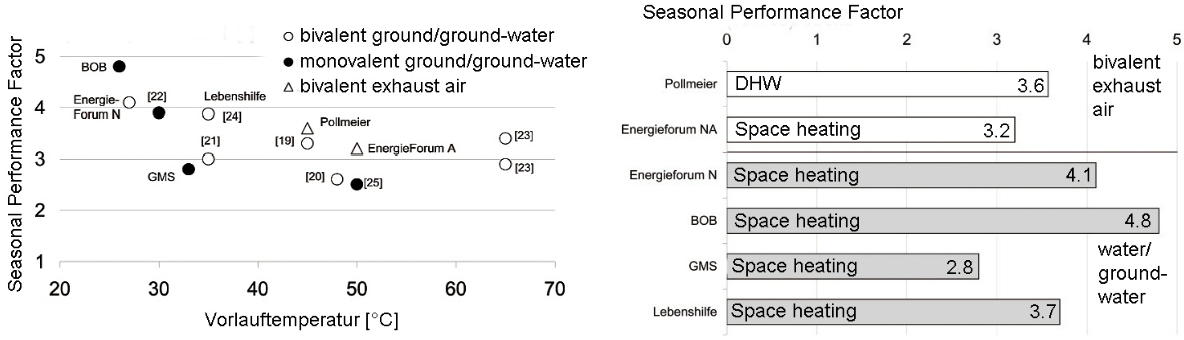


Fig. 8: Overview of field-tested heat pumps and seasonal performance factors of heat pumps in office buildings in Germany (translated from Voss, 2007))

Since many of the office buildings were equipped with heat pumps a particular evaluation of the performance of heat pumps in this application has been accomplished. Fig. 8 left shows an overview of seasonal performance factors dependent on the supply temperature differentiated by the heat source to monovalent and bivalent ground or ground-water heat pumps as well as exhaust air heat pumps. Fig. 8 right shows the same systems dependent on the operation mode.

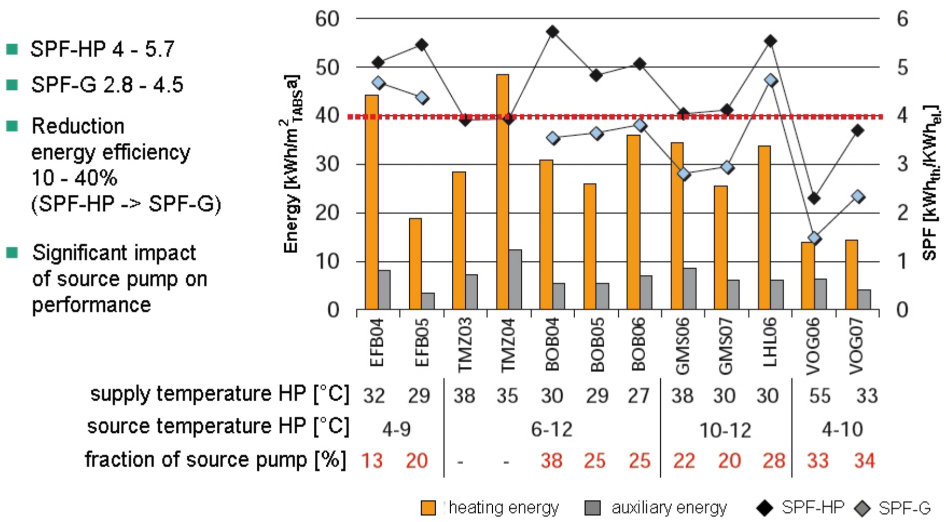


Fig. 9: Field monitoring results of heat pumps installed in office buildings with and without source pump (translated from Kalz, 2009))

Fig. 9 shows experiences of further years of the seasonal performance factors SPF-HP (boundary COP) and the SPF-G (including additionally the source pump). While the heat pump seasonal performance factor ranges from 4-5.7, the SPF-G is significantly lower in the range of 2.8-4.5, i.e. the decrease of the seasonal performance factor by the source pump is in the range of 10%-40%. In small-scale residential buildings, the source fraction is notably lower in the range below 10%. Thus, in larger buildings which are not as standardized as smaller residential units, the auxiliary energy expenditure has to be carefully considered in order to reach good generator and system performance.

2.1.6 Field test direct expansion heat pumps (arsenal research)

System boundary:
The measured system boundary comprises the heat pump and source system as well as a back-up heating. Storage losses and distribution pumps are not included in the SPF.

Outline and results:
The Austrian Institute of Technology (AIT), formerly arsenal research, performed a standard monitoring of direct expansion (DX) heat pumps (Huber, 2007). 9 systems were monitored for one year in order to evaluate the energy flows, the seasonal performance factors and the total equivalent warming impact (TEWI).
Fig. 10 shows the seasonal performance factors of the monitored systems, which all achieve a good performance above 4 with best values above 5. The results confirm that DX systems reach higher SPF values than ground-source brine-to-water heat pumps with an intermediate brine cycle, since in DX-systems no intermediate cycle is used and thereby, the auxiliary energy expenditure for the source pumps is saved. Moreover, the heat exchanger and thereby the temperature drop is avoided. This leads to higher source temperatures.
Direct expansion system used to be very popular in Austria, but in recent years, the fraction of ground-coupled brine-to-water successively increased. Last year, only 10% of the installed heat pumps were of DX type, which is due to the higher installation cost getting a more important argument with increasing number of heat pump systems installed in Austria.

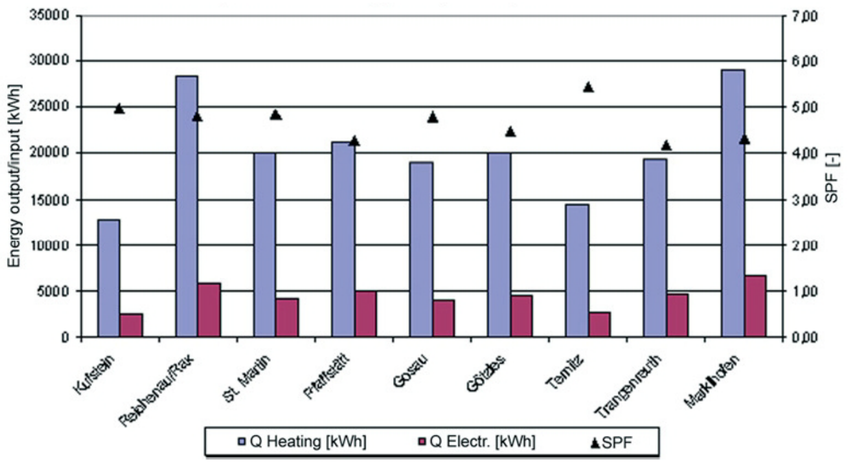


Fig. 10: Field monitoring results of a field test of direct expansion (DX) heat pump (translated from Huber, 2007))

2.1.7 Field test of 20 ground-source heat pumps (Austria) (2005-2008)

System boundary:
The measured system boundary comprises the heat pump and both source and the distribution pumps. Storage losses are not included in the SPF. In order to restrict the system boundary to the source system, a calculatory correction to avoid the balancing of the distribution and storage loading pumps has been applied. The system boundary refers to the generator seasonal performance factor, i.e. the produced heat of the heat pumps for space heating and DHW is considered. Energy generated by installed solar systems is not counted.

Outline and results:
In the Austrian federal state Vorarlberg, a field test of 20 heat pumps with the heat sources ground (borehole heat exchangers) and ground-water has been performed starting in 2005. All systems are designed for space heating and DHW operation and are partly also equipped with solar thermal collectors or solar PV systems. Fig. 11 presents the results of the field

test. Since only the total electrical energy consumptions could be evaluated a measured and the corrected SPF are depicted. Moreover, the minimum SPF requirement applied for subsidies in the years 2005 (SPF > 3.6) and 2009 (SPF > 4) are shown as yellow and blue lines. Since the total electricity includes both the source pump and the storage loading and space heating distribution pump, the SPF values have been recalculated by using a correction factor of 15% for the energy consumptions (assumption: Energy label of the pump B or worse), i.e. the SPF is multiplied by 1.15 (except for the systems 1,2,3) in order to only include the source pump in the SPF value.

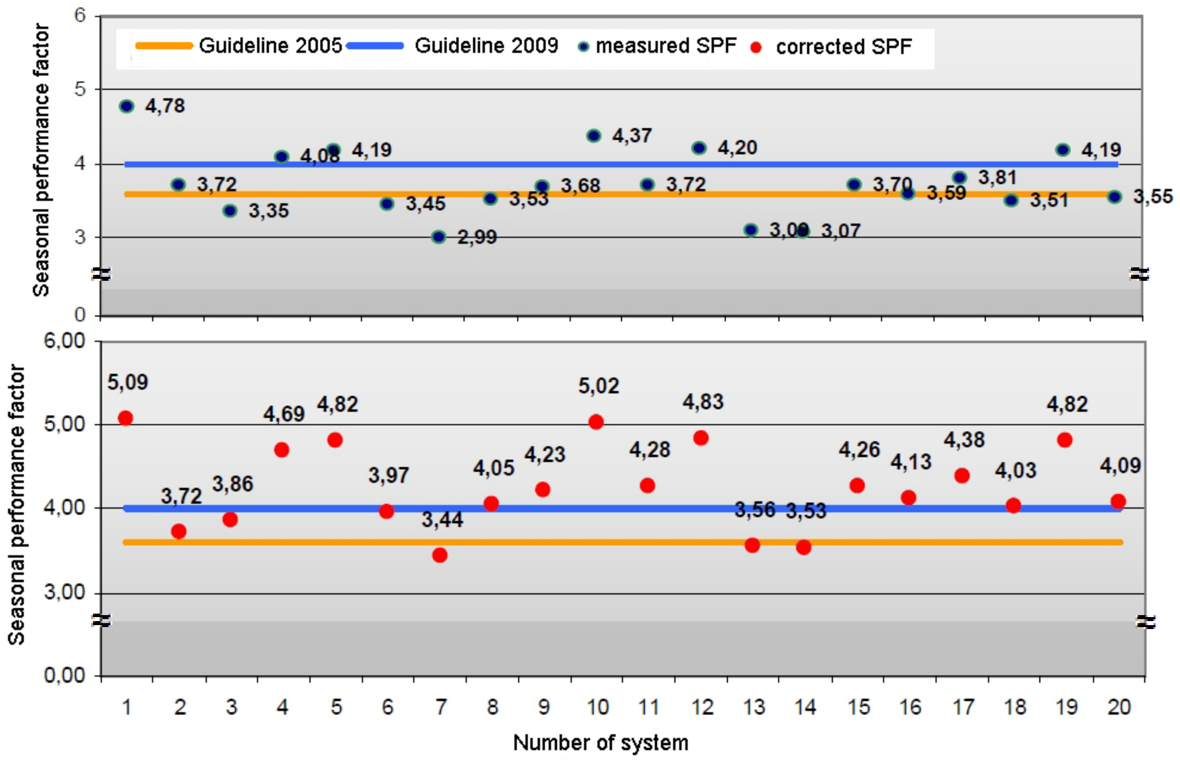


Fig. 11: Field monitoring results of 20 brine-to-water heat pumps in Austria (translated from Vögel, 2009)

The measured results, which include the total auxiliary electrical energy consumption already yields high seasonal performance factors in the range of 3.0 to 4.8. Including the correction factors SPF values range between 3.4 to 5.1. Further information on the field test as well as the characteristics of the heat pump systems (in German) can be found in Vögel (2009).

2.1.8 Field monitoring of heat pumps in the UK

System boundary:
The system performance is defined as the "amount of heat the heat pump produces compared to the amount of electricity needed to run the **entire heating system** (including domestic hot water; supplementary heating; and pumps)". Heat lost from domestic hot water tanks or buffers does not count as useful heat and is not included in the system efficiency.

Outline and results:
The Energy Saving Trust accomplished the first large-scale heat pump field trial in the UK in the year 2009. The year-long field trial was accomplished at 83 sites across the UK. The field test monitored 54 ground-source and 29 air-source heat pumps. Emission systems comprise both under-floor heating and radiator systems. In order to reflect the average building stock, a wide range of houses was monitored in the trial.

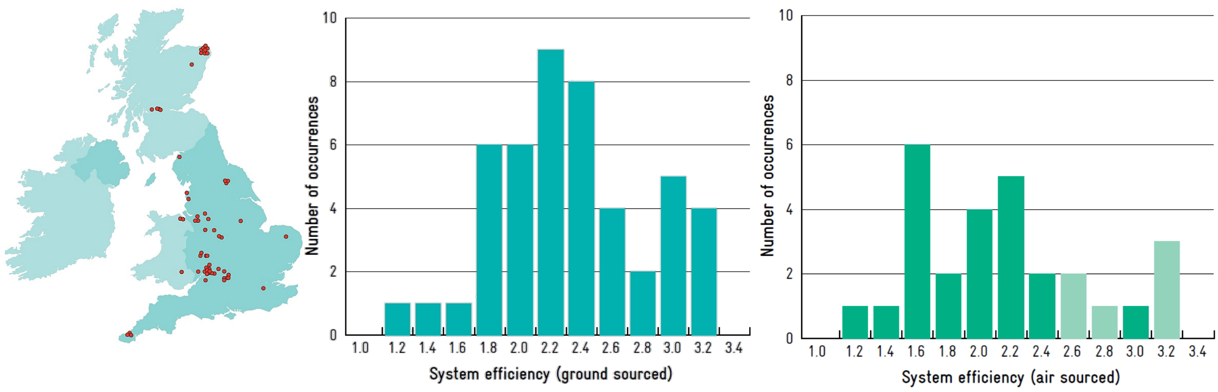


Fig. 12: Field monitoring results of the field test of 83 heat pumps installed in the UK (Energy Saving Trust (2010))

Construction dates range from before 1900 to around 2005. Some, but not all, of the older, solid walled properties had been fitted with solid wall insulation.
Fig. 12 shows the monitoring site in the UK and the results of the field monitoring as frequency of seasonal performance factor distinguished by the heat source, e.g. air-source and ground source systems. Even though the monitoring results show a wide range of seasonal performance factors, the best performing systems show that well-designed and installed heat pumps can operate well in the UK. The 'mid-range' ground-source system efficiencies were between 2.3 and 2.5, with the highest figures above 3.0. The 'mid-range' of measured system efficiencies for air-source heat pumps was near 2.2 and the highest figures above 3.0. The spread of seasonal performance factors confirms that heat pumps are sensitive to installation and commissioning. In general, results are lower than in field test of other European countries. Concerning customer's reaction, most of the households were satisfied with their heat pump, even though some stated difficulties in understanding the instructions how to operate the heat pump. It was proved that user behaviour had an impact on the performance of the heat pump. Concluding, however, well-performing heat pumps can lead to carbon emission savings in the UK and to lower heating cost of the household, in particular when replacing off-the-gas-grid heating systems as direct electric, LPG and oil based systems.

2.1.9 Field monitoring heat pump compact units Germany

System boundary:
13 objects were intensively measured, while further 37 objects were evaluated by logging the electricity meters.

Outline and results:
In co-operation with the Germany utility ENBW, 78 subsidised passive houses were targeted for a measurement compaign in 2001-2003. The passive houses were equipped with heat pumps, solar collectors as well as ground-to-air heat exchangers. Products of various manufacturers were installed. 13 objects were intensively measured, while further 37 objects were evaluated by logging the electricity counters.
Among the heat pump systems also highly integrated ventilation compact units with exhaust air heat pumps for the functions space heating, DHW and ventilation, which are meanwhile popular in German passive houses, have been monitored. On the other hand, modular ground-source heat pumps with solar collector and ventilation systems as modular components have been installed.

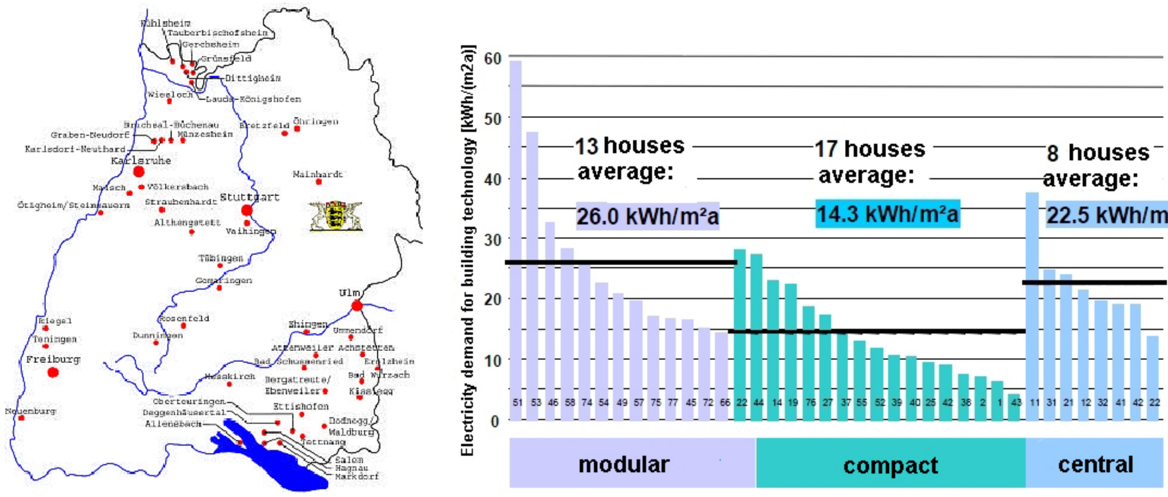


Fig. 13: Comparison of integrated compact units, modular ground-source heat pumps and central heating systems in passive houses in Germany (Bühring, 2003))

Fig. 13 left shows the field test sites. Fig. 13 right shows the electricity consumption of the 38 systems. The average electrical energy consumption of the integrated air-to-air systems is significantly lower than the modular ground-source systems and the central heating systems. There are different explanations for these results. On the one hand, the compact systems might be especially optimised regarding adaptation of system components and control. A second reason is the fact, that the ultra-low energy houses have high DHW shares and in the transitional period the air-source heat pumps may be as efficient as the ground-coupled systems. A third reason could be that the auxiliary energy becomes a larger fraction at very low space heating needs, and the modular ground source heat pumps may have higher auxiliary fractions than the air-source heat pumps.

2.1.10 Field test of ground-source and exhaust air heat pumps in Sweden

System boundary:
The system boundary comprises the entire heating system, e.g. the used heat for the space heating and DHW system divided by a electricity to the heat pump and the source and sink pump. Storage losses are not counted as used heat. In a second boundary, additionally a direct electrical back-up is taken into account.

Outline and results:
In Sweden five ground-source heat pumps (Stenlund and Axell, 2010) and three exhaust air heat pumps (Berg et al., 2010) have been field-monitored in existing buildings in 2004-2005.
The results of the ground-source heat pumps of the year 2003/04 in buildings with radiator systems constructed between 1950 and 1970 yielded an average SPF_{hs} of the space heating system of 2.6, ranging between 2.4 - 2.9 for the five measured systems A-E. The corresponding SPF_{hp} of the heat pump are in the range of 2.5 - 3.1.
The results emphasise the effect of back-up heating, although this was not the only influencing factor, but the most important one. The results were also affected by the dimensioning values. System C had a poor SPF_{hp}, but since almost no supplementary heat was used the SPF_{hs} was not bad in relation to the other systems.
Concerning the field results of three exhaust air heat pumps, the SPF values are in the range of 1.4 – 1.7. Results have been compared by simuations, where a better building insulation and the best available technology has been uesd. Simulations show, that with current technlogy (inverter heat pump) and passive house buildings, the SPF could be increased to about 2.6.

2.2 Outline of national field monitoring projects in Annex 32

Field monitoring of heat pumps in low energy houses has been a focus in Annex 32. In particular by the extensive field monitoring project of standard heat pumps for space heating and DHW in low energy houses as German contribution to Annex 32 and the Austrian field monitoring of 9 standards system and 2 compact units far above 100 heat pumps systems have been documented in field operation. Tab. 1 gives an overview of field tests evaluated in IEA HPP Annex 32.

Tab. 1 Overview of national field test projects in IEA HPP Annex 32

	Institution	#	Functions	Heat source	Heat sink	Characteristic
AT	AIT	9	4 SH, 5 SH&DHW	4 B/W (2 B, 2 H) 3 A/W 1 W/W 1 DX/W (H)	1 radiator 8 floor	Standard
	AIT	2	SH, DHW, V, SC	2 B/W	2 Floor, wall	Compact GS
CA	Hydro-Québec	2	SH, DHW	1 B/W, 1 A/W	2 Floor, Air	Solar
CH	FHNW	2	SH, DHW, SC	2 B/W	2 Floor	Passive cooling
		1	SH, DHW, V	1 A/W	1 Floor	Compact AS
DE	FhG-ISE	97	SH, DHW	26 A/W 68 B/W 3 W/W	5 Radiator 88 Floor 4 combined	Low energy
	FhG-ISE	75	SH, DHW	35 A/W 38 B/W 2 W/W	53 Radiator 20 combined 2 Floor	Retrofit
JP	Uni Hokkaido	2	SH, DHW, V, SC	2 B/W	2 Floor	Cold climate
NO	SINTEF	1	SH & DHW	1 W/W	1 Floor	Propane
total		**191**	**4 SH,187 SH & DHW, 4 SC**	**117 B/W, 66 A/W 7 W/W, 1 DX**	**59 radiator, 108 Floor, 24 combi**	

Legend: AS - Air-source, B - Borehole heat exchanger, DX - direct expansion, GS - Ground-source, H - horizontal ground collector, SC- space cooling, V - Ventilation,

In this report summarising results of heat pump types are discussed. A selection of single well-performing systems and prototypes has been documented as Best Practice Sheets and System Concepts Sheets, respectively. A list of these systems is contained in Appendix B. In the following the field tests within Annex 32 are shortly described.

2.2.1 Austria: 9 standard heat pumps and 2 ground-coupled compact units

System boundaries:

The system boundary of nine standard systems for SH & DHW includes the heat pump, the source system and an eventually installed back-up heater. For the compact units an extended system boundary including also the distribution pumps is applied. Storage losses are not included in the system boundary.

Outline of the field monitoring:

The field monitorings have been carried out by the Austrian Institute of Technology (AIT, formerly arsenal research). Fig. 14 left shows the monitoring plants in Upper Austria, Lower Austria and Styria. The table in Fig. 14 right contains details on the single field monitoring plant. The systems printed in bold letters have been documented as Best Practice Systems.

No.	Location	inhabitants	HP capacity	HP type	Operation modes
1	Weinzierl	4	11.8 kW	B/W	SH
2	Lengenfeld	5	11.8 kW	B/W	SH
3	**Felling**	**4**	**11.8 kW**	**B/W**	**SH&DHW**
4	Behamberg	4	8.0 kW	B/W	SH
5	**St. Peter/Au**	**6**	**19.4 kW**	**W/W**	**SH&DHW**
6	Bad Vöslau	4	10.3 kW	A/W	SH&DHW
7	**Klein Meiseldorf**	**5**	**33.0 kW**	**A/W**	**SH**
8	Ohlsdorf	4	15.1 kW	DX/W	SH
9	**Rutzenmoos**	**2**	**8.3 kW**	**A/W**	**SH&DHW**
10	**Hitzendorf**	**4**	**3.3 kW**	**B/W**	**SH,DHW,SC**
11	**Judendorf**	**4**	**3.3 kW**	**B/W**	**SH,DHW,SC**

Fig. 14: Plant locations and systems of the Austrian field tests (Zottl, Huber and Köfinger, 2010)

The monitoring comprises 9 standard heat pump systems and 2 heat pump compact units, of which 7 are ground-source systems, 3 air-to-water systems and 1 uses a ground-water source. The ground-source systems contain 4 horizontal collector systems, 2 vertical bore-hole heat exchanger systems and one is a direct expansion system with horizontal collector. 5 systems are space heating-only, 4 systems have combined space heating and DHW and the 2 compact units have an additional passive space cooling option by the ground. 3 systems contain a buffer tank for space heating; the other systems are directly coupled to the emission system. All buildings are mediumweight new buildings except for one building, which has been retrofitted and one building, which is a lightweight building.

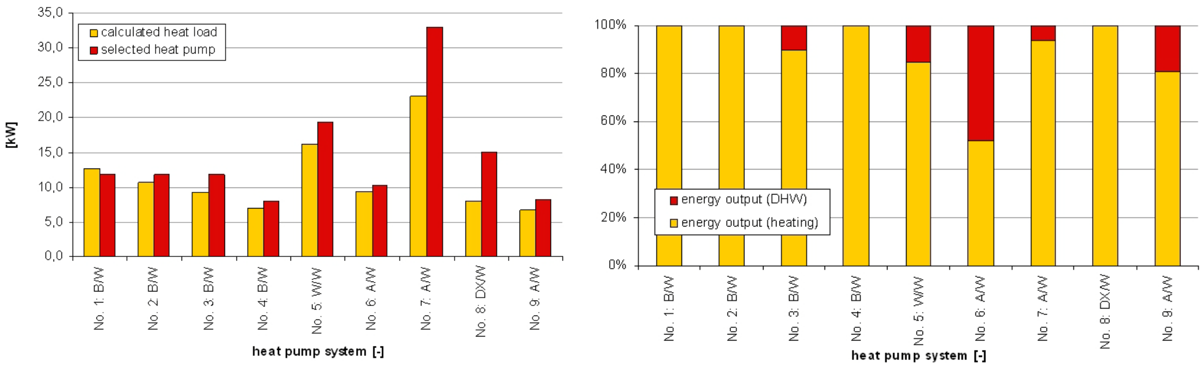


Fig. 15: Comparison of heat pump design and heat load (left) and DHW fraction of the total heat energy needs (Zottl, Huber and Köfinger, 2010)

Fig. 15 left shows a comparison of the heat pump design and the calculated heat load. Most of the systems have a higher design capacity than the calculated heat load. According to this comparison, none or only little back-up operation should occur. Fig. 15 right shows the share of the annual DHW energy consumption. Except for system No. 6, which is an exception due to particular user behaviour, the DHW shares are in the range of 10%-19% with a tendency of higher DHW shares in low energy buildings No. 5 and No. 9. The system boundary used for the standard system was the generator system. For the two compact units, an extended monitoring has been performed which includes also the auxiliary input to the sink pump. Performance of the brine-to-water systems is presented in chap. 3.2.2, performance of the air-to-water heat pumps in chap. 4.3. Details on all systems and the results of the field monitoring are found in Appendix B.

2.2.2 Canada: 2 EQuilibrium house systems

In Canada, two EQuilibrium^{TM} houses as winning concepts of the EQuilibrium Initiative of the Canadian Mortgage and Housing Corporation are in field monitoring. Both houses strive to achieve a net zero energy consumption according to the conditions of the contest. Both

building systems include a roof-integrated solar Photovoltaik/Thermal (BIPV/T) system and a heat pump. In the one system concept the BIPV/T is used for clothes drying, DHW preheating and to heat a concrete floor slap. In the second system the heat of the BIPV/T is either directly transferred to a thermal energy storage or serves as heat source of a heat pump. As back-up, the heat pump can be linked to a second ground source by an intermediate circuit.
Moreover, the building concepts include further efficiency technologies like south-oriented windows and massive construction to store passive solar gains, ventilation heat recovery and waste water heat recovery. Fig. 16 shows the building and a system concept.

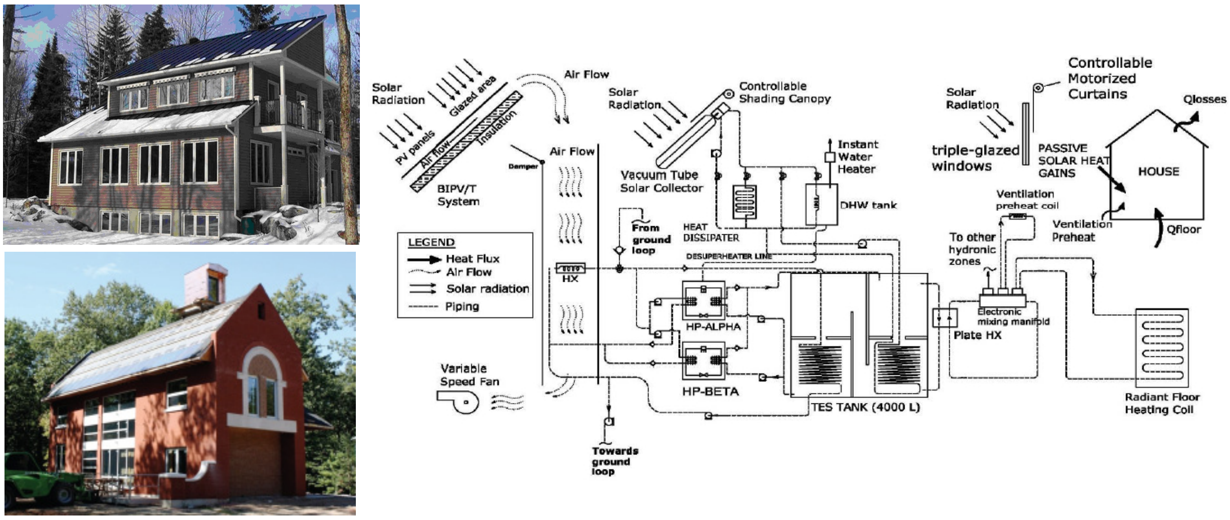


Fig. 16: EQuilibrium™ houses EcoTerra and Alstonvale Net Zero Energy house (left) and concept of the Alstonvale Net Zero Energy house (right, Pogharian and Candanedo, 2008)

The two Canadian systems are documented in two System Concept sheets.

2.2.3 Switzerland: 2 systems with ground-coupled passive cooling

System boundaries:

Two system boundaries have been evaluated. The generator performance factor refers to the produced energy of the heat pump and comprises the heat pump, an eventually installed back-up heater and the heat source system including brine pump and control. The system performance factor refers to the used energy, i.e. it additionally includes the storage losses of an eventually installed heating buffer storage and a DHW storage as well as the storage loading and sink pump.

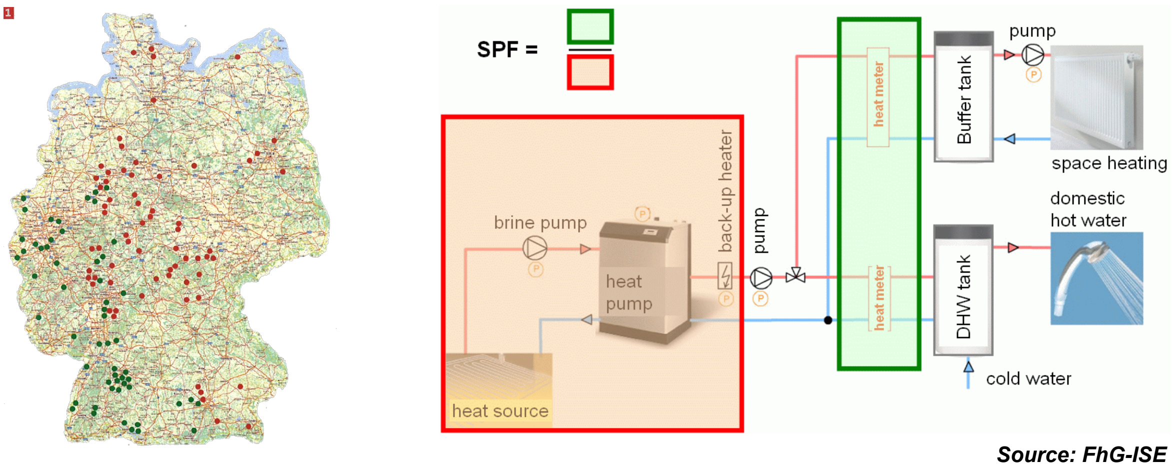
Outline of the field monitoring:

In Switzerland, two ground source heat pumps systems have been field monitored. Both ground-source systems are installed in certified MINERGIE®-buildings, one multi family building according to MINERGIE-P® in Basel and one single-family building according to MINERGIE® in Muolen, St. Gall. Both systems reach a good overall seasonal performance factor of 3.9 and 3.8, respectively. But both buildings have also optimisation potentials, which have partly been implemented during the field monitoring. Results of the Swiss field monitoring with a focus on the ground-coupled passive cooling operation are given in chap. 5.4.
Moreover, both systems have been documented in Best Practice sheets.

2.2.4 Germany: Extensive heat pump field tests in low energy and existing buildings

2.2.4.1 Overview of field tests "HP Efficiency" and "HP in existing buildings"

An extensive field test to analyse the performance of heat pumps in low energy houses („HP-Efficiency“, http://wp-effizienz.ise.fraunhofer.de) is carried out in co-operation with 7 heat pump manufacturers and 2 utilities described in Günther and Miara (2010).
Another field test is dedicated to the investigation of heat pumps as replacement for boilers in existing buildings, which is accomplished in co-operation with the German utility E.ON (HP in existing buildings, http://wp-im-gebaeudebestand.de).



Source: FhG-ISE

Fig. 17: Plant locations and system boundaries of the "HP-Efficiency" and "HP in existing buildings" field tests (Günther and Miara 2010)

Fig. 17 left gives an overview of the plant locations in Germany, where the green dots correspond to the HP Efficiency project and the red dot correspond to the HP in existing buildings project. Fig. 17 right gives an overview on the system boundary in the two field test projects, which corresponds to the seasonal performance factor "generator" based on the produced heat of the generators heat pump and direct electrical back-up heater divided by the electricity consumption of the heat pump, control and source pump.
Tab. 2 gives an overview of the design parameters of the buildings in the field tests.

Tab. 2 Design characteristics of the houses and installed systems in the German field tests

Project	ø A_e [m^2]	SH energy need [kWh/(m^{2}a)]	HP-capacity [kW]	Temperatures [°C]
HP-Efficiency (≈ 100 plants)	ø 192	20-50 (calculated need)	5-10	30-35 (floor heating) 45-65 (Radiator) ≈50 (DHW)
HP existing buildings (≈ 75 plants)	ø 190	ø 182 (fuel consumption)	ø13.8 (B/W) ø14.5 (A/W)	40-45 (floor heating) 45-65 (Radiator) 45-60 (DHW)

2.2.4.2 Outline of buildings in the German field test HP Efficiency

Fig. 18 left shows the heated area of 85 buildings which were evaluated in the field test. The heated area of the houses ranges from 120 to 370 m² with an average of approximately 198 m². The distribution of the measured heating energy consumption is shown in Fig. 18 right. Heating energy consumption ranges in 2008 from 25-130 kWh/(m²a) and is in the range of 32-170 kWh/(m²·a) in 2009. According to this evaluation only 42% of all considered objects (24 of 57 objects in 2009) fullfill the criteria of a low energy house with a space heating energy need lower than 60 kWh/(m²·a), as shown by the grey rectangular for the years 2008 and 2009, respectively.

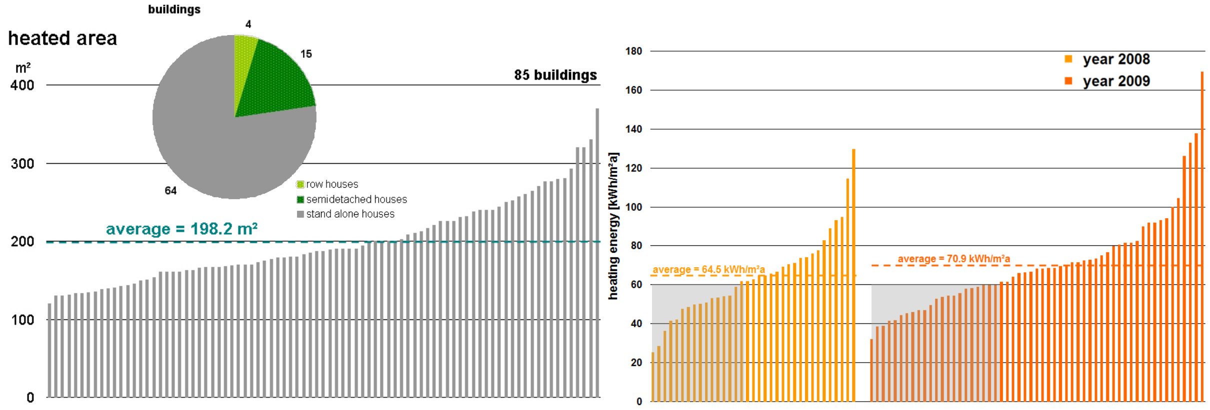


Fig. 18: Heated area (left) and measured space heating energy consumption of objects in 2008 or 2009 (right)

Moreover, the measured space heating energy consumption of the two years has been compared to calculated values according to German Building Directive EnEV (2009).
The average temperatures during the heating period from January to April and September to December for the two examined years are 6.7 °C in 2008 and 7.9 °C in 2009 according to daily average temperature of German Weather Service (DWD). Therefore, the energy consumption in 2009 should be lower. However, the average energy consumption is with 6.4 kWh/(m²·a) slightly higher as shown in Fig. 19 left and some of the monitoring objects show quite a high deviation between the two years, indicating that the energy consumption may be more influenced by user behaviour or other facts. Nevertheless, the average values do not deviate too much between the years although they deviated in the wrong direction. Fig. 19 right depicts the different consumptions sorted by the difference between the calculation and consumption, starting with the lowest consumption. 17 of the 28 evaluted objects show a deviation lower than 12%, the difference between calculated and measured values on the left side adds up to 40 kWh/(m²a), while objects where measured values exceed the calculated values have an averaged difference of 20 kWh/(m²a).

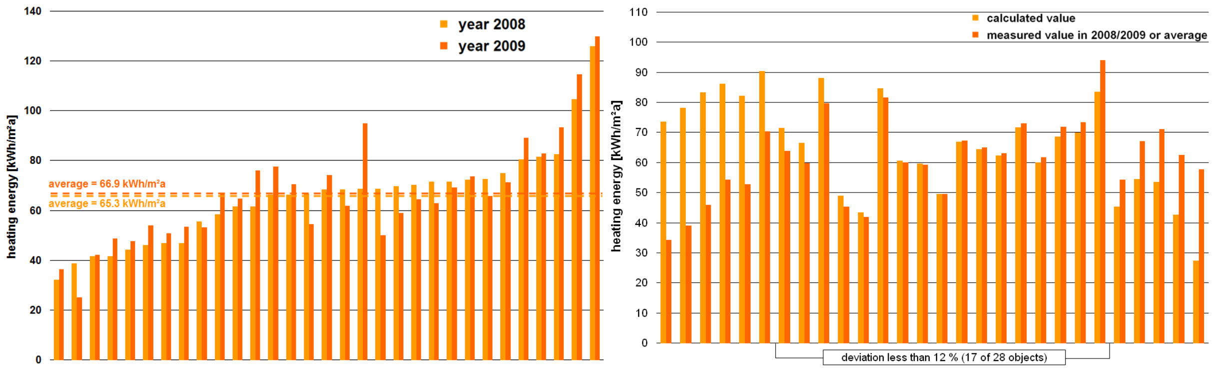
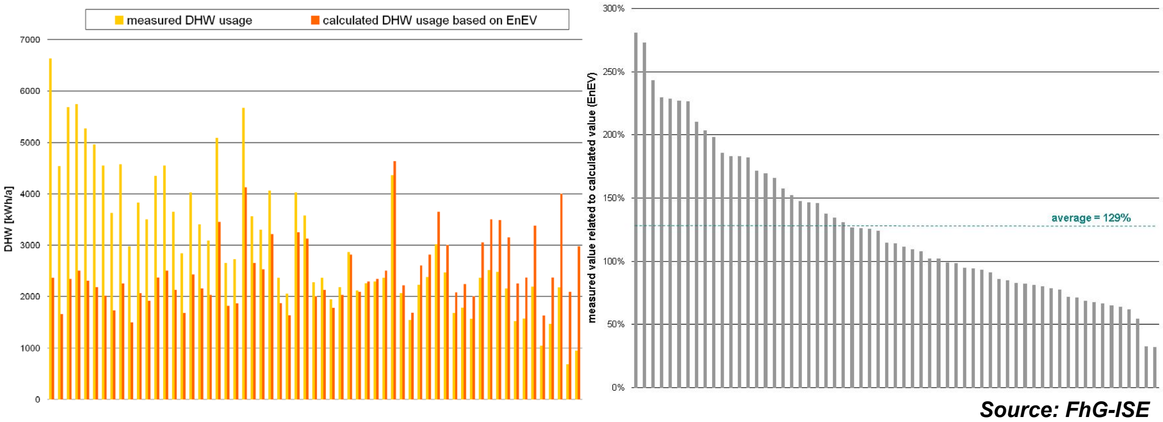


Fig. 19: Comparison of space heating energy demand for the years 2008 and 2009, sorted according energy consumption (left) and difference between measured and calculated values (right)

Fig. 20 shows a comparison between the measured DHW consumption and calculated DHW needs. According to the EnEV (2009) the average DHW usage is fixed to a value of 12.5 kWh/(m²·a), i.e. neither a number of the residents nor any typical consumption habit is considered. On the left hand side of Fig. 20 left objects with higher measured DHW usage are given, the objects with higher calculated DHW usage are found on the right.



Source: FhG-ISE

Fig. 20: Comparison of measured and calculated DHW use based on EnEV (2007)

The objects are sorted by the relatively deviation between the measured and calculated value. Fig. 20 right shows the measured value related to the calculated one. It indicates a huge deviation in both directions, which ranges from 32% up to 281% with an average value of 129%. Considering the EnEV value related to used energy, storage and distribution losses of 30% on average seems reasonable. However, even regarding a fluctuation rate of +/-20%, 57% of the objects would be inadequately calculated.

2.2.4.3 Heat pumps types in the German field test HP Efficiency

Fig. 21 gives an overview of different aspects of the monitored heat pumps in the German field test HP Efficiency.

Since Sept. 2009 all 110 heat pumps in the project have been installed, but the described evaluation only refers to 89 systems, of which the main heat source investigated is the ground (62 systems), which can be further decomposed into 45 vertical borehole heat exchanger systems and 17 systems equipped with horizontal ground collectors. 23 air-source heat pumps comprise 14 with outdoor evaporator set-up, while 9 are located indoor.

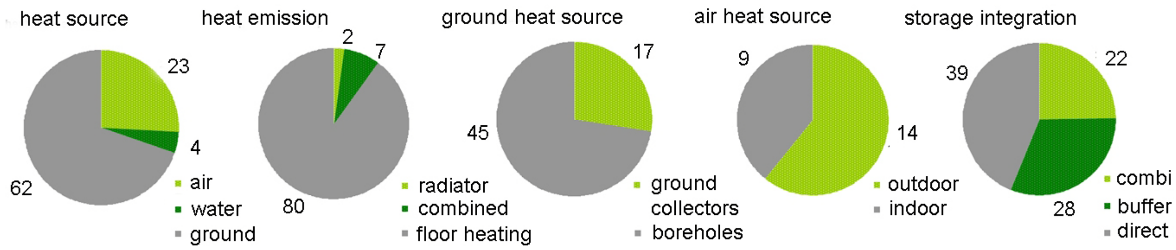


Fig. 21: Heat pump types in the HP-Efficiency field test (from left to right): Heat sources, emission systems, ground-source type and air-source set-up as well as systems

Moreover, four water-source heat pumps have been monitored, but only three are considered in the evaluation. Concerning the system configuration of the emission system, the majority of 80 installations is equipped with floor heating, while merely 2 radiator-only systems are contained. 7 systems comprise a floor heating with additional radiators. 22 systems are connected to the emission by combi-storage, 28 by buffer storage and 39 are directly connected to the floor heating system without storage.

2.2.4.4 COP values of the field monitored systems

Fig. 22 shows the available measured COP values of the field monitored heat pumps. These test results are either based on the standard EN 14511 (2007) or the formerly EN 255-2 (1997), which was partly based on different test points and did not have fixed mass flow rates during testing, which were introduced in the newer standard. Due to these changed test conditions, basically the fixed mass flow conditions, the COP values according to EN 14511 (2007) are up to 5% lower than the old ones according to EN 255-2. On the other hand, COP values improved during the years as shown in Fig. 28.

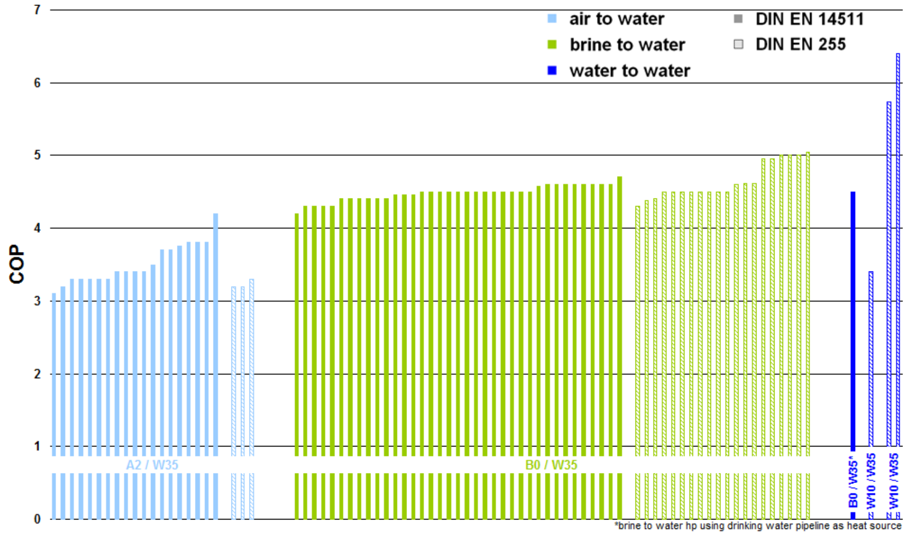


Fig. 22: COP values of the monitored systems based on standards EN 14511 (2007) or EN 255-2 (1997), respectively, for the denoted test points.

The COP values of air-to-water heat pumps based on DIN EN 14511 vary from 3.1 up to 4.2 while the older tests according to EN 255-2 are between 3.2 and 3.3. The COP values of the assessed brine-to-water heat pumps range from 4.2 to 4.7, or 4.3 to 5.0 according to EN 255-2. Besides the standards of measuring, the COPs of water-to-water heat pumps also differ by the temperatures. While one heat pump reaches 3.4 with W2/W35, COPs of 5.7 and 6.4 are reached at W10/W35.

2.2.5 Japan: Inverter-controlled ground source heat pumps in cold climate

Field tests have been accomplished by the University of Hokkaido in Sapporo. Two field tests of inverter-controlled ground-coupled heat pump systems installed in low energy houses in the cold climate zone of the Hokkaido Island have been monitored.

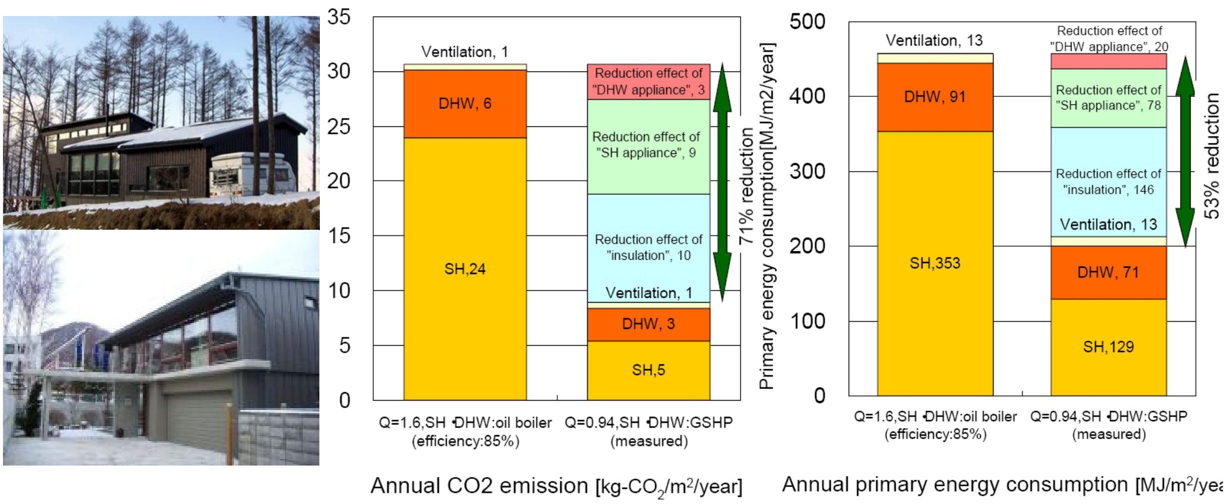


Fig. 23: Field monitoring houses in Hokkaido (left) and environmental benefits by the heat pump applications evaluated in the second field test (Nagano et al., 2009)

Fig. 23 left shows the two field test plants. Both field tests reached a good seasonal performance, in the first field test above 4.5 for the space heating only operation, while the DHW has been produced by a CO_2 heat pump water heater (Eco-Cute system). In the second field test, an overall SPF of the generator system of 3.8 was reached including a 3 m^2 solar evacuated tube collector.
Compared to common houses of the region equipped with oil boiler, significant primary energy and CO_2-eq-emission savings have been evaluated in the field monitoring plants in the second field test plant.
Fig. 23 right shows a decomposition of the savings, for instance, 55% of the CO_2-eq-emission savings are due to the system performance, and 45% come from the thermal insulation of the house.
Each of the two field monitoring projects has been documented as Best Practice Sheet.

2.2.6 Norway: Prototype propane W/W heat pump installed in a passive house

Norway has performed a field test of a 3 kW propane water-to-water heat pump prototype for space heating and DHW in simultaneous mode by desuperheater which has been developed at the NTNU in co-operation with SINTEF Energy research in Trondheim. The system concept of the prototype is described in detail in part 2 of the final report of Annex 32.
For field testing over two heating periods the heat pump prototype was installed in a passive house with a design heat load of 2.9 kW in Flekkefjord in southern Norway. Lake water with a temperature range of 5-15 °C is used as heat source.
The system reached an SPF of 3.7 based on used energy after the storage without considering the pumps and 3.1 including the pumps. These values correspond to about 70% energy saving compared to a direct electric heating system. During testing of the heat pump unit optimisation potentials concerning the evaporator and the expansion valve operation have been detected. Details of the results of the field test are documented as System Concept Sheet.

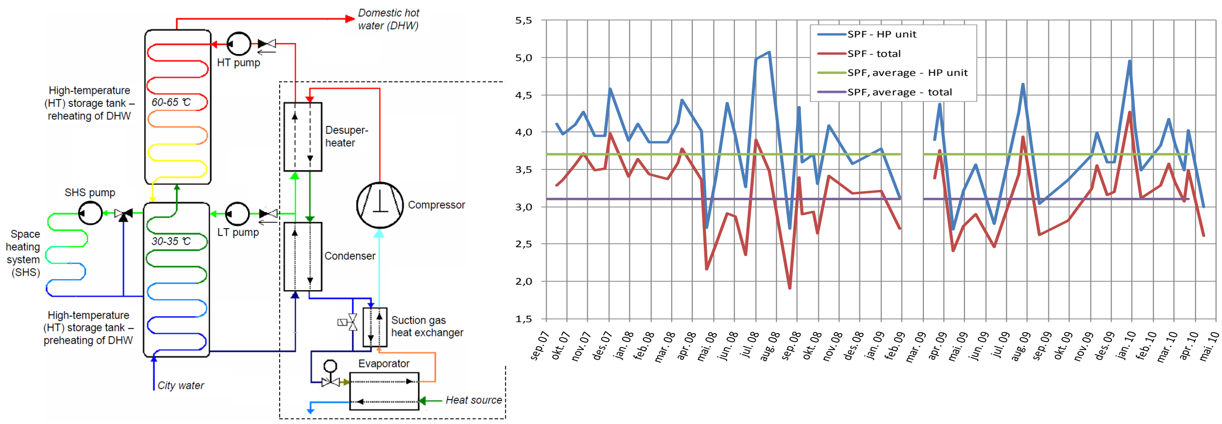


Fig. 24: Norwegian prototype propane water-to-water heat pump (left) and results of the 2 year field monitoring (Justo Alonso and Stene, 2010).

2.2.7 United States of America: IHP prototype and ground-source heat pump field test

The USA developed a highly integrated heat pump (IHP) prototype, which covers all building functions including dehumidification. Both the ground-source and air-source prototype have been lab-tested and simulated. They achieved significant energy savings compared to a benchmark system according to the state-of-the-art minimum DOE efficiency requirements.

Details on the US prototype developments are given in the part 2 of the final report of Annex 32 on the prototype developments.
Currently, field tests of the prototype developments are in preparation in several houses. An outline of the system concept and the preparation of the field tests are documented in a Best Practice Sheet. Fig. 25 left shows an outline of the intended field tests.

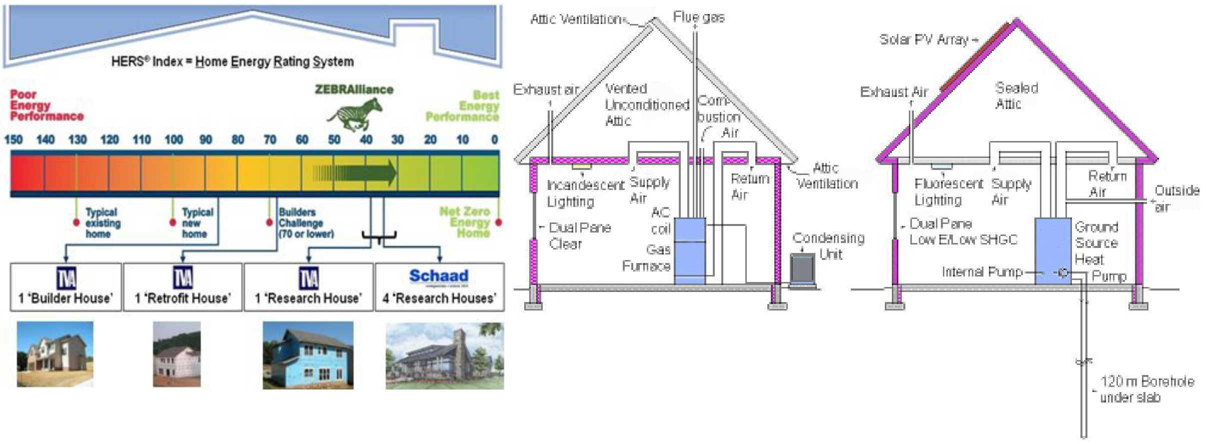


Fig. 25: Outline of the field tests of the US prototype integrated heat pump (left, Baxter, 2009) and house concept for the Habitat for Humanity housing field monitoring (right, Ellis, 2008).

At Hope Crossing, a Central Oklahoma Habitat for Humanity (COHFH) project, single-family brick homes were used for large-scale demonstration of affordable low energy housing.
Oklahoma City is considered to be a mixed-humid climate, with homes requiring significant heating, cooling, and dehumidification throughout the year. In typical single-story COHFH houses, either a central forced-air gas furnace coupled with a split-system AC or split-system central air-source heat pump system is used for heating and cooling. The Hope Crossing homes, however, were built with high-efficiency geothermal heating and cooling systems (GHP). The two system layouts are shown in Fig. 25 right.
All energy loads within the Hope Crossing homes are met with electricity. Additionally, many of the homes have energy-efficient insulation and Low-E windows. Some homes also have solar capabilities, equipped with solar PV panels on their roofs.The geothermal heat pump yielded up to 50% reduction of consumed annual site energy, corresponding to a reduction of 36% annual energy cost, and respective CO_2 emission savings. With additional low energy house construction energy savings could be increased to max. 75%. Both geothermal heat pump and low energy house construction proved to be cost-effective. PV panels can effectively provide summer peak electricity, but are less cost-effective at current PV panel prices. The field monitoring project has been documented in a System Concept Sheet.

2.3 Overview of ongoing field test projects

2.3.1 EU - HP Best Practice Database

A heat pump data base has been set-up to document field test results of heat pump systems. Presently, 53 ground-coupled field-monitored heat pump systems from all over Europe have been included, mainly measured within the EU project Ground-reach.
Information on the heat pump data base is found at

http://www.groundmed.eu/hp_best_practice_database

2.3.2 EU- Project SEPEMO Build (2008 - 2011)

The objective of the European project SEPEMO Build is to standardise the system boundaries for heat pump field monitoring and propose a common methodology to state heat pump performance. In the frame of the project, about 50 field monitoring projects will be contributed by the participants and added to the HP Best Practice Database. Thus, in the frame of the project, the database is extended to air-to-water heat pumps.

An outline of the systems in field monitoring as well as the state and further deliverables of the project can be found on the project website at

http://www.sepemo.eu

2.3.3 Lokale Agenda 21 Lahr Phase 2, Germany (2008-2010)

Based on the experiences of the field test of the 33 standard systems (see chap. 2.1.4) the second phase of the field test is dedicated to 11 innovative heat pump concepts which takes place from 2008 to 2010. The investigated system comprise 5 air-source systems mainly installed in single-family low energy houses, but also in one retrofit and 2 multi-family houses with 8-10 flats.

Heat pumps comprise mainly electrically-driven heat pumps, two with variable speed compressors, one the electronic expansion valve and one unit integrated in a combi-storage. Moreover, one gas-motor driven heat pump is included in the field test.

The 3 ground-water source systems are of higher capacity up to 120 kW for different row-houses.

The ground-source system applies a CO_2 heat pipe as heat source system and contains a variable speed compressor installed in a single-family new building.

First interim results have been published, which summarise ambiguous results. Some of the systems show excellent seasonal performance factors above 5, among those the CO_2 heat pipe, other system show a rather poor performance. Final results will be published in 2011.

Information website: http://www.agenda-energie-lahr.de/WP_FeldtestPhase2.html

2.3.4 Field test WPDirect in Belgium (2008 - 2010)

Fig. 26: Monitoring sites (left), and involved institutions (right) in the Belgium field test (Hoogmaartens, 2010).

In Belgium the heat pump platform of Flanders performs the field test WP-Direct of 15 heat pumps which involves numerous manufacturers and Universities. The field monitoring plants will be measured for 2 years until October- November-December 2011. 15 heat pump systems (1 W/W, 5 brine-to-water (4 borehole, 1 horizontal collector), 6 air-to-water, 3 direct expansion-to-water) installed in 10 new building (constructed after 2005) and 5 retrofitted existing buildings. Calculated space heating demand ranges between 40-90 W/m^2 and the nominal capacity 8-16 kW (one of 24 kW). 2 systems are space heating-only, 13 with combined SH & DHW, 13 systems have floor heating, 1 system with floor heating and radiator heating, and one convector heating, 3 system have both a buffer and a DHW storage, 1 system has only a buffer and 10 have only a DHW-storage.
The location of the field monitoring plants in Belgium and the involved institutions are given in Fig. 26.

Further information in Dutch is found on the website http://www.warmtepomp.info

2.3.5 Extension of the UK Field trial

An extension of the UK field trial (see chap. 2.1.8) is planned for 2010 to 2011 which may include enhanced (fanned) radiators that have not been monitored, yet, and further investigation of domestic hot water production efficiencies.

Further details are given on the website of the Energy Saving Trust.

2.3.6 Field test HP:Monitor in Germany

As continuation of the field test HP-Efficiency in Germany the project HP:Monitor has started in 2009 and will be continued until 2013. About 100 further heat pumps of the involved 12 manufacturers and 1 utility are monitored for at least 30 months to include at least two full heating and summer periods.
The main objective of the field test is the independent and comparative assessment of actual heat pump systems of the single involved manufacturers by means of the seasonal performance factors.
The field test comprises systems, which have not been investigated in the HP Efficiency field monitoring project, e.g. direct expansion systems and systems with capacity control. More-Moreover, installation errors and weak points shall be detected and optimisation potentials shall be identified.

Fig. 27: Project partners of the HP:Monitor field test

The dissemination of the field test results shall contribute to a further acceptance of heat pumps. For the involved utility, opportunities of the application of smart metering techniques is a special interest. Fig. 27 shows monitoring sites and the involved German and Austrian manufacturers as well as the German utility.

Further information in German is found at http://wp-monitor.ise.fraunhofer.de

2.3.7 IEA HPP Annex on Field monitoring

In the Heat Pump Programme (HPP) of the International Energy Agency (IEA) a new project is lauched with the objective to gather and compare existing field monitoring project results based on the system boundaries used in the participating countries. As deliverable a data base of heat pump performance in the participating countries will be set-up or extended with the aim to demonstrate benefits of heat pumps by means of the seasonal performance factors reached.

Further information is found on the website http://www.heatpumpcentre.org

3 RESULTS OF GROUND- AND WATER-SOURCE HEAT PUMPS

3.1 Ground-source heat pumps

The ground is besides air the most commonly used heat source.
The COP of ground-coupled heat pumps has been increasing continuously, from an average of B0/W35 of 3.83 in 1993 to and average of 4.45 in 2008, so presently market available heat pumps tend to perform better than heat pumps of the last decade. The development of COP values from standard testing at the Swiss test centre WPZ and the applied refrigerants in brine-to-water heat pumps is shown in Fig. 28 based on Eschmann (2009).

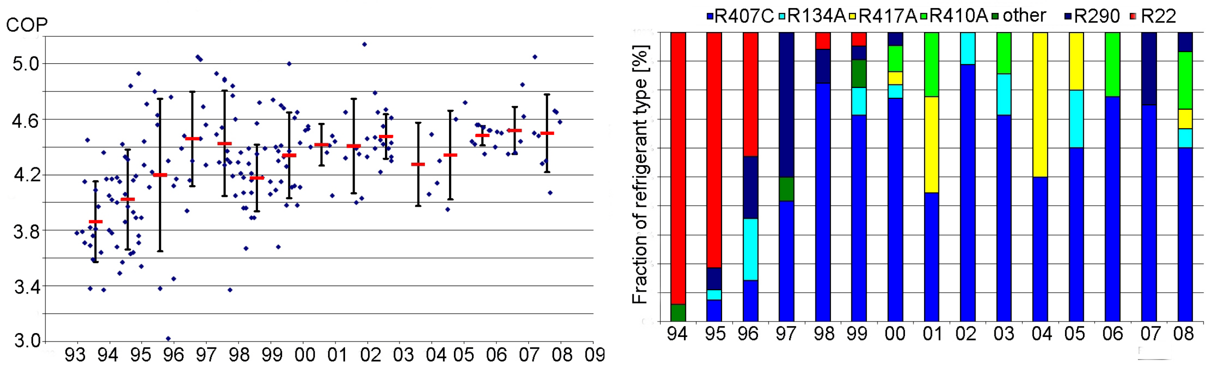


Fig. 28: Development of COP of tested ground-coupled heat pump at the Swiss test centre WPZ (left) and shares of applied refrigerants in the tested heat pumps, translated from Eschmann (2009)

3.2 Field test results of IEA HPP Annex 32

3.2.1 Overview of system configurations

The system concepts of ground-coupled heat pumps covered by the field test contributed to Annex 32 can be differentiated according to the different criteria:

- Heat source
  The field test comprises systems with horizontal collectors and with vertical borehole heat exchangers. No systems with direct expansion or with CO_2 probe were in field monitoring, which may have performance advantages due to the avoidance of the auxiliary energy for the source pump.
- Emission system
  Since virtually all systems are installed in low energy houses, floor heating systems are the predominant heat emission system
- Operation mode
  Most of the heat pumps are operated in alternate space heating and DHW mode. Some systems only cover space heating operation.

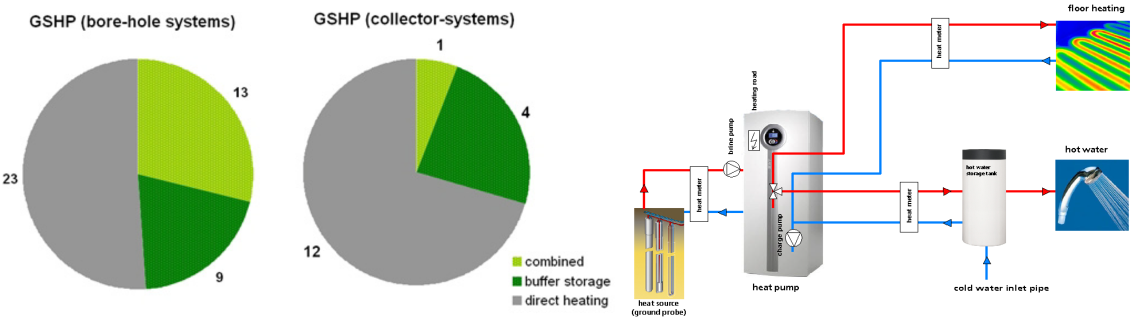


Fig. 29: Variety of system configurations in the German field test HP-Efficiency (based on Günther and Miara, 2010)

Fig. 29 left shows the variety of ground-source heat pumps in the German HP-Efficiency field test. In total, 62 ground-source systems have been evaluated. The majority of 35 systems are directly coupled to the space heating emission system, but also 27 systems are decoupled by a storage, of which 13 systems comprise a combi-storage and 14 systems comprise a buffer storage for the space heating operation. Fig. 29 right shows a sketch of a typical system layout.

3.2.2 Seasonal performance factor of ground-source heat pumps

Fig. 30 depicts the monthly overall seasonal performance factor for space heating and DHW of ground-source heat pumps over the whole measurement period, starting with 32 heat pumps in the beginning of 2008 until December 2009 with 62 heat pumps. The average overall seasonal performance factor of the whole measurement period is 3.9. The number of systems evaluated is displayed as number in the green-grey bars. DHW shares are in the range of 20%. The shares for space heating energy in the summer months are influenced by different phenomena such as a certain constellation in systems with integrated combined storage units, where unclear temperature layering in these combined storage units leads to space heating energy registrations, which in reality are used for DHW.

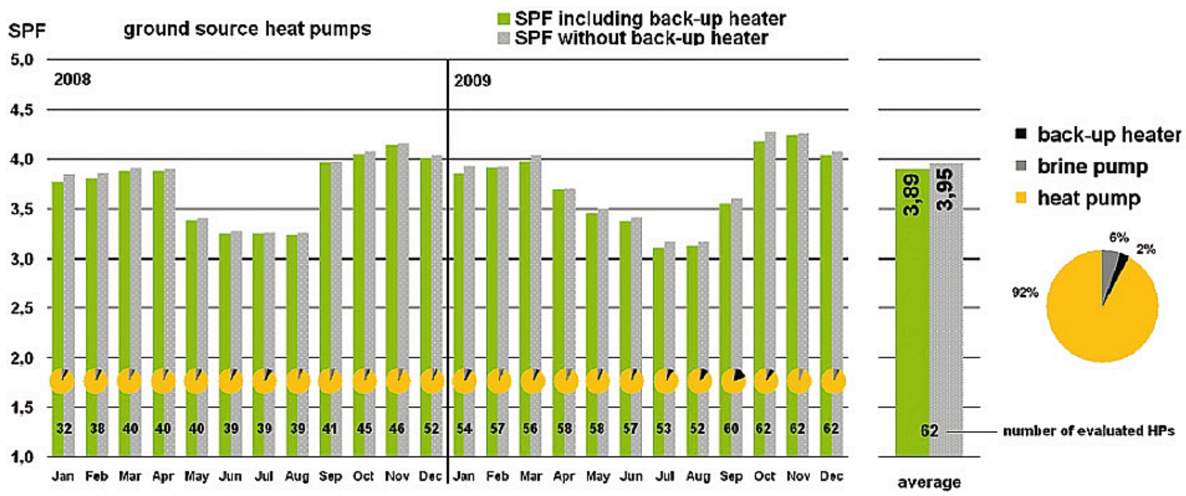


Fig. 30: Evolution of monthly seasonal performance factors of the field monitored brine-to-water heat pumps (Günther and Miara, 2010)

The evolution of the monthly performance factors reflect mainly the temperature levels of the source and sink side, which is detailed in Fig. 31 for the results of 2008.

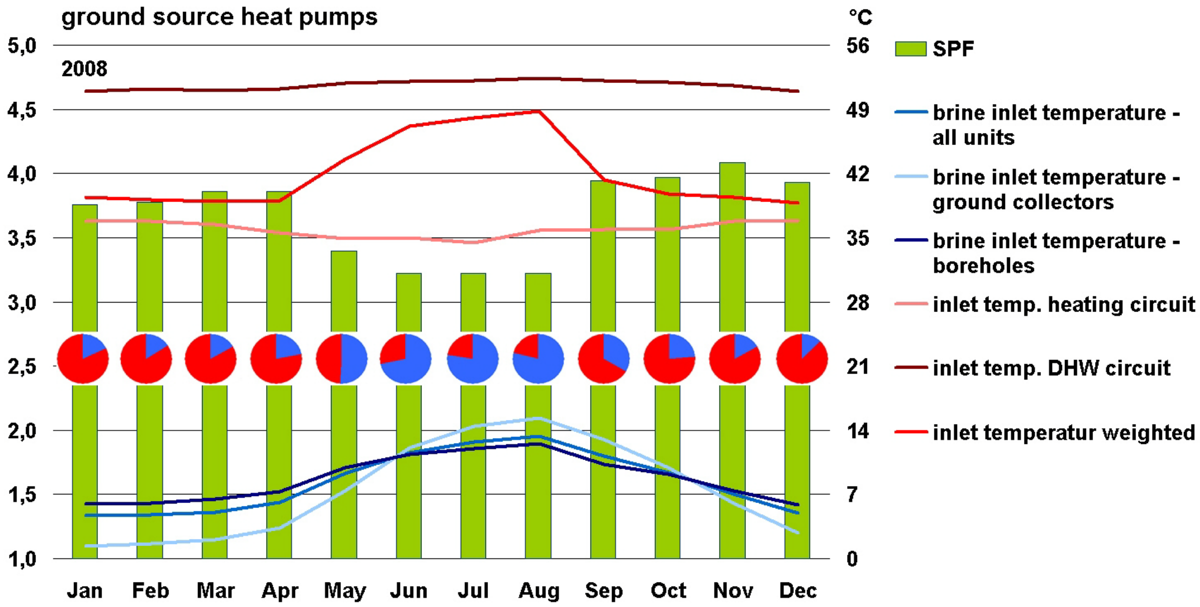


Fig. 31: Evolution of monthly SPF of the field-monitored ground-coupled heat pumps (Günther and Miara, 2010)

In wintertime, the source temperatures are lower, but due to much higher space heating than DHW shares with lower supply temperature requirements than in DHW mode the temperature lift is lower than in DHW-only operation and therefore, performance factors in the winter months are higher. In summer, though, DHW operation with higher supply temperature requirements is dominating, which can be seen by the increase of the weighted effective supply temperature for both modes depicted as red line. This decreases the performance factor despite the higher source temperatures in summertime. The source temperatures depicted in blue are split-up to systems with borehole heat exchanger and ground collectors. It can be seen, that ground collectors yield slightly higher inlet temperatures in summer months from June to October, while the boreholes have higher temperatures in wintertime.

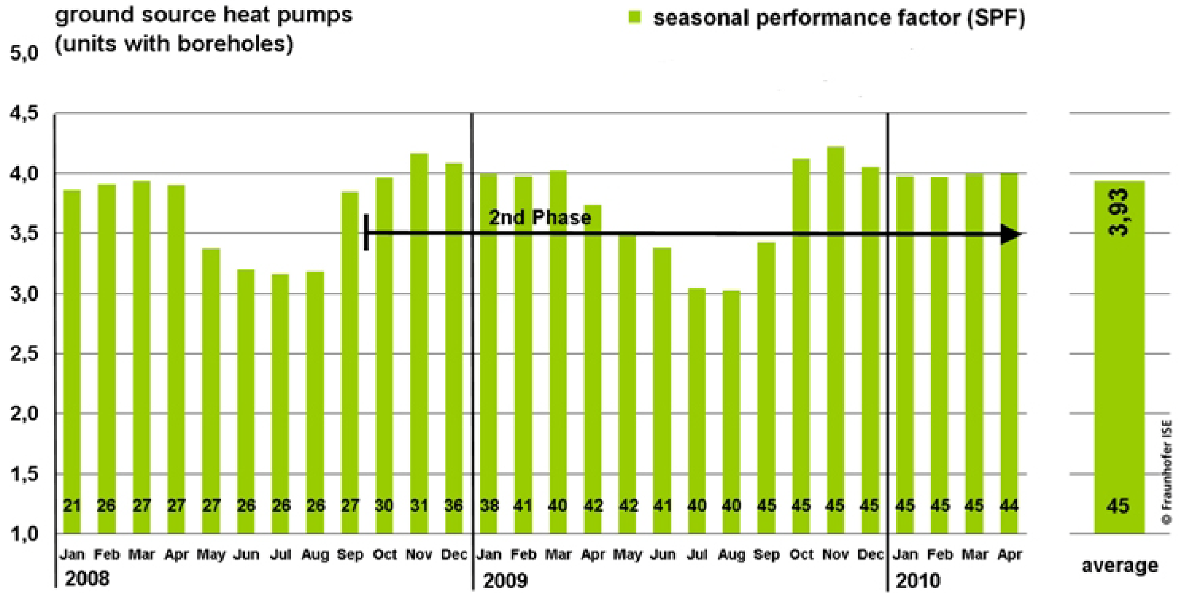


Fig. 32: Annual evolution of monthly seasonal performance factors of the field-monitored ground-coupled heat pumps with borehole heat exchanger (Günther and Miara, 2010)

Fig. 32 and Fig. 33 show the evolutions of ground-coupled heat pump systems separated into the about 45 borehole heat exchangers and about 17 systems with horizontal ground collectors. The systems with borehole heat exchanger outperform the ground-collectors by about 0.2 over the entire period. Mainly in the months at the beginning and end of each year, i.e. in the space heating period, the borehole heat exchanger systems show a higher performance. On the other hand, the ground-collector systems have higher monthly performance factors in the summer month from April to September, which, however, is the period of DHW-only operation with a lower energy fraction than in space heating mode. Thus, the slightly better seasonal performance can be explained by the amount of energy produced in the different periods. Nevertheless, this analysis is limited to the source effects, and there may be differences on the sink side as well as other impacts, which affect the seasonal performance factors.

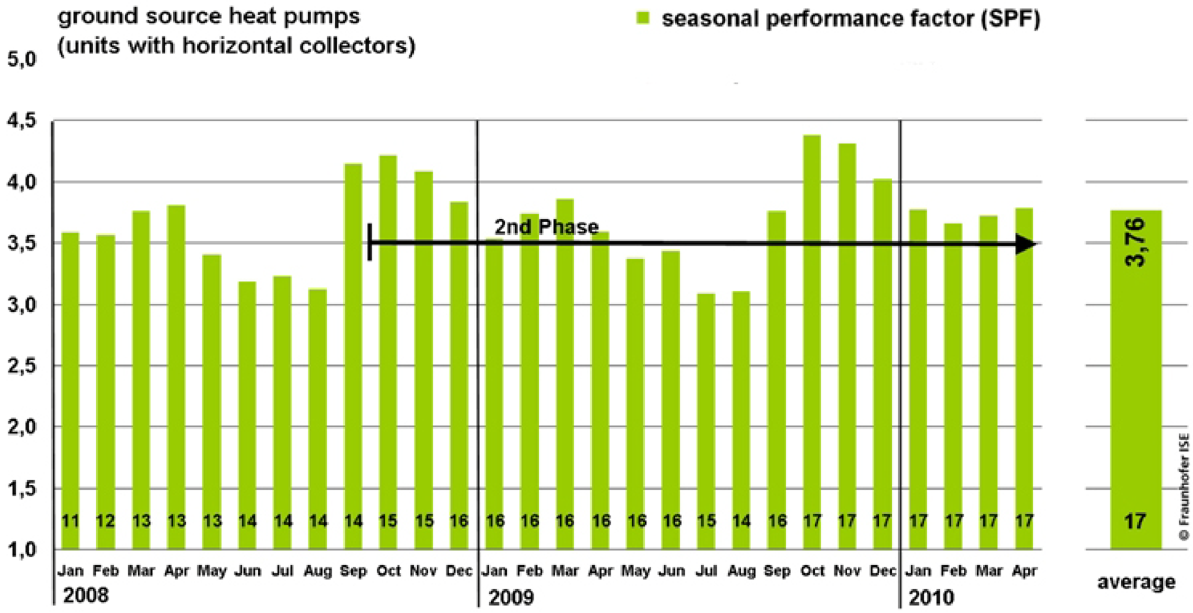


Fig. 33: Annual evolution of monthly seasonal performance factors of the field-monitored ground-coupled heat pumps with horizontal ground collectors (Günther and Miara, 2010)

Fig. 34 shows the seasonal performance factor of all systems with borehole heat exchanger in the year 2009 with the temperature difference between source inlet temperature and sink inlet temperature each in space heating and DHW operation. The SPF values of all borehole heat exchanger systems in 2009 are in the range of 2.7 to 5.5 with an average value 3.9. DHW energy fractions in the year 2009 are in the wide range of 2% to 38%. Concerning the temperature levels, average space heating supply temperature are in the range of 26.8 °C - 45.1 °C with an average value 36.2 °C, which is a rather high average value for floor heating systems. The average inlet temperature for DHW is 51.4 °C. Moreover, the back-up fraction are in the range of 0 (monovalent system) to a rather high fraction of 13%, while the auxiliary energy consumption of the source pump is in the range of 1-11% with an average value of 5%, which is a good value.

In general, the temperature lift is one of the major impacts on the seasonal performance factor. Even though this tendency is confirmed, since at the end of higher performance factors, the temperature difference between the source inlet and the inlet temperature to the emission system in space heating and DHW tend to be lower. There are some exception of system with low temperature lift, which do not reach a high performance factor, for instance just at the beginning system number 63, as well as system 129 and system 38. On the other hand, system 28 and system 108 are in the range of average temperature lift, but reach a very high overall seasonal performance factor of 4.4.

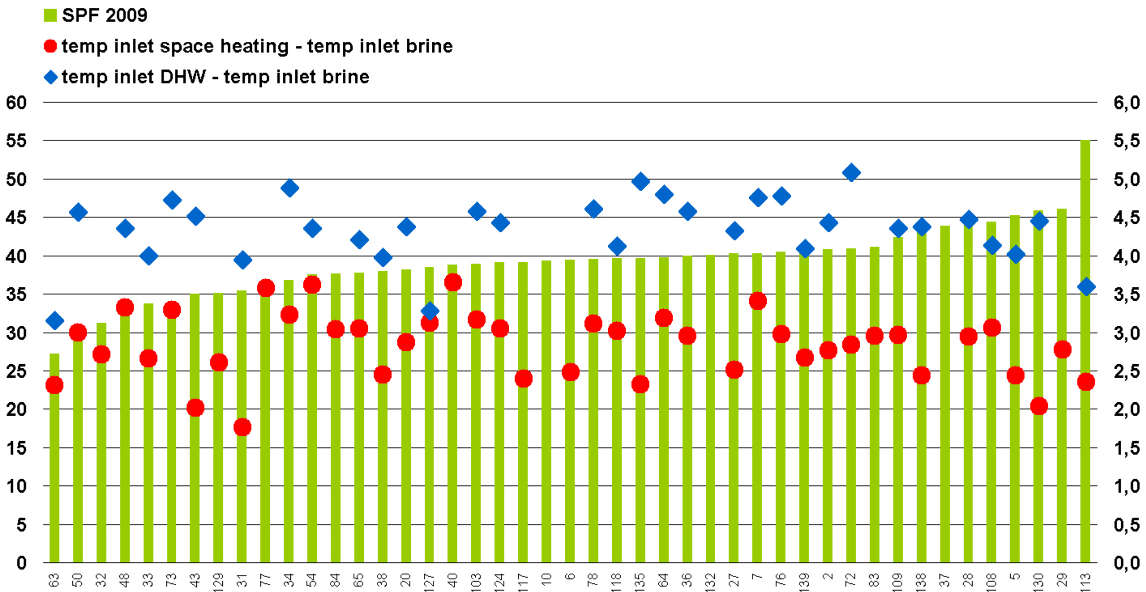


Fig. 34: Seasonal performance factors of the single field-monitored systems with borehole heat exchanger including the temperature lift in space heating and DHW operation for 2009 (Günther and Miara, 2010)

Fig. 35 gives the same evaluation for the horizontal ground collector systems. The SPF values of the ground collector systems are a bit lower in the range of 3.1 to 4.5 with an average value 3.8 in 2009 at DHW energy fractions in the range of 5 to 21%. The average space heating supply temperature is 34.8 °C and ranges from 30.1 °C - 45 °C with an average inlet temperature for DHW operation of 50.9 °C. The back-up operation ranges from monovalent operation to a high fraction of 18%, and auxiliary energy consumption of the source pump is in the range of 2-10% with a good average value of 6%.

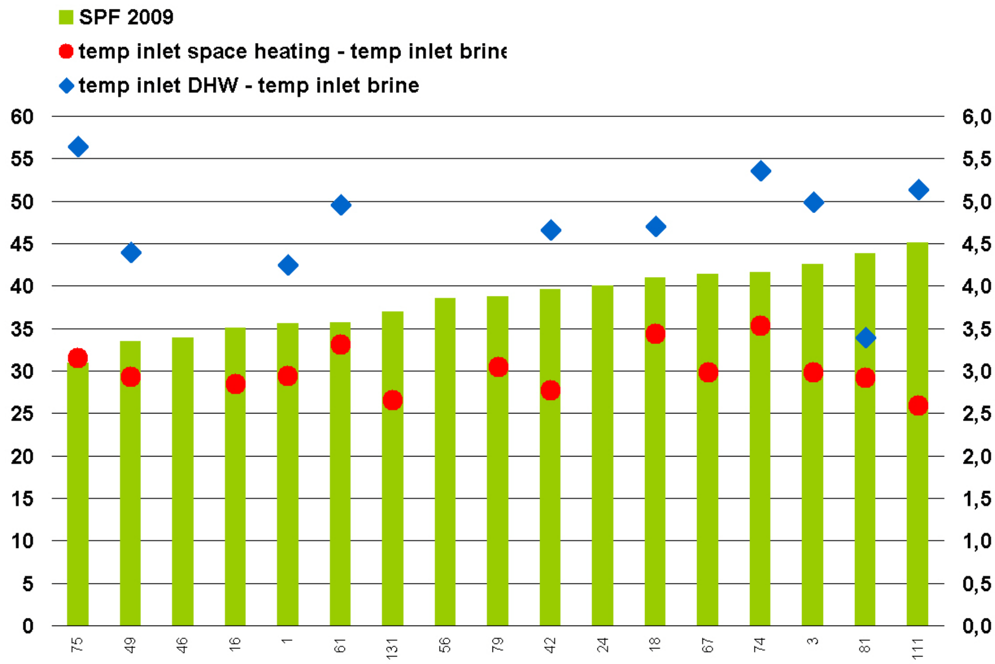


Fig. 35: Seasonal performance factors of the single field monitored system with ground-collector including the temperature lift in space heating and DHW operation for 2009 (Günther and Miara, 2010)

3.2.3 Results of ground-source heat pumps in the Austrian field test

In the Austrian field test four brine-to-water, one direct expansion (DX) and a ground-water source heat pump have been monitored. Two of the systems are used for space heating and DHW production, while four have space heating-only operation. Fig. 36 left presents the DHW fraction of the total heat energy needs of the two systems with combined operation, which is in the range of 10%-20%. All systems are designed monovalently and equipped with floor heating systems. All systems have a direct connection of the heat pump to the floor heating systems. More details on the system configurations are given in Appendix B.2.
Fig. 36 right shows the seasonal performance factors of the ground-coupled systems.
Overall seasonal performance factors for the combined operating systems are in the range of 4. The overall performance of the ground-water-source heat pump is only slightly higher than the ground-source heat pump. While space heating performance of the two combined systems is in the same range, the DHW performance is higher for the ground-water system, but also the DHW share is higher. The seasonal performance factor of the space heating mode is above 4 for all systems.
The space heating performance of the DX system is slightly above 4 and thereby comparatively low, as the auxiliary expenditure of the source pump is missing in the DX system. Therefore, many DX systems show quite high seasonal performance factors, even above 5, as shown in a former field test of the AIT, which is depicted in Fig. 10.
Summarising, the Austrian field test results of the SPF of combined operating systems are in the same SPF range as the German HP-Efficiency field tests.

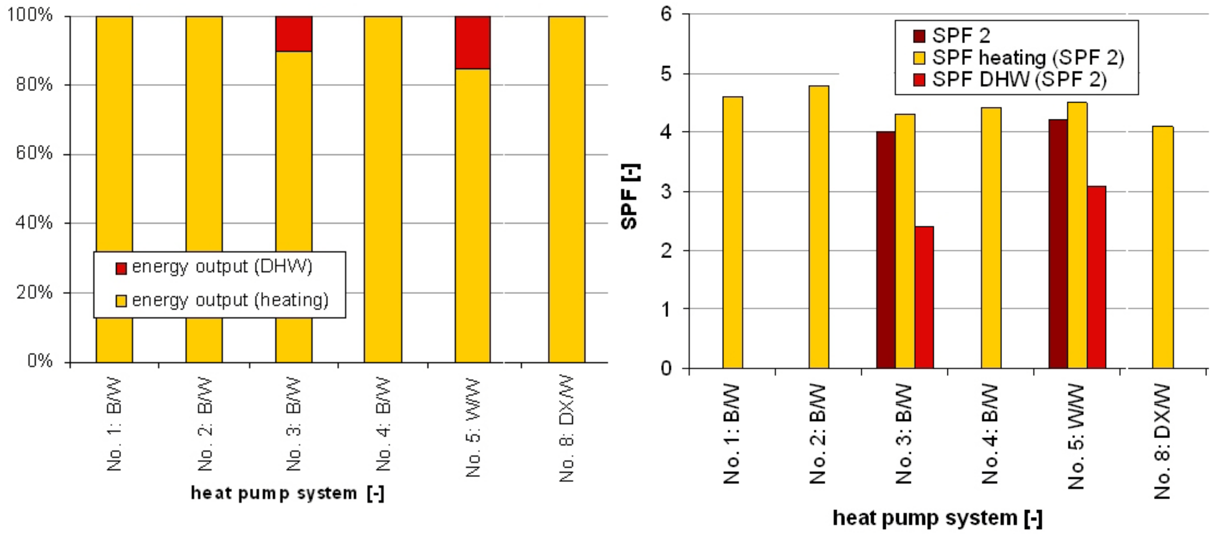


Fig. 36: DHW share and SPF "generator" in the Austrian field test (Zottl et al., 2010)

The seasonal performance is slightly above the average value in the German field test of 3.9.
System No.7 and system No. 9 have been documented in more detail in two Best Practice Sheets.

3.3 Evaluation of source and sink temperatures

In this chapter the characteristics of the source and sink temperatures of different ground-source systems are analysed. While the heat source temperatures are mainly influenced by the type of source system, i.e. the ground collector or borehole heat exchanger or even other variants like ditch collectors or ground basket, which, however, have not been monitored in the frame of the field monitoring projects in Annex 32, the sink system is mainly influenced by the type of emission system and the integration, i.e. the hydraulic complexity in terms of additional storages and number of valves and pumps.

3.3.1 Source temperatures dependent on heat extraction

Fig. 37 shows the brine inlet temperature dependent on the specific annual heat extraction for heat pumps with borehole heat exchanger. The evolution of the source temperature of all systems is shown as grey lines. Moreover, systems were sorted into 4 different classes according to the individual annual specific heat extraction from the ground. The average of the brine inlet temperature for each class is shown as coloured line. Heat pump systems with the lowest specific annual extracted heat of 41-59 kWh/(m·a) exhibit brine inlet temperatures in the year 2008 in the range of 7.5-12 °C. Heat pump systems with the highest specific annual extracted heat of 105-130 kWh/(m·a) show an alternation in the year 2008 from around 3 °C to 12 °C. A first conclusion is that systems with higher heat extraction have constantly lower brine inlet temperatures to the heat pump. However, the comparison shows furthermore that the borehole heat exchanger are generally quite long for the applications, since quite high average temperatures result. Differently formulated, high security factors are applied in the design, since the lowest reached inlet temperatures are between +3 °C and +7.5 °C. None of the monitored systems reaches negative temperatures, i.e. systems could be operated with water as heat transfer fluid, which would have a higher heat capacity enhancing the heat extraction and lower viscosity reducing the necessary pumping energy. In general, the design of longer borehole heat exchanger with lower extractions rates leads to higher source temperatures, but, on the other hand to higher cost for the source and tend to have a higher electrical energy consumption for the source pump.

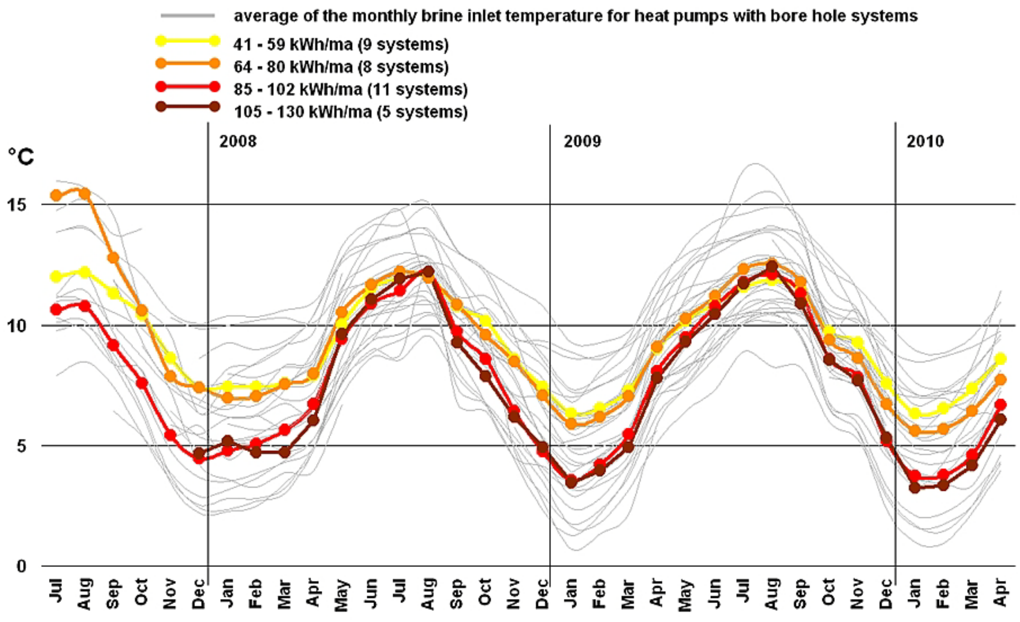


Fig. 37: Dependency of the brine inlet temperature on the specific annual heat extraction (Günther and Miara, 2010)

3.3.2 Source temperatures of different source systems

Fig. 38 shows the brine inlet temperature to the heat pump of ground-coupled heat pumps with boreholes in grey lines, the average of the brine inlet temperature as red line and in comparison the daily-averaged outside air temperature from the German weather service Deutscher Wetter Dienst (DWD) as blue line for the period from July 2007 to March 2010.

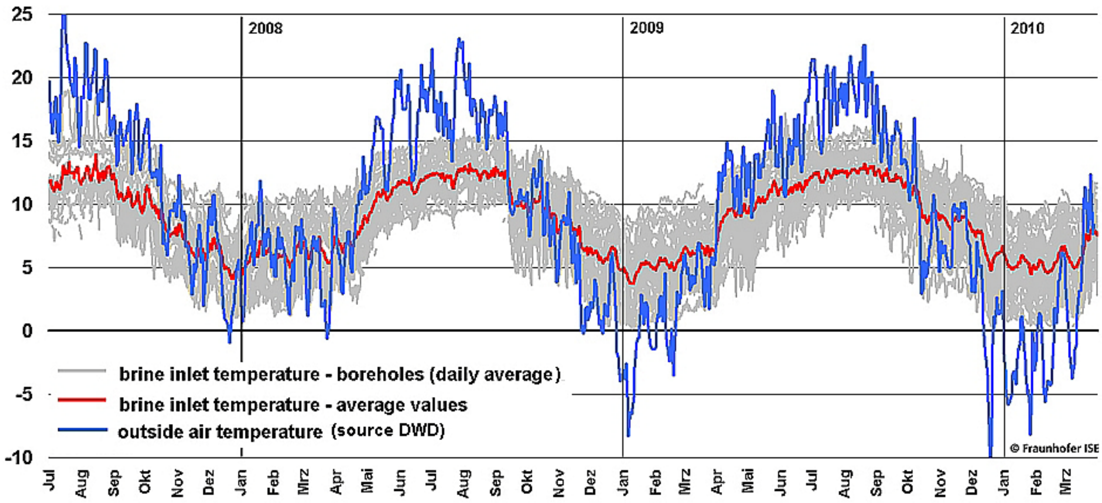


Fig. 38: Brine inlet temperatures of heat pumps with boreholes as heat exchanger (Günther and Miara, 2010)

The average outdoor air temperature forms a sine wave with annual peaks of about 23 °C and minimum values of around -9 °C to -5 °C in January and December each year. According to the correlation of heat pump operation and respective heat extraction of the ground, the brine inlet temperature follows the outdoor air temperature with corresponding peaks. No lag of the brine inlet temperature could be encountered by a detailed study of the temperatures. Thus, averaged inlet temperature of the primary circuit shows the same sine wave with reduced amplitude of 6-9 K, and peaks around 12 °C and minimum values around 4 °C. Even in the cold winter of 2009/2010 the brine inlet temperature does not fall below 0 °C. A settlement at a lower ground temperature level due to the extracted heat could not be observed on average in the first three winters, either, despite there is no recharging of the ground in summertime.

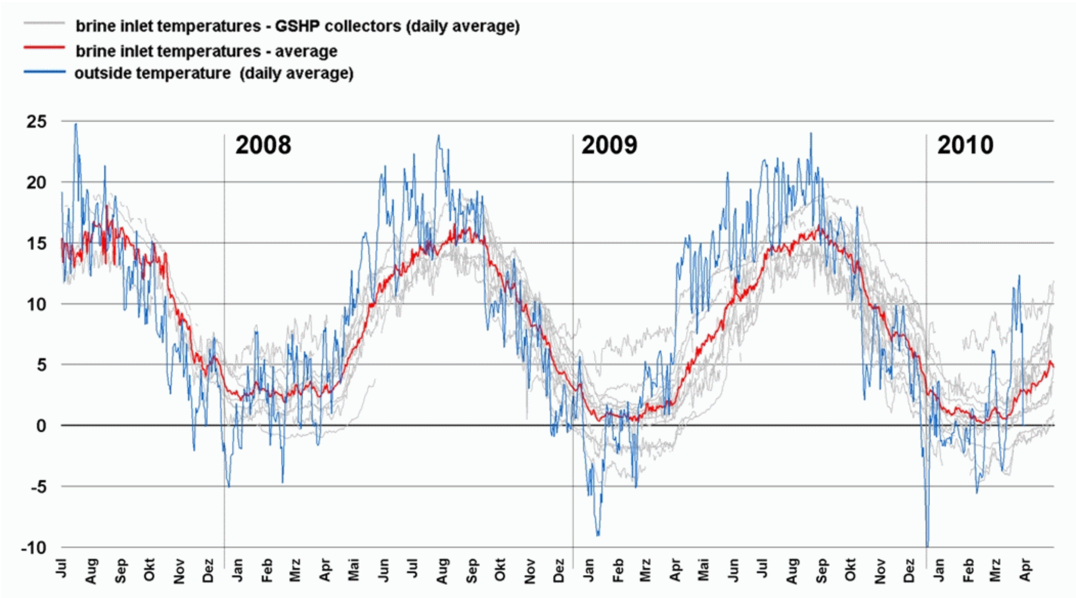


Fig. 39: Evolution of ground temperatures of horizontal ground-collectors (Günther and Miara, 2010)

Fig. 39 shows the brine inlet temperature of heat pumps with horizontal ground-collector, depicted in the same colours as in case of the borehole heat exchangers in Fig. 38. The average brine inlet temperature to the heat pump shows a similar sine wave as the outdoor temperature, but the coupling is closer than in case of the boreholes, since peaks at around 16 °C and minimums at around 2 °C. The amplitude of the sine wave is in the range of 3-6 K. Compared to the outdoor air temperature, higher air temperatures can be found in the period from April to September, while in the winter month, the temperature is about 5-10 K lower than the inlet temperature out of the ground collectors. Thus, the main advantage compared to outdoor air heat pumps is in the heating period, where also the highest energy fractions are required.

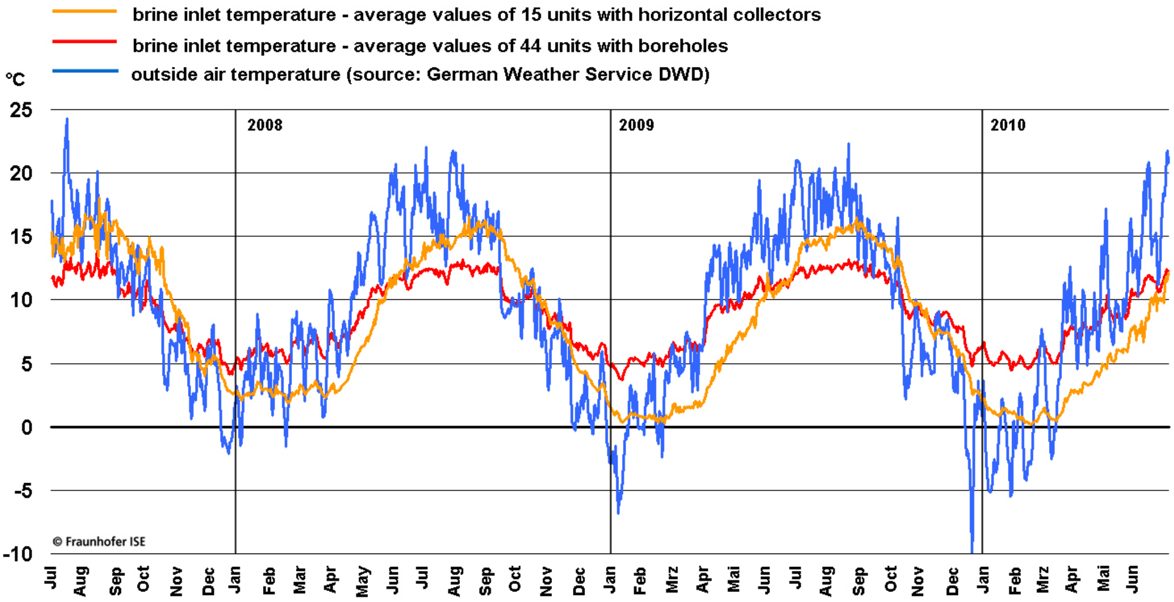


Fig. 40: Evolution of ground temperatures of horizontal ground-collectors (Günther and Miara, 2010)

Fig. 40 shows a comparison of the average inlet temperature to the heat pump of systems equipped with borehole heat exchangers and horizontal ground-collectors. In colder winters like in the year 2010, the ground-collectors are more affected by the outdoor air temperature than the borehole heat exchangers. On the other hand, in summer operation from April to September, inlet temperatures are up to 3 K higher than for system with boreholes. Therefore, the currently higher space heating demand in wintertime leads to higher seasonal performance factors of borehole heat exchanger systems. An increasing DHW share in low and ultra-low energy houses, though, may level out the currently higher seasonal performance factors of borehole heat exchanger systems.

The differences in seasonal performance factors in 2008 and 2009 for ground-coupled heat pumps is shown in Fig. 41 as a comparison of seasonal performance factors of vertical borehole heat exchangers and ground collectors for the single month. In order to receive comparable results, only systems from the 1st installation phase of systems are considered in the graph. The bars in the diagram show the differences of the SPFs for individual periods. The bar is coloured orange, when the averaged SPF of systems with boreholes exceed the averaged SPF of ground collector systems, and in the opposite case, the bars are coloured light blue. The advantage of 0.15 for borehole systems in the period of 2008 disappears in the following year. Both, collector- and borehole systems reach the same SPF, although the averaged value for the entire period shows an advantage for borehole systems of 0.07. Besides an impact of the temperatures of the emission system, the difference depends on the outdoor air temperature, since in colder months, the borehole systems have higher source temperatures and therefore a better SPF, while in the summer months with higher ground temperatures for the collector systems, the SPF of the collector systems is higher. However, due to the energy amounts in the single month, vertical borehole heat exchangers should yield higher seasonal performance factors, which are not confirmed in this comparison.

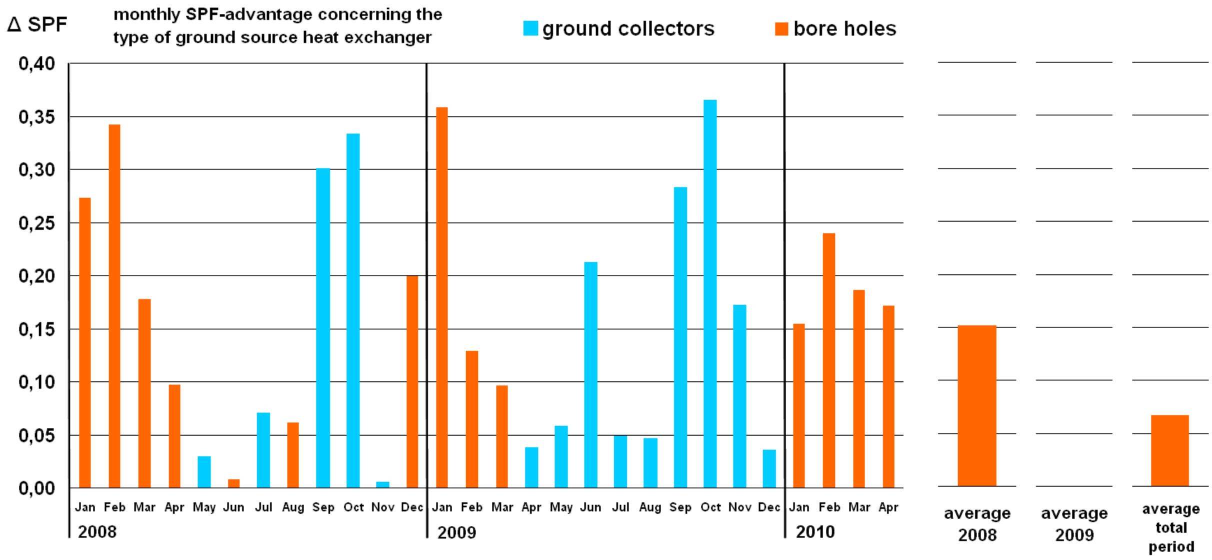


Fig. 41: Comparison of the monthly performance factor of borehole and collector systems (Günther and Miara, 2010)

3.3.3 Seasonal performance for sink temperature due to hydraulics and complexity

The emission temperatures of the system are affected by the hydronic configuration. Therefore, Fig. 42 shows the examination of the SPF according to different hydraulic set-ups and different degrees of complexity. The average SPF of all examined objects yields 3.8 in the period from November 2007 to October 2008. Three different set-ups of the integration of the heat storage have been analysed.

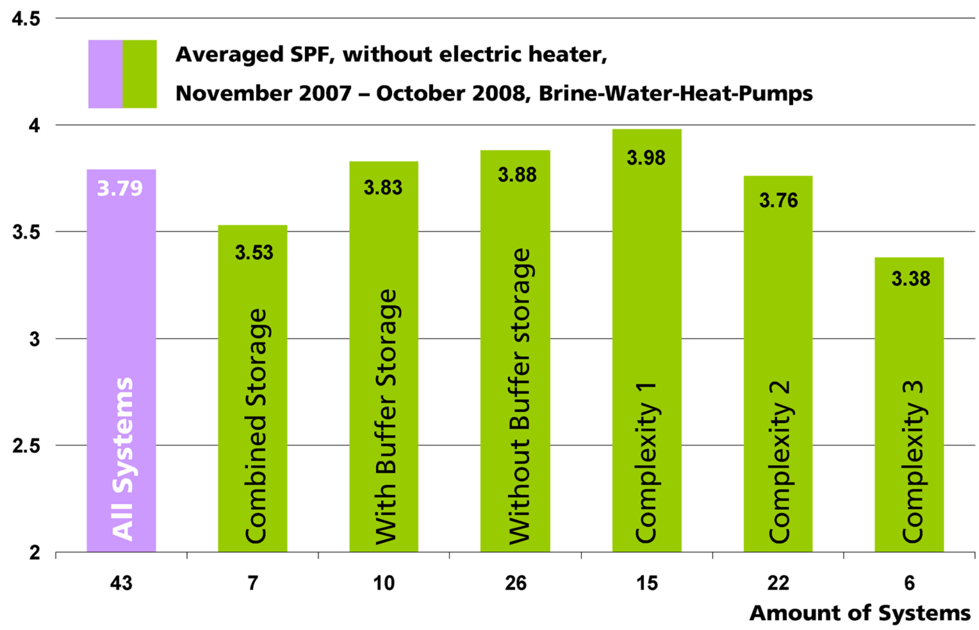


Fig. 42: Average SPF of ground-source heat pumps of different storage integration and degree of complexity (Günther and Miara, 2010)

The lowest SPF of 3.53 is reached by an integrated combined heat storage, i.e. a common storage for space heating and DHW. Heat pump systems with buffer storage, however, yield an average SPF of 3.83 and perform only slightly worse than heat pump systems without buffer storage which reach the highest SPF of 3.88. Another evaluation is made according to three degrees of system complexity evaluated by the number of pumps and valves, storages and additional heating systems besides the heat pump. Systems with a complexity degree of 1, i.e. low complexity, reach an SPF of 3.98, while the SPF decreases to 3.76 for complexity a degree of 2 and a SPF of 3.38 for complexity degree of 3. This confirms the findings of former field tests that complex systems with many valves and integrated storages often do not perform as good as simple hydraulic configurations due to storage and pipe losses, malfunctions and installation errors of valves, additional auxiliary energy consumption of the pumps and complex controls. Thus, the system hydronic should be as simple and robust as possible to guarantee a high performance factor, and system integration should be investigated thoroughly. For standard applications in single family houses, often simple standardised hydronic system configurations show the best real world operation.

3.4 Results of water-source heat pumps

Ground-water is the heat source with the highest and most stable temperature level, in particular in wintertime for space heating mode, but the fraction of ground-water heat pumps is generally low, on the one hand due to the fact that the use of ground-water is restricted to certain areas (no drinking water, ground-water close to the surface, good accessibility etc.) and since the connection of the heat source is expensive. Thus, only 2-3 systems have been monitored in the German field test HP-Efficiency during 2008 and 2009.
Two characteristic hydraulic schemes of the monitored systems are realized, the connection of the heat source with and without a separate heat exchanger as shown in Fig. 43. A separate heat exchanger avoids the direct contact between ground-water of the heat source and the evaporator in the heat pump, but has the disadvantages that the heat exchanger causes additional decrease of temperature level and it requires a second pump which leads to higher auxiliary energy consumption, which both lead to reduced SPF values compared to the other system layout.

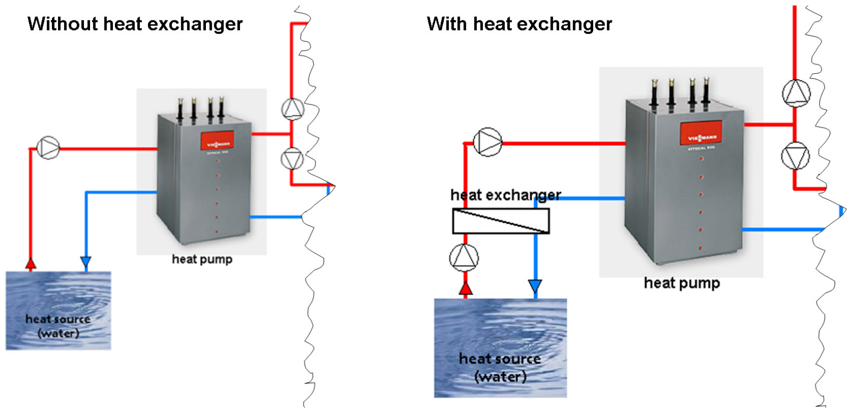


Fig. 43: Two types of connection of the ground-water source to the heat pump (Günther and Miara, 2010)

3.4.1 Seasonal performance factor of water-to-water systems

Fig. 44 shows the monthly-averaged performance factors and the whole period SPF of the 2-3 evaluated systems. Due to the low amount of water source heat pumps, the values only reflect tendencies. The SPF per month shows a similar behaviour in 2008 and 2009 with high values in the heating period from September until April, since supply temperatures are lower and source temperatures stay nearly constant all the year. Lower summer performance factors reflect the higher required sink temperatures for DHW operation.
In 2009, DHW fractions range between 4% and 16% with average DHW inlet temperatures

of 52.6 °C. Space heating inlet temperatures are quite high with 29.9 °C - 48.1 °C at an average of 40.7 °C.
Furthermore, one system has been optimized in 2009, which has a big influence on the overall SPF of all systems, as seen in Fig. 44 and confirms the strong impact of only one system for the small sample of 2-3 systems.
The overall SPF of 3.6 is lower than expected for the good and stable source temperatures of ground water. One reason for this is the relatively high fraction of auxiliary energy of 15%, which is basically due to the consumption of the well pumps.
This is on the one hand due to the difficulty to find adequate well pumps for the capacity of the heat pump, in particular for the small capacity range.

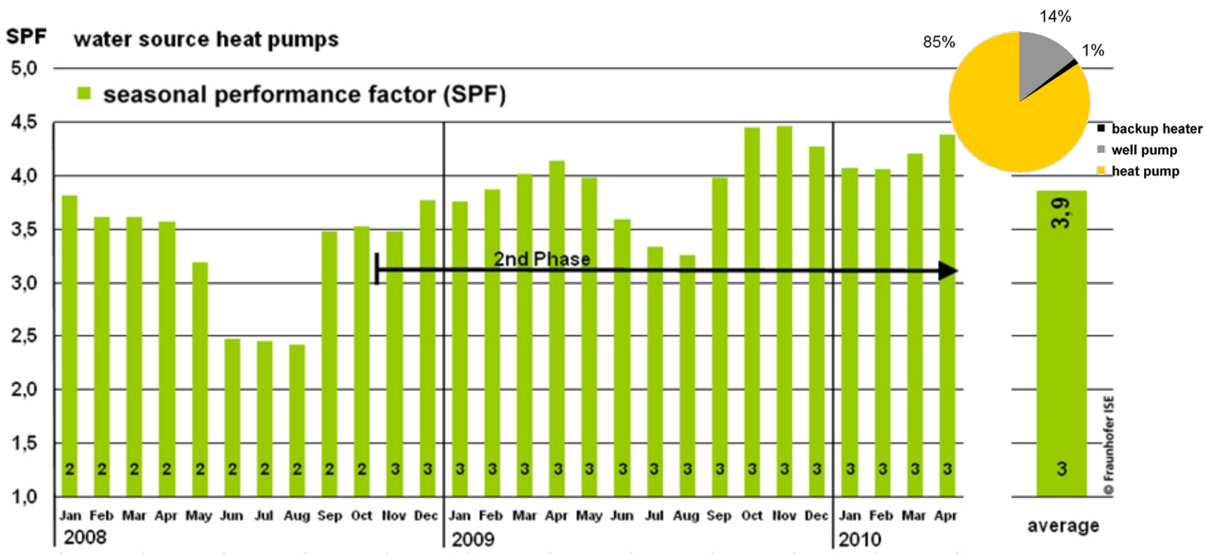


Fig. 44: Evolution of monthly performance factors of the monitored W-W heat pumps

On the other hand, the above mentioned system configuration with an additional heat exchanger leads to the necessity of a second source pump, which doubles the auxiliary consumption. Thus, the use of high efficiency pumps and a thorough pump design is essential in these system configurations.
For the year 2009 the seasonal performance factors are in the range of 3.3 and 5.1. Even though ground-water systems should be able to be operated monovalently back-up shares are up to 4%. The average auxiliary energy share ranges between 15% and 18%.

4 EVALUATION OF AIR-TO-WATER HEAT PUMPS

4.1 Introduction

Besides the ground, air-to-water heat pumps are the most common systems in Europe.
The COP of air-to-water heat pumps has been increasing continuously, from an average of A2/W35 of 2.35 in 1993 to an average of 3.35 in 2008, so presently market available heat pumps should perform better than heat pumps of the last decade. The development of COP values from standard testing at the Swiss test centre WPZ and the applied refrigerants in air-to-water heat pumps is shown in Fig. 45 based on Eschmann (2009).

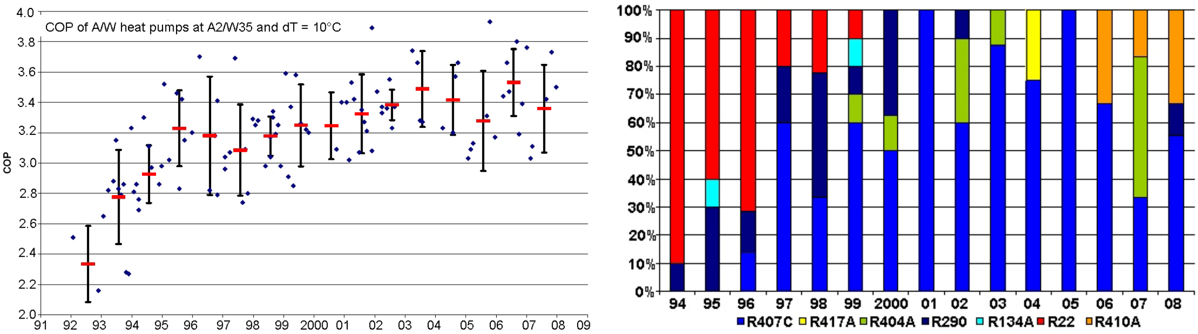


*Fig. 45: Development of COP of tested air-to-water heat pumps at the Swiss test centre WPZ (left) and shares of applied refrigerants in the tested heat pumps (right), translated from Eschmann (2009)^*

4.2 System configurations

Fig. 46 left shows an overview of different system configurations in the German field monitoring "Heat Pump Effiiciency". Both the indoor and outdoor set-up of heat pumps are represented with about 10 or more systems. On the sink side, the majority of the systems include a buffer storage for the heating operation, both as simple buffer storage as well as combined storage used for space heating and DHW. A sample configuration with integrated buffer storage is shown in Fig. 46 right. Reasons given for the integration of a buffer storage by the manufacturers are the more volatile heat source temperatures in air-to-water systems, so a storage gives more flexibility, when the source system is operated. On the other hand, the defrost operation of air-to-water heat pump by refrigerant cycle reversal extracts heat from the buffer storage, so heat extraction from the building in defrost operation is avoided.

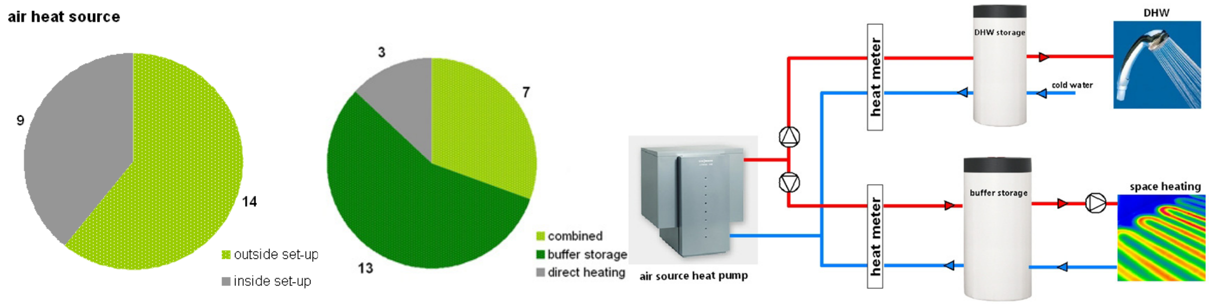


Fig. 46: System configurations of air-source heat pumps in the HP-Effiiciency field test (based on Günther and Miara, 2010)

4.3 Field monitoring results

4.3.1 Heat pump efficiency field results

Fig. 47 provides the average performance factor over 24 months of the evaluated air-source heat pump systems in the German HP-Efficiency project in the years 2008 and 2009. The average SPF of the whole evaluation period of 24 month yields 2.9.

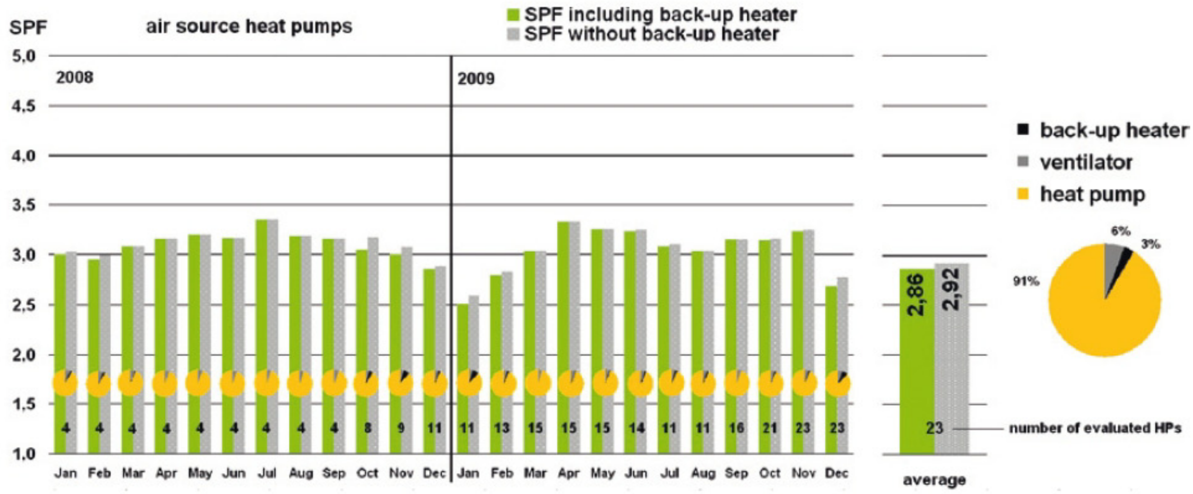


Fig. 47: Annual evolution of monthly seasonal performance factors of the field monitored air-to-water heat pumps (Günther and Miara, 2010)

The average SPF in the single years yielded similar performance factors of 3.0 for 11 systems in 2008 and of 2.9 as average of 20 systems in 2009. Each green/grey bar contains the number of the monthly evaluated objects at the bottom. In 2008, highest performance factors are reached in the summer, but there is higher space heating need also in summer (probably due to drying-out of the building) and only a small number of investigated systems.

In 2009, the highest performance factors are reached in the transitional period in April and May as well as in October and November due to the large fraction of DHW in the summer operation and since outdoor temperature are high in spring and autumn and required supply temperatures of the space heating operation low. In wintertime, the SPF is correlated to the development of the outdoor air temperature and lowest SPF values are evaluated in January and December between 2.5 and 2.7. Moreover, defrosting needs in winter further reduces the performance factor.

With 20 systems most of the systems have direct electrical back-up operation. The average back-up fractions of the entired period is 3%. In 2009 the back-up operation ranges between 0%-10%. The average auxiliary energy fraction for the entire period is 6%. In 2009, the range of auxiliary energy is 1%-10%.

Fig. 48 shows the single seasonal performance factor for each system in the year 2009. The seasonal performance factors range between 2.3 and 3.5 in 2009 at average inlet space heating temperatures of 36.9 °C and DHW inlet temperatures of 50.5 °C. Space heating inlet temperatures are rather high and range between 29.9 °C and 45.3 °C. Minimum DHW shares are with 4% low and reach maximum values of 16%.

Fig. 49 depicts the correlation of the evolution of monthly performance factors to the source temperature of the outside air.

As expected, the performance factors are correlated well to the outdoor air temperature in the winter period and in the summer period, the higher DHW operation get more important.

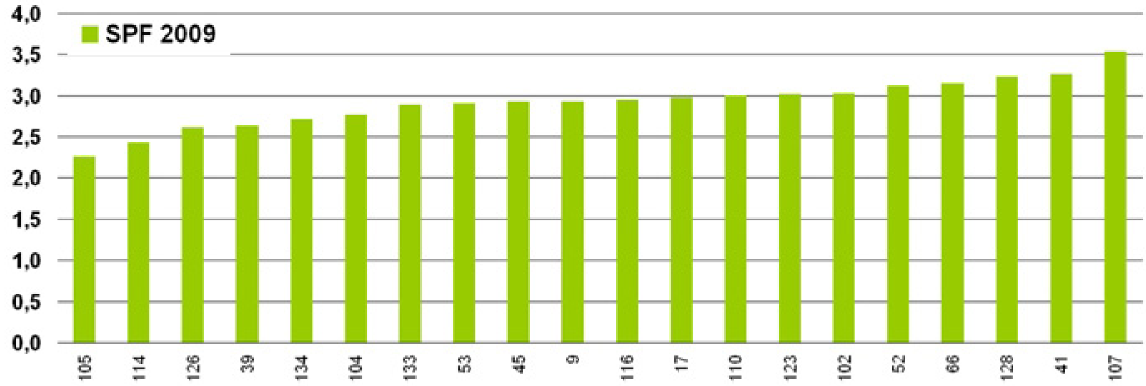


Fig. 48: Seasonal performance factor for all air-to-water heat pumps in the year 2009 monitored in the HP Efficiency project (Günther and Miara, 2010)

Thus, as mentioned before, the maximum performance factors occur in the transitional period.

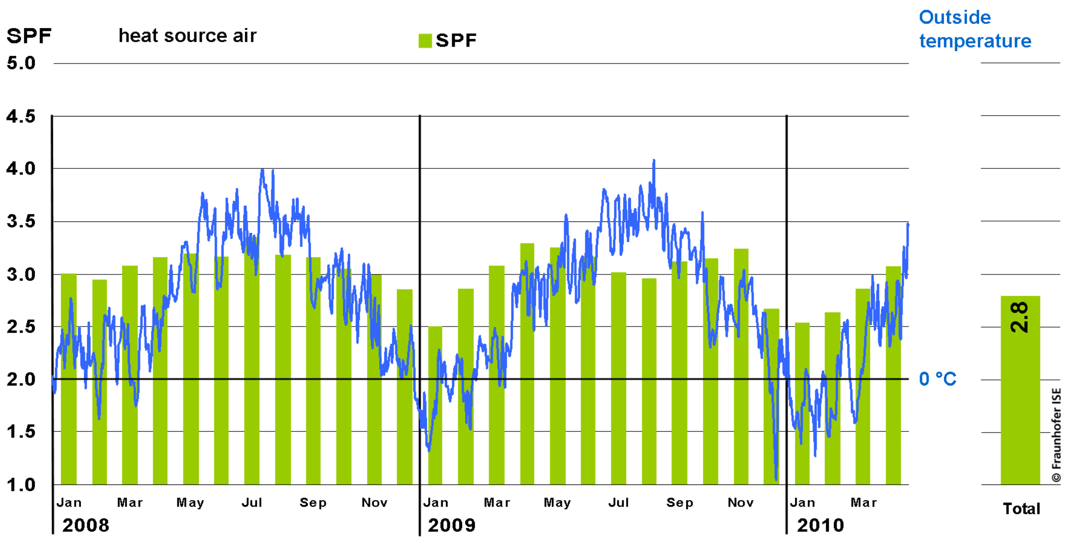


Fig. 49: Evolution of the average seasonal performance factor compared to the outdoor air temperature for air-to-water heat pumps (Günther and Miara, 2010)

4.3.1.1 Design of the heat pump and use of the separate electrical back-up heating

For air-source heat pumps it is currently more common to make a monoenergetic design, where an electrical back-up heater delivers the additionally needed capacity at low outdoor air temperatures.

The analysis of the SPF based on the location and back-up operation is shown in Fig. 50 left.

It indicates that systems perform slightly better when placed inside the building. Due to the small difference, however, it cannot be told, which location of the heat pumps is preferable, since results are influenced by heat losses or gains, respectively, pressure drop of the respective systems and sample of investigated systems, which is with 14 outdoor and 9 indoors systems still low.

The figure also illustrates the SPFs with and without taking into account the electrical back-up heater operation. Systems which are placed outdoor reach a total SPF of 2.8 without the electric heater and 2.75 with the electric heater. Indoor systems reach 0.1 higher SPFs than outdoor setups. Thus, for the considered designs, the back-up heater has a negligible impact on the overall performance.

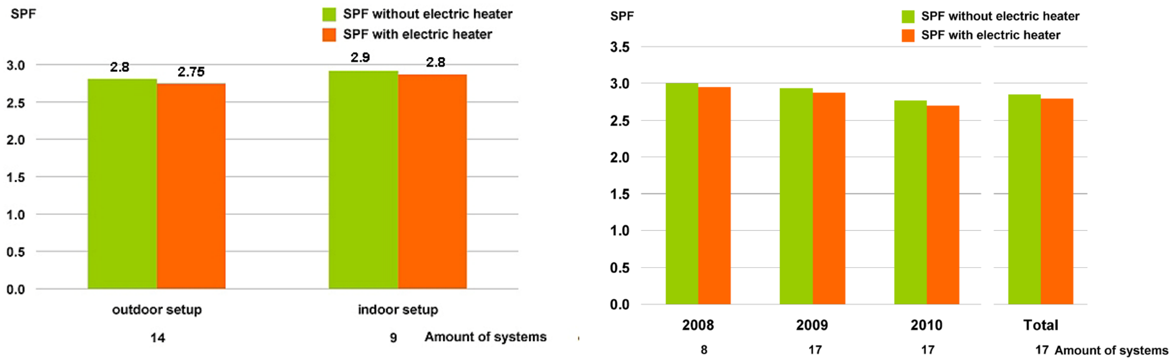


Fig. 50: SPF with and without separate electric heater divided dependent on the setup (Günther and Miara, 2010)

Six systems work without any electrical back-up heating. Fig. 50 right gives the finally installed 17 systems with electrical back-up heating. Since many systems where installed during the construction phase of the building, space heating needs are affected by the drying-out of the building. However, a significantly higher back-up use in the first year could not be observed, maybe due to a milder winter temperatures than in 2010. Back-up heater use is relatively stable and low over the three years, thus it can be concluded, that the heat pumps are more or less operating monovalently, either due to a lower heat load of the building than calculated or due to a respective design of the heat pumps, for instance based on the capacity steps of heat pumps offered on the market.

4.3.2 Results of air-to-water heat pumps in the Austrian field test

In the Austrian field test three air-to-water heat pumps have been monitored. All the systems are used for space heating and DHW production.

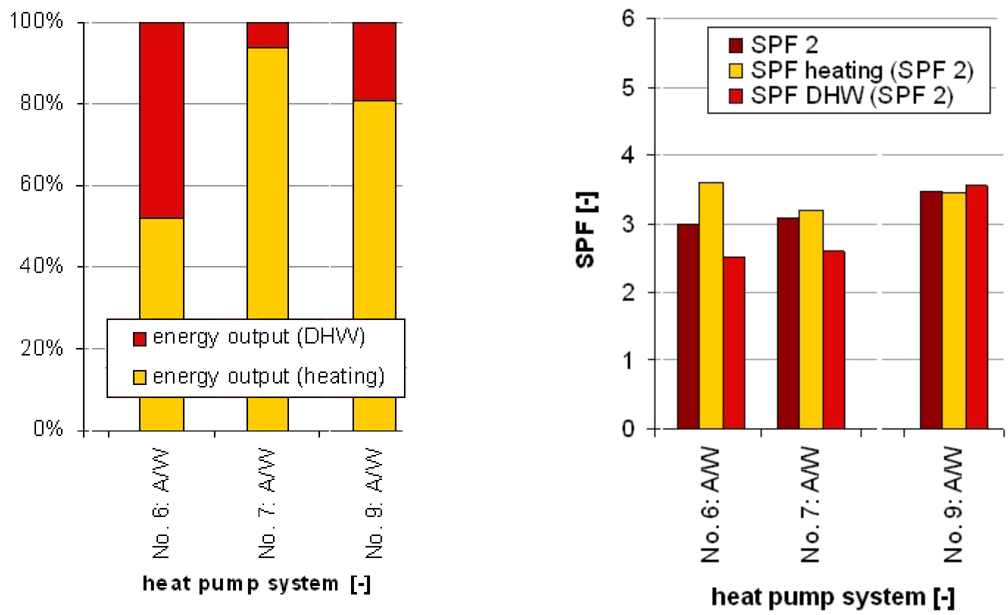


Fig. 51: SPF of the generator system in the Austrian field monitoring project (based on Zottl et al., 2010)

Fig. 51 left presents the DHW fraction of the total heat energy needs, which is in the range of 5%-20% except for the extraordinary fraction of 50% of System No. 6, which is due to a particular user behaviour. System No. 7 is installed in a retrofitted building.
System No. 9 is a split system layout, which has an outdoor evaporator. All considered air-

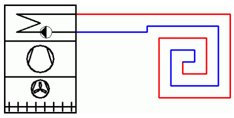
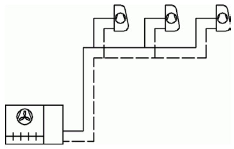
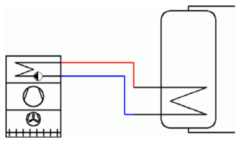
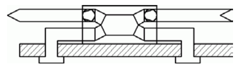
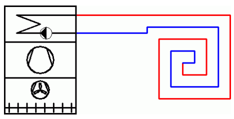
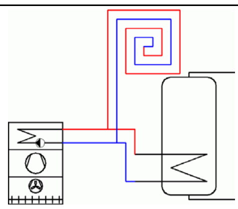
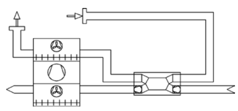
to-water heat pumps are equipped with buffer storage. The explanation of the designer for installing a buffer storage is the defrost operation.
Fig. 51 right shows the seasonal performance factor of the generator. The overall seasonal performance factors are in the range of 3.0 to 3.4. The SPFs in space heating mode range between 3.1 and 3.5, while the DHW performance is between 2.5 and 3.6.
System No. 6 reaches a high overall seasonal performance factor despite the large DHW fraction due to a high space heating SPF. System No. 9 has a very high DHW SPF of 3.6, which is even higher than the space heating SPF, which is very special. The reason is a low DHW temperature of 45°C, which is produced by a fresh water system, so the average supply temperature of the heat pump for DHW was measured with 42°C, so this very low temperature is the explanation for the extraordinary DHW SPF.
Thus, the three Austrian field tests are in the upper SPF range of the German HP-Efficiency field test results.
System No.7 and system No. 9 have been documented in more detail in two Best Practice Sheets.

5 Cooling with Heat pumps in Residential buildings

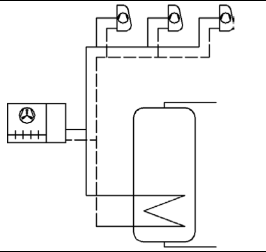
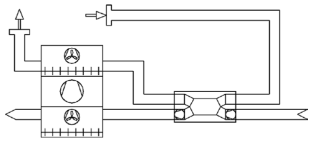
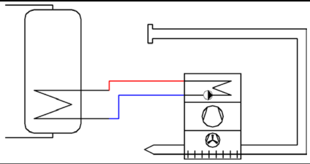
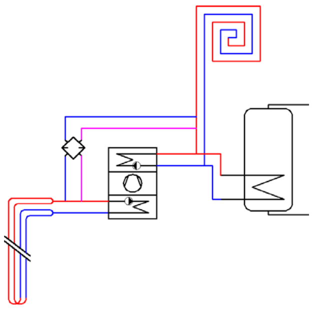
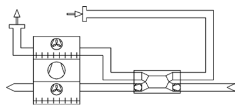
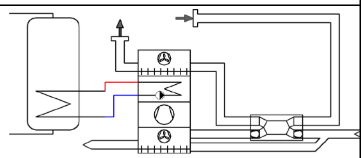
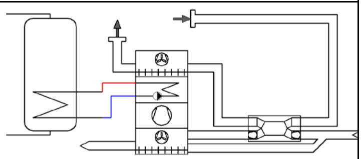
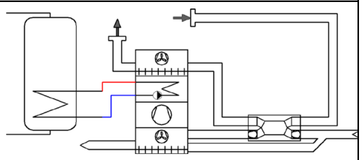
5.1 System variants

Space heating and domestic hot water preparation are common applications for heat pumps in residential buildings and frequently used especially in new buildings. Furthermore, heat pumps running in reverse mode are a common application as air conditioner for space cooling prevalently in non-residential buildings with high internal heat loads. Heat pumps could supply very energy and cost efficient space cooling for residential buildings if the heat source of the heat pump could be used for passive cooling. Tab. 3 shows a compilation of possible functionality combinations, which heat pumps could cover as well as corresponding system examples.

Tab. 3: System configurations on the market

Function	Description	Market Penetration	System Example
H	heat pump for space heating	standard product	
C	air conditioner (split / multi-split)	standard product	
W	heat pump water heater	standard product	
V	ventilation unit	standard product	
HC	reversible heat pump	standard product	
HW	space & hot water heat pump	standard product	
HV	ventilation unit with heat pump	standard product for Minergie-P / Passivhouses	

Tab. 3: System configurations on the market (continued)

Function	Description	Market Penetration	System Example
CW	heat pump water heater with space cooling function or air conditioner with hot water heating function	niche product	
CV	ventilation unit with cooling function	niche product for Minergie-P / Passivhouse	
WV	exhaust air heat pump water heater	standard product	
HCW	reversible HW-HP reversible air conditioner with water heater function HW-HP with pass. cooling	market growth	
HCV	ventilation unit with HP	niche product for residential application	
HWV	HP-Compact unit	standard product for LEH	
CWV	air conditioner compact unit	niche product	
HCWV	multi-function compact unit	market introduction	

H – space heating, C – space cooling, W – domestic hot water, V - ventilation

5.2 System comparison

For the application in residential buildings, four most important system configurations could be defined as shown in Tab. 4. These heat pump systems are always based on heat pump heating systems since this is the more important function for middle Europe. These systems are then adapted so that they could also be used for space cooling. The considered and shown functions are space heating, domestic hot water preparation, ventilation and space cooling.

Tab. 4: Selected system configurations for the system comparison

identifier	function	brief description	chapter
A/A-HP	**HC, W, V**	air/air-heat pump	5.2.1
A/W-HP	**HCW, V**	air/water-heat pump	5.2.2
B/W-HP	**HCW, V**	brine/water-heat pump	5.2.3
HP-CU	**HCWV**	Minergie-P / Passivhouse compact unit	5.2.4

5.2.1 Air/Air-heat pump

Prevalent conventional comfort cooling systems are single or multi split-units using indoor and outdoor convectors and heat rejection to the outside air. Recently these systems are brought to market also for heating and cooling. They are therefore defined as standard system A/A-HP. The first system A/A-HP operates with a heat pump air conditioner for active heating and space cooling. In addition, a central ventilation heat recovery unit supplies fresh air. An extra domestic hot water heat pump generates hot water.

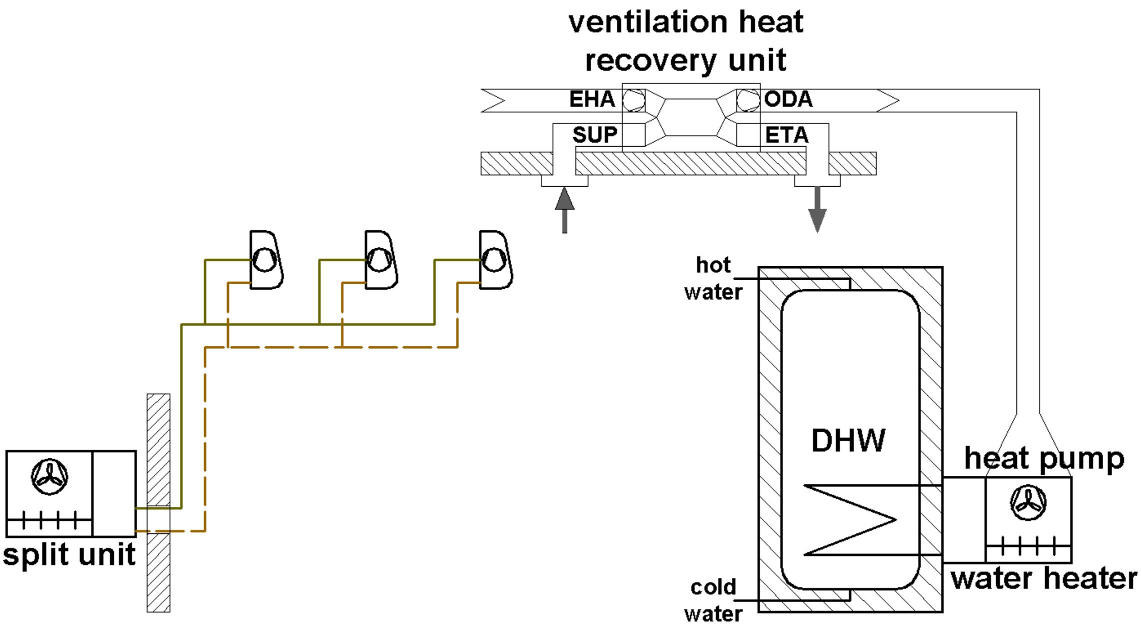


Fig. 52: A/A-HP for space heating with reversed operation mode for space cooling, extra heat pump water heater

5.2.2 Air/Water-heat pump

The A/W-HP system uses an air-to-water heat pump connected to a hydraulic system for floor heating and domestic hot water generation in alternative operation. For space cooling, the heat pump operates in reverse mode. A ventilation heat recovery unit supplies fresh air separately. The A/W-HP system uses the same heat emission system (floor) for space heating and cooling. A higher efficiency could be provided by simultaneous cooling and domestic hot water operation. A/W-HP systems are available with single speed or continuous capacity controlled compressors. Single speed units (c.f. Fig. 53) require a bigger thermal capacity in the heat emission system, if they are operated without technical heat storage.

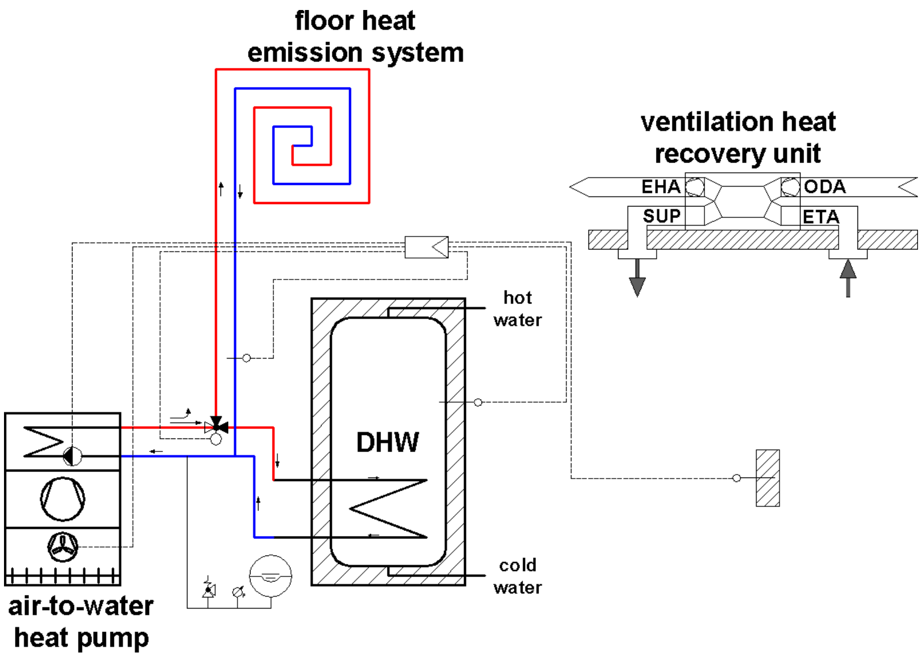


Fig. 53: A/W-HP, single speed compressor with reversed operation mode and common heat emission system of high thermal capacity

Units with continuously capacity controlled compressor could be applied to separate heat emission systems for space heating and cooling like cooled ceilings or convectors.

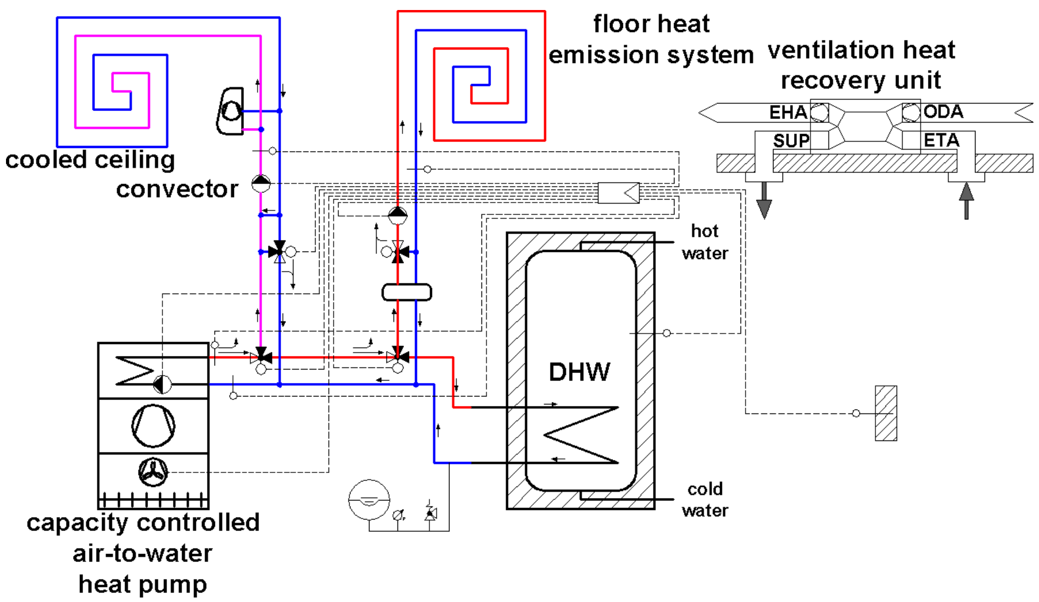


Fig. 54: A/W-HP, continuously capacity controlled compressor with reversed operation mode and separate heat emission systems for space heating and cooling

5.2.3 Brine/Water-heat pump

The B/W-HP system shows the highest potential for an energy efficient heating and cooling and is actually the most widespread system in Switzerland. The heat pump uses a vertical borehole heat exchanger as heat source. Heat is exchanged within the room by a low temperature floor heating system. With this system, the borehole heat exchanger could be coupled directly to the floor heating system with a simple heat exchanger. Additional system as well as operational expenditure for the passive cooling option is comparatively low.

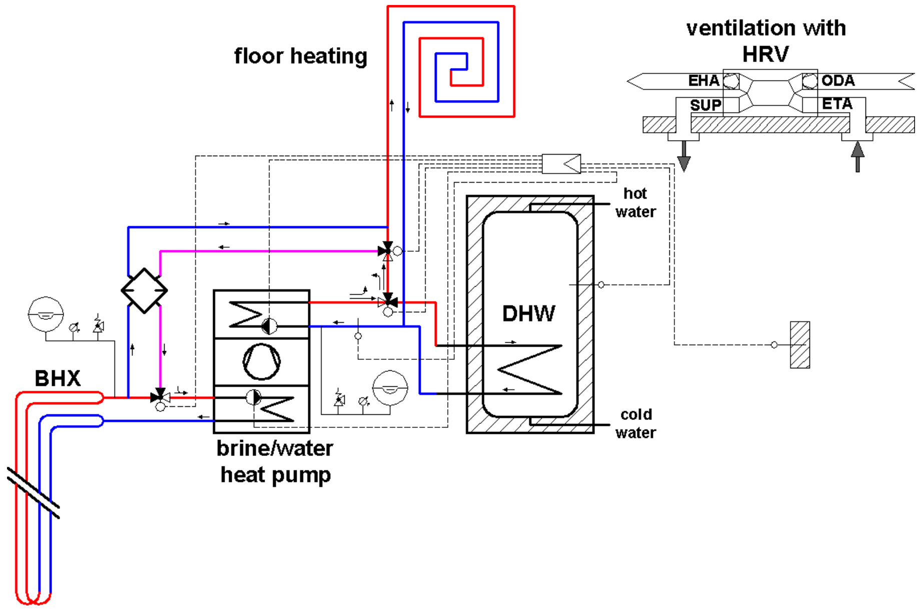


Fig. 55: Standard system B/W-HP, ground coupled heat pump with passive cooling

5.2.4 Minergie-P / Passivhouse heat pump compact unit

The here defined HP-CU system makes only sense applied to ultra low energy houses like Minergie-P® or passive houses, since only very good thermal insulation and a corresponding low heat demand allows to supply the space heat demand with the required ventilation fresh air over the ventilation system. Specially adapted heat pump units have been developed covering the functions space heating, domestic hot water and ventilation. The heat pump uses ventilation exhaust air sometimes mixed with outside air as heat source. Again, the heat pump can operate in reverse mode for cooling, supplying the cooling via ventilation air. An advantage of the system is the ultra-low heating and cooling need during optimised operation of the building, so that even with an active cooling a comparatively low extra electricity demand could be achieved. The HP-CU system uses air as heat source as well as heat sink. Therefore, in the system comparison it will not be displayed separately but assumed to have efficiency between the A/A-HP and A/W-HP.

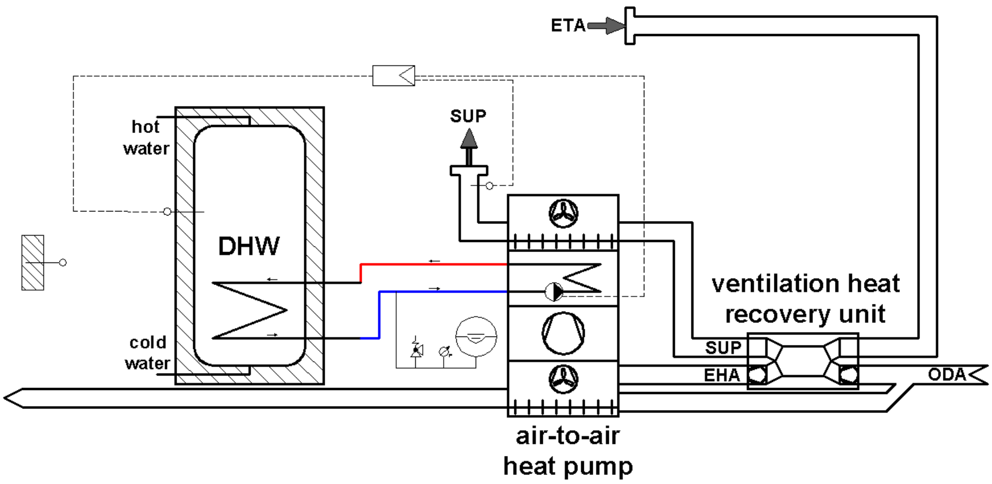


Fig. 56: HP-CU, ventilation heat pump compact unit with reversed operation mode

Tab. 5: Overview of simulations results on different system solutions

System	**A/A-HP**	**A/W-HP**					**B/W-HP**					
characterisation	ME shd025	ME shd075	ME shd075 C sim. W	ME shd025	ME shd025 C sim. W	MEP shd025	ME shd075 without C	ME shd075	ME shd050	ME shd025	MEP shd025 without C	MEP shd025
chapter	5.2.5	5.2.9	5.2.9	5.2.5 5.2.9 5.2.10	5.2.9	5.2.10	5.2.11 5.2.12	5.2.11 5.2.12	5.2.12	5.2.5 5.2.10 5.2.12	5.2.11	5.2.10 5.2.11 5.2.12
temporal share of thermal comfort during heating period in cat. II	100%	93%	93%	92%	92%	96%	95%	96%	96%	96%	96%	97%
temporal share of thermal comfort during cooling period in cat.III	97%	90%	95%	93%	84%	96%	95%	88%	90%	92%	82%	94%
SPF_C	4.1	2.1	2.4	2.3	2.5	2.3	-/-	8.5	10.5	12.9	-/-	11.8
SPF_{HWC}	3.3	3.1	3.3	3.1	3.3	3.0	4.0	4.4	4.5	4.7	4.0	4.7
E_{HWC}	4'510 kWh	4'955 kWh	4'080 kWh	5'325 kWh	4'042 kWh	4'562 kWh	3'324 kWh	3'531 kWh	3'516 kWh	3'490 kWh	2'726 kWh	2'917 kWh
Investment for space cooling function	low	low	low	low	low	low	no	low	low	low	no	low
overall operation expense	moderate	high	moderate	high	moderate	high	low	low	low	low	very low	very low
requirements for the realisation in residential buildings	high	moderate	moderate	moderate	moderate	moderate	low	low	low	low	low	low

Legend for characterisation: ME – Minergie / MEP – Minergie-P / shd025 – shading of all windows up to 25% / shd050 – shading of all windows up to 50% / shd075 – shading of all windows up to 75% / C sim. W – space cooling only simultaneously with DHW operation / without C – no space cooling

5.2.5 Comparison of the systems A/A-HP, A/W-HP, B/W-HP

The comparison of three heat pump systems (with heat sources air and ground, resp.) presented in this section illustrates the behaviour of the systems for the coverage of space heating, domestic hot water and cooling functions.

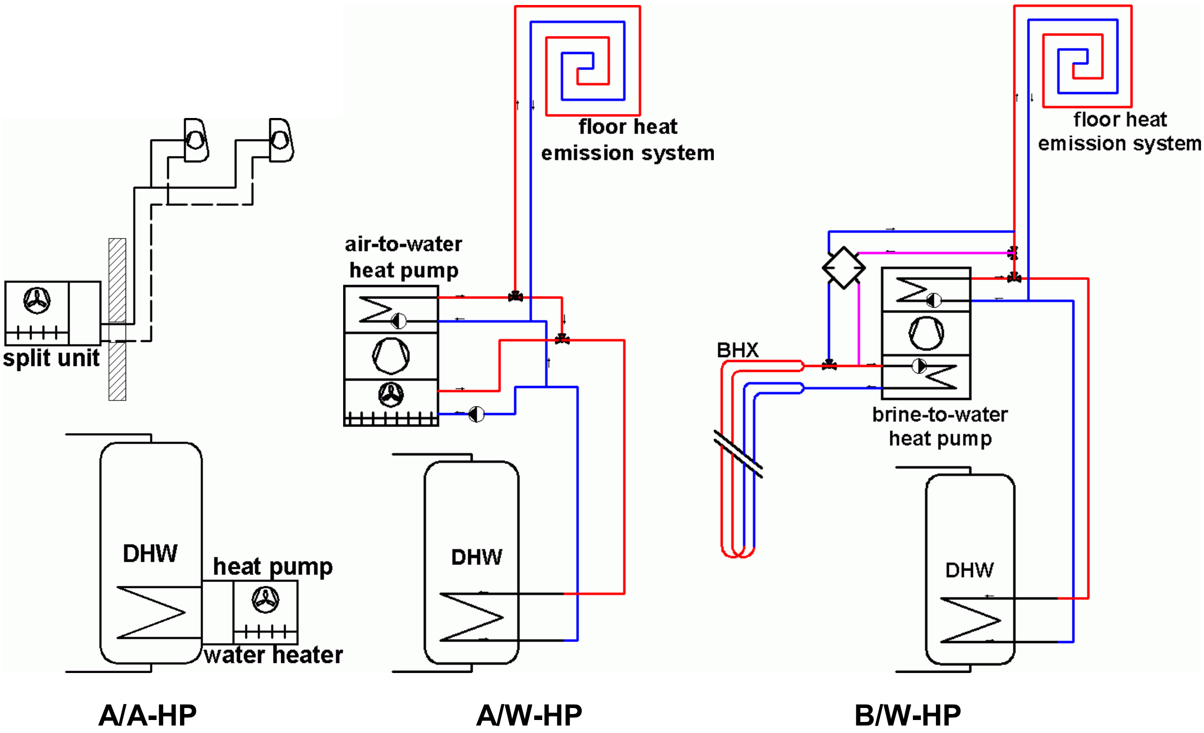


Fig. 57: Schematic of the standard circuit design for the three basic versions A/A-HP, A/W-HP, B/W-HP

For all versions the windows are shaded to a maximal degree of 20%. Besides the building, the heat pump including the controller is represented in the simulation. The systems A/A-HP, A/W-HP and B/W-HP are schematically depicted in Fig. 57. System A/A-HP is based on a multi-split air-to-air conditioner, which is also used for space heating in reverse mode. Domestic hot water preparation is carried out separately by an exhaust air heat pump water heater disposed downstream to the ventilation system. In the system A/W-HP an air-to-water heat pump covers all three functions in active operation. Space heating and cooling is supplied by a low temperature floor heating system. The system B/W-HP uses a borehole heat exchanger as heat source for the space heating and domestic hot water preparation as well as heat sink for passive cooling.

5.2.6 What degree of thermal comfort can be achieved?

Fig. 58 shows the classification of the thermal room climate for the entire year. Altogether a good thermal room climate, also for rather low shading, appears for all versions A/A-HP to B/W-HP with, in each case, only a small fraction of less than 3% in category IV and the highest fraction in category I.

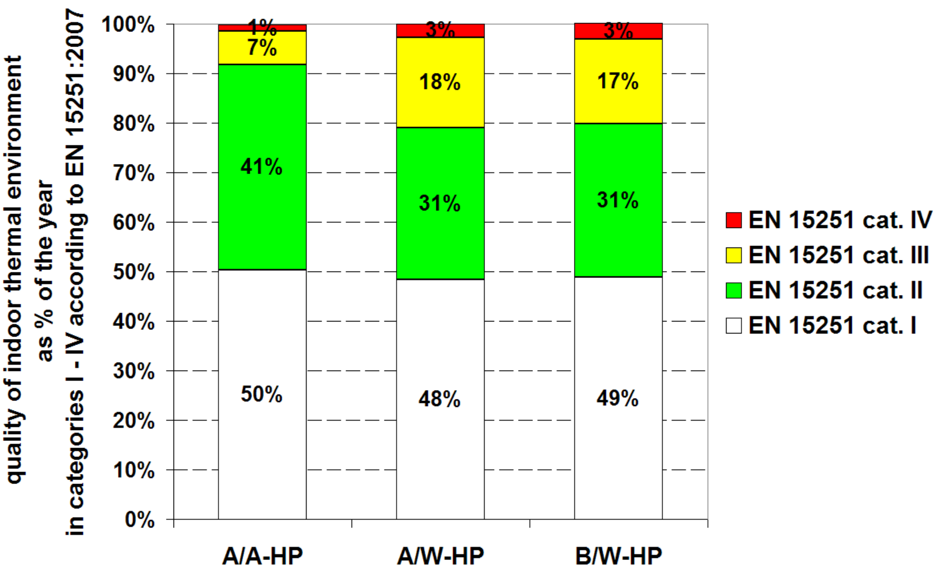


Fig. 58: Classification of the thermal room climate for the entire year

For the winter season Fig. 59 (on the left) shows that in all three versions the thermal comfort lies inside category II for more than 92% of the time. For the summer season (Fig. 59 on the right) higher deviations between the system versions can be seen. Although all systems comply with the upper temperature boundary of category I; all systems show a deviation in downward temperatures. However, this is only partially due to cooling, as the A/A-HP system shows. Besides, the temperatures are also caused by considering the transitional seasons, where neither the heating nor the cooling systems are active.

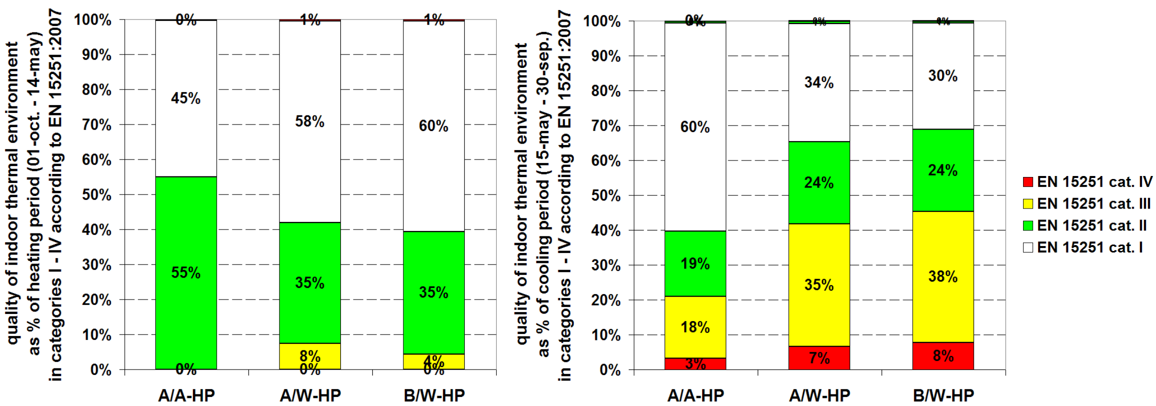


Fig. 59: Classification of the thermal room climate for winter season (Oct. 1st – May 14th) and summer season (May 15th – Sep. 30th, on the right)

The systems A/W-HP and B/W-HP with floor cooling show a somewhat higher tendency towards cooler temperatures. The cause for these different characteristics is indicated in the illustrations of the operative room temperature of the three systems in Fig. 60 to Fig. 62.
For the system A/A-HP, the multi split system, a clear boundary with a constant operative room temperature can be observed. With the supply of the cooling and heating power directly into the room, a simpler adjustment and achievement of the desired room temperature is possible. This is manifested for both winter and summer season. For the other two systems a broader scattering of the room temperature at constant ambient temperature can be observed. This is mostly due to a temporal buffer effect from the construction of the floor and thereby a delayed supply of the cooling and heating power into the room. This high spread of the room temperature at constant ambient temperature demands in reverse a more accurate dimensioning, that the desired temperature range can be maintained.

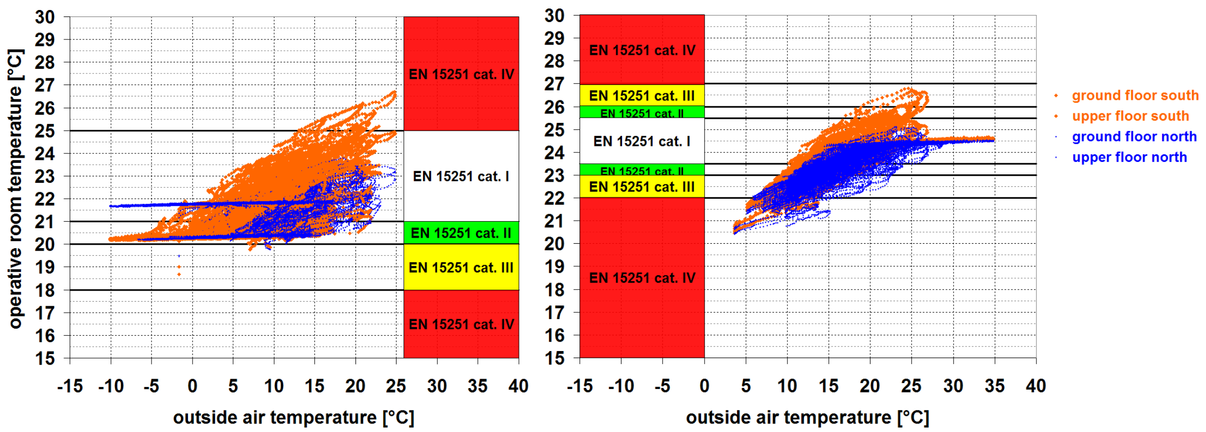


Fig. 60: Distribution of the operative room temperature with respect to ambient temperature for system A/A-HP for winter season on the left (Oct. 1st – May 14th) and summer season on the right (May 15th – Sep. 30th, on the right)

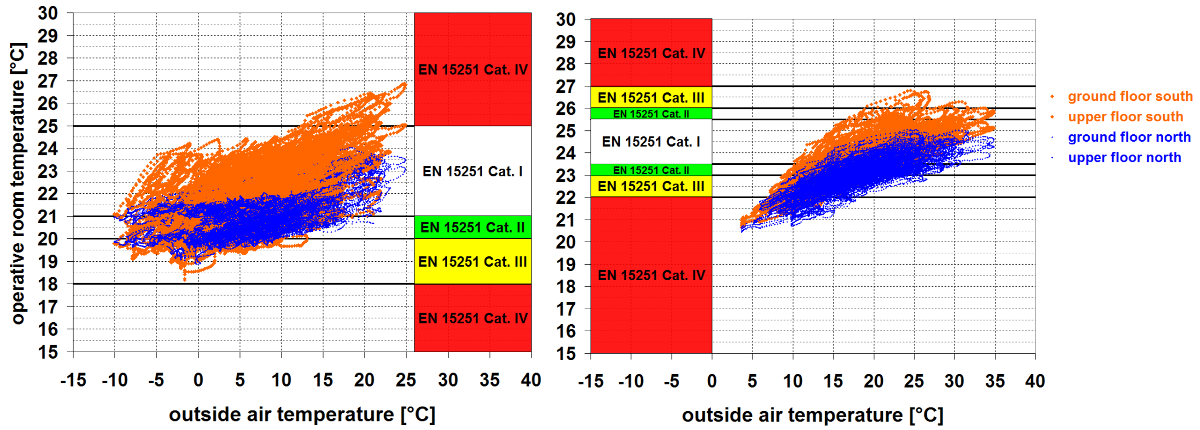


Fig. 61: Distribution of the operative room temperature with respect to ambient temperature for system A/W-HP for winter season on the left (Oct. 1st – May 14th) and summer season on the right (May 15th – Sep. 30th)

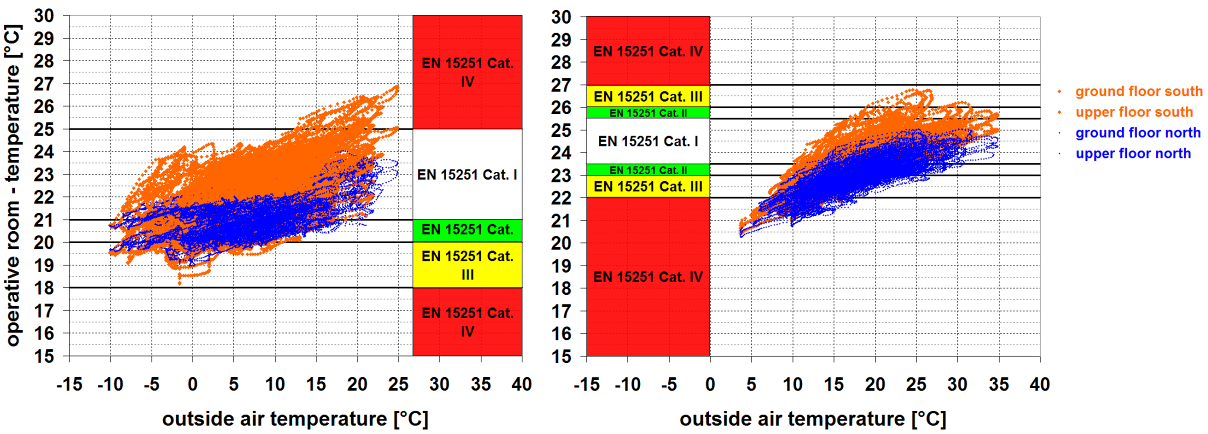


Fig. 62: Distribution of the operative room temperature with respect to ambient temperature for system B/W-HP for winter season on the left (Oct. 1st – May 14th) and summer season on the right (May 15th – Sep. 30th)

Comfort illustration with active and inactive operation

The classification of the thermal room climate according to Fig. 59 shows a part of the operative room temperatures, which lie during summer season below category III after EN 15251 in comfort rating. Fig. *63* shows the cause for this rating therein, that both the active cooling mode and terms of inactive heat pump system, during which the room temperature is reached within free oscillations, are rated in combination. The left side of Fig. *63* shows only the periods with active heat pump operation for the system A/W-HP during the summer season. Here, the operative room temperatures, up to some 15 min values, are within category III after EN 15251. During the periods without active HP operation, c.f. right side of Fig. *63*, downwards deviation of the operative room temperature emerge, caused by colder outside temperature situations during summer season.
Still a coherent illustration and temporally continuous rating is considered as meaningful, because the occupation of a residential building by variable manners of living cannot be foreseen and in addition is not limited.

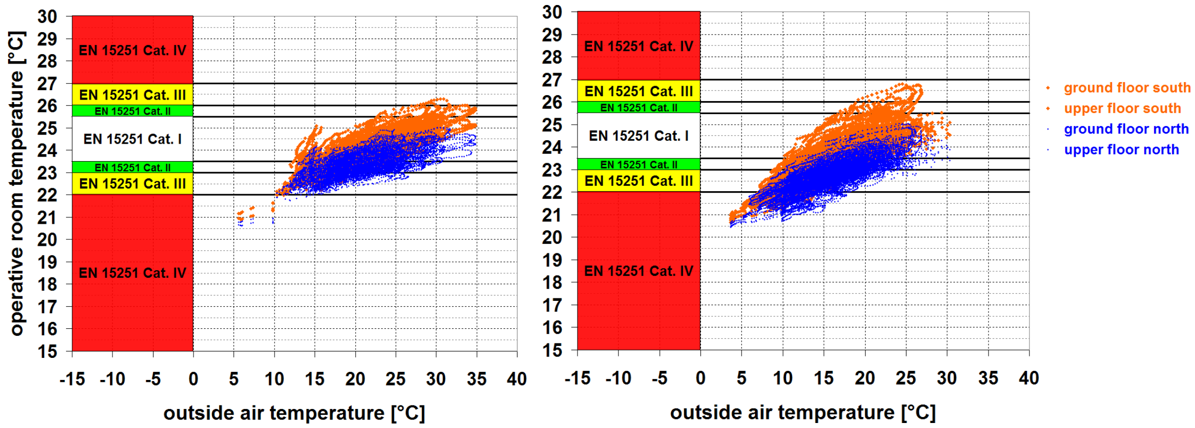


Fig. 63: Juxtaposition of the comfort characteristics for active cooling operation (on the left) and inactive HP-system (on the right) for the system A/W-HP during summer period (May 15th – Sep. 30th)

5.2.7 What is the extent of effort and what efficiency can be achieved?

In Fig. *64* the generated heat and in Fig. 65 the spent electrical energy for the three systems are compared.
The hot water heat demands are the same for all systems and consist of 13.9 kWh/m^2/a useful heat demand plus 2.8 kWh/d of heat losses. The thermal heat demand differs between 8711 kWh/a for the system A/A-HP and 9372 kWh/a for the system B/W-HP as a result of the different room temperatures caused by the control of the systems.

Because of the differences in the distributions of room temperatures, in the cooling energy consumption higher relative differences emerge, where system A/A-HP with 2258 kWh/a has the lowest cooling energy consumption and A/W-HP with 3133 kWh/a the highest.

System A/A-HP has overall the lowest generated useful heat. In aspects of electrical input a notably different picture appears. Here system B/W-HP has the lowest electrical costs both for the individual functions and overall. The system A/A-HP stands out for having the highest electrical energy consumption for hot water preparation and the system A/W-HP for having the highest electrical energy consumption for cooling.

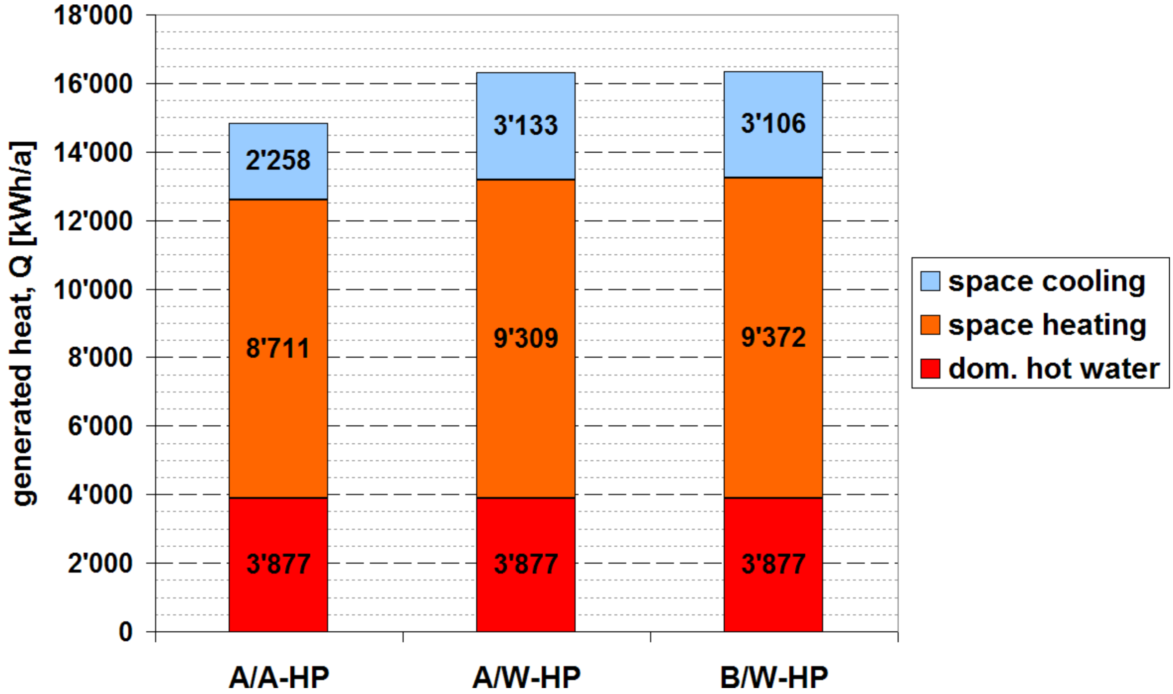


Fig. 64: Comparison of the generated energy for the systems A/A-, A/W- & B/W-HP

The reason for this low electrical energy consumption of the B/W-HP can be seen in the comparison of the heat generator system performance in *Fig. 66* with a high performance of 12.9 for passive cooling.

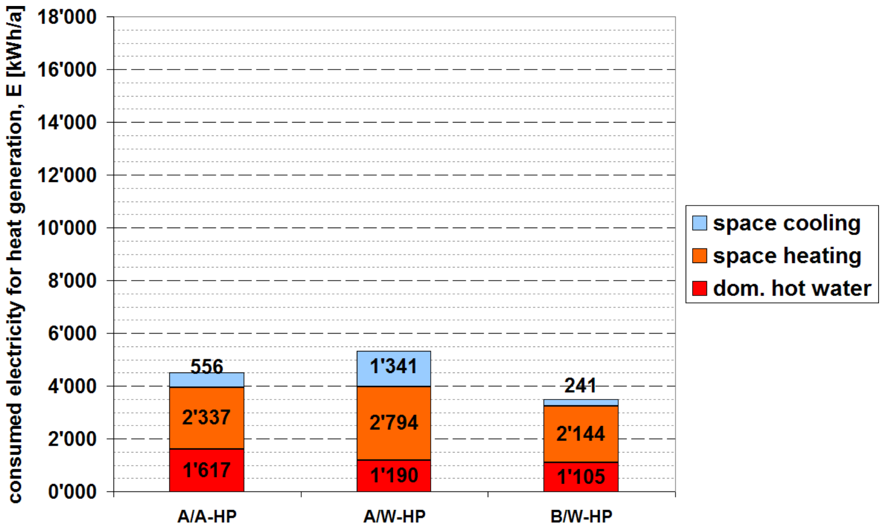


Fig. 65: Comparison of the consumed electrical energy for the systems A/A-, A/W- & B/W-HP

The reason for this low electrical energy consumption of the B/W-HP can be seen in the comparison of the heat generator system performance in *Fig. 66* with a high performance of 12.9 for passive cooling. Also for space heating and DHW the B/W-HP system has the highest system performances, i.e. 3.5 for DHW, 4.4 for space heating and 4.7 overall. For both the total electrical input and the system performance for cooling as well as for all operation modes the system A/W-HP performs worst.

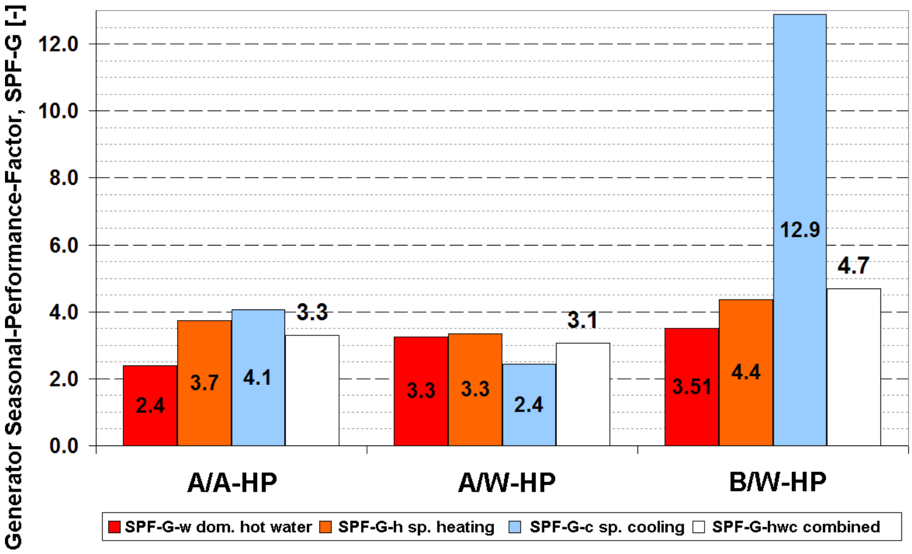


Fig. 66: Comparison of the heat generator system performance for the functions heating, DHW, cooling and their combination for the systems A/A-HP, A/W-HP and B/W-HP

System A/A-HP reaches a high system performance of 4.1 in active cooling mode due to a relatively lower temperature lift, and thus has moderate electrical costs. However, by the low system performance in DHW operation of 2.4 of the exhaust air heat pump water heater behind the ventilation heat recovery and active heat pump in all operation modes, the system performance and the electrical costs of system B/W-HP cannot be reached. The borehole heat exchanger represents a beneficial heat source for all operation modes.

5.2.8 Conclusion

The by far highest efficiency is provided by a passive cooling with a borehole heat exchanger having a heat generator system performance over all modes of operation of 4.7. For normal room usage and low temperature design of the heat supply system, a very high thermal comfort is reached. Additionally, a beneficial effect on the efficiency of estival hot water preparation is achieved. The additional costs for a passive cooling are small.

The overall worst efficiency over all operation modes, i.e. 3.1, is obtained from an air/water heat pump with active cooling in reversed mode. On the one hand the precise achievement of comfortable room temperatures in summer cooling mode with an inertial heat supply system is more difficult, thus yielding higher cooling energy consumption. On the other hand the system indeed has an acceptable DHW system performance of 3.3, though it has by comparison the worst efficiencies in space heating and cooling mode, i.e. 3.3 and 2.3, respectively. An advantage is the lowest infrastructural cost, since all three operation modes are covered by one single system.

A high efficiency in active heat pump operation of 3.7 for heating and 4.1 for cooling is reached by a good multi-split-VRF-air/air-heat pump. With this system the best thermal comfort with respect to the operative room temperature can be reached. The adaptation to changing load conditions concerning temperature and humidity, caused by changing shading or occupation, is carried out easiest. Hence, the lowest cooling energy consumption can be achieved. A disadvantage is the, over the entire building distributed, high amount of refrigerant. Furthermore, the heat delivery to the room by a recirculating air system demands special attention during planning to potential noise emissions and draft occurrences. The DHW preparation with an exhaust air heat pump water heater behind the, for all systems equivalent, mechanical ventilation system with heat recovery, is indeed a low cost device, but has only a low heat generator system performance of 2.4.

5.2.9 What effect has the simultaneous hot water and cooling operation?

A restriction of cooling operation to periods, during which the waste heat can be used for DHW preparation, would only yield a very low cooling effect. The generator system performance during the simultaneous hot water and cooling operation is, for an A/W-HP with 4.2 up to 5.7 for 15 minute values, in fact considerably higher than the else wise reached system performance in the range of 2.5, but still considerably lower than the passive cooling operation and yet in a range, that can be achieved by active cooling with good cooling devices, that do not use waste heat. The advantage in the yearly generator system performance is 0.1 performance points for DHW operation, for cooling operation and also over all operation modes.

5.2.10 What impact has Minergie-P compared to Minergie?

In both building types according to Minergie and Minergie-P standard an almost identical thermal comfort is reached. The better insulation of the building envelope leads to a reduction of space heat demand of about 25%. The space cooling demand reduces by approximately 9% somewhat less. The reduction of the space cooling demand by an improved building envelope has a higher impact for active cooling with an air-to-water heat pump. However, the total electrical energy consumption of a ground-coupled heat pump system is about 35% lower.

5.2.11 Effect of the passive cooling

The passive cooling with borehole heat exchanger and floor heat emission system increases the thermal comfort during cooling season both for high shading and particularly for non consequent shading. The additional electrical cost for the comfort augmentation during cooling season is 6-7% for an average efficiency in passive cooling mode. For a well adjusted system the heating energy consumption increases by 1% at least, whereas the heating energy consumption for DHW decreases by about 5%.

5.2.12 Effect of passive cooling on the borehole heat exchanger as heat source

The usage of the passive cooling with borehole heat exchanger increases the heat generator system performance for DHW operation from 3.3 to 3.6 for else wise identical conditions. Thereby the cold/heat ratio increases from 0% without cooling operation to 26% at intensive cooling usage in the Minergie-P building. The heat generator system performance during heating operation is only marginally affected by the passive cooling operation by 0.04 performance factor points at the most.

5.3 Effect of passive cooling on heating and DHW operation

The effect of the passive cooling on space heating and hot water operation has been investigated in detail on the basis of the comparison of different intensive cooling with borehole heat exchangers. The most important impact on system efficiency is the heat rejection into the borehole heat exchanger in space cooling operation and the heat storage in the ground. The stored heat can then lead to higher outlet temperatures of the borehole heat exchanger during heating and DHW operation with heat usage from the borehole heat exchanger. This effect is quantified by a comparison to simulation results.

5.3.1 Analysis of the effects

Tab. 6 and Tab. 7 show for all three functions (space heating, DHW & space cooling) the correlations of generated and rejected heat, resp., with the heat generator seasonal performances and the average outlet temperatures of the borehole heat exchanger over the whole year. As reference value the cold/heat ratio is listed, which is calculated as the ratio of heat rejected from the room to the borehole heat exchanger to the generated heat with heat extracted from the borehole heat exchanger during heating and DHW operation. The stated temperatures are provided as average of the recorded 15 minute values over the entire time period in active operation for the respective operation mode.
With the increase in cold/heat ratio from 0% without space cooling up to 23% for intensive cooling usage in the Minergie building, the effect on the space heating operation emerges only marginally by an increase of the average outlet temperature of 0.4 K from 3.3 °C to 3.7 °C and a thereby associated increase of the heat generator system performance of less than 1% from 4.36 to 4.37. For the calculation of the heat generator system performance this influence can be neglected for the design. A stronger influence of the heat rejection into the borehole heat exchanger during passive cooling operation is observed in DHW operation.

Tab. 6: Variation of generated heat und resulting heat generator system performance for passive cooling

	B/W-HP 75% ME no C	**B/W-HP 75% ME**	**B/W-HP 50% ME**	**B/W-HP 25% ME**	**B/W-HP 25% MEP**
generated heat					
space cooling	0 kWh/a	2'049 kWh/a	2'532 kWh/a	3'106 kWh/a	2'839 kWh/a
space heating	9'417 kWh/a	9'494 kWh/a	9'454 kWh/a	9'372 kWh/a	6'985 kWh/a
domestic hot water	3'877 kWh/a	3'877 kWh/a	3'877 kWh/a	3'877 kWh/a	3'877 kWh/a
total	13'294 kWh/a	15'420 kWh/a	15'863 kWh/a	16'355 kWh/a	13'701 kWh/a
cold/heat-ratio	0%	15%	19%	23%	26%
annual generator seasonal performance factor					
space cooling	-/-	8.5	10.5	12.9	11.8
space heating	4.36	4.36	4.37	4.37	4.40
domestic hot water	3.33	3.48	3.49	3.51	3.56
total	4.0	4.4	4.5	4.7	4.7

The heat generator system performance augments already for the smallest considered cold/heat ratio of 15% by 0.15 performance factor points from 3.33 to 3.48. With a further increasing cold/heat ratio a further enhancement of the heat generator system performance up to 3.51 emerges. The influence of the borehole heat exchanger on the outlet temperature during DHW operation is particularly notable for days when also cooling is operating. Over the whole year the average outlet temperature increases from 4.5 °C to 5.5 °C rather little. Considerably stronger is the influence on the outlet temperature during cooling days, i.e. days on which the space cooling in the Minergie version with intensive cooling (25% shading) is active. The mean value of the outlet temperature on these cooling days rises from 6.1 °C without cooling up to 10.0 °C at a cold/heat ratio of 23%.

Tab. 7: Influence of passive cooling on the borehole heat exchanger outlet temperature

	B/W-HP 75% ME no C	B/W-HP 75% ME	B/W-HP 50% ME	B/W-HP 25% ME	B/W-HP 25% MEP
cold/heat-ratio	0%	15%	19%	23%	26%
annual average borehole heat exchanger outlet temperatures					
space cooling	-/-	13.0 °C	13.7 °C	14.4 °C	14.4 °C
space heating	3.3 °C	3.5 °C	3.5 °C	3.7 °C	4.4 °C
domestic hot water	4.5 °C	5.0 °C	5.2 °C	5.5 °C	6.1 °C
DHW at space cooling days	6.1 °C	8.7 °C	9.4 °C	10.0 °C	10.0 °C

This substantiates the fact, that a short term usage of the heat which is imported into the borehole heat exchanger can be accomplished; however, the long term storage effect is low. An over the year point of view shows, that the heat import into the borehole heat exchanger from passive cooling has the highest effect on the source temperature of the heat pump for DHW operation during cooling season. In the monthly evolution of the generated heat for space cooling and the increase in the outlet temperature of the borehole heat exchanger for the case of the Minergie building with little shading (25%) in comparison to the not cooled building, shown in Fig. 67*,* the extensive proportionality of the two quantities can be seen.

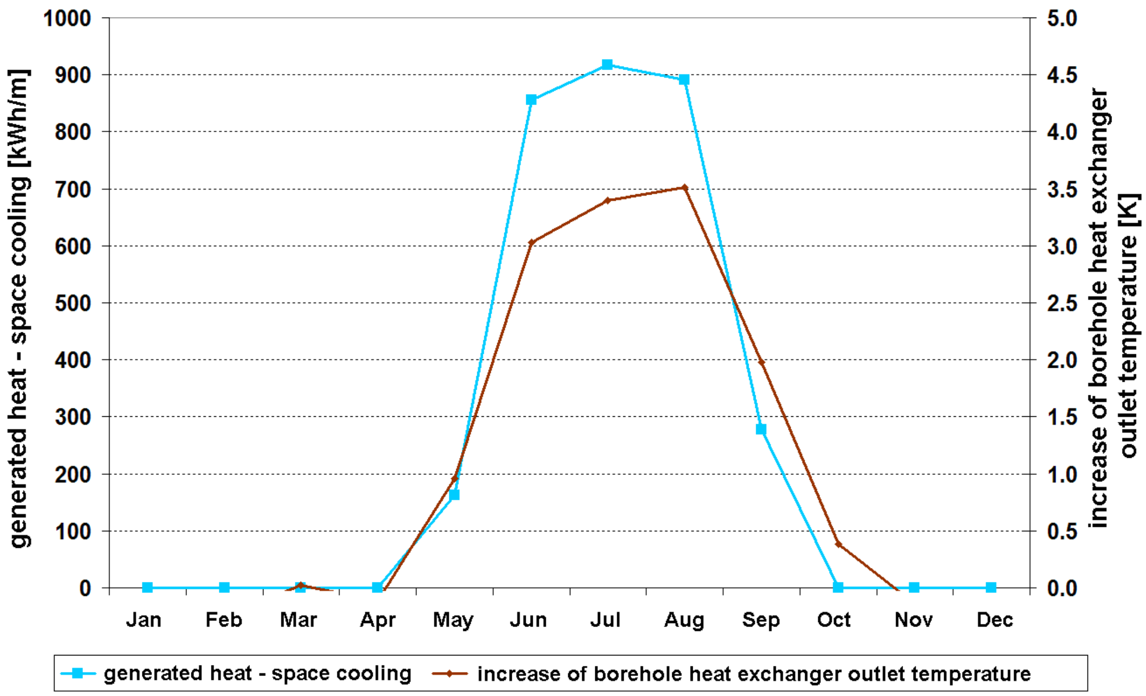


Fig. 67: Monthly correlation of generated heat for space cooling and the increase in the borehole heat exchanger outlet temperature for the case of the MINERGIE building with little shading (25%)

The influence of the imported heat into the borehole heat exchanger from the space cooling on the increase of the outlet temperature of the borehole heat exchanger shows only a small effect from the preceding month in such a way, that the maximum is indeed reached one month later, but being only slightly higher than the previous value and that in October still an increase can be found, but with 0.3 K being rather small. Thus also in a monthly balance only a small heat storage effect can be observed. The increase in the outlet temperature of the borehole heat exchanger in the months from May to October shows only a small phase shift against the imported heat to the borehole heat exchanger.
In Tab. 8 the outlet temperatures of the borehole heat exchanger during DHW operation as monthly averages and the increase compared to the case without cooling are illustrated. In the months from October to April, the building, as can also be seen in Fig. 67, is not cooled. Thus the picture is constrained to the relevant months from May to September.
The increase in the monthly mean values of the borehole heat exchanger outlet temperatures show for all cases a behaviour that is similar to the one shown in Fig. 67, which considers one specific case separately. Thereby the maximal monthly increase of the outlet temperature of the borehole heat exchanger with rising cold/heat ratio increases. To integrate the augmentation of the outlet temperature of the borehole heat exchanger in a calculation scheme for the heat generator system performance during DHW operation, this increase needs to be computed in advance.
Therefore two modelling approaches have been developed. The first model uses an empirical approach for the monthly increase of the borehole heat exchanger outlet temperature based on the cold/heat ratio as indicator for the utilization of the temperature rise potential.

Tab. 8: Correlation of cold/heat-ratio with borehole heat exchanger - outlet temperature in the DHW operation during the cooling period

	B/W-HP 75% ME no C	**B/W-HP 75% ME**	**B/W-HP 50% ME**	**B/W-HP 25% ME**	**B/W-HP 25% MEP**
cold/heat-ratio	0%	15%	19%	23%	26%
generated heat – space cooling					
May	0	96 kWh	130 kWh	163 kWh	154 kWh
June	0	537 kWh	685 kWh	856 kWh	780 kWh
July	0	626 kWh	757 kWh	918 kWh	833 kWh
August	0	612 kWh	738 kWh	892 kWh	808 kWh
September	0	177 kWh	222 kWh	278 kWh	265 kWh
monthly average of the borehole heat exchanger outlet temperature in domestic hot water operation					
May	5.2 °C	5.6 °C	5.9 °C	6.1 °C	6.5 °C
June	5.9 °C	7.8 °C	8.4 °C	8.9 °C	9.0 °C
July	6.1 °C	8.4 °C	9.0 °C	9.5 °C	9.5 °C
August	6.2 °C	8.7 °C	9.2 °C	9.8 °C	9.7 °C
September	6.3 °C	7.5 °C	7.9 °C	8.3 °C	8.4 °C
increase of the monthly average of the borehole heat exchanger outlet temperature in domestic hot water operation					
May	0 K	0.4 K	0.7 K	1.0 K	0.9 K
June	0 K	2.0 K	2.5 K	3.0 K	2.9 K
July	0 K	2.3 K	2.9 K	3.4 K	3.2 K
August	0 K	2.5 K	3.0 K	3.5 K	3.3 K
September	0 K	1.2 K	1.6 K	2.0 K	1.9 K

The maximum temperature rise potential is therein the difference between the undisturbed borehole heat exchanger outlet temperature and the return flow temperature from the floor heat emission system. This model requires less input data than the second model but is also less precise. The second model uses a simplified physical approach. Again the aim is to estimate the monthly increase of the borehole heat exchanger outlet temperature.
Therefore the model assumes that the heat from passive cooling could be stored in the borehole heat exchanger during one day and is lost afterwards. This is a strong simplification, which is dedicated to a very simple calculation. The dynamic heat injection based on a daily cyclic heat load pattern then defines the diameter of the active storage volume around the borehole where the outlet temperature increase is identical to the result of the detailed simulation. The active storage volume based on a daily cyclic heat load pattern has been derived from several simulation variants. The equivalent heat capacity for the active ground heat storage equals 89 Wh/(mK). This corresponds to a diameter of the active ground heat storage d_G of 0.4 m. The total heat capacity of the borehole heat exchanger then for this case then is 8.9 kWh/K. Comparing this simple model to the detailed simulation results shows a very good agreement of the results for the month June to August with high cooling demand, where the deviations are between 0 K and 0.4 K for several simulation variants. The times in the beginning and at the end of the cooling period show bigger deviations of up to 1 K, but have less influence on the seasonal performance calculation results because of the small cooling energy. The overall increase of the borehole heat exchanger outlet temperature in domestic hot water operation for the whole year furthermore show a very good agreement with deviations in the range of 0.1 K to 0.2 K.

5.4 Field test results of ground-coupled passive cooling

Four of the field monitored objects, 2 compact units in the Austrian field test and 2 ground-source heat pumps monitored in Switzerland have been equipped with an additional passive cooling function in summer operation. All four systems have been documented in Best Practice Sheets which contain further information on the field monitoring results. In the following chapters, mainly the passive cooling operation in residential buildings is discussed.

5.4.1 Passive cooling with borehole heat exchangers in Switzerland

In the Swiss contribution to IEA HPP Annex 32 the integration of the passive cooling function in heat pump systems equipped with borehole heat exchangers has been investigated. A simulation study by Dott, Huber and Afjei (2007) yielded SPF of 10-25 for a passive cooling mode. On the one hand these performance factors are based on an overall auxiliary consumption of 150 W for the source pump, the sink pump and the control system. On the other hand, the fraction of rejected heat in the cooling mode reaches values up to 45% of the space heating and DHW demand. Design parameters and hints for the integration of the system derived in this project are given in chap. 5.5.

Tab. 9: Parameter of the houses in the Swiss field test

Field monitoring object	CosyPlace	Muolen
Picture		
Site	Basel (BS), 316 m a.s.	Muolen (SG), 492 m a.s.
Exposition	North-oriented hillside, detached	Plain terrain, detached
Year of commissioning	2007	2008
Building standard	MINERGIE-P®	MINERGIE ®
Energy reference area	1064 m^2	279 m^2
Flats	5 flat (multi-family house)	1 (single family house)
Inhabitants	9 adults, 3 children	2 adults, 1 child
Nom. capacity (B0/W35)	15.5 kW	8.4 kW
Length of boreholes	2 x 130 m	1 x 150 m

The measurement points comprises the data for room indoor parameters, produced and extracted heat in space heating, DHW and cooling, respectively, as well as the electricity consumption of the two heat pump systems. The systems have been installed in the MINERGIE-P® multi-family house CosyPlace in canton Basel-City (BS) and the single-family house in Muolen, canton St. Gallen (SG) according to MINERGIE®. The characteristic values of the two field monitoring plants are summarised in Tab. 9.

5.4.2 Energy and performance

The two systems showed a good performance in the field monitoring, both in space heating & DHW and in the cooling operation. The energy and performance metrics are summarised in Tab. 10.
The system in Muolen is equipped with a so-called NC (Natural Cooling) -Box, which comprises all hydraulic components for the passive cooling and enables a simultaneous DHW and passive cooling operation, where the heat rejected from the building is used as heat source for DHW production.
The values for space heating are in a good performance range in the CosyPlace and correspond to average values of field monitoring objects (see chap. 3.1) in Muolen. The DHW operation is increased in summer from 2.5 to 2.9 due to higher ground temperatures, which are furthermore enhanced by the heat rejection in passive cooling. In Muolen, the change in DHW performance from 2.8 to 3.7 between winter and summer period is even more pronounced due to the simultaneous operation of DHW production and space cooling.
In the CosyPlace the cooling performance factor reaches a value around 8 in the summer period 2008/09 and values around 9 in the previous summer 2007/08. The control of the cooling system was changed in the measurement period of summer 2009, see Fig. 70. After the change almost 2/3 of the total heat was extracted, and a weekly-averaged cooling performance of 15.2 was reached.
In Muolen the cooling performance factor is with 7.3 slightly lower.

Tab. 10: Energy- and performance metrics of the field-monitored buildings

	CosyPlace	**Muolen**
Measurement period	Winter 2008/09, Summer 2009	Winter 2009/10, Summer 2009
Produced heat SH	32'850 kWh	9'186 kWh
Produced heat DHW (winter/summer)	7'258 / 5'552 kWh	1'522 / 765 kWh
Extracted heat cooling	3'637 kWh	1'058 kWh
WNG (SNG) heating	4.3 (4.3)	3.8 (3.7)
WNG (SNG) DHW (winter/summer)	2.5 / 2.9 (1.7 / 1.7)	2.8 / 3.7 (2.7 / 3.5)
WNG (SNG) cooling	8.1 (7.3)	7.3 (7.1)
Electricity DHW/cooling	12'968 kWh	3'657 kWh
WNG overall	3.8	4.1
Average cooling capacity (based on net living space)	10 W/m^2_{NGF}	7.5 W/m^2_{NGF} (max. 29 W/m^2_{NGF})

5.4.3 Expenditure for the passive cooling

Fig. 68 left depicts the produced and rejected heat, respectively, of the heat pump system during the summer period from 1. April 2009 to 30. September 2009. On the right side of the figure, a breakdown of the the respective electricity is shown.
5552 kWh or 73% of the heat is produced for DHW corresponding to a specific consumption of 2.9 kWh/person/day or a DHW tapping of 46 l/Person/day at 50 °C tapping temperature. For space heating 2075 kWh or 27% of the heat is produced. In the space cooling operation 3637 kWh corresponding to 3.3 kWh/m^2 of heat is rejected.

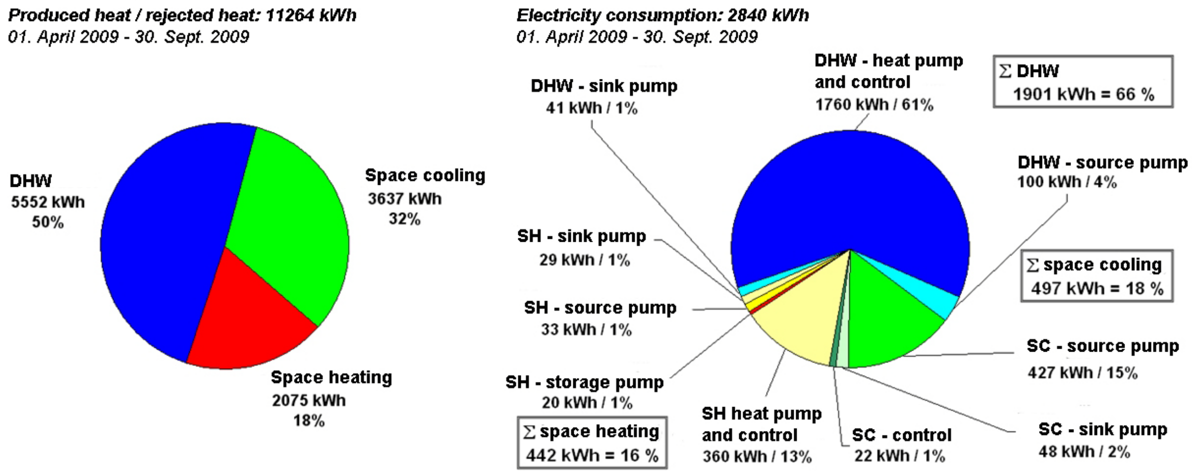


Fig. 68: Shares of produced and rejected heat (left) and electricity consumption of the respective system components (right) of the field monitoring plant Cosy Place in Basel

On the 05.08.2009 the control parameters of the cooling function have been adapted, in particular omitting the limit of the minimal supply temperature of 22 °C, which was lowered to 17 °C. This change in settings led to a nearly doubled cooling energy. Before the adaptation of the controller settings, (01.04.2009 - 04.08.2009) 1232 kWh were extracted, while in the rest of the summer (05.08.2009 - 30.09.2009) 2405 kWh have been rejected.
Regarding the cooling power in this second half of the summer, values of between 4 kW and 5 kW are evaluated. The specific extraction power of the floor emission system in cooling mode is in the range of 10 W/m^2.
The consumed electricity is broken down to 13% or 360 kWh for the heat pump operation in space heating mode, 61% or 1760 kWh for the heat pump in DHW operation and 1% or 22 kWh for the space cooling operation for the system control.
The entire electricity consumption for the passive cooling operation adds-up to 15% or 497 kWh. In the cooling operation, only electricity for the circulation pumps and the control is required. In the field object "CosyPlace" the supply is very low due to a highly efficient, electronically controlled permanent magnet syncron motor pump, which can adapt the power to the varying volume flow rate based on the position of the room thermostats.
This circulation pump has a power consumption only between 30-40 W and thus consumes in space heating operation only about 2 % of the total electricity consumption. In cooling operation about 10 % or 48 kWh are consumed by the circulation pump for the floor emission system. The main consumer, though, is the source pump for the borehole heat exchanger, which consumes 427 kWh.
Fig. 69 shows the energies of the field monitoring plant in Muolen.
In 660 h of operation in cooling mode corresponding to 18% of the summer period from 1. May 2009 until 30. September 2009 totally 1058 kWh heat has been rejected from the single family house. During the same period 765 kWh heat have been produced by the heat pump in DHW operation, which took an operation time of 87.6 h, about 1/6 of this time in simultaneous space cooling and DHW operation.
The respective electricity consumption is depicted in Fig. 69 right. In total 638 kWh have been consumed for a produced or rejected heat of 2007 kWh.
About 20% of the 23 kWh electricity is consumed for the circulation pump in space heating operation, the rest is consumed in DHW operation. The average cooling power in the summer period was 1.6 kW, corresponding to a specific cooling power of 6 W/m$^2_{\text{floor area}}$. Temporarily, a maximum cooling power of 6.4 kW (maximum value of 15 measurement points) corresponding to a specific cooling power of 23 W/m$^2_{\text{floor area}}$ was reached, which was registered after a period without cooling operation.

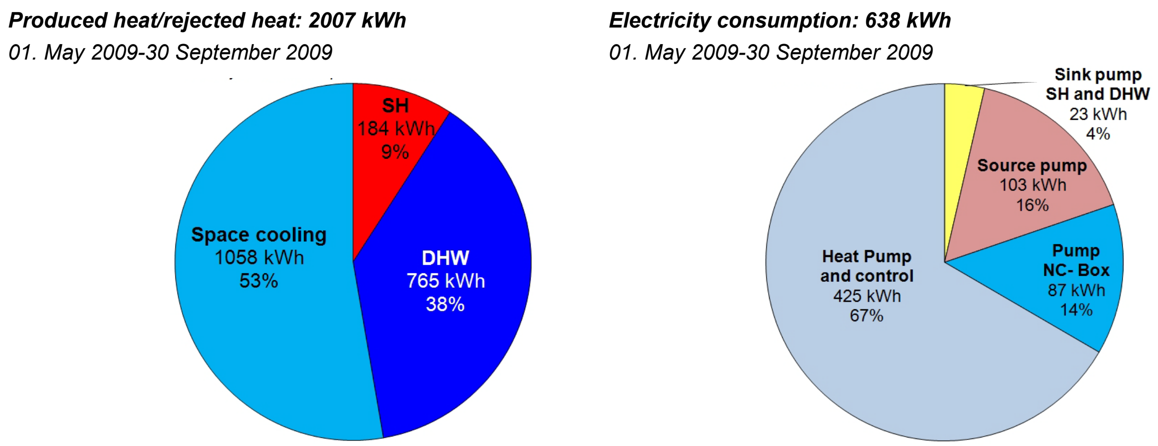


Fig. 69: Shares of produced and rejected heat (left) and electricity consumption of the respective system components (right) of the field monitoring plant in Muolen

The average power rejected at the borehole heat exchanger was 12 W/m.
The seasonal performance factor system (incl. source and sink pump) of the passive cooling was 7.1 and the seasonal performance factor generator (including only the source pump) for the DHW production was 3.6.

5.4.4 Behaviour of the borehole heat exchanger

The dependency of the produced and extracted heat and the brine outlet temperature of the borehole heat exchanger in operation is shown in Fig. 70 for the Cosy Place in summer 2009.

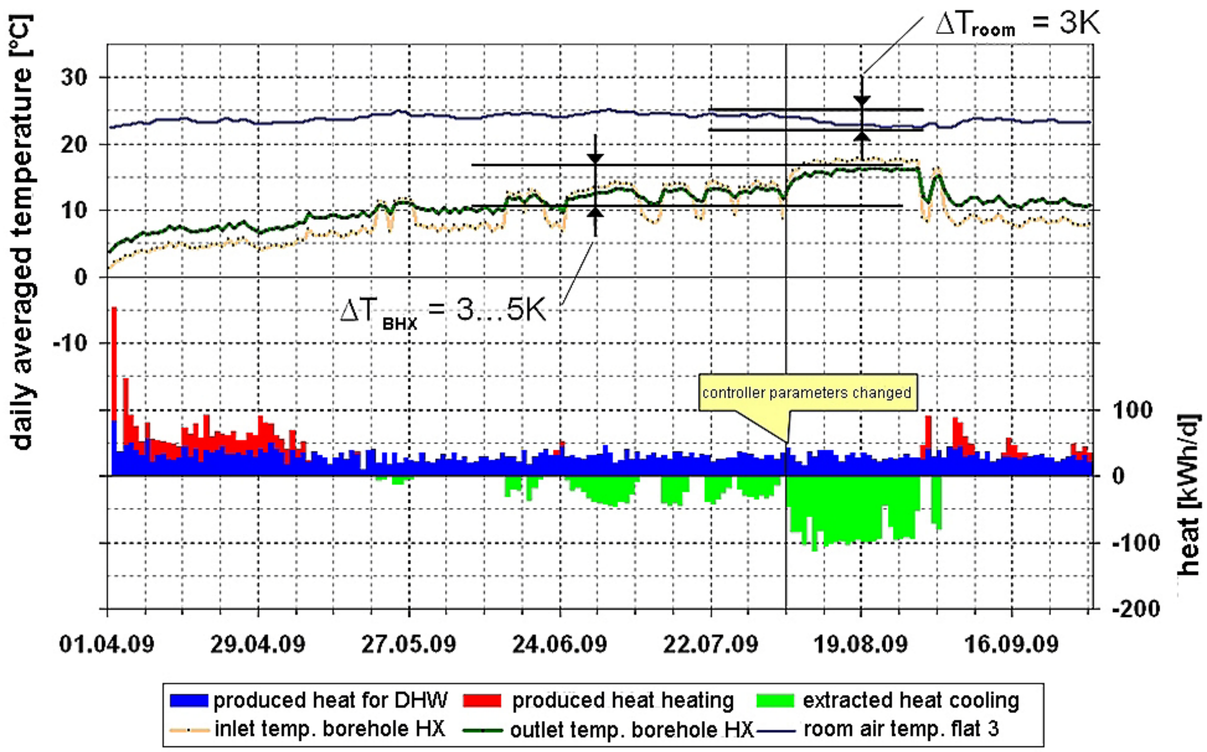


Fig. 70: Evaluation of the produced and extracted heat in the CosyPlace regarding the borehole heat exchanger and the room air temperatures in summer 2009.

The daily average borehole outlet temperature in wintertime is between 0 °C and 7 °C. Due to the lower heat extraction in spring, the outlet temperature increases to about 10 °C in May and June. During cooling operation the temperature further increases by about 3 K and decreases again after the cooling operation. After the change of the controller settings in the beginning of August the heat input to the borehole heat exchanger is further increased, and the daily average outlet temperature rises to 16.4 °C. At the end of the cooling operation in September, the outlet temperature drops again to 10 °C, i.e. a significant long-term storage effect could not be observed.

5.4.5 Impact of the passive cooling on the room air temperature

Fig. 71 shows the distributions of the room air temperatures in the Cosy Place before (left) and after (right) the change of the supply temperature setting in the cooling mode in the summer 2009. Flat 5 has not been cooled the whole summer due a decision of the users, even though flat 5 is the most exposed flat for solar irradiation gains.
Consequently, room air temperatures are highest in flat 5. A comparison of temperature frequencies in flat 5 and the other 2 flats confirms that the temperature with the passive cooling operation is shifted to the conformtable areas, and temperature above 26°C are avoided, which occur in flat 5.
Even temperature on the lower edge of the comfort temperature range are reached after the change of control parameters, since cooling loads are lower than in solar exposed buildings.

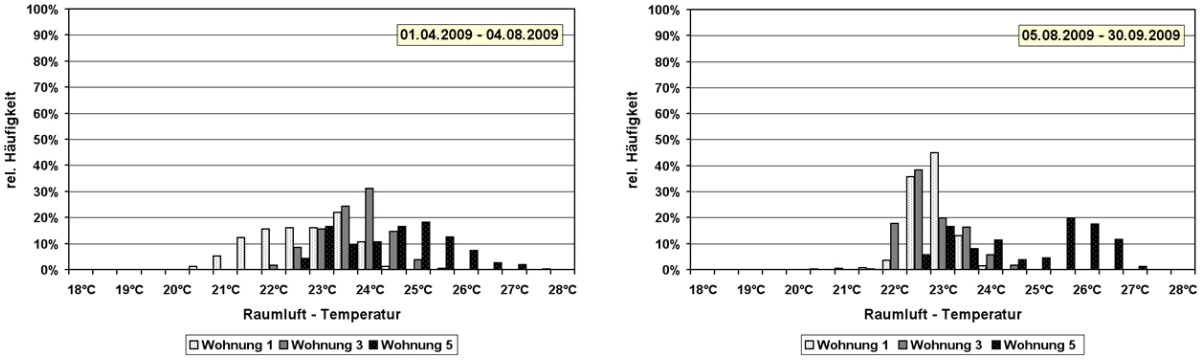


Fig. 71: Room temperatures in the Cosy Place field monitoring plant in summer 2009.

Concluding the passive cooling is well-suited to decrease the room temperature by about 3 K and therefore enhance comfortable temperature conditions.

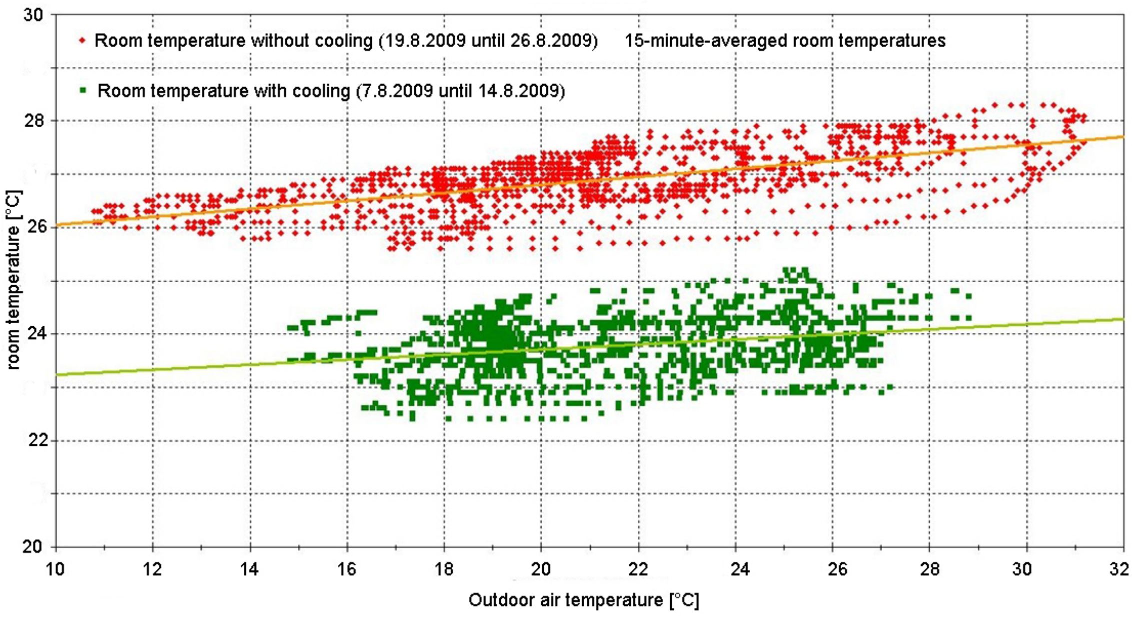
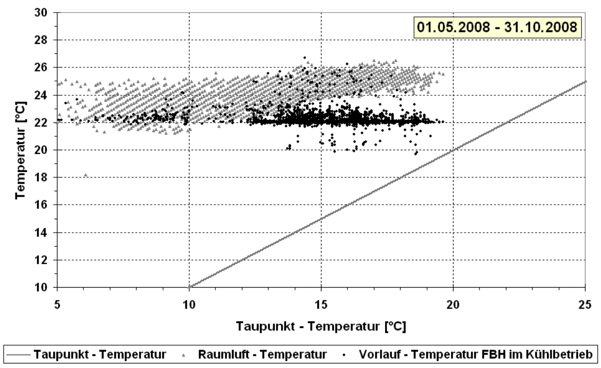


Fig. 72: Comparison of room air temperatures with and without passive cooling operation

For the field monitoring object in Muolen the indoor room air tempereatures in the period without cooling during the holiday absence of the inhabitants could be compared to a period with cooling operation in order to evaluate the impact of the passive cooling operation. Thereby, the periods were chosen in such a way that the outdoor air temperatures are approximately the same in both periods.
Results depicted in Fig. 72 confirm that the indoor air temperatures can be lowered by about 3 K with the passive cooling operation. After the activation of the passive cooling after the holiday absence, the cooling operated about 2 days without interruption and on the second day, the indoor air temperature was already lowered by 2 K. Within this period about 50% more heat was extracted than the average value for the entire summer period.

5.4.6 Condensation risk at the room emission system

The condensation risk is estimated based of the surface temperature of the floor emission system. Even though it is not measured directly it can be estimated by the room temperature and the supply temperature of the floor circuit, since the floor surface temperature is inbetween the room temperature and the supply temperature. The evaluation was made separately for both summer periods.
In Fig. 73 the room air temperatures (grey dots) and the supply temperature of the floor circuit (black dots) in the cooling operation are plotted vs. the dewpoint temperature (grey line). The dew point temperature is derived by the relative humidity of the room air. The grey line is therefore the condensation limit. The surface temperature lays between the grey and the black dots. Since even the supply temperature of the floor circuit is always higher than the condensation limit, the condensation risk for this period is negligible.
Even after the decrease of the supply temperature in August 2009, no condensation risk on the floor surface results.
Fig. 73 right shows the lower room temperatures and decreased supply temperatures, which fall below the condensation limit for a few hours by 1-2 K. However, these low values are only reached in single hours, mainly in the first half hour after the start of the cooling operation due to the cold borehole heat exchanger and are not continuously reached. Moreover, the room air temperature lies above 22 °C in the total summer period and thus, the surface temperature is above the supply temperature. At free pipes and valves, however, condensation could occur under these boundary conditions. Thus, a lower supply temperature should be avoided, also due to comfort considerations.
These evaluations confirm the findings of the simulation study, that the comfort boundary condition on the floor surface is stronger than the condensation risk for locations in the Swiss middle land.



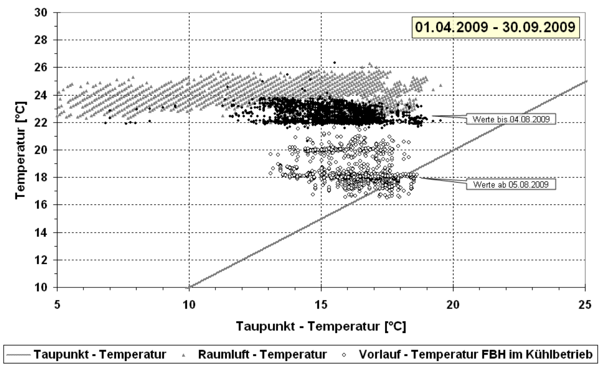


Fig. 73: 15 min.-average room temperatures and supply temperatures of the floor heating circuit in flat 3 in summer 2008 (left) and in summer 2009 (right)

5.4.7 Ground-coupled compact units with passive cooling option in Austria

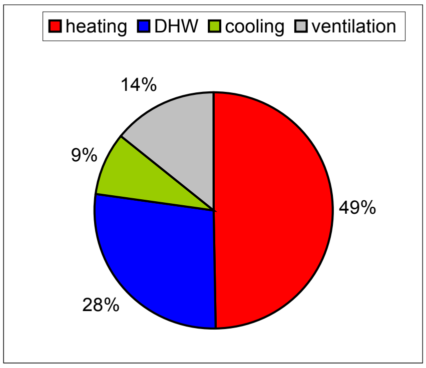
In the Austrian field monitoring project (Zottl, Huber and Köfinger (2010)), two compact units were monitored.

The compact units of a nominal heating capacity (B0/W35) of 3.3 kW equipped with ground collectors and installed in passive houses cover the functions space heating, DHW, ventilation and a passive space cooling. Field monitoring took place in the period from Oct. 2008 - Oct. 2009.

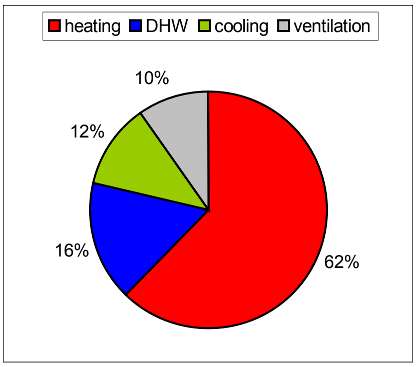
Tab. 11: Parameters of houses with ground-coupled compact units in the Austrian field test

Field monitoring object	**Hitzendorf**	**Judendorf-Straßengel**
Picture		
Site	Hitzendorf, Styria	Judendorf, Styria
Exposition	Detached single family house	Detached single family house
Year of commissioning	2008	2008
Building standard	Passive house	Passive house
Energy reference area	180 m^2	210 m^2
Inhabitants	2 adults, 2 children	2 adults, 2 children
Nom. capacity (B0/W35)	3.3 kW	3.3 kW
Size horizontal collector	130 m	200 m
SPF generator (SH/DHW/SC)	4.1 (4.7/ 3.7/ 4.7)	4.3 (4.3/ 3.6/ 9.0)

Tab. 11 gives an overview of the field monitoring plants. Both houses are single family passive houses with each 4 inhabitants and the same compact unit installed. The overall seasonal performance factors reach high values above 4. The performance factor in passive cooling mode is with 9.0 in a normal range, while the factor 4.7 is rather low.



Space Heating:	6302 kWh	Ventilation:	1804 kWh
Hot water:	3511 kWh	Cooling:	1083 kWh
Total Output:	**12700 kWh**	**Energy Input:**	**3403 kWh**



Space Heating:	5823 kWh	Ventilation:	919 kWh
Hot water:	1539 kWh	Cooling:	1088 kWh
Total Output:	**9369 kWh**	**Energy Input:**	**2640 kWh**

Fig. 74: Energy fractions of the two Austrian compact units (Zottl et al., 2010)

This can be explained by higher average ground temperatures evaluated for the system in Hitzendorf, which increases the necessary running time to extract the heat energy from the building, and therefore, the performance factor in the cooling mode is affected. In the second field monitoring plant, lower average ground temperatures have been measured, and the performance factor is significantly higher, as can be expected.
Fig. 74 shows the energy fractions for the year-round operation. In both houses, the main fraction is dedicated to space heating. The DHW fraction is between 20% and 35% of the space heating consumption. Space cooling contributes about 10% the total energy production, similar to the ventilation function, which is in the range of 10%-15%.
The passive cooling of the system installed in Judendorf-Straßengel has been analysed in more detail. Fig. 75 left shows the performance factors for the cooling mode including the distribution of supply temperatures. The cooling period is defined from May to September in which 50% of the cooling demand (1089 kWh) was registered in July. The compact unit was operating 850 hours for passive cooling.

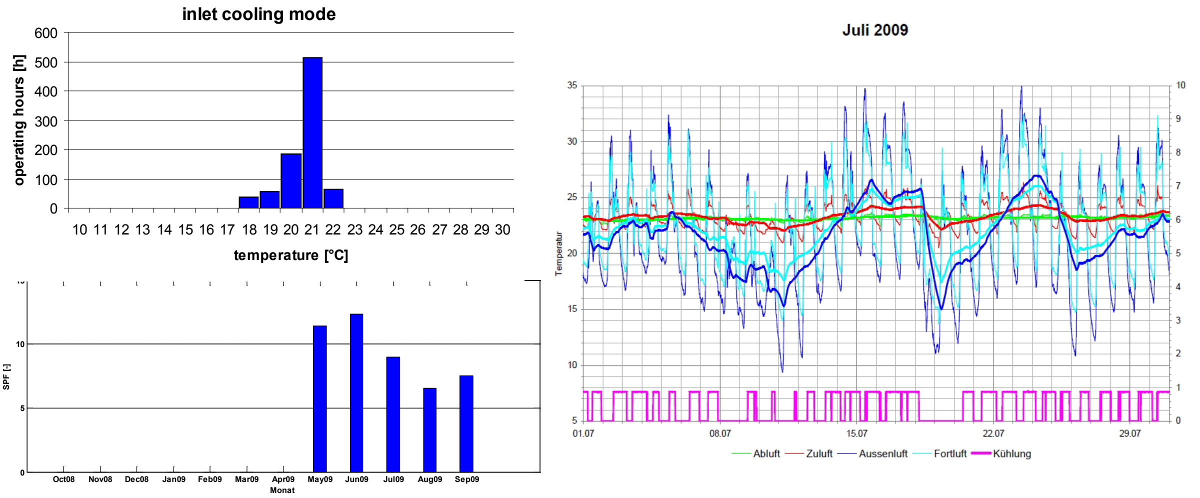


Fig. 75: Performance factor and temperatures (left) passive cooling operation in July 2009 (right) in the field monitoring plant Judendorf-Strassengel

For approximately 500 hours of this period, the inlet temperature was 21°C. The tendency to a smaller performance factor during the cooling period is caused by the temperature rise inside the brine collector, affected by an additional regeneration by passive cooling and the outdoor temperature.
With a decreasing temperature difference between heat source (building) and heat sink (collector) the efficiency of the cooling mode decreases, too. According to Dott, Afjei and Huber (2007) the simulated seasonal performance factor of passive cooling with borehole heat exchangers is in a range of 10 and 25 depending on the cooling load. In the field monitoring plant, however, performance factors are between 7 and 12.5 which tends to be lower than simulated values. This is due to the higher ground temperature of the “working space” collector in the summer time compared borehole heat exchangers.
In Fig. 75 right, the cooling operation during July 2009 is given in more detail. The thin curves display the values of outdoor air-, supply air-, return air- and exhaust air temperatures. Moreover, the trend lines have been drawn as thick line in the respective colour. 50% of the days in July 2009 had an outdoor temperature above 30°C. By passive cooling the indoor air temperature (green curve) reached maximum values of only 23°C. The peaks of the supply air temperature (red curve) are caused by the pre-heating of the ambient air during the domestic hot water production. The trend of the extract air (green curve), though, is not negatively affected by this additional heat input during the summer.

5.5 Design recommendation for a passive cooling function

The following design recommendations have been derived in Dott, Afjei and Huber (2007).

5.5.1 Hydraulic system layout

Concerning the hydraulic layout, different marketable system configurations have been evaluated which led to a simplified standard layout. The objective was to derive the best configuration in terms of simplicity and robustness.
Two standard layouts and related investigations are given in the following chapters, one for a passive ground cooling-only and one for simultaneous space cooling/DHW operation and one for a sole passive ground cooling option.

5.5.1.1 System layout for passive and simultaneous cooling operation

Fig. 76 shows the layout enabling both passive cooling with the borehole heat exchanger and by simultaneous operation of the heat pump for space cooling and DHW production.

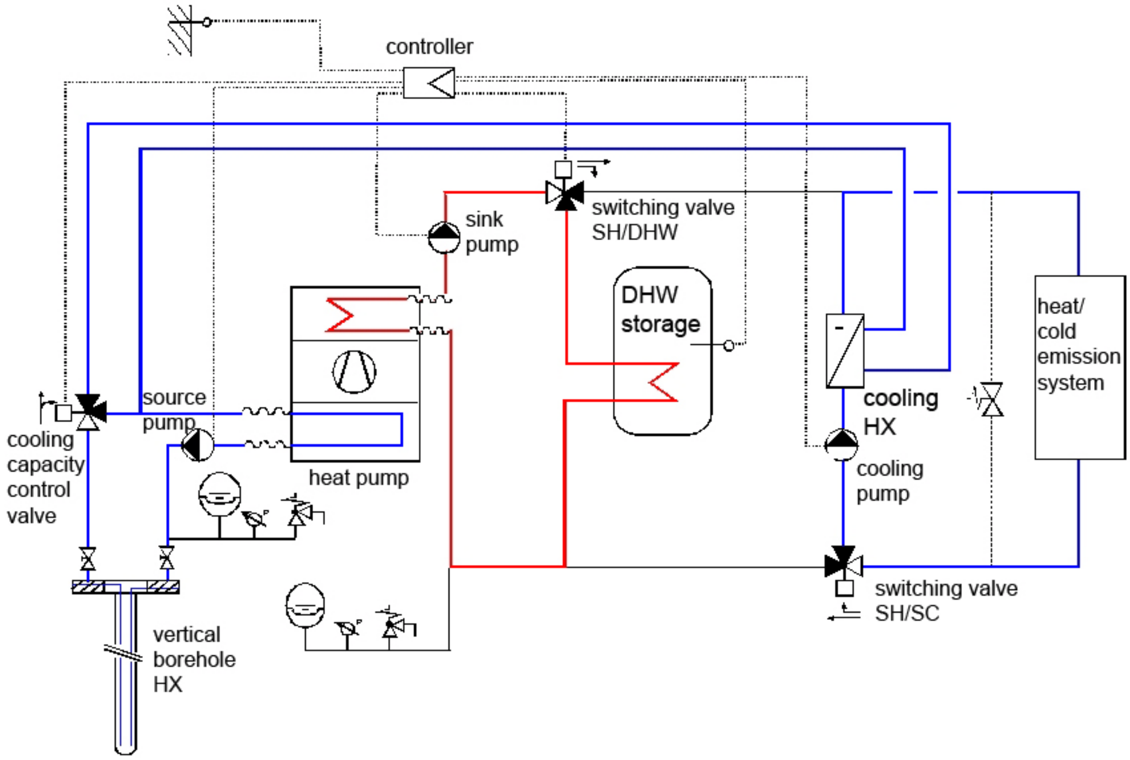


Fig. 76: Hydraulic system layout including an option of passive cooling and active cooling by simultaneous space cooling/DHW operation

5.5.1.2 System layout for passive ground cooling operation

Without an option of simultaneous operation of space cooling and DHW, the system layout depicted in Fig. 76 can be further simplified, as shown in Fig. 77, where the pump and the valve in the emission system are no longer required.

<u>Recommendation hydraulic layout:</u>

Since the simulations show only a marginal benefit of the simultaneous operation, the simpler configuration with the only passive ground cooling option is recommended.

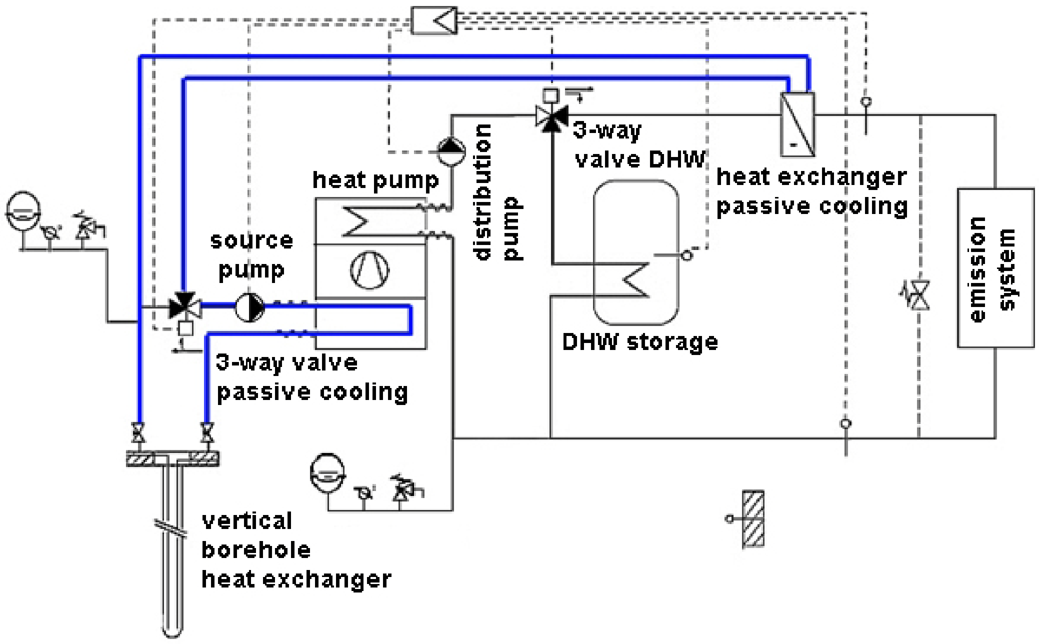


Fig. 77: Simplified hydraulic system layout without simultaneous space cooling/DHW option

5.5.1.3 Integration of the borehole heat exchanger

The evaluation of the two possible serial configurations depicted in Fig. 78 left and Fig. 78 right revealed that the flow direction heat pump, borehole heat exchanger and floor heat exchanger delivers a higher COP. In situations with capacity control, i.e. in cases with simultaneous DHW operation which delivers a higher cooling capacity than needed in the room, all temperature differences are fixed by this capacity. At the same start conditions of the ground the same temperatures for the ground heat exchanger result. However, in the above given order, the inlet temperature to the heat pump evaporator is by the temperature change in the floor heat exchanger higher and thus yields a higher COP for the DHW operation. Moreover, this integration minimises the risk of freezing in the cooling heat exchanger in case of low cooling load due to the buffering in the ground. Therefore, an integration of the floor heat exchanger in the order "heat pump, vertical borehole heat exchanger and floor heat exchanger" is preferable as shown in Fig. 78 left.

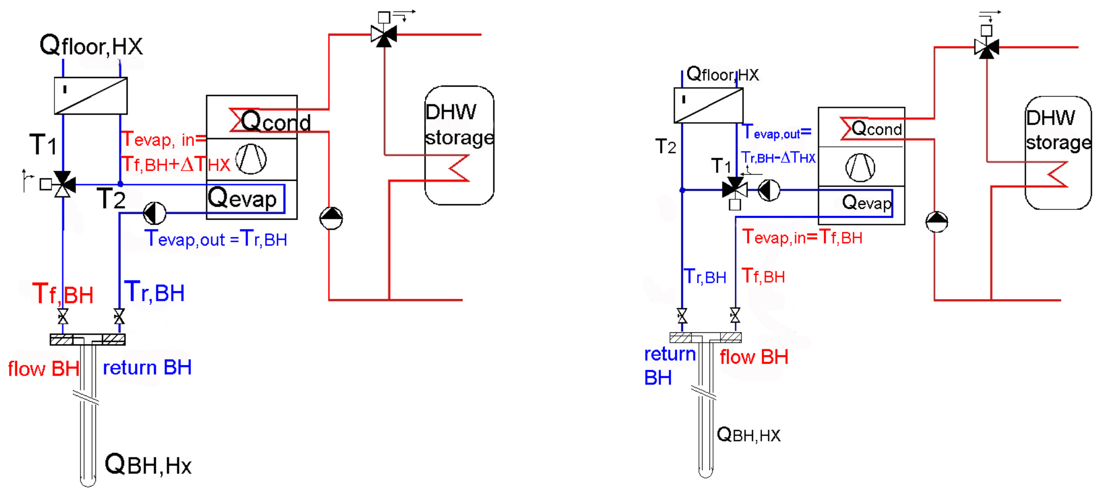


Fig. 78: Variants for the integration of the vertical borehole heat exchanger

<u>Recommendation for the hydraulic integration of vertical borehole heat exchanger:</u>

The vertical borehole heat exchanger shall always be integrated in serial configuration in the order of the flow heat pump evaporator – vertical borehole heat exchanger – floor cooling heat exchanger, since the configuration maximises the COP and minimises the danger of frost in the floor heat exchanger

5.5.2 Design of the system components

The design recommendations for the components are based on an evaluation of the meteorological conditions of the Swiss middle land characterised by the weather data set of Zurich Meteoschweiz.

5.5.2.1 Vertical borehole heat exchanger

With a design of the vertical borehole heat exchanger (double U-tube type) according to the space heating operation, about 90% of the cooling energy for the summer operation could be covered by direct ground cooling without active cooling. This means that a conventional design of a single vertical borehole heat exchanger for the space heating is sufficient for the cooling operation, too, and active cooling is not needed.

<u>Design recommendation borehole heat exchanger</u>

Usually no extra design of the borehole heat exchanger is needed for the space cooling operation, so the vertical borehole heat exchanger shall be designed according to the space heating requirement

5.5.2.2 Floor cooling heat exchanger (cooling heat exchanger)

The cooling capacity of the vertical borehole heat exchanger depends on the required flow temperature for the cooling operation, i.e. the higher the flow temperature, the larger the capacity of the borehole heat exchanger for the cooling operation. Thus, the crucial point for the fraction of covered space cooling energy need is the design of the cooling heat exchanger. At a temperature difference of 1 K in the heat exchanger 94% of the cooling energy can be covered (design borehole cooling capacity 26 W/m). In case of a temperature difference of 3 K, only 66% can be covered (design borehole cooling capacity of 13 W/m). Therefore, the design of the floor heat exchanger is more important than a particular hydraulic configuration.

<u>Design recommendation floor cooling heat exchanger</u>

The cooling heat exchanger for the coupling of the ground and the floor shall be designed to the lowest temperature difference possible in order to minimise the loss of temperature level of the heat transfer and maximise the covered space cooling energy need by passive cooling operation. However, since the performance depends on the electrical power consumption of the circulation pump, as well, which depends on the pressure drop, an optimisation of the thermal characteristic and the pressure drop yields the optimum design of the heat exchanger.

5.5.3 System performance and sensitivities

For a Minergie® building with a heat load of 4.5 kW the electrical power consumption and the energy consumption during passive cooling operation for high, average and low efficiency are juxtaposed in Tab. 12. Here low efficiency stands for little efficient, but today still used components, average efficiency for good products, and high efficiency for very good market available technologies. The connection between the heat sink and the heat supply systems demands normally a circulating pump for each hydraulic loop and additional components for control and auxiliary power unit (controller, actuators, sensors and, as the case may be, additional circulating pumps). Tab. 12 shows the composition of the electrical energy consumption for a runtime of 500 and 1500 hours, respectively. With high efficient components the electrical power consumption can easily be reduced from 130 W for an average efficiency to 60 W, i.e. less than half. Although, low efficient components or as well as an over dimensioning (e.g. of the heat pump) can easily yield a rise in the power consumption to 210 W.

Tab. 12: Variation and sensitivity of electrical costs and efficiency during passive cooling operation

	Efficiency	high	average	low
1	circulating pump borehole heat exchanger	45 Watt	100 Watt	150 Watt
2	control	5 Watt	15 Watt	30 Watt
3	circulating pump floor heat emission system	10 Watt	15 Watt	30 Watt
4	**Sum el. power consumption**	**60 Watt**	**130 Watt**	**210 Watt**
5	el. energy consumption cutting of temperature peaks (runtime 500 h)	30 kWh/a	65 kWh/a	105 kWh/a
6	el. energy consumption mild continuous cooling (runtime 1500 h)	90 kWh/a	195 kWh/a	315 kWh/a
7	rejected heat from space cooling with mild continuous cooling	2500 kWh/a	2500 kWh/a	2500 kWh/a
8	**Generator seasonal performance factor**	**33.3**	**14.5**	**9.3**
9	**System seasonal performance factor**	**27.8**	**12.8**	**7.9**

From this rather vast variance of the electrical power consumption a similarly broad range of the heat generator seasonal performance and system seasonal performance is obtained. An assumed mild continuous cooling with the passive cooling system leads to a runtime of 1500 hours per year and a heat removal of approximately 2500 kWh/a. From the simultaneous variation of the efficiencies of all components, according to line 4 in Tab. 12, the heat generator seasonal performance varies between 9.3 for a low efficiency up to 33.3 for a high efficiency of all components as well as the system seasonal performance with values between 7.9 and 27.8.

Design recommendation system performance

A passive ground cooling with vertical borehole heat exchanger performs a coupling of the indoor space to the natural cooling source ground. For good seasonal performance values the coupling by a heat exchanger with low temperature difference, preferably in the range of 1 K between the source and the sink side is crucial. On the other hand, the hydronic expenditure affects the performance, in particular for the BHX source pump. The main influence on the passive cooling seasonal performance there has the circulating pump of the borehole heat exchanger. In the floor heat emission system, high efficient circulating pumps are quite widespread. In the borehole circuit low efficient circulating pumps are still of common use and highly efficient pump can rise the efficiency significantly. With highly efficient pumps the performance factor can easily be double from 15 to 30.

5.5.4 Control considerations

Standard systems for space heating and -cooling on the market are equipped with switchable thermostatic valves. That means that an action by the user is required to switch the control valves from heating to cooling operation, since in space heating operation, the thermostatic valves close, when the room temperature increases. However, in space cooling operation, a reverse operation is required, i.e. the thermostatic valves have to open in case of an increasing room temperature.
In the investigated system no manual activation of the user is required due to the self-regulation effect, i.e. the space heating and cooling is always activated. Concerning the control of this automated operation the following issues have to be considered:

- In the intermediate season both space heating and space cooling need can occur, leading to an intermittent heating and cooling operation. In order to prevent a counter-heating effect due to the high inertia of floor emission system, a dead time of 12 hours after the heating operation should be provided. Otherwise, it might occur that the space heating and -cooling may only act on the thermal mass of the floor emission system, and the room temperature will not be much affected, causing higher energy consumption in both modes.
- Dew point control is normally not required for Swiss boundary conditions. This is due to the fact that the minimum comfortable surface temperature of the floor is a stronger boundary condition than the dew point temperature. A moderate cooling curve can further minimize the condensation risk.
- A capacity control of the space cooling operation is useful, e.g. by a cooling curve. Thereby, the capacity of the borehole heat exchanger can be best adapted to the cooling needs in the room and therefore, the cooling need can be covered by passive cooling to a large extend.

<u>Control of passive ground-coupled floor cooling systems</u>

A moderate cooling curve which increases supply temperature with increasing outdoor air temperatures can maximise the capacity of the borehole for passive cooling operation, minimises the risk of condensation on the floor and yields adequate indoor comfort. Under Swiss boundary conditions a dew point control is usually not required. However, rooms with higher moisture content (e.g. kitchen, bathroom) shall not be cooled by the floor.

5.5.5 Reachable comfort

Simulations showed that the passive ground cooling potential is sufficient for keeping the comfort boundary concerning the operative temperature with an adequate shading concept and design of the system components.

<u>Reachable comfort of passive ground-coupled floor cooling systems</u>

A passive ground cooling with vertical borehole heat exchanger coupled to a low-temperature floor heating system can reduce the room temperature by 2 to 4 K depending on the applied shading. In combination with a passive air cooling with maximum air exchange rate of 1 ACH and consequent shading, operative temperatures can be kept below the boundary of 28°C. The design of the components for the space heating operation is sufficient for the space cooling operation, as well. Concerning local comfort a floor temperature between 20°C and 29°C shall be kept. Rooms with higher comfort requirements, e.g. bathrooms due to barefoot walking, shall not be cooled by the floor.

6 COMPARATIVE EVALUATION OF HEAT PUMPS

In this chapter, comparison of the field monitoring results to other field test results and characteristic numbers of the heat pump operation as well as a comparison of the environmental impact are given.

6.1 Comparison to performance in existing buildings

In this chapter, the results of the HP-Efficiency field test are compared to the German field test in existing buildings, which has been introduced in chap. 2.2.4.1. In comparison to the "HP-Efficiency" project with new dwellings of the construction year 2006, the buildings in the project "HP in existing buildings" comprises 44% of buildings before 1970, 34% of buildings constructed between 1971 and 1990 and 22% of buildings constructed after 1990. The average living area is 190 m^2 and the average energy consumption 180 kWh/(m^2a), which is based on the oil consumption of the last 5 years for space heating and DHW including distribution losses.

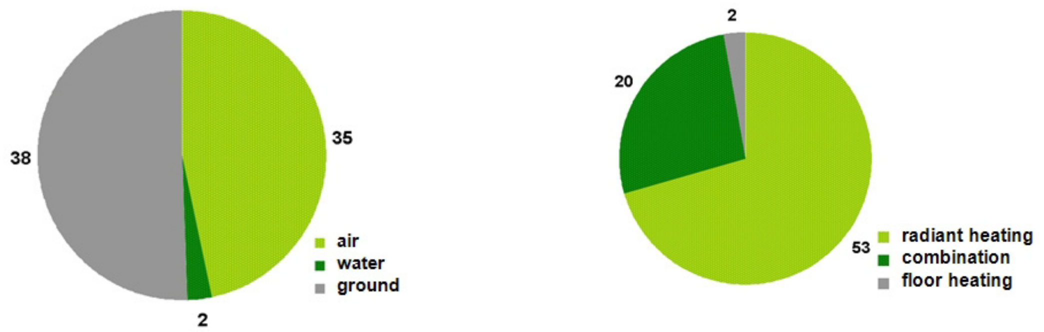


Fig. 79: Heat sources and heat emission systems in German field test "HP in existing buildings" (Miara, 2008)

Contrary to the new buildings equipped with 95% floor heating systems, 66% of the existing buildings are equipped with radiators, but also a significant fraction of 28% of the non-retrofitted buildings are equipped with a combination of floor heating systems and radiators connected to a storage and a small fraction of 6% with floor heating systems only. The fractions of source and emission systems are summarised in Fig. 79.

Fig. 80 depicts the dependency of the daily average performance factor in the space heating operation mode on the temperature lift of the heat pump, i.e. the difference between the supply inlet temperature and the source inlet temperature. The brine-to-water heat pumps are in the range of daily performance factors of 4.5 at 30 K temperature lift and 3.0 as 45 K with an average value of 3.8. Air-source heat pumps range between 3.5 at 25 K and 2.0 at 60 K with an average value of 3.0. The figure underlines the strong dependency of the heat pump performance factor on the temperature lift between source and supply temperature and the strong performance increase with lower temperature lifts.

Fig. 81 shows the evolution of the monthly performance factors of ground-source heat pumps in existing buildings. Due to the higher space heating energy needs in existing buildings, the performance is dominated by the space heating mode. However, temperatures levels of space heating design temperatures and DHW temperatures are in a similar range, and therefore, the deviation between summer and winter performance factors are moderate.

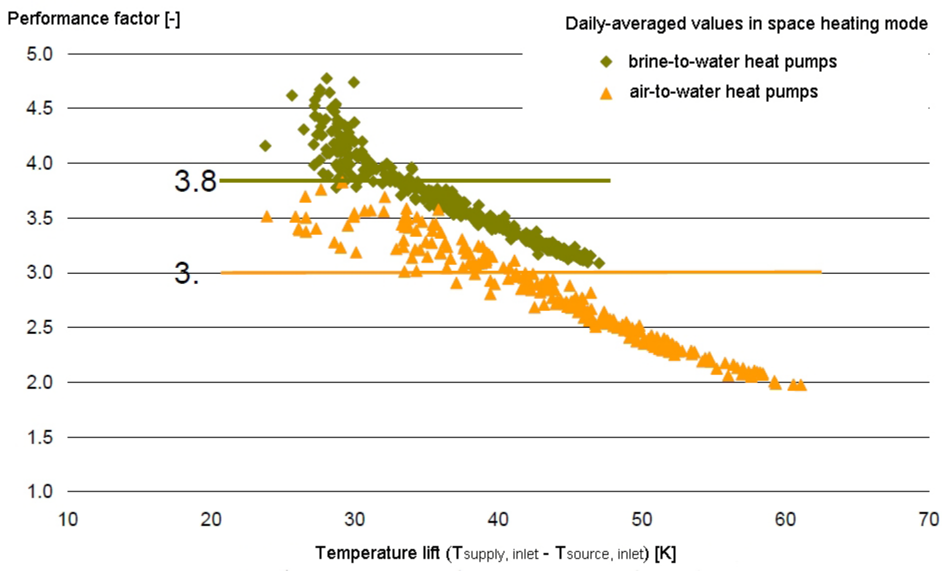


Fig. 80: Daily performance factors dependent on the temperature lift in existing buildings (Russ, 2008)

However, in the transitional season, when space heating supply temperatures decrease, the performance factor reaches the maximum at about 3.5, while in summer with higher DHW supply temperatures, minimum seasonal performance factors at 3.0 are reached. The average seasonal performance factor of the 34 systems is 3.3. DHW fractions are about 12% and electrical back-up heating is with 2% of the total electricity consumption negligible. The fraction of auxiliary energy for the heat source is low due to the higher energy needs in existing buildings.

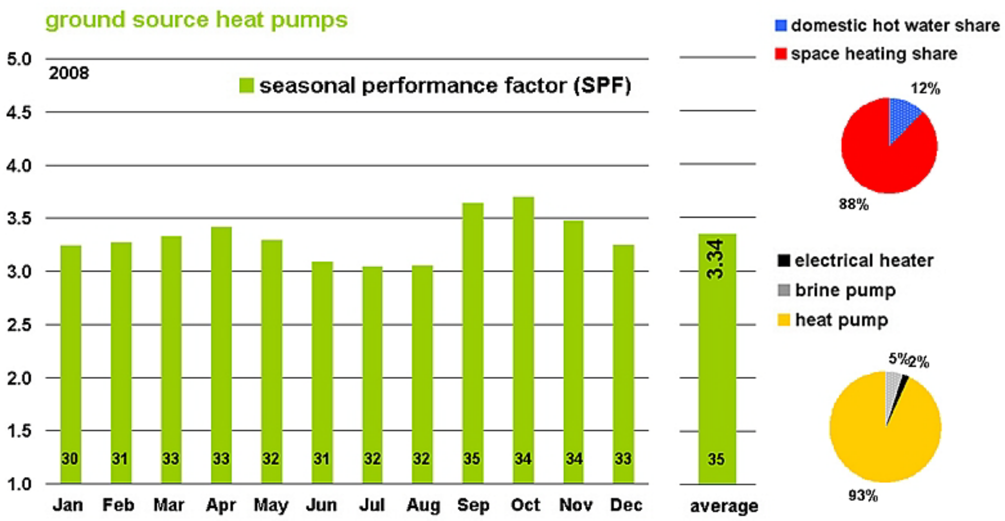


Fig. 81: Evolution of Seasonal Performance Factors for ground-source heat pumps (Miara, 2008)

Fig. 82 presents the monthly performance factors for the air-to-water heat pumps. In this case, the maximum performance factors with values up to 3.0 are reached in the summer period due to the significantly higher outdoor air source temperatures in this period. Winter values are around 2.5, and the average seasonal performance factor is evaluated to 2.6 for the 34 heat pump systems. The DHW fraction is with 14% in the same range as for the ground-source systems, and the back-up fraction is with 1% negligible, too. The auxiliary energy for the source systems is with 3% also low.

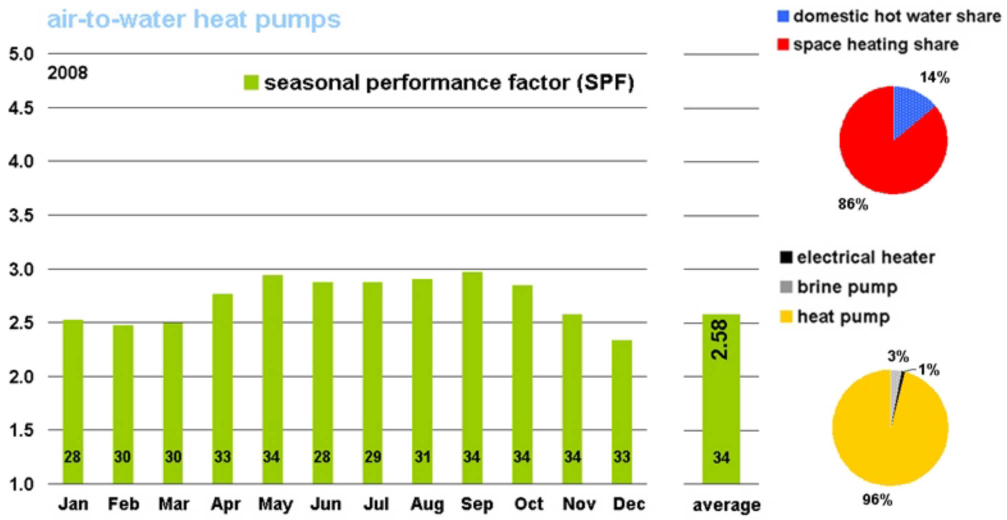


Fig. 82   Evolution of Seasonal Performance Factors for air-source heat pumps (Miara, 2008)

Fig. 83 depicts the comparison of the systems in existing buildings and the heat pumps in the low energy buildings in 2008. Concerning the seasonal performance factors the brine-to-water heat pumps reach the highest performance factors, even higher than the water-to-water systems. Brine-to-water systems have a seasonal performance factor of 3.8 in the low energy houses and are 0.5 higher than in existing buildings, which reflect the lower supply temperature in the low energy houses.

For the air-to-water systems the difference in 2008 is in the same range, but the three year average is 2.8, so the difference decreases to 0.2. A reason could be, that the potentials for a lower temperature lifts are not entirely implemented.

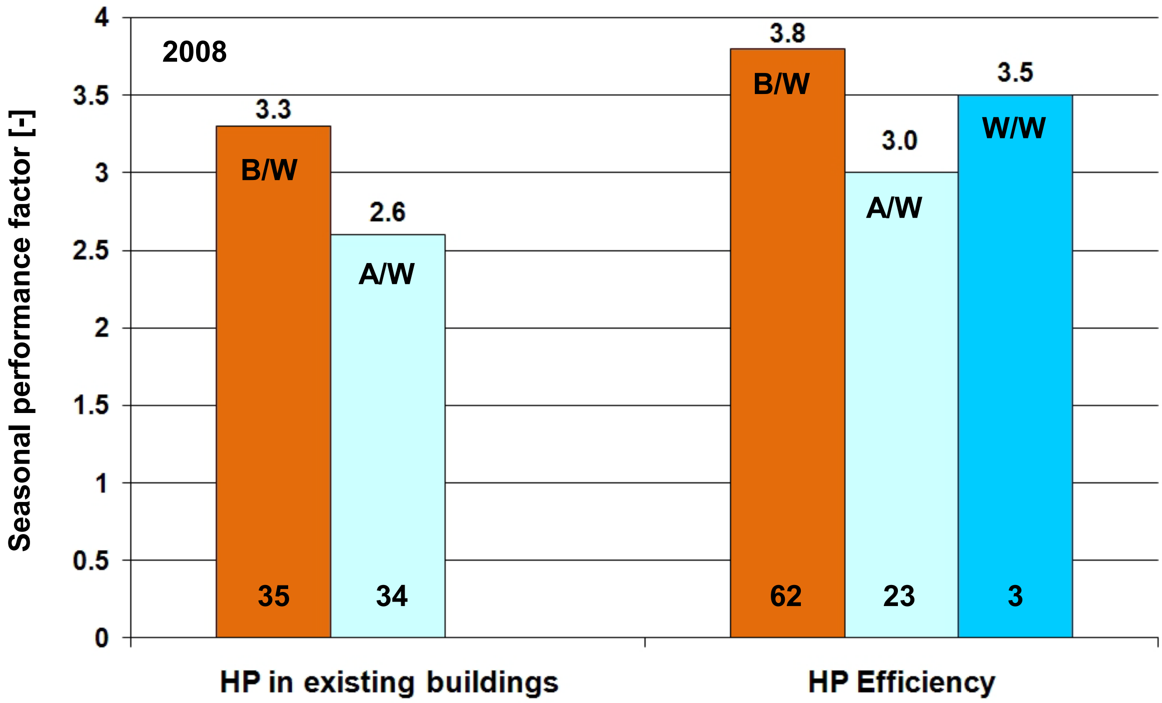


Fig. 83: Comparison of SPF for heat pumps installed in new and existing buildings in the German field tests

The SPF of the ground-water heat pumps in the HP-efficiency project is lower than in systems with borehole heat exchangers. However, only 3 ground-water heat pumps have been measured. One reason is the high fraction of electrical energy for the source which is with 15% more than double of the 6% average source auxiliary energy consumptions in the case of borehole heat exchangers. Another reason is the partly relatively high source temperatures due to the design of the boreholes systems which reach about 10 °C in some cases and thereby a similar temperature level as the ground-water systems, see chap. 3.3.

Tab. 13: Comparison of DHW fraction, Back-up fraction and source auxiliary energy fraction for the German HP in existing buildings and HP Efficiency field tests

Fractional Energy	**HP in existing buildings**		**HP Efficiency**		
	B/W	A/W	B/W	A/W	W/W
Back-up fraction	2%	1%	2%	2%	2%
Auxiliary fraction source	5%	3%	6%	7%	15%
DHW fraction	14%	12%	22%	28%	18%

Tab. 13 summarises the fractions of back-up energy use, auxiliary energy expenditure for the source system and DHW.
As can be expected, the DHW fraction in existing buildings is lower than in low energy buildings due to the higher space heating energy needs.
Fig. 84 presents the DHW shares of the "HP-Efficiency" and "HP in existing buildings" in more detail for the years 2008 and 2009, where the blue share represents the average DHW share for each month. It confirms the higher share of DHW for the HP efficiency project which is typical for the low energy houses with lower space heating energy needs due to the better insulation standard. The shares for space heating energy in the summer months are influenced by different phenomena such as a certain constellation in systems with integrated combined storage units which lead to unclear temperature layering and to space heating energy registrations which is in reality used for DHW. Thus, the actual DHW shares may even be higher than depicted in Fig. 84.

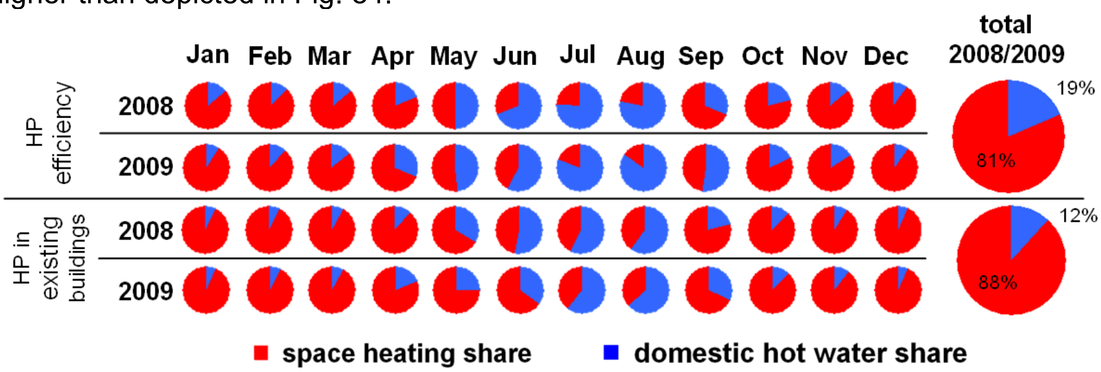


Fig. 84: Comparison of the DHW and space heating shares in the two monitoring projects "HP-Efficiency" and "HP-in existing buildings" (Günther and Miara, 2010)

6.2 Environmental Impact

The environmental impact of electrically-driven heat pumps in comparison to boilers using natural gas or fuel oil can be evaluated based on total energy expenditure expressed as primary energy or based on CO_2-equivalent emissions relevant for an assessment of climate change impact. The Best Practice Sheets listed in Appendix B contain an evaluation of the environmental impact of the single heat pump systems compared to a gas or oil boiler.
The comparison to other techniques requires a fundamental data base and clear boundaries of balancing. Fig. 85 shows data based on the software GEMIS 4.5 (2005) used in Germany,

which provides a suitable data base, enabling the consideration of the entire process-chain of different applications of energy transformation.
The figure shows an example of the break even point of SPF values based on primary energy factors (left) and CO_2-equivalent emissions (right) based on the data base GEMIS 4.5 (2005). For instance, based on the primary energy factor of 2.46 for electricity for the EU 27 countries and natural gas of 1.12, a minimum seasonal performance factor of 2.2 of a heat pump would be required for an equal primary energy evaluation. Concerning the same CO_2-equivalent emissions, a minimum SPF compared to natural gas of 2.0 or 0.8 compared to black coal would be required.

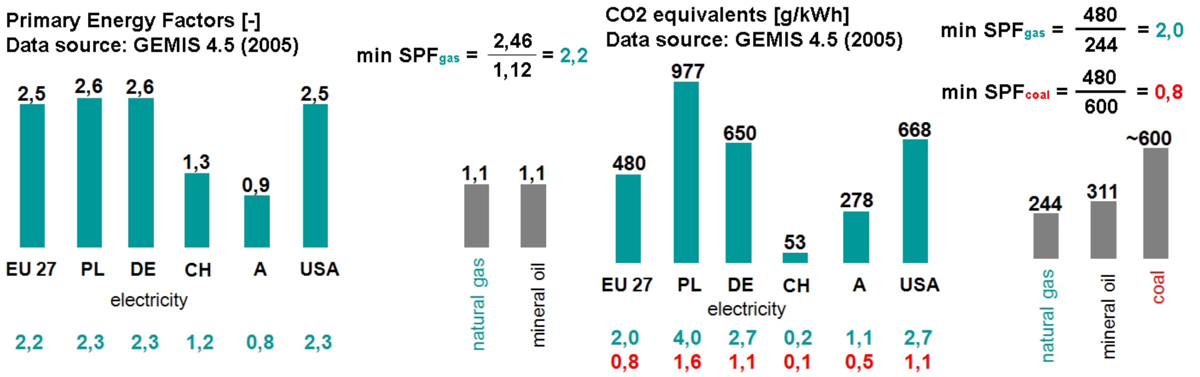


Fig. 85: Examples for minimum required SPF values in comparison to natural gas, mineral oil and coal based on primary energy factors (left) and CO_2 equivalent emissions (right) of different countries of the GEMIS 4.5 data base (Miara, 2009)

Fig. 85 also points out, that primary energy factors and CO_2-equivalent emissions strongly depend on the electricity production of the respective country. Furthermore, this figure shows the required minimum SPF values. Moreover, different data bases may be relevant for the respective countries. For instance in Switzerland, factors are defined in SIA 2031 (2009), which is based on the Ecoinvent data base (http://www.ecoinvent.ch) and defines a primary energy factor for electricity of 2.97 and 1.15 for natural gas, resulting in a minimum required SPF values of about 2.6 for primary energy equality. As a consequence, the used factors for the calculation have to be indicated, as well, since otherwise, the results for primary energy and CO_2-eq-emissions cannot be interpreted or compared.

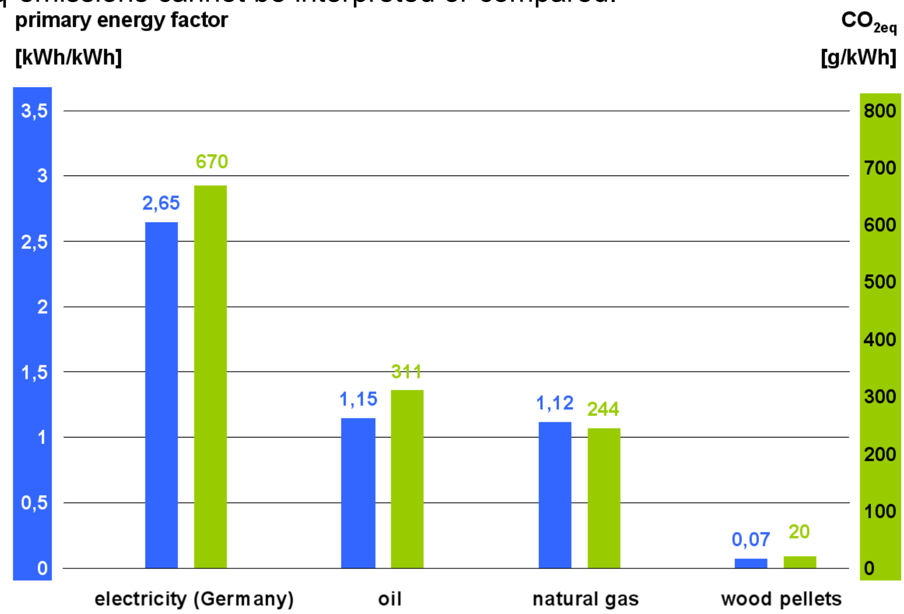


Fig. 86: Primary energy and CO_2-eq-emission factors used in Germany (Günther and Miara, 2010)

Fig. 86 gives more detailed primary energy factors and CO_2-eq-emissions of different energy carriers in Germany. As can be seen, values of oil are slightly higher, compared to the measurements of gas, whereas the impact of wood pellets with its characteristics of renewable energy sources is marginal. Fig. 87 depicts a respective comparison of the average seasonal performance factors evaluated of the HP-Efficiency field monitoring in comparison to the different heat generators and their individual impact on the environment. Air-source heat pumps of an average SPF of 2.8 cause a 28% higher consumption of primary energy than a ground-source heat pump with an average SPF of 3.9.
This is due to direct proportionality between the SPFs and the values of environmental impact. Nevertheless, even for air-source heat pumps a lower value of primary energy expenditure is attained compared to the conventional heating systems, which are using fuel oil or natural gas. For instance, the largest difference occurs for the comparison of ground source heat pumps to oil-condensing boilers, which have 88% higher primary energy expenditure factor and about 100% higher emissions of climate relevant gases. In this comparison, even the worse type of heat pumps outperforms the best type of conventional fuel system (primary energy difference +31%, CO_2 equivalent +13%).
Based on the German factors, however, wood pellet heating systems make the difference. Their low share of included fossil fuels in the process chain leads to the lowest impact on the environment in terms of primary energy factor and CO_2 equivalents with values of 0.08 and 23 g/kWh.

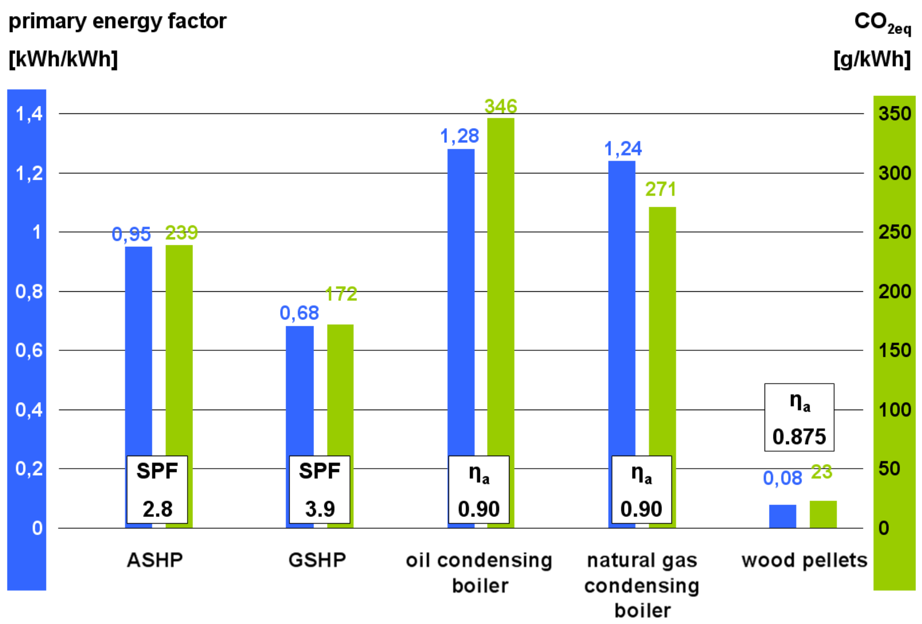
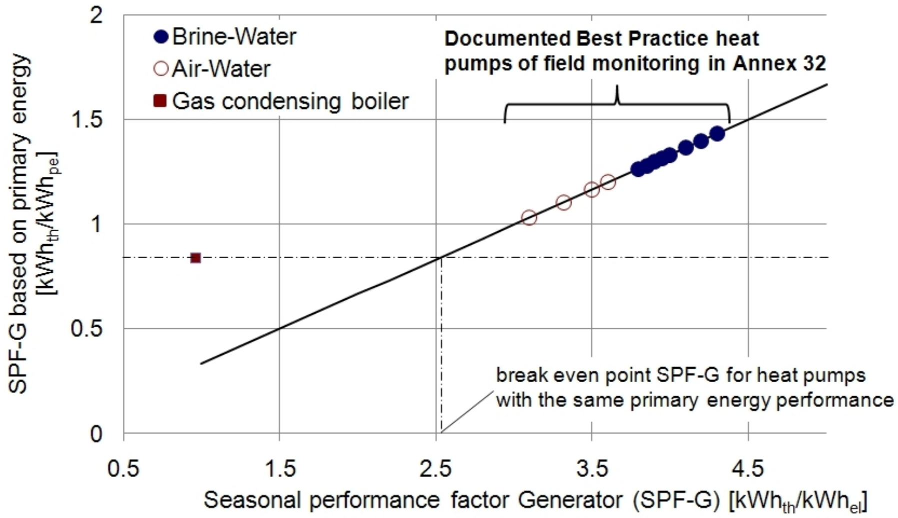


Fig. 87: Primary energy expenditure factor and CO_2-eq-emission for different heat generator and respective SPF or efficiencies (Günther and Miara, 2010)

However, this is only the evaluation for Germany with low primary energy factors for wood pellets. For example in Switzerland, a primary energy factor of 1.22 and a CO_2 equivalent emissions of 36 g/kWh are used for wood pellets and 2.97 and 155 g/kWh for electricity. These values lead to break even points based on a pellet boiler efficiency of η = 0.875 of a seasonal performance of 2.13 for primary energy equality and 3.76 for equal CO_2-eq.-emissions, which can be achieved by adequate designed ground-source heat pumps.
Summarising, Fig. 88 shows the SPF based on primary energy, which has been calculated as quotient of the SPF and the primary energy factor, of the air-to-water (red hollow dots) and brine-to-water (blue bold dots) heat pumps documented in Best Practice Sheets listed in Appendix A.

These SPF-values are compared to a gas condensing boiler with an efficiency of 0.96 according to the average efficiency of a field test of 59 condensing boilers in Germany reported in Wolff et al. (2004).
As can be seen, of the seasonal performance factors of the generator system based on primary energy of all documented Best Practice systems are significantly higher than the primary energy efficiency of the condensing gas boiler and thus contribute to primary energy savings.



Based on primary energy factor according to SIA 2031[1] (2009): natural gas: 1.15; Electricity: 3.0
Performance factor gas-condensing boiler: 0.96 (Average field test 59 gas-condensing boiler, based on H_u[2])

[1] SIA Merkblatt 2031 (2009), Energieausweis für Gebäude, SIA, Zurich [2] Wolff, D., Teuber, P., Budde, J., Jagnow, K. (2004). Felduntersuchung: Betriebsverhalten von Heizanlagen mit Gas-Brennwertkesseln, DBU Abschlussbericht, FH Wolfenbüttel

Fig. 88: Comparison of the Seasonal Performance Factor based on primary energy of documented Best Practice Systems to a condensing boiler for primary energy factors used in Switzerland (natural gas 1.15, electricity 2.97)

7 RECOMMENDATIONS FOR SYSTEM DESIGN

In this chapter weak points in heat pump installations found in the field monitoring projects and recommendations for the system layout and design, control, installation and commissioning are summarised. Results of the field monitoring in the frame of Annex 32 thereby partly confirm results of former field monitoring projects, in particular the FAWA field test (Erb, Hubacher and Ehrbar, 2004). In connection to the FAWA project, the STASCH (standard systems configuration) design guidelines (Gabathuler et. al., 2002) were developed, which are referenced, as well. Some malfunctions found in the field tests are due to installation problems. Therefore, the initiatives like the certified heat pump installer of the EHPA (http://www.ehpa.org/european-certified-hp-installer) and labels for heat pump and borehole heat exchanger installations as the EHPA quality label (http://www.ehpa.org/ehpa-quality-label) as well as the monitoring activities and performance requirements by various subsidy programs (e.g. see chap. 2.1.7) can be considered as useful steps towards better performing heat pump installations.
This chapter shortly summarises the experiences gathered in the mentioned field test projects.

7.1 Heat pump functionality and identification of weak points

Several malfunctions have been documented in the field monitoring projects which are described in this chapter in order to draw conclusions on improvements concerning design, control and commissioning, which is discussed in the following chapters.

Heat pump

- In general, heat pumps are robust heat generators with a good functionality.
- Most of the field tests described in chapter 2.1 confirm generally good user contentment with their systems, which was also confirmed in the HP Efficiency field monitoring project.
- Heat pumps have a good availability and little maintenance requirements, as proven in the Best Practice system follow-up of the FAWA field test as described in chapter 2.1.2.
- In most of the field tests no problems with the heat pump operation itself have been documented.

Circulation pumps

- In the HP Efficiency field test well pumps for water source heat pumps consume 15% of the total electrical energy usage. There seems to be a discrepancy between the capacity of the heat pump and the capacity of the well pump for the water-to-water systems with low capacity. According to Günther and Miara (2010) it seems that for the examined water-to-water heat pumps with low capacity the market is not able to provide appropriate well pumps, yet. Brine pumps used for ground-source heat pumps have a share in the range of only 6% in this regard.
- Over-dimensioned source pumps in case of B/W-heat pump auxiliary consumption were encountered in the FAWA field test

Storages

- a sub-optimal integration of combi-storages and solar systems was found in the HP Effiiciency field test. In extreme cases the part of the storage, which serves as heating buffer was heated-up to the DHW temperatures.

Valves

- Not completely closing three-way-valves as well as

- missing or wrong positioned check-valves were found. These may lead to continuous discharge of the DHW storage during the heating mode and to unnecessary energy loss.
- Insufficient thermal insulation of the fittings and pipe connections led to heat losses and higher temperature requirements for the heat pump to reach the set-point temperature.

Back-up heating

- It was found in the HP Efficiency field test that although the fraction of electrical back-up use is low in total, a lot of unnecessary use was noticed.
- Sometimes, there was operation of electrical heaters without any reason, in particular in brine-to-water heat pump systems, which usually provide enough capacity without a back-up heater.

Control

- It has been evaluated that the adjustment of the heating curve was not accomplished in the best way. It has been noticed that inlet temperatures were set higher than the design inlet temperatures.
- Some ground-source heat pumps have wrongly adjusted brine pumps.
- Partly, the operation of the heat pump was not corresponding to circulation pumps of the distributions system, and even continuously running circulation pumps of the distribution system during the entire heating period have been detected. Also a constantly running heating circuit pump in summertime was detected.

Information to the building owners

- It was observed that residents had very different attitudes towards their heating system. A few of them did not even know about the function of heat pumps, while others were interested in all aspects and tried to improve the performance as much as possible. The majority can be assigned to the first group which is mainly due of missing information about the heat pump and its characteristics.

7.2 Design of the systems and system integration

This chapter summarises the design recommendations of the system components for the most common heat pump types investigated in the field monitoring projects. Results are mainly based on the FAWA field monitoring project (Erb, Hubacher and Ehrbar, 2004) and the developed design guidelines STASCH (Gabathuler et al., 2002) as well as recommendations derived in the German field test HP-Efficiency (Günther and Miara, 2010). Further discussion of design issues for integrated heat pump systems are found in chap. 5.

- **Integration of the DHW operation is advantageous**

As shown in chap. 2.1.1 results of the FAWA field test confirm that an integration of the DHW operation by the heat pump is advantageous compared to direct electrical DHW heating. In the HP-Efficiency field test only integrated systems were considered which delivered good overall performance results. Only few reasons exist for not integrating the DHW operation according to the STASCH guidelines:

  - Explicit customer wish for decentral DHW heating
  - the DHW demand is too low for a coupling to the heat pump
  - a retrofit of an existing DHW heating is not worthwhile

The most common integration in Europe is done by a parallel DHW storage, where the heat pump is switched from space heating to DHW operation (alternate mode). Design and control recommendations for integration of the DHW operation in alternate mode given in FAWA include

- Simple hydronic system with internal heat exchanger in the DHW system yielded the best performance of the systems installed in Switzerland.
- The storage should be designed rather small (capacity to store the demand of 1 day).
- A demand-controlled storage loading, e.g. by a hysteresis control according to the storage temperatures should be applied. If a time dependent control is chosen, only a short necessary time interval should be dedicated to the DHW operation.

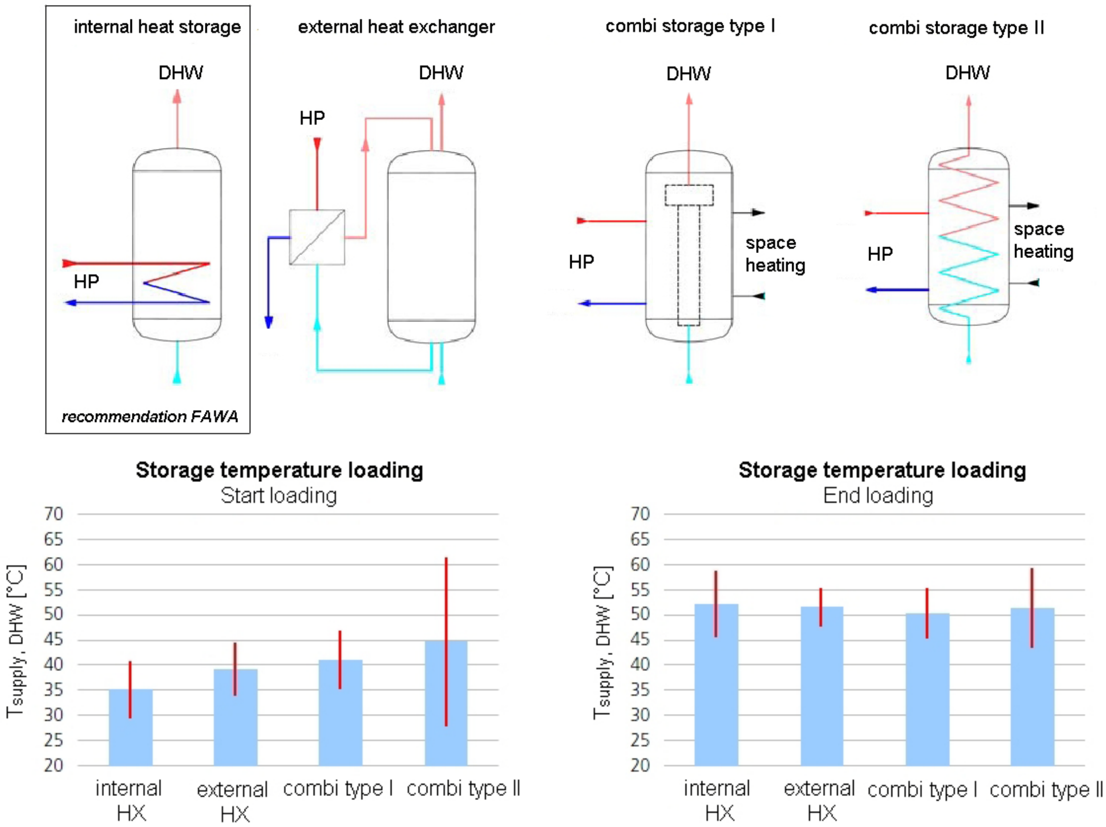


Fig. 89: Results for the storage integration in the FAWA field test (blue: average temperatures, red: range of temperatures found) (Erb et al., 2004)

- **Design should be made to the lowest possible temperature lift**

As stated before the main impact on heat pump performance is the temperature lift between the source and the sink side, so the design should try to minimize the temperature lift whereever possible. This can be achieved by an adequate design of source and sink system. Source systems with a high temperature level in the heating period should be chosen, but cost and availability considerations have to be taken into account, as well.
On the other hand a low temperature lift can be reached by a design of the space heating emission system to low supply temperature (low-ex design). Average design supply temperature found in the FAWA field test still reach high values of 41 °C which was only 5 K lower than in the retrofit case. An average decrease of the supply temperature by 1 °C increases the SPF by about 1.5%. Also in the HP Efficiency field test, high average temperatures over the heating period of ≈36°C have been found.
Floor heating systems designed to supply temperatures below 30°C have the advantage of

- Maximising the heat pump COP and thereby the SPF.
- Using the self-regulation effect.
- The possibility of simplifying the hydronic configuration by avoiding the necessity of thermostatic valves due to the self-regulation effect (possible cost savings).

According to the actual cantonal regulations MuKEn (2008) in Switzerland, floor heating systems have to be design to supply temperature lower than 35 °C.

- **The system configuration should by as simple as possible**

In the FAWA and HP Efficiency field test it was found that simple systems with few components tend to perform better than more complex system configurations with many pumps, valves and storages. Storages and heat exchangers may increase the required temperature level to be produced by the heat pump due to necessary temperature differences for the heat exchange or due to heat losses. Therefore, system integration should be considered thoroughly and the necessity of intermediate cycles and storages should be checked.

- **Heating buffer storages should by avoided whereever possible**

Concerning the integration of a buffer storage it is recommended to avoid heating buffer storages in floor heating systems according to the FAWA field test, since they raise the cost of the system, need additional installation space and make the system more complex. Thus, buffer storage should only be installed when absolutely required. Avoidance of buffer storages is possible

- in floor heating systems since the capacity is large enough to prevent cyclic operation of the heat pump (FAWA, STASCH)
- in system with less than 50% of the heat emission by radiators which still guarantees enough capacity (STASCH)
- with a maximum of 40% of the heat emission system equipped with thermostatic valves (STASCH)
- with only one heat emission system connected to the heat pump (STASCH)

Buffer storages may be required to prevent cyclic operation of the heat pump

- in heating systems with low thermal inertia like radiators and convectors (STASCH)
- in case of heat emission systems equipped to more than 40% with thermostatic valves (STASCH)
- in heating systems serving multiply distribution circuits (STASCH)

or

- for bridging the shut down periods of power supply by the utilities (HP Efficiency)
- for defrost operation of air-to-water heat pumps with cycle reversal to prevent extraction of heat from the building (HP Efficiency, Austrian field test)
- for bivalent systems with solar thermal collector or wood in space heating operation (FAWA)
- for DHW integration by a common heating buffer storage as "fresh water system" with instantaneous DHW preparation, which is a common system configuration in the Austrian field test.

- **Combi-storages should be installed only if absolutely required**

Both in the FAWA and the HP-Efficiency field test systems with combi-storage did not perform as well as expected. In the HP Efficiency field test it was evaluated that combi storages raise the temperature requirement and, if not installed properly, the space heating temperature may be as high as the required DHW temperatures. As result of the FAWA field test it is stated that the integration of combi-storages shall only be considered for bivalent systems with a second energy like wood or solar thermal energy.

- **Design of the heat pumps tends to be too large**

In the FAWA project 30% of the investigated heat pumps had a load factor below 50%. Thus, the capacity of the heat pump shall be chosen carefully and it is recommended not to include safety factors in the heat pump design, in particular for brine-to-water systems.

- **Correctly designed ground-source heat pumps do not need an electrical back-up heater**

In both the FAWA field test and the HP-Efficiency field test it was found that brine-to-water heat pumps should be designed monovalently. In this case, the installation of the direct electrical back-up heater is avoided. If an electrical back-up heater is included in the system installation, it should be deactivated. Back-up heating could be considered in cases in which drying-out of the building is required, i.e. in the first year after the finishing of the building, in order to avoid damage of the borehole heat exchanger due to excess heat extraction. But afterwards, it should be switched-off. However, the drying-out of the building could also be done by external mobile heaters. In the FAWA field test it was evaluated that also air-to-water heat pump in moderate climate conditions like the Swiss middleland can be operated monovalently. This is confirmed in the German field test where average back-up fractions are with 3% very low.

- **Design of ground-source systems according to the energy extraction**

In the FAWA field test it was found that brine-to-water heat pump systems are often overdimensioned which is a drawback regarding the cost. It was found that, if the borehole heat exchanger systems (BHX) are designed by a specific extraction power of the heat pump with the common value of 50 W/m, it is often too long, since heat pumps are overdimensioned. According to FAWA a recommended value for the design of a ground-coupled heat pump system is the annual heat energy to be extracted of 80 kWh/m according to the equation:

$$\text{length of borehole heatexchanger} = \frac{\text{heat energy demand} - \dfrac{\text{heat energy demand}}{\text{expected SPF (or average COP)}}}{80\ \text{kWh/(m} \cdot \text{a)}}$$

However, the design of the BHX strongly depends on the ground characteristics. If no detailed information on the ground properties is available, a considerable insecurity results concerning the design.
In case of guaranteed borehole temperature above 0 °C in long borehole heat exchangers a switch of the source fluid from brine to water can increase the performance as shown in Fig. 4 left.

- **Design of source pumps in borehole heat exchanger systems**

In the FAWA field test, installed source pumps were often overdimensioned which had a negative effect on the efficiency. It shall be designed according to the pressure drop to avoid overdimensioning. Moreover, highly efficient pumps (level A+) are stated to be economically feasible and paid back in short time.

7.3 Control of the system

Concerning the control of the system the following optimisation potentials were found in the field tests

- **Supply temperatures are often higher than required**

In both the FAWA and in the HP-Efficiency field test supply temperatures encountered in the real operation were higher than necessary according to the heating needs. Thus, a considerable optimisation potential exists regarding a correct setpoint of the heating curve. In the FAWA field test, it was found that control with additional room temperature information has further advantages compared to an only outdoor air temperature based control by a heating curve. A detailed briefing for the house owner may be very useful.

- **Reduction of auxiliary energy by reduced runtime of the pumps**

Instead of pumps running through the entire heating period, the pumps control should be improved by an intermittent pump operation, which could be achieved by periodical start-up in order to evaluate the return temperatures as criterion to start the heat pump.

7.4 Installation and commissioning of the system

Due to the experience during the measurement, the following recommendations regarding the installation and commissioning of the systems can be concluded.

- Missing or not continuous thermal insulation of piping and fitting to storages and valves increases the requirements for the supply temperature. Thus, it should be taken care to a continuous and thorough thermal insulation of the components during the installation or commissioning of the system, in particular in systems with combi-storages.
- A hydronic balance shall be accomplished.
- The loading strategies for storages should be checked regarding the control of supply temperature for heating systems, in particular in the case of combi-storages.
- In case of B/W-heat pumps direct electrical back-up heaters should be deactivated.
- The user of the system should be provided with information on the systems.

7.5 Design of integrated heat pumps with natural refrigerants

This chapter gives design recommendations for heat pump systems which cover the space heating and DHW function, in particular using a system layout for simultaneous space heating and DHW operation and natural refrigerants propane (R 290) and CO_2 (R 744) according to Justo Alonso and Stene (2010)

7.5.1 Introduction

Integrated heat pump systems cover the entire space heating and domestic hot water (DHW) heating demand in a residence. A heat pump system with shuttle-valve operation has alternating operation for DHW heating and space heating, i.e. the system only operates in heating mode, while simultaneous space heating and DHW heating can be achieved e.g. by desuperheaters or cascade heat pump layouts.

The annual energy demand for DHW heating in low-energy and passive houses may constitute about 50% and higher of the total annual heat demand of the residence. The temperature requirement for DHW may reach 60 °C in order to avoid growth of legionella bacteria while the temperature requirement for hydronic heat distribution systems for space heating in low-energy and passive houses typically range from 30 °C to 50 °C depending on the emission system type.

7.5.2 General Design Recommendations

Due to the low total annual heating demand in low-energy and ultra-low energy houses, the major design challenge is to obtain high energy efficiency at acceptable investment and installation costs. In order to achieve a high Seasonal Performance Factor (SPF) and long lifetime, the heat pump system should have the following characteristics:

- Heating capacity – The heat pump should be designed to cover the entire heat demand of the house, i.e. the annual DHW heating demand and space heating demand.
- DHE heating – The heat pump should cover the DHW heating demand at the required temperature level without reheating by a low-efficiency supplementary heating systems, e.g. electric immersion heaters. Moreover, the heat pump unit and the DHW storage tank(s) should be optimized for both small and large DHW heating demands, i.e. both small and large DHW draw-offs.
- Coefficient of Performance – The heat pump system should be designed to achieve high COP at all operating conditions, and special attention should be paid to the DHW heating mode and combined heating mode due to a relatively high DHW temperature level. It is also important to minimize energy use for integrated pumps, utilize energy-efficient fan control and evaporator defrosting for air-to-water systems etc.
- Reliable operation – It should be focused at moderate start/stops for the compressor during intermittent operation, maximum limit for the discharge gas temperature etc.
- Working fluid – In order to be an environmentally benign technology, integrated heat pumps should use a working fluid with zero ODP-value, zero or negligible GWP-value and excellent thermophysical properties, e.g. carbon dioxide, CO_2 (R744) or hydrocarbons such as propane (R290) or propylene (R1270).

7.5.3 Subcritical Heat Pump Cycle

- A possible design of a high-efficiency subcritical heat pump cycle using a negligible-GWP working fluid (propane, R290) is given in Fig. 90 left. In order to achieve a high Seasonal Performance Factor for a single-stage subcritical heat pump cycle for combined space heating and DHW heating, the following general design criteria should be met (Zijdemans, 2007).
- Suction gas heat exchanger (internal heat exchanger) – The suction gas heat exchanger transfers heat from the liquid high-pressure working fluid to the low-pressure suction gas, and increases the discharge gas temperature and with that the superheat.
- Desuperheater for reheating the DHW to the desired temperature level – The desuperheater, which can be a separate counter-flow heat exchanger or a heat exchanger coil in the DHW storage tank, utilizes the hot discharge gas from the compressor for DHW heating at 60-70 °C. When the desuperheater is combined with a suction gas heat exchanger, about 30-40% of the total heating capacity of the heat pump unit can be supplied by cooling of discharge gas at a higher temperature level than the condensation temperature. This means that high-temperature DHW can be provided at a moderate condensation temperature (high COP).
- Condenser heat for preheating of DHW – When condenser heat is used for preheating of DHW and hot discharge gas is used for reheating the DHW in a desuperheater, the heat pump can cover the entire DHW heating demand even in periods when there is no space heating demand.
- Accumulation tank for the space heating system – An accumulation tank for thermal storage of low-temperature heat from the condenser will be able to cover the space heating demand and preheat DHW even during periods with a considerable draw-off from the DHW tank.

- Component design – Due to the low heating capacity of integrated heat pumps for low and ultra-low energy houses, it is of crucial importance to find/design high-quality and optimized components which have a sufficiently low capacity including compressor, evaporator, condenser, desuperheater, suction gas heat exchanger and expansion device.

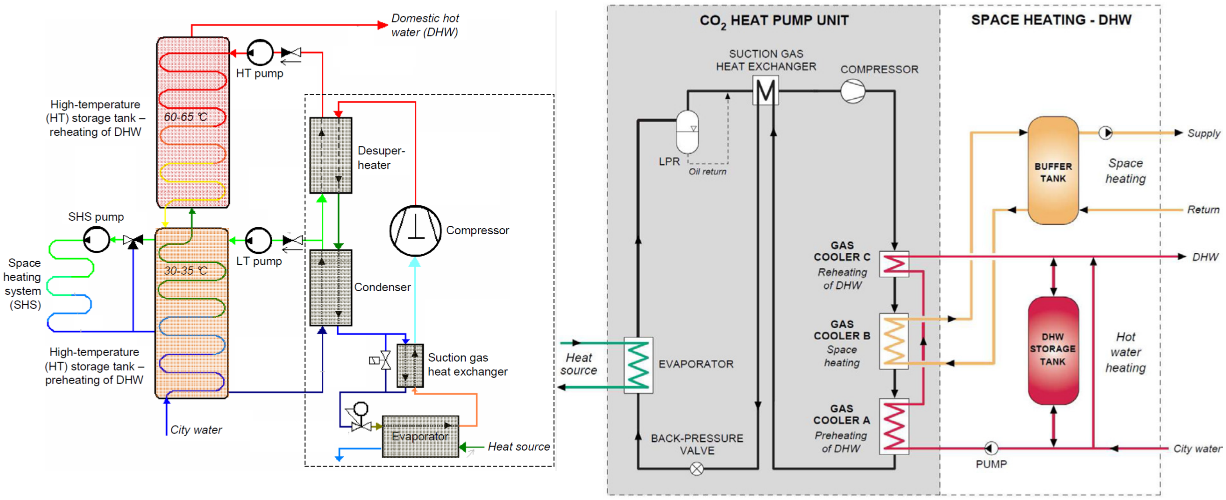


Fig. 90: Examples of principle system configurations of a subcritical R290 (left, Zijdemans, 2007) and transcritical R744 (right, Stene 2008) integrated heat pump system.

Further results on the prototype subcritical propane heat pump system and field test results are given in the prototype report of Annex 32 and a System Concept Sheet.

7.5.4 Transcritical CO_2 Heat Pump Cycle

A possible design of a high-efficiency transcritical heat pump cycle using a working fluid (CO_2, R744) of GWP = 1 is shown in Fig. 90 right. In order to achieve a high Seasonal Performance Factor (SPF) for single-stage integrated CO_2 heat pump systems the following general design criteria should be met (Stene, 2004/2008).

- Low-temperature heat distribution system – Utilize a low-temperature heat distribution for space heating with a maximum return temperature of 30-35 °C. The lower the return temperature in the heat distribution system, the lower the CO_2 outlet temperature from the gas cooler and the higher the COP of the CO_2 heat pump.
- Accumulator tank for the space heating system – An accumulator tank will act as a thermal buffer between the CO_2 heat pump and the space heating load and increase system COP.
- Bipartite or tripartite gas cooler design – Design the heat pump unit with more than one gas cooler in order to utilize the considerable temperature glide of the CO_2 in the gas cooler space heating and DHW heating. The gas cooler(s) should always have counter-flow operation in order to obtain maximum temperature decrease of the CO_2 before the expansion (throttling) valve.
- DHW storage tank design – Use a DHW storage tank design that minimizes mixing of hot and cold water and minimizes heat conduction between water volumes at different temperature levels during tapping of DHW and thermal charging of the tank. The application of diffusers at the tank inlets will reduce inlet water velocity and minimize mixing of hot and cold water. Moreover, a storage tank with a small diameter-to-height ratio will minimize heat conduction between hot and cold water volumes inside the storage tank.

- Component design – Due to the low heating capacity of integrated heat pumps for low-energy and passive houses, it is of crucial importance to find/design high-quality and optimized components which have sufficiently low capacity including compressor, evaporator, gas cooler (single type, bipartite, tripartite), suction gas heat exchanger, low-pressure receiver (LPR) and expansion device (back-pressure valve). Optimized design of the oil return system is also of crucial importance in order to achieve trouble-free operation and long lifetime.

8 CONCLUSIONS

Some conclusions of the results of field monitoring project accomplished within Annex 32 and existing field monitoring results are drawn in this chapter.

Performance factors

The different field monitoring of heat pump systems which took place during the last 10 years show a different picture of the performance of heat pumps in real operation. Both excellent performance above 5 as well as poor performance slightly above 1 has been registered. However, these numbers may lead to misunderstandings, since different system boundary are used and are mainly deliberately chosen by the executing organization. This situation shall be overcome with European project SEPEMO and a new project in the Heat Pump Programme of the IEA by a comparison of existing field test and the establishment of a data base of heat pump field test results based on defined system boundaries.
Results of field tests of heat pumps in low energy houses in IEA HPP Annex 32 confirm that the measured heat pumps achieve good performance factors of about 3 for air-to-water heat pumps and in the range of 4 for brine-to-water heat pumps and thereby contribute to primary energy savings and reduction of CO_2-eq.-emissions compared to gas condensing boilers.

Operational experience

Despite the good performance factors in field tests within Annex 32 also malfunctions have been detected. A comparison to systems installed in existing buildings confirm that in particular for air-to-water heat pumps the difference in seasonal performance is not large, but just 0.2 SPF points. This means that also in low energy houses, opportunities of low temperature heat emission systems with low supply temperature are not fully implemented in practical operation, what is confirmed by an average supply temperature of 36 °C for floor heating systems with air-source heat pumps in low energy houses.
Based on the experience of malfunction it would be useful to have an easy-to-use possibility to check the system behaviour. This could for example be realised by a smart metering approach, where the user can easily access e.g. the consumption of the back-up heating and the actual performance factor of the heat pump.

Ground-coupled passive cooling in residential buildings

Passive cooling in residential buildings is mainly applied in the luxury housing market segment to date. However, the results of the simulations show that a passive cooling function can be easily integrated and even retrofitted in ground-source heat pump systems. Additional costs for the extension of the cooling function are moderate, as well.
Simulated seasonal performance factors are in the range of 10-25. Reasons for lower seasonal performance factors in the field monitoring projects of around 8 are both due to a higher auxiliary consumptions owing to less efficient pumps and a lower extracted cooling energy. Monitoring results prove that passive cooling is feasible in residential buildings, as well, where the indoor air temperature can be decreased by 2-4 K. Performance factors can still be increased.

9 ACKNOWLEDGEMENT

The operating agent as editor of this report is very grateful for the valuable contributions of all participants of the IEA HPP Annex 32 and for the constructive discussion and co-operation. It has to be emphasised that the Annex 32 is a co-operative research project and results are taken from national contributions.

The operating agent would like to thank the Swiss Office of Energy (SFOE) for funding and supporting the project, in particular the research programme manager Prof. Dr. Thomas Kopp for advising in the Annex 32 project and the Swiss national project within the Annex 32.

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APPENDIX

The Appendix

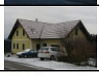
- **A: List of Best Practice Sheets and System Concepts Sheets**

- **B: List of detailed results of the field monitoring in Austria (AIT)**

A LIST OF BEST PRACTICE SHEETS ON THE WEBSITE

Single systems of the field test accomplished in Annex 32, which have shown a good operational behaviour and a high seasonal performance factor, have been documented as Best practice systems for the download on the website.
Tab. A 1 gives and overview of the field monitoring systems on the website.

Tab. A 1: Overview of documented Best Practice systems in Annex 32

Country	Object	Description of Best-Practice heat pump system
AT		Retrofitted SFH with A/W-HP for radiator heating and DHW (Meiseldorf)
AT		SFH with B/W-HP for floor heating and fresh water system (Felling)
AT		SFH with A/W-HP for floor heating and DHW (Rutzenmoos)
AT		SFH with W/W-HP with floor heating and fresh water system (St. Peter)
AT		ground-source compact unit in passive house incl. passive SC (Judendorf)
AT		ground-source compact unit in passive house incl. passive SC (Hitzendorf)
CH		ventilation compact unit for SH, DHW and Vin MINERGIE®-Haus (Gelterkinden)
CH		B/W-HP for SH, DHW and passive SC in MINERGIE-P® MFH (Basel)
CH		B/W-HP for SH, DHW and passive SC in MINERGIE® SFH (Muolen)
DE		B/W-HP with borehole heat exchanger for SH and DHW in SFH (Köngen)
DE		B/W-HP with horizontal collector and energy fence for SH/DHW (Spechbach)
DE		S/W-WP with borehole heat exchanger for SH and DHW in SFH (Gaggenau)
DE		Outdoor air/W-HP for floor heating and exhaust air-HP for DHW (Buchenbach)
DE		B/W-HP with borehole heat exchanger for SH and DHW (Neuffen)
JP		Inverter-controlled B/W HP, A/W-Eco-cute and V in SFH (Naganuma, Hokkaido)
JP		Inverter-controlled B/W-HP for SH, DHW, solar collector, V in SFH (Sapporo)
US		16 B/W HP in SFH (Hope Crossing, Oklahoma City Habitat for Humanity)

Single systems in prototype status or upcoming systems concepts, which are expected to be introduced in the market soon, have been documented in System Concept Sheets for download on the website.
Tab. A 2 gives and overview of the field monitoring systems on the website.

Tab. A 2: Overview of documented System Concept Sheets in Annex 32

Country	Object	Description of heat pump system concept
CA		Concept EcoTerra EQuilibrium NZEB with BIPV/T and B/W-HP (Eastman)
CA		Concept Alstonvale NZEB with BIPV/T and L/W- HP (Hudson, Québec)
NL		Demand controlled V with B/W-HP for SH, DHW and SC (Ypenburg, Den Haag)
NL		B/W-Heat pumps for SFH with solar collectors (Delfgauw, Delft)
NL		B/W Heat pump for LE-appartments (De Tas, Biddinghuizen)
NO		Prototype 3 kW W/W propane-HP for SH, DHW in passive house (Flekkefjord)
US		B/A and A/A IHP-prototype for SH, DHW, V,SC , de-/humidification in NZEB (Oak Ridge)

B DETAILS ON THE FIELD MONITORING PROJECTS

B.1 Statistical evaluation of the German field test

Tab. B 1: Stastics of the field test HP Efficiency in Germany (Miara, Günther, 2010)

Heat source	2008												
	SPF max [-]	SPF min [-]	Back-up share max [%]	Back-up share min [%]	Brine pump/fan share max [%]	Brine pump/fan share average [%]	Brine pump/fan share min [%]	DHW share max [%]	DHW share min [%]	max inlet temperature SH [°C]	average inlet temperature SH [°C]	min inlet temperature SH [°C]	average inlet temperature DHW [°C]
ground	5,3	3,0	17	0	11	6	2	43	2	47,1	36,0	28,6	51,4
horizontal collectors	4,3	3,2	14	0	10	6	2	21	5	45,6	35,4	30,4	51,6
boreholes	5,3	3,0	17	0	11	6	1	43	2	47,1	36,6	28,6	51,2
outside air	3,4	2,5	16	0	11	7	2	43	3	44,2	36,7	30,6	48,7
ground water	3,7	3,4	1	0	18	16	14	16	6	47,3	41,4	36,2	52,3

Heat source	2009												
	SPF max [-]	SPF min [-]	Back-up share max [%]	Back-up share min [%]	Brine pump/fan share max [%]	Brine pump/fan share average [%]	Brine pump/fan share min [%]	DHW share max [%]	DHW share min [%]	max inlet temperature SH [°C]	average inlet temperature SH [°C]	min inlet temperature SH [°C]	average inlet temperature DHW [°C]
ground	5,5	2,7	18	0	11	6	1	38	2	45,1	35,6	26,8	51,3
horizontal collectors	4,5	3,1	18	0	10	6	2	21	5	45,0	34,8	30,1	50,9
boreholes	5,5	2,7	13	0	11	5	1	38	2	45,1	36,2	26,8	51,4
outside air	3,5	2,3	11	0	10	6	1	37	3	45,3	36,9	29,9	50,5
ground water	5,1	3,3	4	0	18	15	15	16	4	48,1	40,7	29,9	52,6

B.2 Detailed results of the Austrian field test

The Austrian field test consists of 9 standard heat pump systems for space heating or for combined alternate space heating and DHW and of 2 ground-coupled heat pump compact units which cover the functions space heating, DHW, ventilation and passive ground coupled space cooling. Tab. B 2 gives details on the results of the 9 standard systems and Fig. B 1 to Fig. B 9. Tab. B 3 shows the results for the compact units and Fig. B 10 to Fig. B 11 give details on the system configuration.

Tab. B 2: Overview of the field test results of standard heat pump system for space heating and DHW in the Austrian project at AIT (Zottl et al. 2010)

	No.	-	**No. 1: B/W**	**No. 2: B/W**	**No. 3: B/W**	**No. 4: B/W**	**No. 5: W/W**	**No. 6: A/W**	**No. 7: A/W**	**No. 8: DX/W**	**No. 9: A/W**
Object	Location	-	Weinzierl	Lengenfeld	**Felling**	**Behamberg**	**St. Peter/Au**	Bad Vöslau	Klein Meisel-dorf	Ohlsdorf	**Rutzenmoos**
	Heat load	kW	12,7	10,6	9,3	7,0	16,2	9,3	23,0	8,0	6,7
	Heating area	m²	295	210	272	190	264	160	500	189	309
	Specific heat load	W/m²	43	50	34	37	61	58	46	42	22
	Specific heat energy demand	kWh/m² a				45	50				28
	Inhabitants	-	4	5	4	4	6	4	5	4	2
	Building construction	-	medium	medium	medium	light	medium	medium	medium	medium	medium
	Commissioning	-	2006	2004	2004	2004	2004	2004	2007	2001	2006
Heat pump	Heat pump capacity	kW	11,8	11,8	11,8	8,0	19,4	10,3	33,0	15,1	8,3
	A/W	-						x	x		x
	B/W	-	x	x	x	x					
	W/W	-					x				
	DX/W	-								x	
Heat source	Horizontal collector	-	x			x				x	
	Vertical collector	-		x	x						
Heat sink	Floor heating	m²	295,0	207,0	272,0	145,0	241,0	115,0		154,3	309,0
	Wall heating	m²		10,0							
	Radiators	-							x		
	DHW with heat pump	-			x		x	x			x
	DHW with separate heat pump	-	x	x		x			x	x	
	Comfort ventilation	-	x	x	-	-	-	-	-	-	x
Monitoring results	Energy										
	Energy output	kWh	16840	11308	15191	11318	19250	22835	54008	11821	13036
	Energy output (heating)	kWh	16840	11308	13624	11318	16273	11871	50511	11821	10567
	Energy output (DHW)	kWh			1567		2977	10964	3497		2469
	SPF- seasonal performance factor										
	SPF 2	-	-	-	4,0	-	4,2	3,0	3,1	-	3,5
	SPF heating (SPF 2)	-	4,6	4,8	4,3	4,4	4,5	3,6	3,2	4,1	3,4
	SPF DHW (SPF 2)	-	-	-	2,4	-	3,1	2,5	2,6	-	3,6
	Temperatures										
	Av. heat pump outlet temperature	°C	28,6	30,4	33,7	30,7	36,9	31,6	34,3	35,1	31,2
	Av. heat source outlet temperature	°C	6,0	5,9	8,5	4,1	13,8	-	-	-	-
	Av. outdoor temp. (heating days)	°C	4,4	5,2	3,8	4,1	4,6	4,9	4,0	5,4	5,6
	Av. indoor temp. (heating days)	°C	23,2	20,9	22,5	23,7	21,6	23,3	18,4	23,2	21,8

Heat pump system No. 1 - Weinzierl

Heat Pump: In the heat pump system a brine-to-water heat pump is installed. The heat pump is filled with 2,35 kg of the refrigerant R407C and operates with a scroll compressor. According to the type plate, the heat pump has a heating capacity of 11,8 kW at the operation point B0/W35.

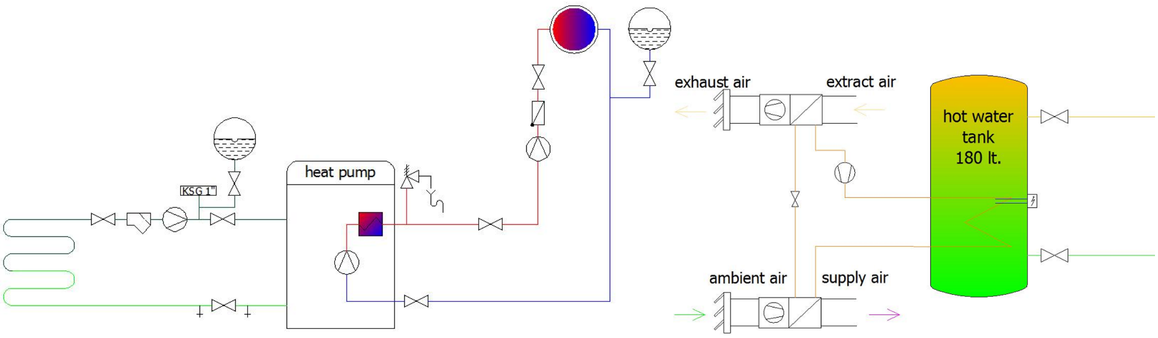


Fig. B 1: Scheme of the heat pump system "Weinzierl"

Heat sink system: The installation is operated using no backup heating system and is connected to the heat distribution system without any buffer storage. The circulation pump of the heat distribution system has an electric power consumption of 67 W. The heat sink system is designed as a floor heating with a heated area of 295 m^2 and as a wall heating with a heated area of 5 m^2.

Heat source system: The heat source system is a horizontal collector with an area of 380 m^2. The collector is arranged in 16 parallel collector circuits with a length of 75 m each. The collector pipes are installed in a depth of 1,4 m and have a diameter of 20 mm. The specific heat abstraction capacity was 25 W/m^2. For the circulation of the brine, a pump with an electric power consumption of 115 W is used.

Domestic hot water preparation: The hot water is prepared in combination with a comfort ventilation unit. This device, with active heat recovery and a 180 litre domestic hot water tank, has a 1 kW electrical auxiliary heating. The DHW temperature was set at 55°C. This separate DHW production was not taken into account during the measurements and evaluation of the system.

Heat pump system No. 2 - Lengenfeld

Heat Pump: In the heat pump system a brine-to-water heat pump is installed. According to the type plate the heat pump is filled with 2,35 kg of the refrigerant R407C and operates with a scroll compressor. The heat pump has a heating capacity of 11,8 kW at the operation point B0/W35.

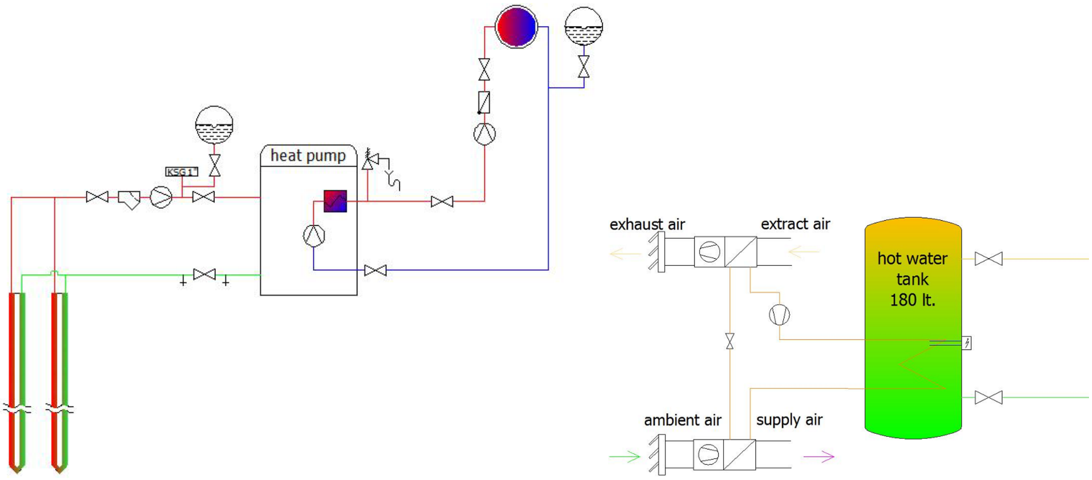


Fig. B 2: Scheme of the heat pump system "Lengenfeld"

Heat sink system: The installation operates without a backup heating system and is connected to the system without any buffer storage. The circulation pump of the heat distribution system has an electric power consumption of 93 W. The heat sink system is designed as a floor heating with a heated area of 210 m^2 and as a wall heating with a heated area of 10,5 m^2.

Heat source system: The heat source system is a simplex heat exchanger with a pipe diameter of 40mm. Two BHE were installed, each with a length of 76 m. The specific heat abstraction capacity was 61 W/m. For circulating the brine a pump with an electric power consumption of 130 W was used.

Domestic hot water preparation: The hot water was prepared in combination with a comfort ventilation unit. This device, with active heat recovery and a 180 litre domestic hot water tank, has a 1 kW electrical auxiliary heating. The DHW temperature was set at 55°C. This separate DHW production was not taken into account during the measurements and evaluation of the system.

Heat pump system No. 3 – Felling

Heat Pump: In the heat pump system a brine-to-water heat pump is installed. According to the type plate the heat pump is filled with 2,35 kg of the refrigerant R407C and operates with a scroll compressor. The heat pump has a heating capacity of 11,8 kW at the operation point B0/W35.

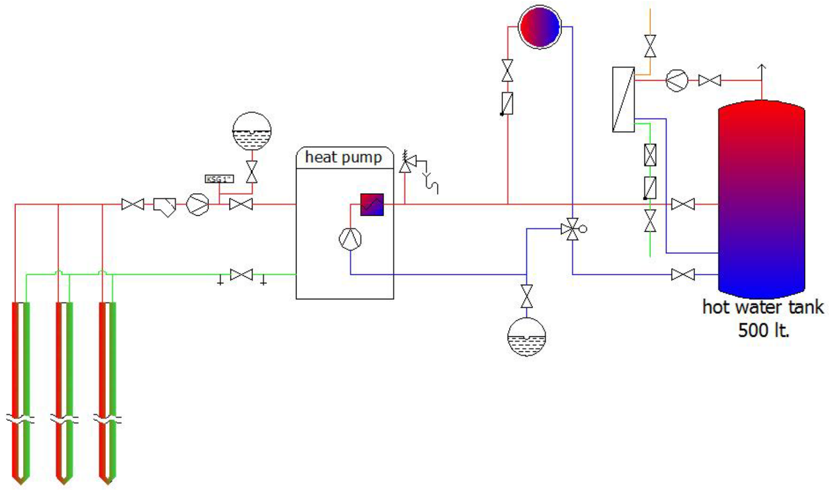


Fig. B 3: Scheme of the heat pump system "Felling"

Heat sink system: The installation is operated using no backup heating system and is connected to the heat distribution system without any buffer storage. The circulation pump of the heat distribution system has an electric power consumption of 55 W. During the design of the installation the maximum supply temperature was set at 35 °C and a temperature difference of 5 K in the heating system. The building has a specific heat load of 34 W/m^2. The heat supply system is designed as a floor heating with a heated area of 272 m^2.

Heat source system: The heat source is a simplex heat exchanger with a pipe diameter of 40 mm. Three BHE were installed, with a length of 58 m, 53 m and 49 m. The specific heat abstraction capacity was 57,5 W/m. For the circulation of the brine a pump with an electric power consumption of 130 W is used.

Domestic hot water preparation: The hot water is prepared with the heat pump which is also used for space heating combined with a 500 litre domestic hot water tank and is operated as a fresh water system. The DHW temperature was set at 48°C and the measured average temperature was 51.6°C.

Heat pump system No. 4 - Behamberg

Heat Pump: In the heat pump system a brine-to-water heat pump is installed. The heat pump is filled with 1,95 kg of the refrigerant R407C and operates with a scroll compressor. According to the type plate the heat pump has a heating capacity of 8 kW at the operation point B0/W35.

Heat sink system: The installation is operated using no backup heating system and is connected to the heat distribution system without any buffer storage. The circulation pump of the heat distribution system has an electric power consumption of 90 W. The heat sink system is designed as a floor heating with a heated area of 145 m^2.

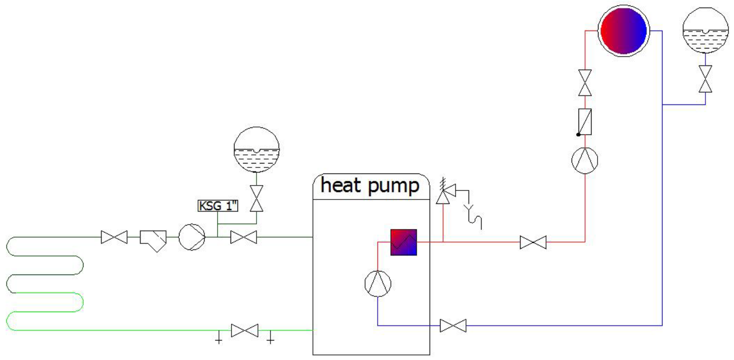


Fig. B 4: Scheme of the heat pump system "Behamberg"

Heat source system: The heat source system is a horizontal collector with an area of 200 m^2. The collector is arranged in eight parallel collector circuits with a length of 75 m each. The collector pipes with a diameter of 20 mm are installed in a depth of 1,3 m. The specific heat abstraction capacity was 31 W/m^2. For the circulation of the brine, a pump with an electric power consumption of 400 W is used.

Domestic hot water preparation: The domestic hot water production is done by an electric hot water tank. This device was not measured because of its independence from the heat pump.

Heat pump system No. 5 – St. Peter/Au

Heat Pump: In the system a water-to-water heat pump is installed. The heat pump is filled with 2,65 kg of the refrigerant R407C and operates with a scroll compressor. The heat pump has a heating capacity of 19,4 kW at the operation point W10/W35.

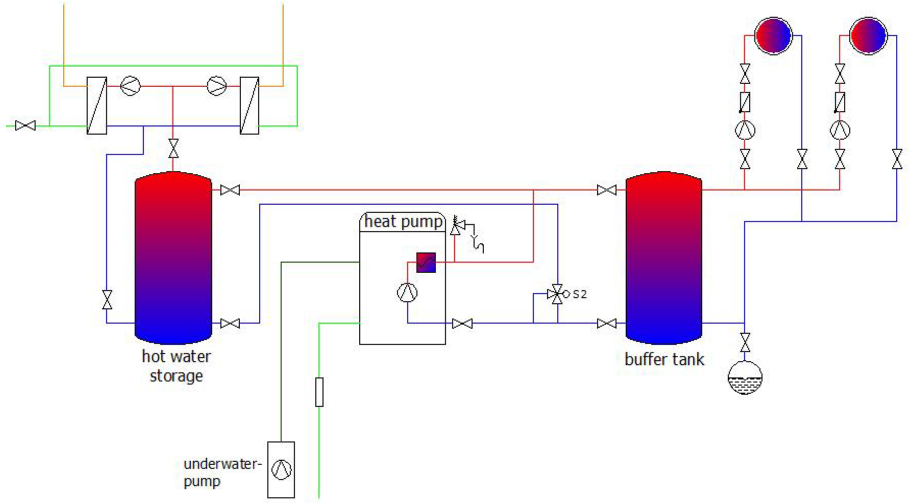


Fig. B 5: Scheme of the heat pump system "St. Peter/Au"

Heat sink system: The installation is operating without a backup heating system and is integrated to the heat distribution system with a buffer storage tank. The circulation pump of the heat distribution system has an electric power consumption of 90 W. The heat sink system is designed as a floor heating with a heated area of 241 m^2.

Heat source system: The heat pump system uses ground-water as heat source. Ground water is extracted out of a well and afterwards it is led in an absorbing well.

Domestic hot water preparation: The hot water is prepared with the heat pump which is also used for space heating combined with a 500 litre domestic hot water tank and is operated as a fresh water system. For each accommodation unit one module is installed. The DHW temperature was set at 48 °C and the measured average temperature was 54.9 °C.

Heat pump system No. 6 – Bad Vöslau

Heat Pump: An air-to-water heat pump is installed in this system. The heat pump is filled with 4,1 kg of the refrigerant R407C and operates with a scroll compressor. The heat pump has a heating capacity of 10,3 kW at the operation point A2/W35.

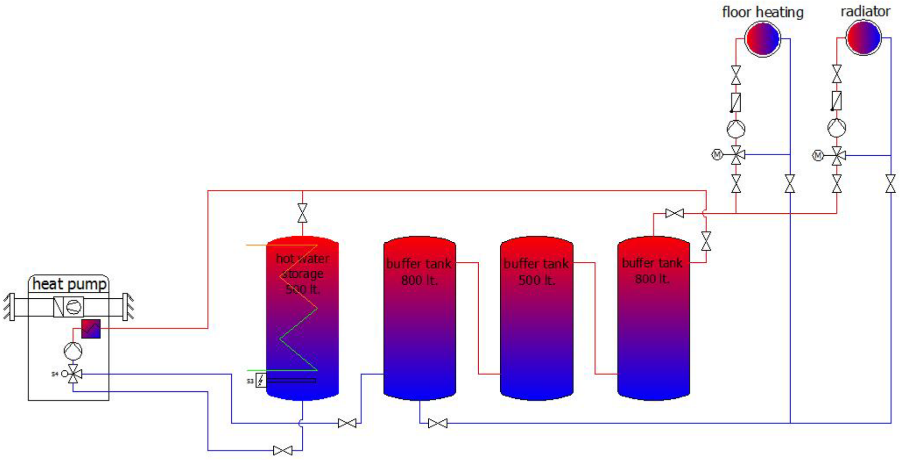


Fig. B 6: Scheme of the heat pump system "Bad Vöslau"

Heat sink system: The installation is operated using no backup heating system and is connected to the heat distribution system with a buffer storage tank. The circulation pump for the floor heating has an electric power consumption of 45 W on level 2. In the radiator heating circuit the circulation pump has an electrical power consumption of 30 W on level 1. The charging of the buffer storage tank is done by a circulation pump with an electrical power consumption of 93 W. The heat sink system is designed as a floor heating with a heated area of 160 m^2 and as radiators.

Heat source system: The used heat pump is an indoor compact unit and uses air as heat source. The outside air is led to the heat pump via a 2 m long flexible tube with a diameter of 500 mm.

Domestic hot water preparation: The hot water is prepared with the heat pump which is also used for space heating combined with a 500 litre domestic hot water storage tank. Additionally there is an electric backup-heater with a capacity of 6kW integrated. The DHW temperature was set at 58° and the measured average temperature was 58.5°C.

Heat pump system No. 7 – Klein Meiseldorf

Heat pump: An air-to-water heat pump is installed in this system. The heat pump is filled with 12,2 kg of the refrigerant R404A and operates with two parallel connected scroll compressors. The heat pump has a heating capacity of 33 kW at the operation point A2/W35.

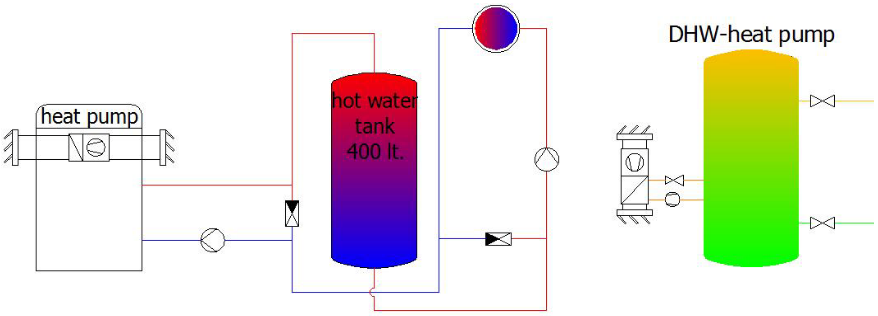


Fig. B 7: Scheme of the heat pump system "Klein Meiseldorf"

Heat sink system: The installation is operated using no backup heating system and has integrated a buffer storage tank with a capacity of 400 litres. The heat sink system is connected to radiators for a heated area of 400 m^2.

Heat source system: The used heat pump is a compact outdoor heat pump and uses air as heat source.

Domestic hot water preparation: The hot water is prepared by a separate air/water heat pump.

Heat pump system No. 8 - Ohlsdorf

Heat Pump: In the heat pump system a ground coupled direct expansion-to-water heat pump is installed. The heat pump is filled with 3,8 kg of the refrigerant R290 (Propane) and operates with a reciprocating compressor. The heat pump is equipped with a frequency converter and can be operated on two capacity levels. The heat pump has a heating capacity of 7 kW in step 1 and 14 kW in step 2. According to the type plate, the heat pump has a heating capacity of 15,1 kW at the operation point E4/W35.

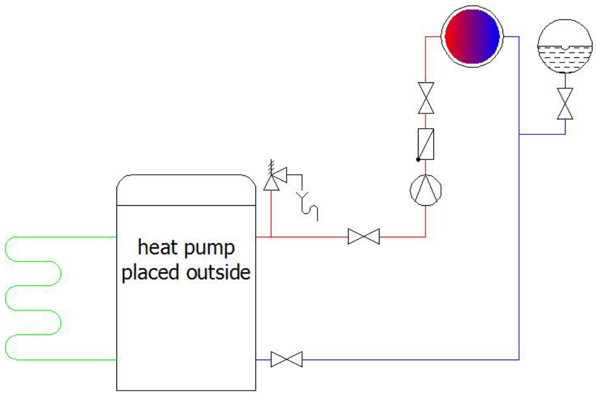


Fig. B 8: Scheme of the heat pump system "Ohlsdorf"

Heat sink system: The installation is operated using no backup heating system and is connected to the heat distribution system without any buffer storage. The circulation pump of the heat distribution system has an electric power consumption of 40 W. The specific heat load of the building is 42 W/m^2. During the design of the installation the maximum supply temperature was set at 35 °C and the return temperature was set at 30 °C. The heat sink system is designed as a floor heating with a heated area of 154 m^2.

Heat source system: The heat source system is a horizontal collector with an area of 270 m^2. The collector is arranged in six parallel refrigerant circuits with a length of 75 m each. The collector pipes with a diameter of 12 mm were installed in a depth of 1,2 m. The specific heat abstraction capacity was 22 W/m^2.

Domestic hot water preparation: The domestic hot water is heated by a separate air-to-water heat pump. The heating capacity of this heat pump is 1.850 W and operates with the refrigerant R134a. The hot water storage tank has a volume of 300 litres and the temperature was set at 55 °C.

Heat pump system No. 9 - Rutzenmoos

Heat Pump: An air-to-water heat pump is installed in this system. The heat pump is filled with 4 kg of the refrigerant R404A and operates with a scroll compressor. The heat pump has a heating capacity of 8,3 kW at the operation point A2/W35. Additionally, an 2,7 kW electric back up heater is integrated.

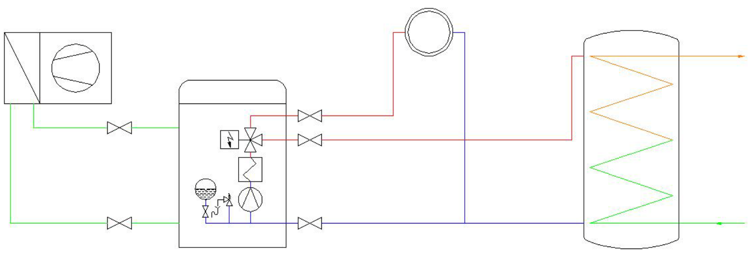


Fig. B 9: Scheme of the heat pump system "Rutzenmoos"

Heat sink system: The installation is operated using a backup heating system and is connected to the heat distribution system without any buffer storage. The circulation pump of the heat distribution system has an electric power consumption of 80 W. During the design of the installation the maximum supply temperature was set at 40 °C. The heat supply system is designed as a floor heating with a heated area of 309 m².

Heat source system: Air is used as heat source. The heat pump system is an air to water split heat pump with an outside stationed evaporator.

Domestic hot water preparation: The hot water was prepared with the heat pump which is also used for space heating. It heats a hot water storage tank with a volume of 800 litres. Domestic hot water is prepared according to the instantaneous water heater principle with an adjusted temperature of 45°C, the measured average temperature was 42.5°C.

Compact units

Tab. B 3: Overview of the field test results of ground-coupled compact units for SH, DHW, ventilation and passive SCin the Austrian project at AIT (Zottl et al., 2010)

	No.	-	No. 10:B/W	No. 11:B/W
Object	Location	-	Hitzendorf	Judendorf
	Heat load	kW	3,8	3,5
	Heating area	m²	180	210
	Specific heat load	W/m²	21	17
	Inhabitants	-	4	4
	Building construction	-	me-dium	me-dium
	Commissioning	-	2008	2008
Heat pump	Heat pump capacity	kW	6,3	6,3
	A/W	-		
	B/W	-	x	x
	W/W	-		
	DX/W	-		
Heat source	Horizontal collector	-	x	x
	Vertical collector	-		
Heat sink	Floor heating	m²		130,0
	Wall heating	m²	80,0	
	Radiators	-	x	x
	DHW with heat pump	-		
	DHW with separate heat pump	-	x	x
Monitoring results	Energy			
	Energy output	kWh	12700	9369
	Energy output (heating)	kWh	6302	5823
	Energy output (ventilation)	kWh	1804	918
	Energy output (DHW)	kWh	3511	1539
	Energy output (cooling)	kWh	1083	1089
	SPF- seasonal performance factor			
	SPF 1		4,8	4,6
	SPF 2	-	4,3	4,1
	SPF heating (SPF 2)		4,7	4,3
	SPF DHW (SPF 2)	-	3,6	3,7
	SPF 3	-	3,7	3,4
	SPF (cooling)		4,7	9,0
	Temperatures			
	Average heat pump outlet temperature	°C	35,4	29,2
	Average heat source outlet temperature	°C	6,6	2,9
	Average outdoor temp. (heating days)	°C	4,4	4,7
	Average indoor temp. (heating days)	°C	23,2	21,9

Compact unit No. 10 - Hitzendorf

This compact unit combines the functions of ventilation, DHW, space heating and passive cooling in one unit.

Heat Pump: In the heat pump system a brine-to-water heat pump is installed. According to the type plate the heat pump is filled with the refrigerant R134a and operates with a scroll compressor. The heat pump has a heating capacity of 3,3 kW at the operation point B0/W35.

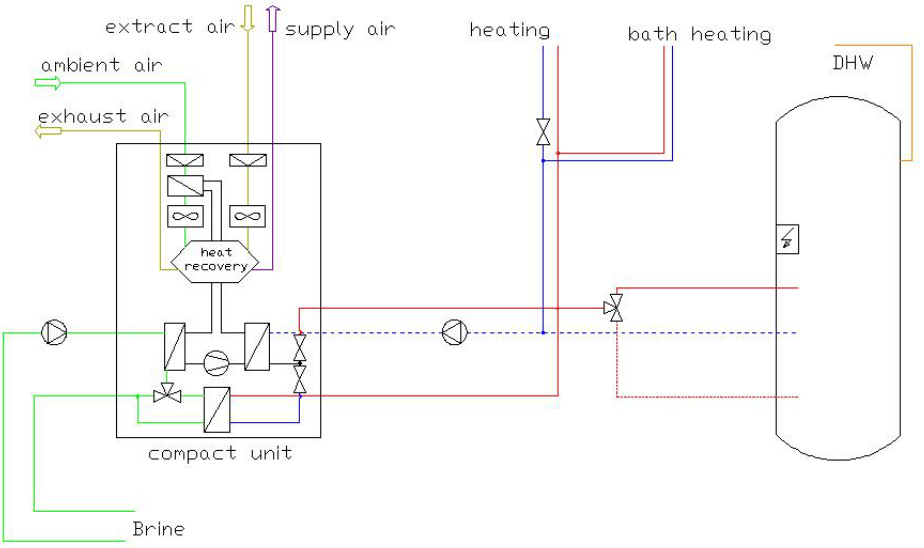


Fig. B 10: Scheme of the compact unit (Hitzendorf)

Heat sink system: The heat supply system is designed as a wall heating with a heated area of 80 m^2.

Heat source system: The heat source is a horizontal collector placed in the working space around the building. The collector is arranged in 4 strings one upon another with a total length of 350 m. Collector pipes with a diameter of 1 inch were installed on 4 levels with a distance of 70 cm.

Domestic hot water preparation: The hot water was prepared with the heat pump which is also used for space heating. It heats a hot water storage tank with a volume of 560 litres. Domestic hot water is prepared according to the instantaneous water heater principle. A part load of the DHW heat energy is used for the bath-heating.

Ventilation: The nominal air quantity is 160m^3/h and the maximal air quantity at 170 Pa (extern) is 235m^3/h.

Cooling: Passive cooling is done by the use of the heat source and heat sink system through a serially integrated heat-exchanger.

Compact unit No. 11 - Judendorf

This compact unit combines the functions of ventilation, DHW, space heating and passive cooling in one unit.

Heat Pump: In the heat pump system a brine-to-water heat pump is installed. According to the type plate, the heat pump is filled with the refrigerant R134a and operates with a scroll compressor. The heat pump has a heating capacity of 3,3 kW at the operation point B0/W35.

Heat sink system: The heat supply system is designed as a floor heating with a heated area of 130 m^2.

Heat source system: The heat source is a horizontal collector placed in the working space around the building. The collector is arranged in strings one upon another with a total length of 200 m. The collector pipes with a diameter of 1 inch were installed on levels with a distance of 70cm.

Domestic hot water preparation: The hot water was prepared with the heat pump which is also used for space heating. It heats a hot water storage tank with a volume of 300 litres. Domestic hot water is prepared according to the instantaneous water heater principle.

Ventilation: The nominal air quantity is 160m^3/h and the maximal air quantity at 170 Pa (extern) is 235m^3/h.

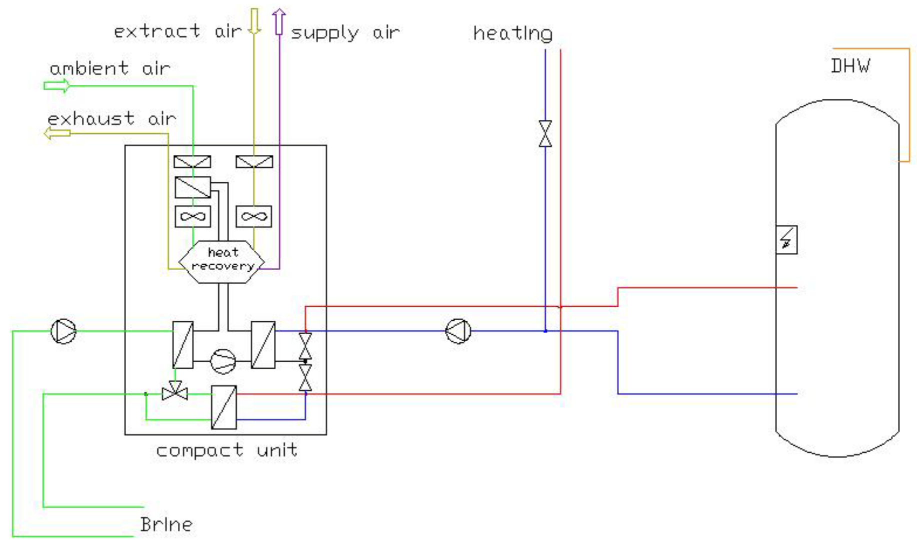


Fig. B 11: Scheme of the compact unit (Judendorf)

Cooling: Passive cooling is done by the use of the heat source and heat sink system through a serially integrated heat-exchange.

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